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30 June 2013

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ENGINEERING AND DESIGN

Mechanical and Electrical Design for Lock and Dam Operating Equipment

ENGINEER MANUAL

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DEPARTMENT OF THE ARMY
U. S. Army Corps of Engineers
Washington D. C. 20314-1000

EM 1110-2-2610

Manual
No. 1110-2-2610

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Engineering and Design
MECHANICAL AND ELECTRICAL DESIGN FOR
LOCK AND DAM OPERATING EQUIPMENT

1. Purpose. This manual provides guidance in the mechanical and electrical design of navigation lock and dam operating equipment and control systems for both new construction and the rehabilitation of existing projects.

2. Applicability. This manual applies to all HQUSACE elements, major subordinate commands, districts, laboratories, and field operating activities having responsibilities for the design and construction of civil works projects

3. Distribution Statement. This manual is approved for public release with unlimited distribution.

4. References. References are in Appendix A. References in Appendix A, Part 1, are provided as attachments in electronic PDF format.

5. Background.

a. Function of lock gates and valves. Lock gates and valves serve a number of different functions, depending on location and conditions. A navigation lock requires operable closure gates at both ends of the lock so the water level in the lock chamber can be varied to coincide with the upper and lower approach channels. Mechanical and electrical equipment provides the means to accomplish this. The various chapters of this manual describe a number of different types of mechanical and electrical equipment. A navigable floodgate is the same as a lock gate, except there is only a set of gates and machinery and no lock chamber. These are typically sector gates and vertical lift gates. The machinery to drive lock gates and valves can be of a variety of designs, depending on existing site infrastructure and designer or operator preferences. The common attributes that should be considered in all cases and are emphasized in this manual are reliability, longevity, and ease of maintenance.

b. Function of navigation dam gates. Navigation dam gates are the movable portion of a navigation dam that is used to regulate the upstream pool to maintain a minimum

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This manual supersedes EM 1110-2-2610, dated 12 December 2003, and the electrical and mechanical portions of EM 1110-2-2703, dated 30 June 1994.

navigation depth. Again, the mechanical and electrical equipment provides the means to accomplish this. Navigation dam gates are generally non-navigable, except when they are not required for pool maintenance and can be lowered completely and there are no pier obstructions (i.e. wicket gates). Reliability, longevity, and ease of maintenance are also emphasized for dam gate machinery.

FOR THE COMMANDER:

Cheryl L. Partee

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CHERYL L. PARTEE
Chief of Staff

DEPARTMENT OF THE ARMY EM 1110-2-2610
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CHAPTER 1

Introduction

1-1. Purpose. This manual provides guidance for the mechanical and electrical design of navigation lock and dam operating equipment and control systems for both new construction and the rehabilitation of existing projects. In some cases, information is provided on equipment that is rarely used for new installations. Such information is given not to advocate the replacement of serviceable equipment, especially where not economically or functionally justified, but to provide information to help facilitate rehabilitation, to lend historical perspective, or to give some insight into innovations in lock machinery design in other countries. Chapters are included to provide information on mechanical components, hydraulic drives, miter gate operating machinery, sector gate operating machinery, filling and emptying valves and machinery, vertical lift gate operating machinery, and dam gate operating machinery. Chapter 8 provides guidance on some gates less commonly used in the United States, including barge gates and pocket gates or rolling gates. Chapter 10 provides guidance on systems ancillary to the lock gate operating machinery. Electrical chapters supply guidance on power distribution, equipment and machinery controls, and electrical support systems. Operation, maintenance and inspection information as it relates to equipment design and layout, is provided in Chapter 14.

1-2. Applicability. This manual applies to all HQUSACE elements, major subordinate commands, districts, laboratories, and field operating activities having responsibilities for the design and construction of civil works projects.

1-3. References. References are in Appendix A. Appendix A is broken into two parts. Part 1 includes references that may be difficult to locate and/or out of print. Electronic versions (PDF) of these references are included as attachments. Part 2 includes references to current engineering manuals, industry standards, text books, etc. Appendix A also includes a list of applicable Unified Facilities Guide Specifications (UFGS). These can be found at the Whole Building Design Guide website of the National Institute of Building Sciences.

1-4. Plates. Plates presenting general information, typical details, mechanical design data, and sample computations are in Appendix B. These plates are referred to herein as Plate B-1, B-2, etc.

1-5. Calculations. Sample calculations supporting information in the manual are in Appendix C and are referred to herein as Plate C-1, C-2, etc. These calculations typically span multiple pages but when mentioned in the text will be referred to by the first plate number in the series.

1-6. General. The lock and dam gate operating equipment information presented in this document is a revision and update of the information presented in the 30 June 1994 version of EM 1110-2-2703 and the 12 December 2003 version of EM 1110-2-2610. The information presented herein is, for the most part, based on years of actual experience of similar equipment and systems currently in use. The only exceptions are the information presented for the hydraulic-cylinder-operated wicket gate and the hinged crest gate. The wicket gate information is based on a full-scale facility built to test different materials and operating arrangements, (see report titled “Results of the Olmsted Hydraulic Operated Wicket Dam” in Appendix A for additional information). Information about the hinged crest gate came from Montgomery Point Lock and Dam.

a. Function of Lock Gates and Valves. Lock gates and valves serve a number of different functions, depending on location and conditions. Locks often are adjacent to or in close proximity to a dam, but this is not always the case. While the major use of lock gates is to form the damming surface across the lock chamber, they may also serve as guard gates, for filling and emptying the lock chamber, for passing ice and debris, to dewater the lock chamber, and to provide access from one lock wall to the other by means of walkways or bridge ways installed on top of the gates. A navigation lock requires closure gates at both ends of the lock so that the water level in the lock chamber can be varied to coincide with the upper and lower approach channels. The sequence of “locking” a vessel upstream is to: first, lower the water level in the lock to the downstream water level; second, open the lower gate and move the vessel into the lock chamber, third, close the lower gate and fill the lock chamber to the level of the upper pool; and finally, open the upstream gate and move the vessel out of the lock. Lockage of a vessel downstream involves a similar sequence in reverse order. A navigable floodgate is the same as a lock gate except there is only a set of gates and machinery and no lock chamber. These are typically sector gates. The machinery to drive lock gates and valves can be of a variety of designs depending on existing site infrastructure and designer or operator preferences. The common attributes that should be considered in all cases, as emphasized in this manual, are reliability, longevity, and ease of maintenance.

b. Function of Navigation Dam Gates. Navigation dam gates are the movable portion of a navigation dam that is used to regulate the upstream pool to maintain a minimum navigation depth. Navigation dam gates are generally non-navigable except when they are not required for pool maintenance and can be lowered completely and there are no pier obstructions, (i.e., wicket gates). Again, this manual emphasizes the importance of the design for reliability, longevity, and ease of maintenance, regardless of the gate or machinery design.

c. Types of Gate Machinery Covered.

(1) Miter gates and machinery. A large percentage of the locks in the United States are equipped with double-leaf miter gates, which are used for moderate and high-lift locks. The construction and operation of these gates are fairly simple, and these gates can be opened or closed more rapidly than any other type of gate. Maintenance costs generally are low. A disadvantage of this gate is that it cannot be used in an emergency situation, to close off flow with an appreciable unbalanced head. Further miter gate description and structural design information are in EM 1110-2-2105.

(a) When miter gates are open, they fit into recesses in the wall. The bottom of the recess should extend below the gate bottom to preclude operating difficulties from silt and debris collection. Enlarged recesses sometimes are used to facilitate the removal of accumulated ice. An air bubbler system is recommended to help clear ice and debris from gate recesses (see Chapter 10 for a description of a typical air bubbler recess flusher).

(b) Miter gate machinery arrangement typically falls into two broad categories: a directly connected linear actuator and a linkage connected to the gate and driven by a linear or rotary actuator. Actuators can be either direct electric or hydraulic fluid power driven.

(2) Sector gates and machinery. A sector gate is similar in shape to a tainter gate except that it is oriented to rotate about a vertical axis and is supported at the top and bottom in a manner similar to that of a miter gate. Like miter gates, sector gates are used in pairs, meeting at the center of the lock when closed and swinging into recesses in the lock wails when open. The trunnions are located in the lock walls, and the skin plates face in the direction of the normally higher pool level. Further sector gate description and structural design information are presented in EM 1110-2-2105.

(a) Sector gates are used at both ends of locks in tidal reaches of rivers or canals where the lifts are low and where the gates might be subjected to reversal of heads. Since these gates can be opened and closed under head, they can be used to close off flow in an emergency. The gates swing apart and water flows into or out of the lock through the center opening between the gates. In some cases, flow is admitted through culverts to improve filling characteristics or where ice or drift might not permit adequate flow between the gates.

(b) Because the turbulence area at the upper end of a lock filled by a sector gate, created by flow into the lock through the gate opening, impacts the lockage of vessels, the length of the lock chambers must be increased proportionately. This turbulence can shift the vessel and possibly break mooring lines. Model tests indicate about 100 ft of additional length is required. Like other end-filling systems, sector gates cannot be used for filling and emptying high-lift locks unless the filling and emptying rates are greatly reduced. The practical lift limitation is usually about 10 ft, although gates with higher lifts have been built.

(c) The disadvantages of the sector gates are high construction cost, long opening and closing times, and larger wall recesses.

(d) Sector gates typically are driven by rotary actuators, electric or hydraulic motors, driving gear reducers that in turn drive a pinion mated with a rack bolted to the radius of the gate. Alternate arrangements pull the gate in and out of recess with wire ropes wrapped on a drum or utilize a hydraulic cylinder directly connected to the top frame of the gate near the hinge.

(3) Vertical lift gates and machinery. Vertical lift gates may be used at both ends of a lock or at only one end in combination with a miter gate at the other. They can be raised or lowered under low-to-moderate heads but are not used when there is reverse head. The operation time of older gates is much slower, and maintenance costs are higher than those of miter gates, but they can be used in emergency closure. The newer gates, however, are capable of achieving operating speeds equal to, or even faster than, miter gates. Further vertical lift gate description and design information is in EM 1110-2-2105.

(a) A vertical lift gate installation at the upstream end of a lock normally consists of a single or double-leaf submergible gate, which rises vertically to close off the lock chamber from the upper pool. When the lock is filled, the gate is opened by sliding the leaf vertically downward until the top of the leaf is at or below the top of the upper sill.

(b) In some cases, a double-leaf vertical lift gate can be used. The upper leaf can have a curved crest, which permits overflow to supplement flow from the primary filling system when the lock chamber is nearly full. This type of gate also can be used for skimming ice and debris.

(c) When a vertical lift gate is used at the downstream end of a lock, it is raised vertically to a height above the lower pool level so that vessels can pass underneath. The gate leaf is suspended from towers on the lock walls and might be equipped with counterweights to reduce the power hoist size. Lock gates of this type are practical only for very high locks and where required vertical clearance can be provided under the gate in its raised position.

(d) Vertical lift gates most commonly are driven by a electric-motor-driven wire rope hoist that lifts both sides of the gate with a single hoist drum connected by reeving and a tunnel under the lock. Another drive type utilizes hydraulic cylinders on both sides of the gate. A variation of this type is hydraulic cylinders on each side of the gate connected through wire ropes and reeving. Another hoist arrangement uses a bullwheel or friction wheel driving a wire rope connected to the gate and a counterweight to reduce the hoisting loads. With these types or any other type that uses an independent or semi-independent hoist on each side of the gate, synchronization of the two hoists becomes something that must be accounted for in the design.

(4) Tainter gates. Tainter gates are a type of radial arm gate in which the convex side of the gate faces upstream. The hydraulic forces act through the radius of the gate and are concentrated on the pivot point or trunnion. Further tainter gate description and design information is in EM 1110-2-2105.

(a) Tainter gates as navigation dam gates. Tainter gates are widely used on navigation dams to control the upstream pool. Tainter gates require a lower hoist capacity and have a relatively faster operating speed than other types of dam gates. Also, because side seals are used, gate slots are not required. This reduces problems associated with cavitation, debris collection, and ice buildup.

(b) Submergible tainter gates as lock gates. The locks of The Dalles Dam, some Lower Snake River projects, and the Upper St. Anthony Falls Locks have submergible tainter gates. This type of gate is raised to close the lock chamber and lowered into the lock chamber to open it. The gates are also used to pass water through the lock during flood conditions. The end frames are recessed into the lock wall so no part of the end frame projects into the passageway. This type of gate was chosen for these projects because it is structurally efficient and is estimated to be lighter in weight and less costly than a double-leaf miter gate. The tainter gate also permits the length of the approach channel to be reduced by the leaf width of the miter gate. There are two potential problems in the operation of this type of gate: skewing of the gate during opening and closing, and vulnerability to damage if hit by lock traffic. However, with good design practices and lock management, these problems will be minimal.

(c) Tainter gate machinery. Tainter gates traditionally have been operated by hoists located in machinery houses above the gates and connected to the gates with wire rope or chain. Chain is no longer used in new installations because of past maintenance problems. More recently, hydraulic cylinder hoists have been utilized. Some challenges have included the vulnerability of the cylinder rods to impact damage and to corrosion caused mainly by a combination of improperly selected rod materials and the infrequency of use of some gates.

(5) Rolling/sliding gates and machinery.

(a) Rolling/sliding gates. Rolling or sliding gates for navigation locks are stored in the lock wall perpendicular to the axis of the lock and move across the lock to create a damming surface. This type of gate becomes an option when the width of the lock exceeds that practical for miter gates or when the area available for gate monoliths is limited. Although rolling or sliding gates were commonly used in the United States in the early 20th century, they now are primarily used in Europe on ship locks and for the new locks that are part of the Panama Canal expansion.

(b) Rolling/sliding gate machinery. Rolling gates typically utilize multiple-wheel trucks rolling on tracks on the bottom of the gate or, in the case of a wheel barrow gate, on the bottom of one end and at the top of the gate at the other end. Buoyancy tanks often are built into the gate to decrease the wheel loads and required hoist capacity. Sliding gates do not use wheels but slide on hydrodynamic bearings. This type has been successful in Europe. The gate machinery typically consists of electric-motor-driven hoists that drive double-wire rope drums either pulling the gate open or closed.

(6) Culvert valves and machinery.

(a) Culvert valves. Culvert valves are used to control the flow of water through the lock's filling and emptying system to raise or lower the lock water elevation. A variety of valves have been used for this purpose. For large locks, the most efficient has been the reverse tainter valve, which is similar to the tainter gate but with the convex side of the valve facing downstream. Other culvert valves used have included regular orientation tainter gates, butterfly valves, vertical slide gates, caterpillar gates, and stoney gate valves. The choice depends on site constraints, culvert size, and pool elevations. Further information on the hydraulic design of lock culvert valves can be found in EM 1110-2-1610.

(b) Culvert valve machinery. Machinery for opening and closing the culvert valves can be driven either electrically or hydraulically. Hydraulic forces must be designed to ensure that adequate hoist power is available under all conditions to both open and close the valve under all conditions.

d. Control Systems. Chapter 12 features information on control systems. This information provides the means to develop control systems for new projects and for the replacement or upgrading of existing systems. It is written for navigation lock and dam application, but much of the same technology and design information can be adapted to other Corps civil works projects.

e. Relationship to Other Manuals. This manual supersedes all previous versions of EM 1110-2-2610 and the mechanical and electrical information in EM 1110-2-2703. It should be used in conjunction with EM 1110-2-2105 and all other referenced engineering manuals for the design of gates, operating machinery, and control systems. Other manuals applicable to the design of navigation locks and dams are listed in Appendix A.

1-7. Mandatory Requirements and Deviation from Design Criteria. This manual provides guidance for the protection of U.S. Army Corps of Engineers (USACE) structures. In certain cases, guidance requirements, because of their criticality to project safety and performance, are considered to be mandatory as discussed in ER 1110-2-1150. Those cases will be identified as “mandatory,” or the word “shall” will be used in place of “should.”

a. When the project delivery team (PDT) determines that designing to less than the mandatory requirements stated in this manual is appropriate and reasonable, a cover letter along with the supporting cost and risk analysis documentation shall be submitted to USACE-HQ requesting exemption. See the paragraph below for requirements for dam safety studies, major rehabilitation studies, and reliability (issue evaluation) studies.

b. Dam studies, Major Rehabilitation Studies, and Issue Evaluation studies will require an evaluation and risk assessment of the entire lock and dam structure or portions thereof. The mechanical and electrical components may be excluded, screened out, or sometimes the results of these studies may require an analysis of lock and dam machinery. If an analysis is required, it should include a review of the design criteria utilized to build the machinery. If it is found that the design criteria for the machinery has changed since it was originally constructed or last upgraded and is not consistent with current criteria, further investigation may be warranted. This further investigation, if required, should include an assessment of the performance of the machinery over the life of the project under all conditions. Furthermore, an assessment of the risk accepted by not upgrading the machinery to meet new criteria should be evaluated within the context of the study being conducted. This performance and risk assessment should be documented. The old and current design criteria should also be documented and deviations from current criteria should be noted where warranted. A decision to upgrade the machinery solely on the need to meet current criteria should be based on this assessment. A budgetary cost analysis should be provided as part of the study to bring machinery and components up to current standards. If upgrades to mandatory requirements are not recommended based on performance and risk assessment, then concurrence must be obtained from USACE-HQ. Deviations from mandatory safety related criteria such as interlocks should be documented separately.

c. Operation and Maintenance Work. Often, portions of mechanical and electrical machinery are replaced such as motors or a gear or a gear box. The existing project design criteria may be followed. New design criteria should be followed to the extent possible but is not mandated. If Operation and Maintenance funding is utilized to replace the entire set of lock machinery or dam machinery, then new design criteria should be followed.

CHAPTER 2

Mechanical Components

2-1. <u>General Description and Application</u>.

a. General Design Criteria. The components and design criteria described herein are applicable to components the designer might deem appropriate for use in both electromechanical and combination hydraulic-mechanical systems used to operate navigational locks and spillway/dam gates. For a more thorough discussion of components strictly applicable to hydraulic drive systems, see Chapter 3. The list of components provided is not intended to be all inclusive for the designer, rather it is intended to present a list of commonly utilized components that have design parameters established to meet a broad range of civil works projects. This chapter applies to all HQUSACE elements, major subordinate commands, districts, laboratories, and field operating activities having responsibilities for the design and construction of both new and rehabilitated civil works projects. Additional specific guidance for the material selection of various mechanical system components is in ERDC/CERL TR-02-7, Advanced Materials Selection Guide for Lock, Dam and Hydroelectric Plant Components, January 2002. Guidance for reliability analysis of navigation lock and dam mechanical and electrical equipment is contained in ETL 1110-2-560. This ETL has expired but the information will be incorporated into the new EC 6062, Engineering Risk and Reliability, that is currently under development.

b. Machinery Component Criteria. All components of lock and spillway/dam operating equipment should be designed for the maximum normal full load,100% rated, torque of the electric motor, or the maximum effective hydraulic actuator pressure, with a minimum factor of safety of five (5.0), based on the ultimate tensile strength of the material when designed by the Corps of Engineers. Where OEM products are specified and published rating data is available, the designer must use caution in blindly applying factors of safety to OEM components that already have inherent service factors provided for in their design. This is to avoid having the designer grossly oversize a component through the application of safety factors upon safety factors.

(1) All components should be designed for a unit stress not to exceed 75% of the yield strength of the material, using the locked rotor torque rating of the electric motor, or the maximum hydraulic actuator pressure available through the control system. A fracture mechanics/fatigue analysis also should be performed to help identify localized stress concentrations that will govern the percentage of yield strength established for the lower limit strength. Components that might fail in buckling compression should be designed for a minimum factor of safety of three (3.0), using the Euler or J.B. Johnson formulas, as appropriate. The factor of safety shall be applied to the maximum load on the member and the critical buckling load. In almost all cases, the end fixity coefficient for pin-ended columns should be used. These criteria determine the maximum allowable stresses for all components. Components used as fuses, such as some shear bolts, keys, torque-limiting couplings, etc., will not be designed to these criteria.

c. Standard Manufactured Products. All standard manufactured products should be selected based upon the manufacturer's published catalog ratings, or actual data procured by correspondence with all known major manufacturers of that type of component. The intent is to provide open competition for standard manufactured items, while permitting the designer to use available data to provide a fully functional design. Plans and specifications should be presented in a manner that defines a range of performance obtainable by a majority of the manufacturers. The designer should be aware of product delivery times through correspondence with known major manufacturers to allow for a realistic time line. Delivery information will aid in estimating the construction schedules for the prescribed contract period and development of any required interim completion dates. Where applicable, the designer should consider identifying the design basis for any specialty equipment incorporated in the design. The design basis should be used with the salient characteristics included in the drawings and specifications for the benefit of the contractor. The purpose of identifying the design basis is to leave no doubt for the construction contractor as to the designer's intent.

d. Component Efficiencies. The following operating efficiencies should be used as a design guide:

- Silent Chain (including oil-retaining case) 97%

- V-belts (including drive/driven sheaves) 90%

- Spur Gear Reduction Unit (up to):
  1:1 to 16:1 88%,
  16:1 to 40:1 84%,
  40:1 to 150:1 78%,
  Helical, Herringbone or Planetary Reduction Unit,
  Single Reduction 97%,
  Double Reduction 95%,
  Triple Reduction 90%,
  Quadruple Reduction & Worm Gear Reduction Unit 1,
  Spur Gear Set 97%,
  Bevel Gear Set 95%.

- Bearings
  Ball and Roller 98%
  Bronze Plain Bearings (> 5 rpm) 95%
  Bronze Plain Bearings (< 5 rpm) 93%

Certified starting and operating efficiencies should be obtained from manufacturers' data for the normal operating speeds. Special operating conditions, such as high or low ambient temperatures, or lubricant heaters, should be coordinated with the manufacturers' engineering departments. The lowest efficiency obtained from a minimum of three standard manufacturers should be used.

e. Mechanical and Structural Coordination and Equipment Forces. Forces from equipment start up can be large and can impart stresses to structures that are not designed for such forces. The designer needs to be aware of the forces not only at start up but under overload conditions and how that affects structural components. Controls, limit devices, and interlocks not only protect machinery but also ideally would eliminate many unintentional consequences to structural components. This includes damage to expensive structural gates and machinery foundations. Assets are better managed and protected with properly designed controls and safety interlocks.

2-2. Machinery Components.

a. Bearings.

(1) Antifriction Bearings. Ball, roller, tapered roller and spherical roller bearings should be selected in accordance with the manufacturer’s published catalog ratings of the group, type, and size required. The manufacturer's ratings for loads and speeds shall be used in determining the bearing capacity. Service and installation factors shall be in accordance with the bearing manufacturer's recommendations. The L-10 bearing life shall be a minimum of 75,000 hours, based on the largest full load motor horsepower provided by the specified motor. Bearings, which remain static for extended periods, should be designed with greater safety factors, using the Basic Static Load Rating (Cor). The B-10 life and load ratings calculation methods are defined in specifications ISO 281 or JIS B1519. All bearings shall be equipped with labyrinth seals, to exclude foreign matter and retain lubrication without leakage under both static and dynamic operating conditions. Only one fixed mount bearing should be used on shafts with multi-bearing installations to permit thermal expansion in the axial direction. Manufacturers should be consulted for typical axial capacities of the bearings.

(2) Plain Bearings. Plain bearings, also called sleeve bearings or bushings, should be designed for a maximum normal bearing pressure of 6.9 MPa (1000 psi), except for bearings operating below five (5) rpm. Under special, slow speed, uniform load conditions, the bearing pressure may be designed for up to 27.6 MPa (4000 psi). Common bearing materials to be specified for their strength, high load carrying capacities, good wear, and corrosion resistance include the tin bronze alloys. Alloy C90500 is suitable for most slow-to-moderate speed applications, in accordance with ASTM B271, ASTM B505, or ASTM B584. The tin bronzes should be used with reliable grease lubrication systems because they do not imbed contamination particles well and do not tolerate shaft misalignment. Examples of plain bearings for miter gate and trunnion pin connections are shown in Figures 2-1 and 2-2. Mating shaft material should be specified with a hardness of 300-400 BHN. Special materials, or pressure designs, should be coordinated with several manufacturers to ensure adequate competition is available.

(a) Where an easily machined bronze or the environment and lubrication might be questionable, the designer may elect to utilize high-leaded tin bronzes. Alloy C93200 is an exceptional bronze for medium loads and speeds. The alloys C93800 and C94300 are 15% and 25% lead. Their high lead percentages allow them to easily conform to misaligned shafts and embedded contaminants. They provide a high level of lubricity under poor lubrication conditions and can be used with unhardened shafts. A high lead poured and scraped babbitt bearing for a low rpm pump is shown in Figure 2-3.

(b) The length to diameter ratio (L/D) should be designed close to unity (1.0), considering the bearing pressure required, in order to minimize wear and misalignment. Some consideration should be given to spherical plain bearings for such things as tainter gate trunnions, bellcranks, struts, and other partial-rotation, slow-motion joints that require an extra degree of freedom. Spherical bearings minimize wear between pins and bushings by accommodating modest misalignment.

(c) Grease grooves should be designed and incorporated into the bearings to provide redundant grease pathways. The pathways should be capable of delivering the grease around the entire circumference of the bearing and mating surface without relying on rotation of the components. Delivery of the grease through the pin to the bearing grease grooves in lieu of delivery through the bearing housing should be the preferred method where feasible. Pathways through the bearing housing can become cut off or comprised, should the bearing ever rotate within the housing during the life of the bearing. Extended grease lines for pintles, bushings, and bearings need to be oversized to compensate for grease hardening over the life of the project and to ensure adequate grease delivery. The grease lines should be rigid in areas prone to damage from debris and ice. Schedule-80 stainless steel rigid piping has proven adequate in many installations. The piping should be routed to provide the maximum protection to the line while also maintaining accessibility for replacement by divers, should damage occur. The use of flexible lines should be minimized and shall be armored and rated for the maximum anticipated pressure of the greasing system.

Figure 2-1. Plain bearing

Figure 2-2. Plain bearing

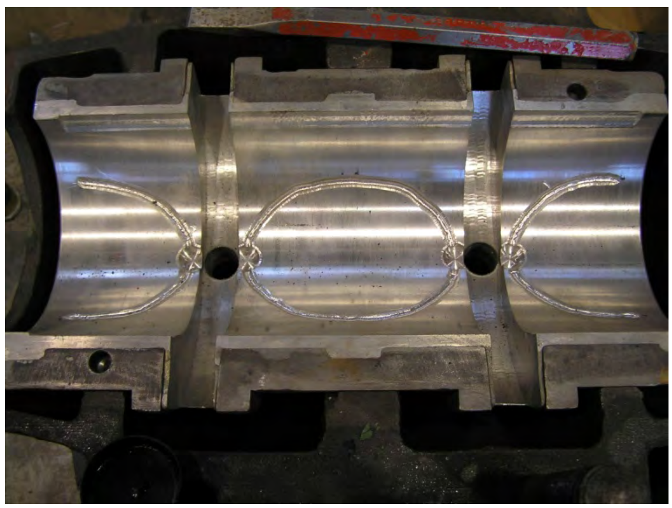


Figure 2-3. Babbitt plain bearing

(3) Self-lubricated Bearings. Self-lubricated bearing materials, also known as composites or greaseless bearings, have been produced for years. The manufacturer's materials often have wide variations in the properties and quality of materials. The designers of self-lubricated bearings should review the assessment documents of failures at different Corps projects to consider lessons learned when initiating a new design. The failures within the Corps have been in applications where the material has been used for pintle bearings and other sleeve-type bearings. Some examples of failed gudgeon pin and pintle bushings are shown in Figures 2-4 and 2-5. When considering the use of self-lubricating bearing materials, the design must be engineered thoroughly by the Corps designers or by the specific bearing manufacturer and the designer should not rely on the general contractor to furnish an adequately designed bearing. If the bearing is to be designed by the bearing manufacturer, sufficient data must be furnished to the manufacturer by the government to ensure a successful design. The bearing

design should be tested by a third-party independent laboratory. Self-lubricating bearing systems should be restricted to those that have been tested and approved under the test procedure "Performance Evaluation of Self Lubricating Bushings" by Powertech Labs Inc., 12388-88th Ave., Surrey, B.C., Canada V3W 7R7. The self-lubricating composite material should not contain any graphite if used for submerged applications. The government should incorporate into the contract sufficient checks and balances to allow the government to maintain input and review of the design process.

Figure 2-4. Failed gudgeon pin bushing

Figure 2-5. Failed pintle bushing

(a) The materials that define self-lubricating bearings are made of polyester composites fabricated into bearing surfaces, then secured to a base material or conventionally retained within a bearing housing. The composite materials can be machined and typically are fitted into a bronze housing through shrink/interference fit or secured with epoxy. They also can be produced as plugs or disks to be secured in a bore by press-fit or mechanical fasteners and can be bonded mechanically to metallic substrates for sliding bearing surfaces. Composites that use impregnated graphite for lubrication should be used only in absolutely dry applications.

(b) The U.S. Army Engineer Research and Development Center (ERDC) and its Construction Engineering Research Lab (CERL) have conducted a number of projects to study the performance of self-lubricating materials. The first was for hydropower applications, and more recently completed was a report for navigational lock and dam applications. Some manufacturers' materials and arrangements have worked better than others for different applications and environments. The results of these findings are described within the reports. When considering the use of self-lubricated materials, the reports produced by ERDC/CERL should be reviewed extensively by the designer before design or selection of a greaseless bearing material. The reports to be referenced include ERDC/CERL SR-04-8, Field Evaluation of Self-Lubricated Mechanical Components for Civil Works Navigation Structures, June 2004, and CERL TR 99/104, Greaseless Bushings for Hydropower Applications: Program, Testing, and Results, December 1999. Both reports are detailed in Appendix A. They identify specific Corps lock and dam projects and composite manufacturers that have used self-lubricated bearing designs. Most installations are relatively recent and are being evaluated for long-term performance. Designers are encouraged to contact the Corps districts identified in the reports for updated feedback on the performance of specific applications. The better-performing self-lubricated bearing manufacturers identified in the report should be consulted for recommendations to select the best bushing arrangement. They also might be capable of assisting with identification of the appropriate design criteria best suited for each application. Additional guidance for the selection and properties of various greaseless bearing manufacturers is in ERDC/CERL TR-02-7, Advanced Materials Selection Guide for Lock, Dam and Hydroelectric Plant Components, January 2002 and EM 1424, Lubricants and Hydraulic Fluids. Successful use of composites for bearing materials and pintle bushings is evolving and designers considering their use should proceed with caution.

(c) Selecting the correct self-lubricated bearing material and identifying the proper design criteria/parameters for each application is critical to ensure a successful installation. The designer must be able to calculate, specify, or convey the following parameters when proceeding with a self-lubricated bearing design. Each bearing must be engineered for the specific service conditions of the application. Wet or dry application, contact stresses, composite thickness, coefficient of friction, surface finishes, mating surface material and hardness, interference fit, percentage water/oil swell in composite material, running clearance fits, and stick-slip phenomena must be known or considered to utilize self-lubricated bearing designs. Ambient and service temperature conditions also must be considered in the design for understanding the effects on the bearing materials due to the coefficient of thermal expansion. The coefficient of thermal expansion of self-lubricating bearing materials is generally much greater than that of ferrous housings in which they are installed. These differences in expansion and contraction must be accounted for in the design. Bore closure between the rotating components can occur from the swell effect of bearing materials subject to large temperature increases. The opposite also can occur in low temperatures, resulting in the bearings shrinking within the housing and losing any press-fit tolerances. Figure 2-6 shows a tainter valve trunnion bearing made of greaseless bearing material.

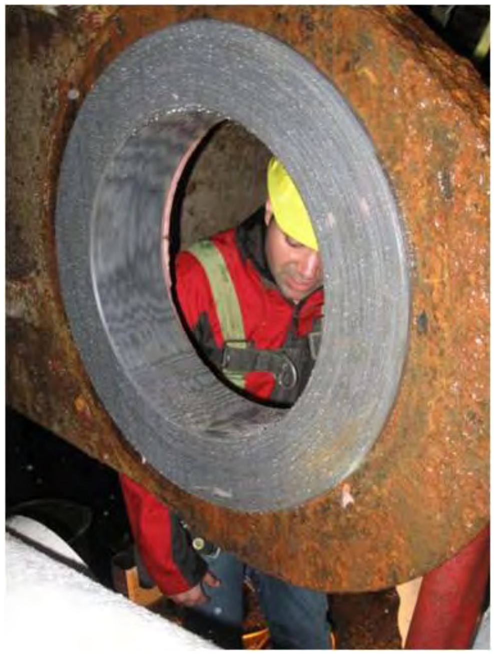


Figure 2-6. Greaseless tainter valve trunnion bearing

(d) Design contact stresses can be considerably higher for self-lubricated bearing composites when compared to conventional bronze bearings and, for dynamic bearing conditions, should be designed for a maximum bearing contact pressure range of 34.5 MPa (5000 psi) to 69 MPa (10,000 psi). Figure 2-7 shows an adjustable quoin and miter block design where contact stresses are high and adjustability must be maintained to compensate for wear. Design bearing pressures will be dependent upon the characteristics of the composite material being considered, the type of movement intended for the bearing, and the rotational or sliding speed at the bearing surface. Slower rotational speeds will tolerate bearing pressures at the higher end of this range. Static bearing applications are allowed to have even higher bearing contact pressures between 103 MPa and 138 MPa (15,000 to 20,000 psi) and are more dependent upon the crush strength of the composite material. In no case should a composite bearing material have crush strength less than 345 MPa (50,000 psi). Figures 2-8 and 2-9 show a greaseless bearing design with a spherical bearing that allows the floating mooring bit roller wheel an additional degree of freedom to allow for full wheel contact within the guide slots under heavily loaded conditions. Bearing pressures for hydroturbine applications utilizing self-lubricating bearing materials are recommended to be more conservative, with a range of 17 MPa (2500 psi) to 21 MPa (3000 psi) preferred.

(e) When compared with greased bronze in the same application, self-lubricated materials that provide lower coefficients of friction (0.06 to 0.10) should be selected. Composites that have static and dynamic coefficients of friction equal to each other result in smoother operating equipment, even if the friction load is high. When the static and dynamic friction coefficients are closer in value, a lower difference in strain energy exists as the system transitions from a static to dynamic condition. This helps to reduce the stick-slip, or stiction, phenomenon, allowing for a smoother operating system with less vibration, noise, and possible damage to the equipment. Wear rates (mils per 100 hr operation) also must be considered for each specific application. Independent testing of a manufacturer's materials and published data are highly recommended to confirm acceptable results can be achieved when compared to greased bronze in the same application and environment.

(f) Self-lubricated materials are well suited to provide a service life of 30 to 50 years when held free from contaminants with relatively thin bushing surface material thicknesses. To provide the highest load carrying capacity, on average, the specified bearing thickness should be in the range of 0.020 to 0.060 in. The designer must be aware that thicker bearing surface material does not always equate to longer service life. Wear rates amongst the various manufacturers are not all equal in the same environment. It might be more prudent for the designer to provide a system that excludes contaminants from the system to increase the bearing longevity versus arbitrarily increasing the thickness of the bearing surface material.

(g) Suitable mating materials for self-lubricated composites should be limited to the use of corrosion-resistant steels or heat-treated, medium-to-high strength steels with chrome plating. The most commonly used steel is the heat-treated 17-4 PH precipitation hardened stainless steel: ASTM A564, A693, Grade 630, UNS S17400. In saltwater or brackish water environments, 316 stainless steel, ASTM A276, Condition A, UNS S31600, is the best suited material for use. The mating steel hardness required varies widely amongst composite manufacturers and, in general, all manufacturers require a harder surface be furnished with higher loads and speeds. Therefore, mating surfaces should be consulted with a specific manufacturer. Hardness specified in the range of R_c 28-32 (271-301 BHN) will cover the broadest group of manufacturer's requirements. The required surface finish also varies widely amongst the composite manufacturers from R_a 0.1 μm (4 μin.) to R_a 6.35 μm (250 μin.). A specified surface finish of R_a0.4 μm (16 μin.) or better will fall into the broadest group of manufacturers' requirements, but consultation with various manufacturers for their specific requirements is recommended.

(h) Self-lubricated composite bearing length-to-diameter (L/D) ratio for thick-walled bearings is recommended to be in the range of 1.0 to 2.0. The ratio of 1.25 often is stated by many manufacturers as the preferred ratio. For thin-walled bushings, the L/D ratio of 0.35 to 1.0 is preferred. The manufacturer always should be consulted before finalizing a design.

(i) The bearing-to-shaft clearances recommended for thick-walled bearings generally is provided with a clearance of 0.001 to 0.002 in./in. of shaft diameter and, for thin-walled bearings, this range is reduced to a clearance ranging from slight interference up to 0.001 in./in. of shaft diameter. For shafts greater than 5 in. diameter, the clearances will require a reduction adjustment per the manufacturer's recommendations because the larger clearances for these diameters are not required.

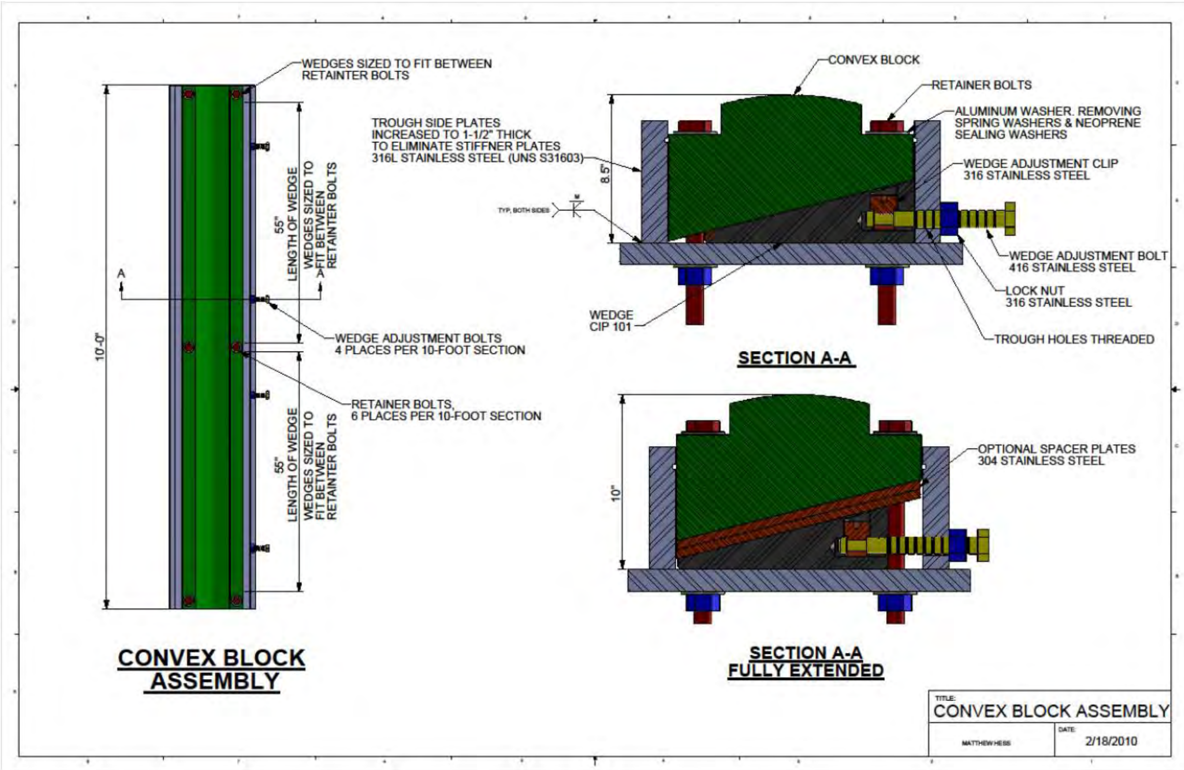


Figure 2-7. Adjustable quoin and miter block

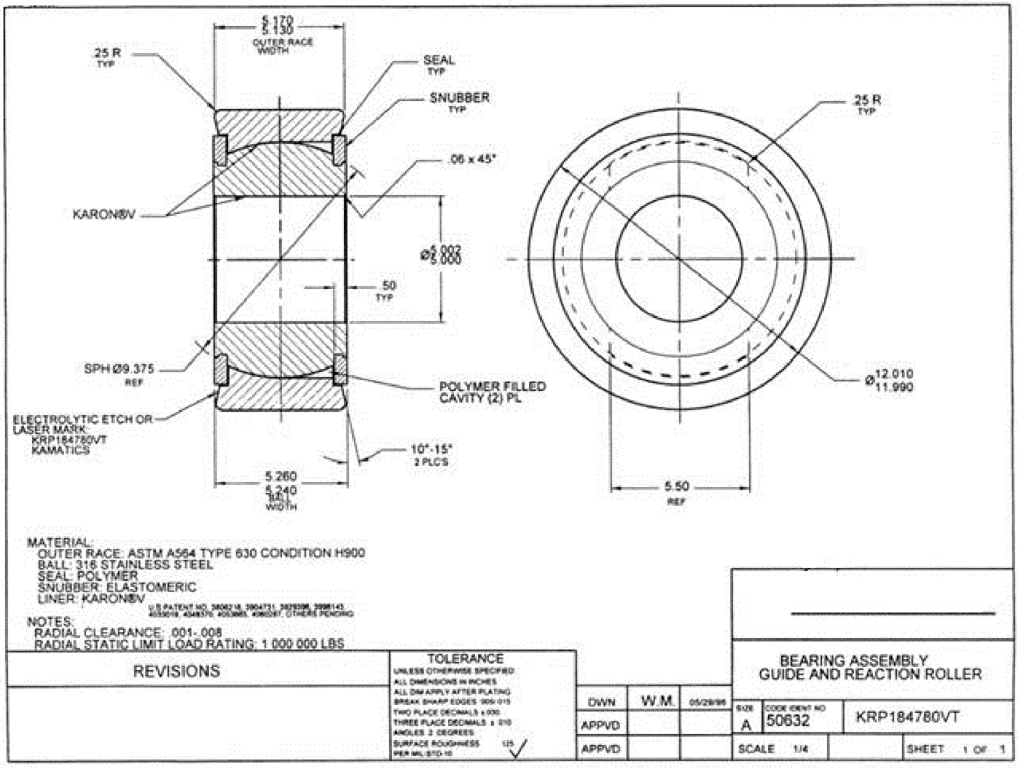


Figure 2-8. Greaseless floating mooring bit roller design

Figure 2-9. Greaseless floating mooring bit roller

(j) When self-lubricated composites are submerged in water or petroleum-based products, the material has a tendency to absorb the fluid immersed within and swell in the bore is likely to occur. The designer must account for the smaller diameter and operating clearance for the installed condition compared to that of the original condition in the dry. Without recognition of this occurrence and adjustment to the design dimensions, the bearing likely is to encounter binding, stick-slip, overheating, and premature failure. Consultation with the bearing manufacturer is highly recommended to avoid costly failures and repairs. Swell due to thermal expansion/contraction also must be accounted for in the design, but is usually insignificant in wet applications where submerged in a large heat sink.

(k) Sleeve-type bearings commonly are retained by shrink/interference fits or epoxy bonded to their housing with the epoxy bond method being preferred. When a pressed interference or shrink fit is utilized, the bearing manufacturer must be consulted to properly engineer fit tolerances and recognize that some bearing materials might be susceptible to material relaxation after installation. Epoxy-bonded bearings, as shown in Figure 2-10, must have proper surface preparation to ensure adequate bonding strengths are achieved and most commonly are specified with bonding radial clearances of 0.020 to 0.30 in. Designers must be aware that epoxy-bonded bearings require some method of centering and aligning of the bearing within the bore before installing the epoxy. The use of stainless steel wires is one method of centering the bearing within the bore.

Figure 2-10. Epoxy-bonded bearing

(l) Self-lubricating composites are susceptible to failure under misalignment and excessive bearing clearances. This results in edge loading of the material. Failures due to bearing fracture, bond failure, and material de-lamination can occur. The best methods to prevent this type of occurrence are to ensure proper shaft alignment, provide bearings with recommended running clearances, and design bearings with edge chamfers to reduce the possibility of edge failure.

(m) Self-lubricated bearing materials installed in submerged applications heavily laden with contaminants and/or prone to frequent flood conditions should have lip sills or o-ring seals incorporated into the design to exclude contaminants. The seals will protect the bearing running surfaces from accelerated wear or bearing failure. Bearing retainers also might be prudent in certain applications to prevent the possibility of bearing creep or extrusion of the bearing from the housing bore.

(4) Pillow Blocks. Pillow blocks should be cast iron, cast steel, or fabricated from forged steel. Pillow blocks should be designed to provide full radial and axial capacity in all directions. Mounting bolts, nuts, etc., should be rated for the bearing's full Basic Dynamic Load Rating Capacity in all directions, including upward through the cap. This rating is obtained from the bearing manufacturer's published data for commercially available bearings or from formula calculations available in specifications ISO 281 or JIS B1519 for custom bearings. Ball-bearing pillow blocks and flange blocks shall have a two-bolt base. Roller-bearing pillow blocks shall have a four bolt base. Spherical roller bearings shall be either of the fixed or expansion type, as required. End caps shall be provided on open-ended shafts. Roller-bearing housing caps shall be recessed into or dowelled onto the bases and secured with not fewer than four bolts, SAE Grade 8. Slotted mounting holes may be used for the base, as required, but dowel pins or keeper bars should be permanently installed after final alignment and testing. Only one fixed mount pillow block should be used on shafts with multiple pillow-block installations to permit thermal expansion in the axial direction. Some examples of pillow-block installations are in Figures 2-11 and 2-12.

Figure 2-11. Pillow block

Figure 2-12. Pillow block

(5) Pintle Bushing. Chapter 4 provides specific discussion on miter gate pintle bushings. Chapter 5 provides discussion on sector gate pintle bushings. Pintle bushings for lock gates traditionally have been grease-lubricated aluminum bronze, as shown in Figure 2-13. The aluminum bronze alloy used is typically C95400, meeting the requirements of ASTM B148 or ASTM B271. Plate B-21 provides recommended grease groove and seal details. The aluminum bronze bushing is press fit into the pintle socket and secured by bolting to the socket. The amount of press fit of the bushing in the socket should be kept minimal (0.001 to 0.003 in.) to prevent any alteration of the machined fit between the bushing and pintle ball after assembly. The bearing surface should be finished truly hemispherical and the pintle ball fitted to the bushing by scraping, or lapped until uniform contact is attained over the entire bearing surface. This can be determined by testing with carbon paper or similar media transfer technique. The pintle and bushing need to be match marked. Surface finishes should be shown on the drawings in accordance with ASME B46.1. Determining compliance with surface requirements typically is done by sense of feel and visual inspection of the work and comparing it to the Roughness Comparison Specimens of ASME B46.1. Plates B-23 and B-24 provide additional pintle information. Grease-lubricated bronze continues to work well, but environmental issues created by pumping grease to the pintle bushing is causing a shift in recent designs toward the self-lubricated pintle bushings. The self-lubricated composite materials also can be designed with much larger bearing pressures than conventional bronze for large gate loads. The pintle typically is manufactured of cast steel with bearing surfaces of stainless steel deposited in weld passes to a thickness of not less than 4.8 mm (0.1875 in.) and machined to the required shape.

(a) Recent designs have been completed with self-lubricating material installed onto hemi-spherical or near-spherical pintle sockets with matching stainless steel pintle balls. The self-lubricated material is shaped into pucks or disks recessed and secured to the socket bushing or the pintle ball. See Figures 2-14 and 2-15 for examples of greaseless bearing pintle/bushing designs. Conductivity indicator wear pins should be incorporated into the bearing surfaces to allow the project personnel to periodically test for bearing surface wear and to schedule replacement.

Figure 2-13. Aluminum bronze pintle bushing

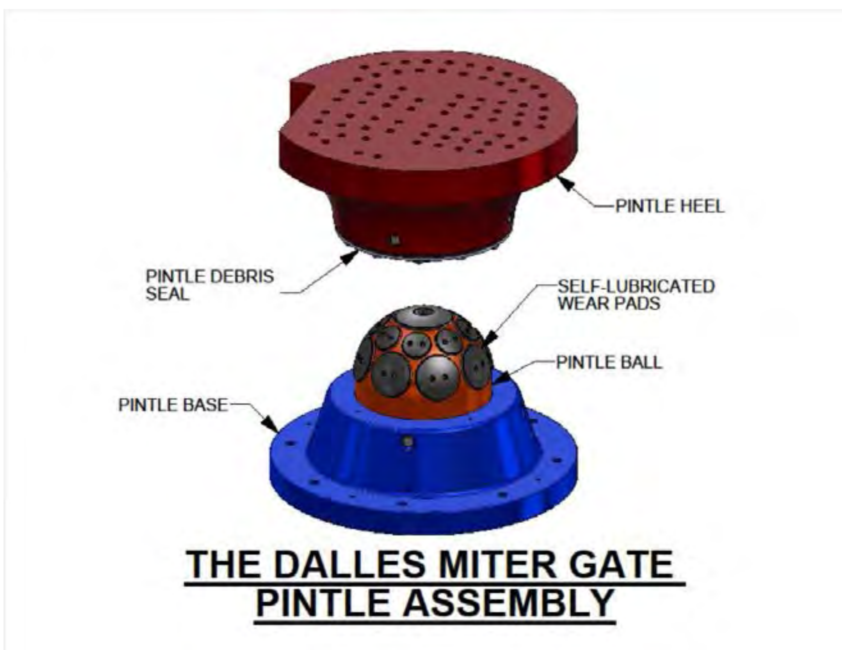


Figure 2-14. Greaseless pintle bushing

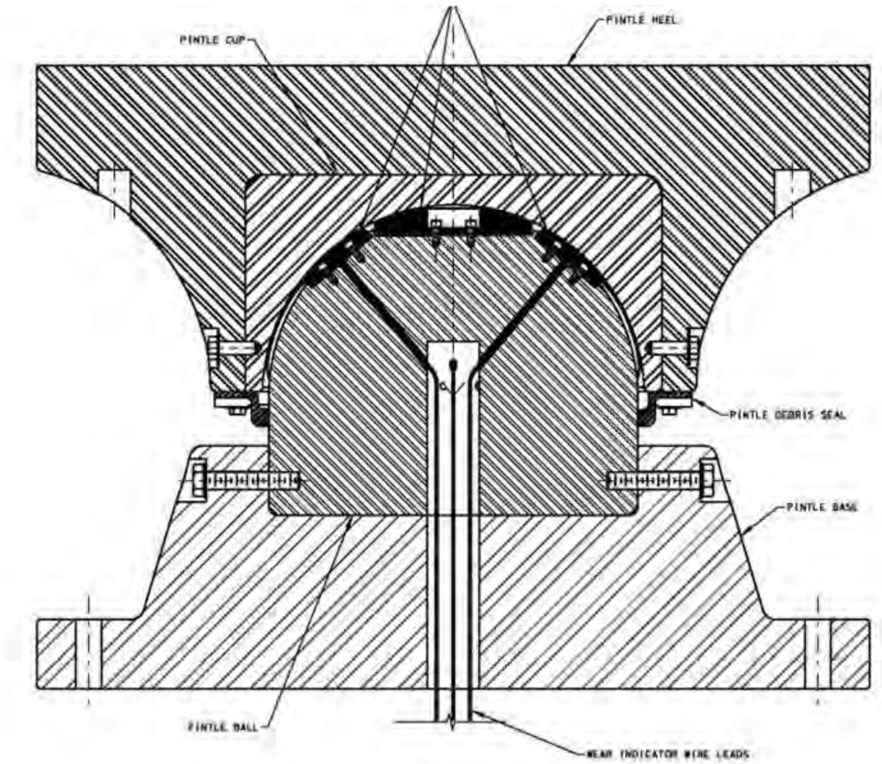


Figure 2-15. Greaseless pintle bushing

b. Brakes. Holding brakes should be the shoe-type, spring set, with DC magnet-operated, AC-rectified solenoid, or AC hydraulic thruster release. The DC type actuates by electromagnets, and the AC-rectified type typically uses a solenoid and plunger as shown in Figures 2-16 and 2-17. The AC thruster type, shown in Figure 2-18, uses a small hydraulic pump and hydraulic actuated cylinder. Brakes should have a minimum continuous duty torque rating of 150% of the maximum full load torque rating of the electric motor, or hydraulic actuator, as applied to the brake wheel. Special consideration in brake selection shall be provided by the designer to prevent runaway speeds from developing in gates with the potential for free fall. The brake set reaction time must be minimized to prevent the downward momentum of the gate from exceeding the holding torque of the brake or causing excessive glazing or wear of the brake shoes. Fuses should not be used in the brake control circuit. Brakes should be mounted in watertight and dust-tight enclosures, with heaters for moisture protection, manual release devices, limit switches as applicable, and indicators and electrical connections, as required by the operating environment. Brakes shall be self-compensating to adjust for shoe wear. The designer should ensure enclosures are designed with removable enclosure panels for access to perform inspection and maintenance of the brake components. UFGS 35 20 20, Electrical Equipment for Gate Hoist, provides design parameters for specifying brakes.

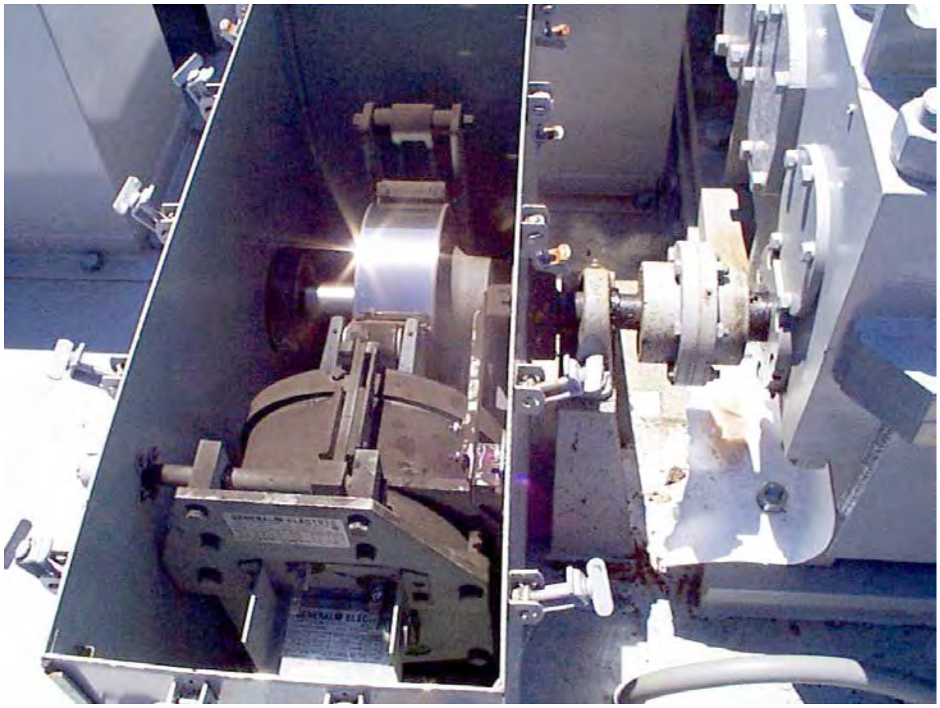


Figure 2-16. DC brake

Figure 2-17. AC rectified brake

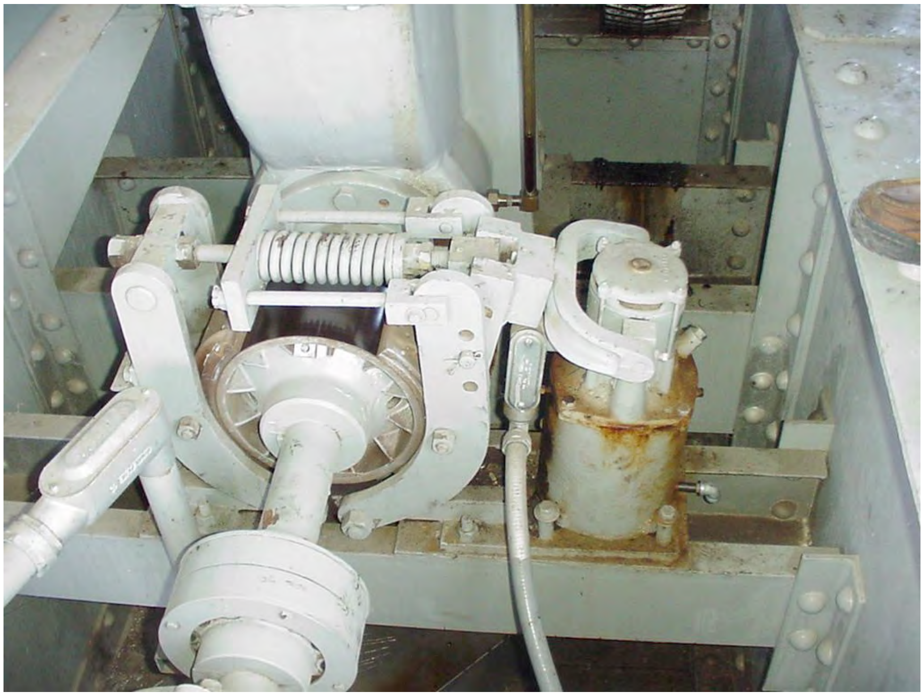


Figure 2-18. AC thrustor brake

c. Couplings. There are many different types of shaft couplings. The torque ratings, angular misalignment capabilities, axial float capabilities, and shock-absorption characteristics are the main factors that differentiate them. Some of the more common types used on inland navigation projects will be mentioned here. These include flexible disk couplings, elastomeric couplings, chain couplings, gear couplings, grid couplings, and jaw-type and rigid couplings. A coupling is a device for joining two rotating shafts. The most basic form is rigid and can accommodate no angular misalignment or shaft float. Other coupling designs are available to accommodate some amount of angular misalignment or axial shaft movement. Couplings also can be selected to absorb shock loading between shafts and to protect against a momentary over-torque condition. It is the engineer's responsibility to select the correct coupling design for each application, anticipating the conditions in which the coupling must continue to perform reliably. Couplings should be selected using the manufacturer's published ratings. Exceeding the limitations of any particular coupling has led to the premature failure of the coupling or damage to machinery.

(1) Flexible Disk. Flexible disk couplings are only suitable for specific applications such as precise control equipment (turbines, etc.). Flexible disk couplings require no lubrication and their components are external to allow for visual inspection. They are suitable for only slight misalignment, and the replacement of the disk pack is not as easy as the elastomeric coupling inserts. They are suitable for heavy-duty, slow-to-medium-speed applications. Flexible disk couplings can be used where high-starting torques, shock loads, torque reversals, or continuous alternating torque is encountered. Flexible disk couplings transmit torque and provide for angular and axial misalignment between shafts with a coupling comprised of shaft-mounted hubs connected through

flexible disk packs with spacer or sleeve assemblies. Because these couplings do not require lubrication, maintenance costs can be considered comparably low. They are easy to inspect for proper operation. Disadvantages include a relatively high initial cost. They also require more precision on alignment and assembly.

(2) Elastomeric. Elastomeric couplings incorporate a flexible synthetic insert between coupling halves, either providing a flexible cushion between jaws or a direct torsion connection. These are available in a variety of designs and capabilities. The elastomer in Figure 2-19 is secured at each bolt. Each elastomer sleeve consists of a hollow cylinder through which the bolts join each half of the coupling. The elastomer serves to accommodate severe angular misalignment and to cushion shock over loading. Advantages include low maintenance costs, because lubrication is not required, and ease of inspection when the coupling is accessible. The couplings also can be fitted to existing coupling applications easily. There are power limitations due to the varying modulus of elasticity of the flexible elements. Environmental conditions such as temperature and humidity, chemical attack, and UV exposure can have an adverse impact on the life and performance of the insert material for this type of coupling. Designers must take these factors into consideration during selection.

Figure 2-19. Elastomeric coupling

(3) Chain. Chain couplings provide for inexpensive coupling alternatives in the medium torque range. They can be exposed or enclosed within hub/sleeve assemblies to retain lubrication. The chain couplings transmit torque through sprocket hubs mounted on the shaft ends that are coupled by double-width roller chain. A chain coupling drawing is presented in Plate B-6. Chain couplings do not tolerate large misalignment.

(4) Gear. Gear-type couplings provide the highest torque-carrying capabilities with small, compact designs and are capable of accommodating small amounts of angular and slight parallel misalignment. Drawings of two differently sized gear couplings are

presented in Plate B-7. Gear couplings can be more tolerant of axial growth and shrinkage of shafts, which can be advantageous for the wide temperature variations common on inland navigation projects. Gear couplings tolerate the misalignment through the clearance between the outside teeth of the hubs and the inside teeth of the sleeves. The tooth-to-tooth sliding motion caused by misalignment can be detrimental if it is a permanent condition and can lead to premature failure. Misalignment for gear couplings shall be minimized by not exceeding the manufacturer's recommendations for installation limits pertaining to gap-hub separation, angular alignment, and parallel offset alignment measurements. Gear-type couplings must maintain their internal lubrication for successful operation. External grease fittings often are replaced with a plug for safety reasons and must be installed temporarily during periodic maintenance lubrication. Gears shall be machined in accordance with applicable ANSI/AGMA standards. Couplings shall be of sufficient capacity to develop the full strength of the shafting that they connect and shall be pressed and keyed thereon. The key fits shall be in accordance with ASME B17.1, Class II. Gear couplings should have steel housings and hubs, with integral lips to contain the seals and retain the sleeves, with lubrication. Sleeves should be fastened such that they cannot slip or loosen. The mating sleeve flanges join both halves of the coupling to transmit the torque between the shafts. The proper bolt type and torque must be installed in the flanges to avoid premature bolt bending fatigue and shear failures. Gear couplings that use snap rings to hold the sleeves should not be permitted. Internal shaft hubs must be bored to match their mating shaft diameters and matched keyways provided. This interface most often requires interference fits specified to develop full load-carrying capacity. Reversing loads often require special considerations in the design. The designer must carefully identify and match bore diameters to shaft sizes with special contract language to ensure the contractor and equipment manufacturer are aware of this requirement. Bolts shall be SAE Grade 8. Spare couplings purchased should not have their final bores specified and should be rough bored with final boring completed after the shaft sizes can be measured. An example of a gear-type coupling is shown in Figure 2-20.

Figure 2-20. Gear-type coupling

(5) Grid. Grid couplings are similar to gear-type couplings with the use of an interlocking steel grid between two slotted shaft hubs to transmit the torque. A disassembled grid coupling and mating hubs are shown in Figures 2-21 and 2-22. The grid is contained within a sealed housing to retain the lubrication for the coupling. The grid flexes in a sliding action within the hubs to transmit torque and compensate for misalignment. Grid couplings are also capable of reducing vibration by absorbing impact energy. The grid couplings are easy to install and maintain.

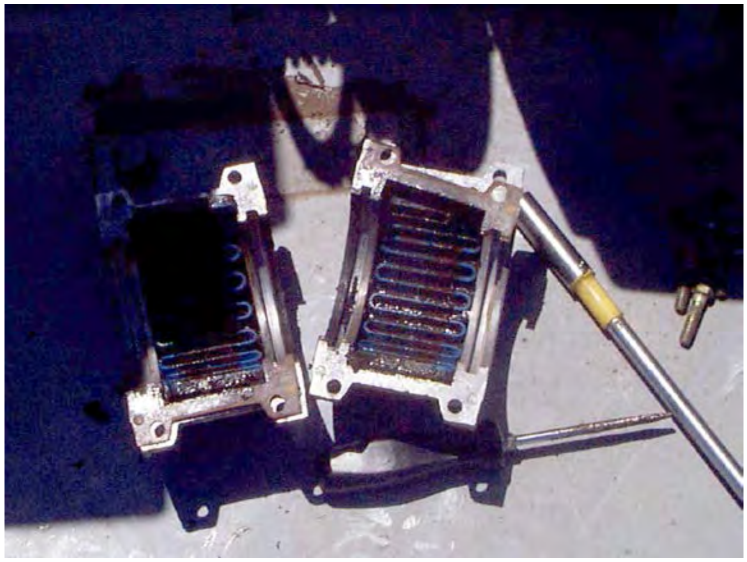


Figure 2-21. Grid coupling

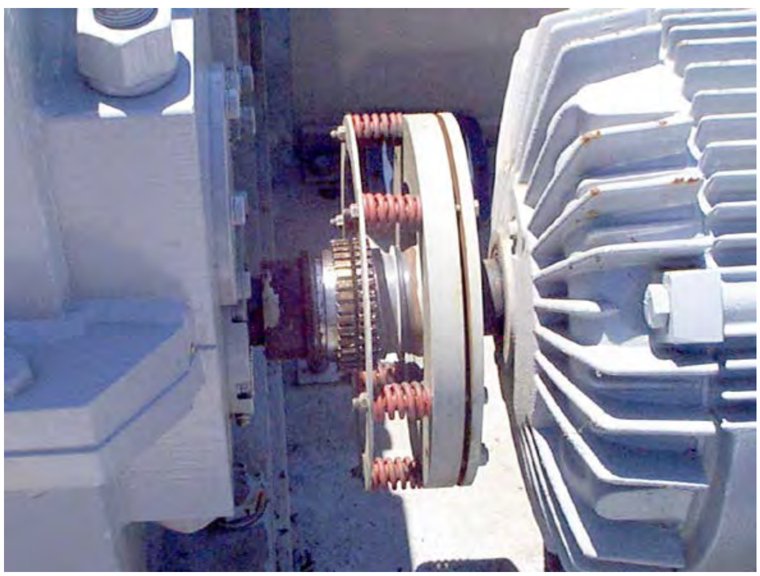


Figure 2-22. Grid coupling

(6) Jaw. Jaw-type couplings have two metallic hubs interconnected by a center insert referred to as a spider. The hub jaws engage the lobes on the spider to form the torque transmitting connection. The spiders are fabricated from a variety of different materials and hardness to fit the application. As the damping ability of the coupling increases, the load-carrying capacity is decreased. Two jaw-type couplings are shown in Figures 2-23 and 2-24. The jaw-type couplings are common for small shaft sizes and lighter loads. Misalignment with this type of coupling must be minimized because rapid deterioration of the spider could occur. One advantage of jaw couplings is they are fail-safe. The hub jaws will continue to engage and transmit torque even if the spider assembly fails.

Figure 2-23. Jaw-type coupling detail

Figure 2-24. Jaw-type coupling

(7) Rigid. Rigid couplings are the simplest form used to transmit torque in a power transmission arrangement. They will have either bolted flanges, keyed shaft with sleeved hub, or a clamp design to connect shaft ends. Rigid couplings have no capabilities to compensate for misalignment or axial expansion. Designers should use caution and consider the application carefully before installing rigid-type couplings. Perfect alignment must be provided and maintained for the use of this type of coupling. If misalignment is present, the load will be transferred to the shafts and bearings, likely resulting in premature failure.

d. Force Control Limit Switches. Force control limit switches can be used to control overloads in single- or multi-part wire rope hoists. The device can be installed between the wire rope and the dead-end anchorage connection. Another alternative is the use of electronic load cells that can be furnished as pins, links, or various other configurations. The electronic load cells can provide continuous monitoring of systems and can easily be incorporated into overall or stand-alone PLC systems to provide similar set point control, as discussed below, for force control limit switches. High and low force-control limits can be set with backup trip switches for each limit. The designer should provide switches for high load conditions such as locked rotor torque or gate seizing or obstruction. One switch is set slightly above the normal operating load, and a second is set higher as an emergency backup device. A low load switch can be set for the load encountered in raising due to a seized sheave. Low load, at the end anchorage, would be indicative of an overload in the portion of the wire rope from the sheave nest to the drum. The low load switches also should be capable of recognizing a slack cable condition. This condition likely would occur while lowering the gate and the gate encounters an obstruction or becomes wedged in the operating slot. The low load switches then would prevent the possibility of the wire rope coming off the sheave or dropping the gate and shock loading the machinery. The main limit switches interrupt the control circuit, while the backup switches de-energize the hoist. The forces are determined by calculating the wire rope tension that would be produced by the maximum load used in the design of machinery components. Backup switches typically are set at approximately 900 kg (2,000 lbs) differential from the primary switch. Pressure limit switches in the hydraulic circuits, shown in Figure 2-25, of wire rope hoisting arrangements with sheaves mounted to the hydraulic actuator rams also have been used successfully for operational control. These are set below the hydraulic pressure reliefs valves for high overload conditions and can monitor for slack cable conditions under low pressure when not affected by hydraulic system counterbalance valves.

Figure 2-25. Hydraulic pressure limit switch

e. Torque Limiting Devices. Slip clutches can be used to protect gate operating mechanisms by limiting the maximum motor torque applied to them. Multi-plate, slip-type clutches with fiber-type friction disks are recommended and shown in Figure 2-26 and Plate B-5. They should be adjustable and use coil springs. The coil springs must be properly adjusted and maintained to compensate for any slippage wear in the friction plates. To protect the machinery and components, the torque coupling shall be adjustable and set to slip at the designer's predetermined torque value. The torque slip should have a minimum adjustment range of plus or minus 20% of the specified load, depending on the designer's requirements. The coupling should continue to slip until the torque drops below this level. The torque slip range should be controlled by a spring-type mechanism that can be adjusted by means of tightening or loosening the through bolts. Torque couplings should be equipped with a means of sealed lubrication. Most slip-type torque-limiting couplings require a break-in period after assembly to the motor and shafts, and the designer should specify such in the execution section of the specifications. Slip clutches normally are rated for 200% of the maximum calculated torque requirement, and should be adjusted to initiate slipping at a setting slightly above the normal operating torque requirement. However, the manufacturer's recommendations should be considered for both sizing and adjustment. The torque capacity requirement is minimized if the clutch is located on the motor side of the speed reducer. Heat-rejection capacity is not an important consideration, as the device would not be expected to slip, except for very short periods. The designer should provide protection from the weather and oil or grease contaminating the friction elements. Ball and detent torque limiters are another type of limiting device that can accurately disengage the machinery at a pre-set torque value. The ball and detent type can fully disengage or reset once the overload condition is removed. To protect the machinery and components, the torque coupling shall be adjustable and set to slip at the designer's predetermined torque value. The coupling will continue to slip until the torque drops below this level. They also can also be equipped with limit switches for control signaling of an overload condition.

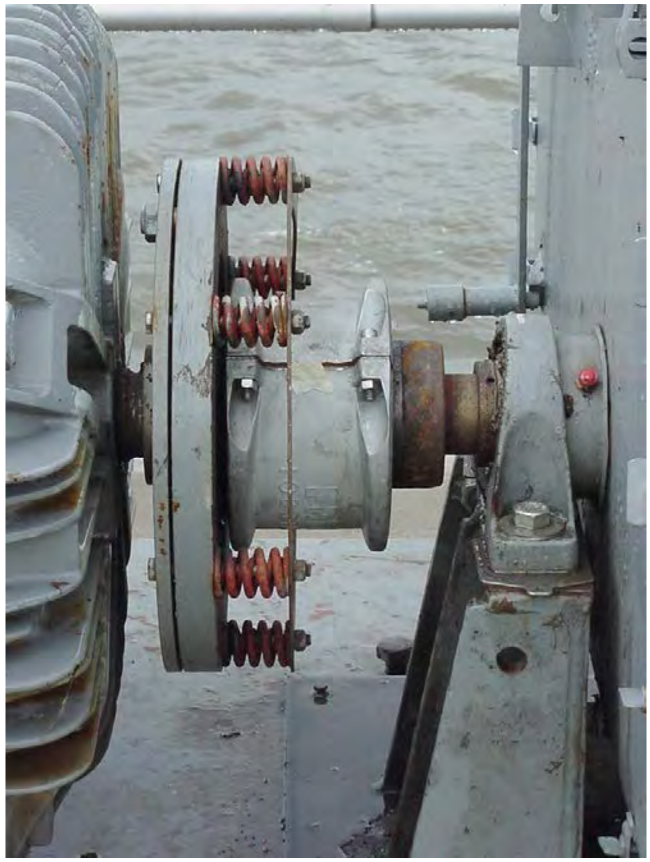


Figure 2-26. Slip clutch

f. Open Spur Gears. Open gears should have spur teeth of the involute form, in compliance with applicable American Gear Manufacturers Association standards. An oil bath spur gear with evident pinion wear is shown in Figure 2-27, and a greased open spur gear set is shown in Figure 2-28. The designer should provide in its operation and maintenance manual lubrication requirements for all open gear designs. Basic strength should be based upon the static load from the Lewis equation, as modified by the Barth Equation (design stress = Lewis stress x 600/(600 + pitch-line velocity-fpm). Large spur gears should be designed with forged steel rims in accordance with ASTM A290, while the hubs and arms can be cast (ASTM A148) or fabricated from rolled steel plate. A large cast steel gear is shown in Plate B-20. Gearing shall be cut from solid steel and may be integrally cast with the hub, drum or shaft. Large spur gears can be split radially, along two of the support arms, in order to permit more convenient handling and removal. The split line can be fastened by high-strength bolting materials, designed for the maximum loads on the gear. Pinions should be fabricated from ASTM A291 forged steel. Pinion gear teeth should have a hardness of approximately 50 Brinell Hardness Number (BHN) greater than the driven gear teeth to equalize wear. A pinion gear is shown in Plate B-8. Gears should not be overhung on shafts, including speed reducer shafts. The designer should specify the minimum contact area between pinion and spur gears. The specifications should establish what misalignment is acceptable, define tolerances on the gear sets, and define what minimum contact area should be achieved. The factors of safety in paragraph 2.1(b) shall not be exceeded.

Figure 2-27. Oil bath spur gear with wear

Figure 2-28. Greased open gear set

g. Round Link Chain. Round link chain hoists utilize both pocket wheel and grooved drum lifting mechanisms. A close-up top view of a dual pocket wheel and chain is shown in Figure 2-29. These types of hoists have been used primarily to replace existing roller chain hoists. The selection of a chain-handling device depends entirely on the type of chain to be used. There are many types of chains available for various applications of lifting service. The links of the round link chain are not actually round, but have round ends and approximately parallel sides. The calibrated links are designed specifically to be used with a pocket wheel that drives the chain properly, loads each link in tension and bearing, and eliminates the bending stresses in the links that occur when a grooved drum is used. This type of chain is applied widely to both low-speed and high-speed lifting and is both abrasion and corrosion resistant.

Figure 2-29. Dual pocket wheel and chain

(1) Pocket Wheel. The pocket wheel, shown in Plates B-70 and B-71, is a universally applied lifting mechanism to handle and hold a round link chain to the limit of the chain's breaking strength (326,000 lbs). A pocket wheel is designed to properly load the chain in tension and bearing without inducing the bending loads predominant in a grooved drum. The wheel may be either a ring forging of alloy steel or a weldment. Two spare dual chain pocket wheels with accompanying spur gear and shaft are shown in Figure 2-30.

Figure 2-30. Dual chain pocket wheels with spur gear

(2) Design Standard. Round link chain used in chain hoists, shown in Plate B-72, is made from an alloy steel. Although the materials and heat treatment might vary among manufacturers, AISI 8620 is common for this chain. This material is heat treated to the required tensile strength. Proof testing and visual inspection of each link of chain

should be specified. The hardness of this chain from different manufacturers also might vary, but a figure of 300 BHN is considered average with higher values desired as specified below. The higher hardness of this material provides improved wear qualities over low alloy chain. Low alloy chain of equal breaking strength has a greater energy absorbing capacity for greater shock load capacity over that of high alloy chain. A standard specification for a ring forging:

- Material ASTM A290, Class K, or AISI 8620 cast steel
- Tensile strength, minimum 1,170 Mpa (170,000 psi)
- Yield strength, minimum 1,000 Mpa (145,000 psi)
- Brinnel Hardness Number (BHN) range of 340 to 400

(a) The standards for the design of a pocket wheel are derived indirectly from the dimensions included in DIN 22252, Part 1 (High-tensile Round-link Steel Chains for Mining: Testing). This standard covers the dimensions and tolerances for chain that is compatible with pocket wheels. Preliminary design calculations for pocket wheels using a specific chain size are necessary in order to determine that the unit size is compatible with any physical space limitations imposed by the gate machinery location.

(b) An additional auxiliary item required for a pocket wheel mechanism is a chain locker. The general arrangement of a tainter gate pocket wheel chain hoist with divided chain storage locker as viewed from beneath is shown in Figure 2-31 and Plate B-70. The size of a chain locker should be such that it adequately contains the slack length of chain when the gate is in the fully raised position. Chain locker volume should be a minimum of the product of the diameter of the chain in inches squared, multiplied by the length of the chain in fathoms, multiplied by 0.85. A sample calculation in Appendix C shows various dimensions of chain lockers required for a 14-m (46-ft) length chain, 38 mm (1.5 in.) in diameter. In pocket wheel designs with dual hoisting chains on a single pocket wheel, the chain locker should be equipped with a center divider to keep each chain separate and prevent the chains from piling up on one another.

Figure 2-31. Pocket wheel chain hoist with chain locker

(3) Availability. While no manufacturer will have a standard off-the-shelf product that will fit a given application exactly, the technology to build a pocket wheel to a given design criteria is available. The pocket wheel has been successfully installed at locks and dams on the Upper Mississippi and Illinois rivers. The designer should be aware of long lead times in the fabrication of large pocket wheels when assembling the contract documents and establishing construction schedules.

(4) Assembly Test. After the first set of machinery is fabricated and completed, a factory assembly lift test should be required as part of the contractor's responsibility for the gate-lifting machinery. These tests should include not only a design load test, but an overload test that proves that the maximum specified motor torque will not deform the chain nor allow the chain to slip on the pocket wheel. The field hoisting assembly and pocket wheel elevation views are shown in Figures 2-32 and 2-33, and Plate B-70.

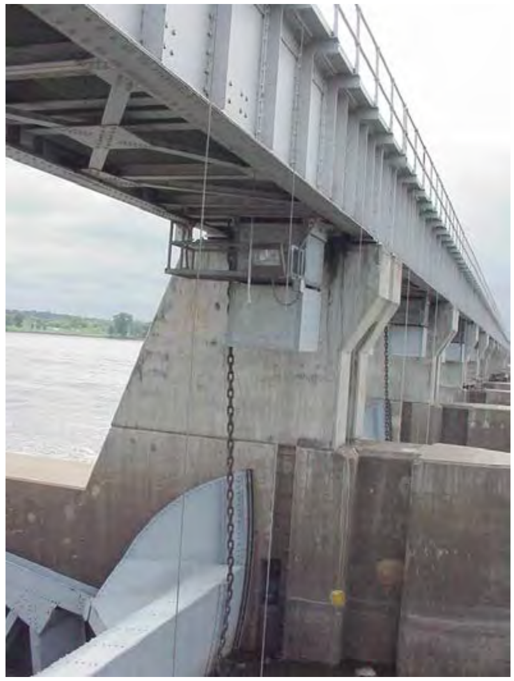


Figure 2-32. Pocket wheel hoisting assembly

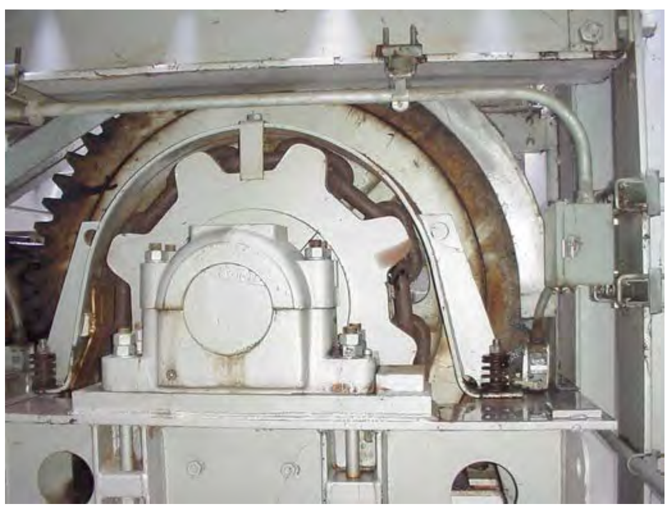


Figure 2-33. Pocket wheel elevation

(5) Grooved Drum. Another type of round link chain-lifting mechanism is a cylindrical grooved drum. An example of a grooved drum arrangement is shown in Figure 2-34. This design includes a cast or fabricated cylinder with a helical groove that is either cast or machined into the surface. The groove is designed to accept every other chain link and must be sized large enough to wind the entire length of chain around the drum in a single row. One advantage of a grooved drum is that it requires no chain locker since it stores the chain on the drum similar to a wire rope drum. The diameter of the grooved drum should not be less than 25 or 30 times the diameter of the bar used for the chain links. For applications where there is no additional room for chain

storage, the use of a grooved drum might be indicated. An added advantage of this type of drum is that it can accept a deformed link without becoming jammed. The main disadvantage of the grooved drum is the manner in which it loads the chain links. Each link is loaded in both tension and bending. This loading situation puts undue stresses on the links, especially since chain links are not meant to be loaded in bending. A sample stress calculation is in Appendix C. It shows a 38-mm (1.5-in.) chain being loaded on a 1.0-m (41.49-in.) diameter drum. The results of the calculation show that, even under a normal tensile load, the chain is loaded at or near its yield point. Chain is typically known by its breaking strength, and a proof load is normally specified. Actual stress levels in a chain under loading are determined by an involved analysis dependent on criteria such as chain geometry, hardness, material properties, etc.

Figure 2-34. Grooved round link chain drum

(6) Compatibility with Existing Material. The type of chain that can be used for gate hoisting should be compatible with the existing gate and hoist component materials to prevent undue wear and abrasion. Chain guides must be fabricated and installed on curved face gates that have the gate connection at the bottom of the gate to allow the chain to lay flat over the curvature face of the gate. Hoisting chains mainly bear against the chain guides or plates at every other chain link parallel to the face of the gate. The chain guides or bearing plates should be sized specifically for the chains that will be installed. The chain guides and bearing plates should be compatible in size and hardness with the lifting chain selected. Galvanic corrosion due to dissimilar metals also should be considered during the design process.

(7) Tolerances. Manufacturing standards for calibrated round link chain require that each length of chain meets certain tolerances with regard to link size and breaking strength. High-alloy hoist chain is manufactured to length and width, plus or minus 0.51

to 0.76 mm (.02 to .03 in.). These tolerances are an international standard so that all chain, regardless of manufacturer, will be suitable for the intended use. The DIN standards for strength testing of this chain are rigorous and include tensile, bending, and shock tests.

(8) Abrasion. For chain to be suitable for dam gate-lifting service, it must be resistant to abrasion caused by silt trapped in the submerged links. Chain used for dam gate lifting probably will never be washed or cleaned because it would be difficult and impractical to do so. High-alloy conveyor chain must be specified to be abrasion resistant.

(9) Shock. To resist shock loads, a material must be strong and be able to absorb the energy imparted to it by the shock load. When carbon steel is alloyed and heat treated to increase strength, its energy absorbing capacity doesn't increase proportionately. Thus, for equal breaking strength, lower alloy material normally will be more resistant to shock loading than higher alloy material. Round link alloy chain is tested for shock loads by the manufacturer. To reduce the possibility of shock loads, the designer should incorporate a pocket wheel guard with limit switches. The guard should be mounted over the pocket wheel on spring mounts to allow the guard to detect when the chain has debris caught in it or the chain tries to ride out of the pocket wheel. The limit switch for the chain guard shown in Figure 2-35 has been tripped due to debris caught in the chain and passing through the pocket wheel. The lever arm limit switches detect the movement of the guard and can shut down the operation of the machinery before a shock-loading condition occurs.

Figure 2-35. Chain guard limit switch

(10) Distortion. To be compatible with a host chain pocket wheel, the round link chain must be capable of being loaded without being significantly distorted. If the chain distorted, the links no longer would fit the lifting device pockets. Round link chain resists distortion because the sides of the link are designed to remain parallel. If these links

distort, they tend to elongate; but they only can do so after they have exceeded the elastic limit of the material. The design criteria for lifting chain requires that the minimum breaking strength of the chain be no less than five times the design load, and that the lifting machinery shall in no case impart to the chain a load that will exceed 75% of the yield strength of the steel in the chain in both tension and bending. When a chain is selected within these design limits, link distortion will not be a factor.

(11) Corrosion. The round link chain discussed in this chapter normally should be protected against corrosion. Hoist chain is available with a variety of special corrosion coatings such as special paints, Tectyl 846-10 MIL Specification MIL-PRF-16173E, Grade 4, Class I, or hot galvanizing. Hot galvanizing should not be used for chain in this type of application unless the reduced strength due to reheating is taken into account. The galvanization might also affect the chain’s ability to ride in the pocket wheel correctly. Specifying the corrosion coating is the designer’s responsibility and will be dependent upon the specific application and environment for which the chain is intended. It should be noted that the existing chains on the prototype pocket wheels at Lock and Dam 20 on the Mississippi River have worked very well for a period simulating 50 years of operation with no corrosion coating at all. However, mid-1980s field installations on the Illinois River have revealed moderate corrosion and seized links in the lower sections of chain normally submerged. These sections of corroded chain normally are not raised out of the water or capable of being exercised through the pocket wheel.

(12) Replacement. Replacement of round link chain described in this manual should not be a problem in the future. This type of chain is widely used around the world.

(13) Chain Costs. For comparison purposes and cost estimates, the cost figures for chain in the round link hoist category can be obtained from various manufacturers and from previously constructed Corps projects. Chains should meet the strength and durability requirements for gate-lifting service at civil works projects. Differences in cost could influence the chain selection criteria in projects with large quantities of chain required. For example, the Mississippi River Lock and Dam 20 project required almost a mile of chain to power all the tainter gates.

h. Engineered Steel Chain. In addition to round link chain, engineered steel chain also should be considered as a replacement for existing roller chain. Roller chains (using pins, rollers, and sidebars) have been a source of operation, maintenance, and environmental problems at gated dams and spillways owned and operated by the Corps for many years. Original roller chain designs were difficult to lubricate, causing bearing surfaces to corrode and bind, preventing smooth operation of the chain over the sprocket. As a result, spillway gates could not be operated, chains failed, and gates were dropped. This created both a dam safety problem and a hazard for operating personnel. The chain design that’s described herein has solved the problems associated with previous roller chain designs for both tainter and roller gates at John Martin Dam, Robert Byrd Lock and Dam, and others in the Corps’ St. Paul District. It has been in use at these sites since 1997, with no problems reported. Discussed next will be material selection, corrosion prevention, first cost, and life cycle cost. An engineering analysis of this type of chain is presented and maintenance issues are examined.

(1) Terminology.

(a) It is important to show how a lifting chain for a tainter gate is different from a bicycle chain, beyond the obvious size and strength differences. There are several standards and a chain manufacturer's association that classifies various types. The chain industry, chain manufacturers, and American Chain Association (ACA) make a distinction between roller chain and engineered steel chain. In general, roller chain is used for power transmission between sprockets at moderate-to-high speeds. The chain speed, sprocket design, and kinematics between the sprocket and chain are crucial. Roller chain is manufactured per ANSI/ASME B29.1 (see standards paragraph below). The tension members between pins (side plates) are called link plates. This type of chain generally is produced in large quantities. The size and strength ratings are relatively low.

(b) Engineered steel chain is intended for a wider variety of applications, including materials handling, conveying, and other industrial uses. The engineered steel chain usually is manufactured in smaller quantities; has greater strength, more corrosion resistance, and greater shock resistance; and is designed to be used in severe environments. The chain is manufactured per several standards including ANSI/ASME B29.10 and B29.15. The side plates are called sidebars. The pin-bushing area is called the chain joint. The sidebars establish the chain pitch (see Figure 2-36). The ACA defines tension linkage chain as a chain application in which the main function is to move a load slowly and intermittently through a short distance, or to hold a load. These types of chains are well suited for Corps projects to hoist and support gate loads. The function of a tension linkage chain is to transmit a moving force using chain tension, hence the nomenclature. Lifting chain for roller and tainter gates, thus, falls under the category of a tension linkage, engineered steel chain. Note these terms and their meanings:

- Pitch is the distance between the centers of adjacent chain joint members or center-to-center distance between adjacent pins.
- Sidebars are tension members connecting the chain joints
- Pins connect one link section to another. Pins are the shear members between the inner and outer sidebars.

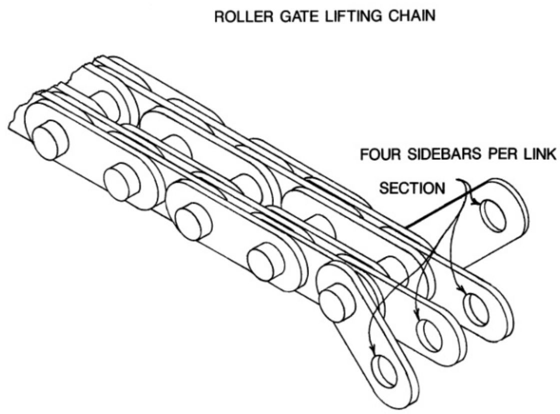


Figure 2-36. Chain assembly

(2) Standards. The American National Standards Institute (ANSI) and American Society of Mechanical Engineers (ASME) both publish standards for chain, as stated above. There are manufacturers that make a true metric chain. Also, the European DIN (German) Standards and the International Standards Organization (ISO) categorize metric chain.

(a) Many of the original tainter gate chain designs (from the 1930s) for the Upper Mississippi River projects used offset sidebar roller chain as opposed to straight sidebar roller chain. The primary benefit of offset sidebar type of chain is the links are all identical. Offset sidebar chain can be used in odd or even number pitches. The primary advantage of using straight sidebar chain is that the chain is easier to manufacture and, for a given sidebar plate thickness, the straight sidebars will have more strength. Straight sidebar chain consists of inside and outside links. Sections of this chain type must be used in an even number of pitches (lengths). This chain also can be constructed without rollers. However, in gate-lifting applications, the rollers are necessary for reducing friction as the chain is going over the sprocket.

(b) It should be noted that the majority of the ANSI/ASME standards concern chain used in power transmission rather than lifting applications. However, this difference is generally irrelevant. The loading on the chain is basically the same in both applications where the chain is going over a sprocket. The biggest difference between power transmission and lifting application is likely to be the speed. In lifting applications the chain travel will be extremely slow. ANSI/ASME B29.1M, Precision Power Transmission Roller Chains, Attachments, and Sprockets, lists a series of standard roller chains. This standard only classifies chain with pitch dimensions up to 3.0 in. ANSI B29.1 assigns standard number designations to chain based on pitch, chain width, and roller diameter. The chain sizes given in B29.1 are generally inadequate for the majority of tainter gate and roller gate lifting applications. The primary benefit of this standard is that any chain manufactured according to it will fit over any corresponding sprocket manufactured to the standard. The chain of one manufacturer will replace the chain of another manufacturer. ANSI/ASME B29.10M, Heavy Duty Offset Sidebar Power Transmission Roller Chains and Sprocket Teeth, only standardizes offset sidebar chain. This standard is an engineered steel chain standard and includes chain with pitch dimensions up to 7 in. (177.8 mm) and a minimum ultimate strength of 425,000 lbs (1890 kN).

(3) Material Selection. Material selection is likely the single most important feature of the lifting chain design. The type of material used will impact the strength, corrosion resistance, and overall life of the chain. Proper material selection must be made to ensure a 50-year life for the lifting chains. Lifting chains normally are subjected to all weather and river conditions. For projects that maintain upper pool, the portion of the chain that connects to the gate will be submerged in the river. This will subject the chain to silt and debris. Because the dam gates rarely are moved completely out of the water, the lower section of chain will be submerged for a majority of its service life. This lower portion of chain also will be subjected to sandblasting and paint over spray when the dam gates are being painted.

(a) Recent lifting chain design utilizes aluminum bronze sidebars and stainless steel pins (see Figure 2-37). Both materials should provide adequate corrosion protection to allow the chain to last 50 years. Aluminum bronze is manufactured per ASTM B505, and 62,000 psi (427,586 kPa) minimum yield strength is specified.

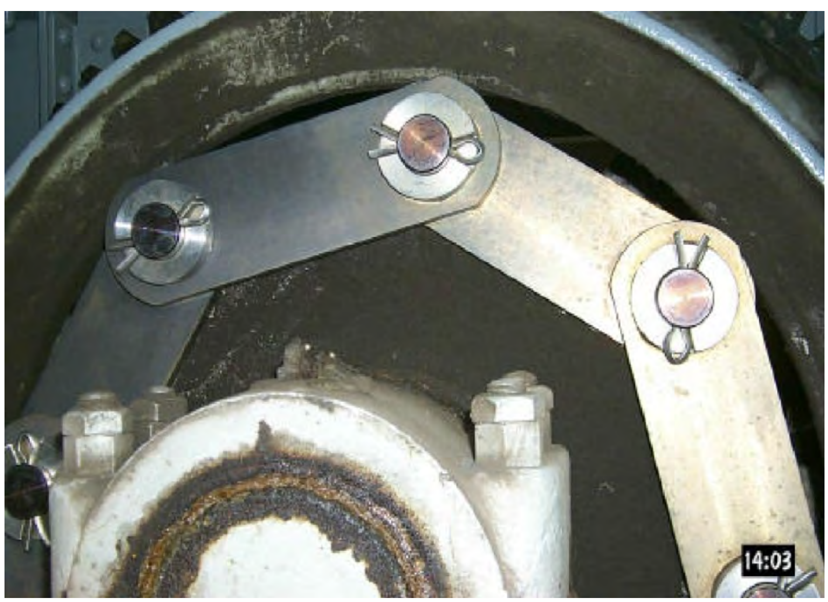


Figure 2-37. Typical engineered chain installation

(b) The stainless steel pins are manufactured per ASTM A564, Type XM-25, Condition H1050. This stainless steel is equivalent to Type 304 stainless steel for corrosion resistance. The primary disadvantage of using this type of stainless steel is it was developed by one manufacturer and is not readily available from others. Other options for the stainless steel include using ASTM A564, Type 630 (17Cr-4Ni). This material is close in properties to XM-25 and is available from more manufacturers. There are some disadvantages of using Type 630 stainless versus XM-25. The Type 630 stainless is more difficult to machine and must be age hardened prior to using. A comparison of the stainless steels is provided below in Table 2-1. Plates B-01, B-02, and B-03 show engineered steel chain assemblies and components. Figure 2-38 shows a chain using aluminum bronze sidebars and stainless steel pins installed on a tainter gate. This chain had been in service more than 10 years.

	ASTM A A564, Type XM-25, H1050	ASTM A564, Type 630, H1025
Min Tensile	145,000 psi	155,000 psi
Min Yield	135,000 psi	145,000 psi
BHN	321	331
BHN is Brinell Hardness Number		

Table 2-1. Engineered steel chain material comparison

(4) Cost. The cost of the lifting chain primarily will be a function of the materials used. Although carbon steel materials will have the lowest initial cost, it is likely that the underwater portions of the chain will need replacement over 50 years. The stainless steel and aluminum bronze chain design, thus, will have a lower life cycle cost including maintenance costs.

Figure 2-38. Engineered chain installation on a tainter gate

(5) Chain Design. Chain design is based on a 50-year service life. Several design considerations need to be analyzed to ensure this 50-year life. Strength and material selection are probably the most important. As discussed, the material selection will dictate how much the chain will corrode over 50 years, in particular the lower section of the chain. There are other design considerations that must be analyzed. These include yield strength, shear strength, fatigue strength, bearing stress at the chain joint, bearing clearance at the chain joint, and shock loading.

(a) The ANSI/ASME standards define minimum ultimate strength (MUS) as the tensile load in pounds (or kilonewtons) at which a chain, in the condition at the time it left the factory, might break in a single load application. The yield strength of the chain should be 40 to 60% of the MUS. The chain also should be designed for shock loading. An example of this would be when a gate falls under a slack chain condition. The ACA Design and Applications Handbook lists a service factor of 1.4 to 1.7 for heavy shock loading. Several of the lock sites have broken chain when a gate has been dropped under slack chain conditions or when slack chain was generated to provide additional momentum for breaking a frozen gate loose. Though these practices are not recommended by the designers, the lifting chain likely will be subjected to these conditions sometime over the course of its service life.

(b) Since the lifting speed of dam gates is very slow, the chain/sprocket design is not paramount. The main factor for the chain is the ability to hoist and hold the load from the dam gates and perform under all service conditions. The interface between the pin and sidebar of the chain (or the chain joint) will be the highest stress area of the chain. A chain failure will result from either a sidebar or pin failure in this area. At the chain joint area, the sidebar will be in tension and shear. Corrosion in the chain joint area might cause the pin not to rotate as the chain is going over the sprocket, causing damage to the gate hoist machinery. The bearing clearance necessary in the chain joint will depend on the materials used for the sidebar and pin. A minimum clearance of .005 in. is used in the current chain designs. This value should be doubled or tripled if steel sidebars and pins are being used. The pin undergoes bending stress in the center between sidebars and shear stress at the chain joint. Both values need to be calculated.

(c) An appropriate design standard is necessary to adequately design the chain joint area and determine a bearing stress. The American Association of State Highway and Transportation Officials (AASHTO) standard for bridges can be used for this purpose. In particular, the design constraints for pins, rollers, and rockers for bridges can be utilized. This standard makes a distinction between bearing stress on pins subject to rotation versus non-rotating pins. The chain joint should be classified as a rotating joint as opposed to a non-rotating joint. The standard also sets an allowable shear stress (Fv) of 40% of yield for the pin. The AASHTO standard also helps determine whether a bushing or bearing is required. The AASHTO Specification for Highway Bridges, 15th Edition, 1992, Section 10 (Structural Steel), Part C (Allowable Stress Design), Table 10.32.1A, permits an allowable bearing stress of 80% of yield for pins not subject to rotation. This specification allows a bearing stress of 40% of yield for pins subject to rotation. The standard states the effective bearing area of a pin shall be the diameter multiplied by the thickness of material on which it bears (e.g., the sidebar). The AASHTO standard for pins, rollers, and rockers is meant to eliminate galling in the pin/rocker area (i.e., the chain joint). The AASHTO standard implies that stress values below 40% of yield strength will avoid galling and that a bushing or bearing is not required. The standard only recognizes structural steel and alloy steel materials. Galling results from metal-to-metal contact. When a cohesive force between two metals exceeds the strength of either metal, adhesion or cold welding will occur. Under high stresses, cold welding will occur more rapidly and over a wider area. For instance, galling likely will occur when the chain is loaded up to and beyond yield limits. Galling is also a particular concern when stainless steels are mated with other stainless steels. To avoid galling, the 40% of yield value might need to be lowered in applications where the stainless steel chain (sidebar and pin) is used without bearings or bushings. The surface finish at the chain joint also will affect the rate of galling. The smoother the surface finish, the less likely galling will occur. Chain designs using aluminum bronze sidebars and stainless steel pins will act like a bushing/pin interface. These two metals have good compatibility in terms of their bearing properties. These materials also have a fairly low corrosion potential (from dissimilar metal corrosion or galvanic corrosion). The lower the potential difference, the less likely galvanic corrosion will occur. The Metals Handbook, Volume 1, Properties and Selection of Metals, 8th Edition, American Society for Metals, lists a potential difference of 79 mV between aluminum bronze and 304 stainless steel in dilute sea water. This compares to 904 mV between zinc and copper.

(d) Fatigue strength of the chain also should be considered in the chain design, though the chain speed is slow. Fatigue strength is not likely to be the limiting factor in chain design at many Corps projects. Each specific application should be verified by the specific site conditions and projected cycles of operation. For the Corps' St. Paul District, it was assumed that the gates will be raised and lowered three times per week, which means over a 50-year period the chains will be cycled nearly 10,000 times. As each chain link section goes over the sprocket, it will be subjected to maximum tension. The link section then will be slack as it goes over the sprocket and is coiled in the chain rack.

(6) Maintainability. A primary goal of chain designs should be either to eliminate or reduce the amount of maintenance necessary on the gate-lifting chains. For projects that normally maintain an upper pool, a reasonable assumption can be made that it takes a crew of four people one week to bulkhead a single gate, temporarily support the dam gate, and grease the lifting chains (two per tainter gate). Switching to a non-lubricated chain offers a significant cost savings over 50 years. When compared to replacing existing chain with wire rope, chain replacement offers several advantages. First, the existing gate-lifting machinery could be reused. Using chain instead of wire rope also requires less maintenance and replacement over 50 years. Wire rope must be lubricated on a regular basis. Any damaged part of chain can be replaced individually, while wire rope must be completely replaced.

(a) Many original (1930s design) lifting chains for the tainter gates were lubricated in a number of different ways. All these lubrication methods allowed oil and grease to enter the water. Some of the lock sites lubricated the chain with 30W motor oil. Other sites used diesel fuel or waste oil. None of these methods allowed any lubricant into the chain joint because the bearing clearances were too tight. Grease lubrication systems worked well initially, but there are a number of them now that will not accept grease. This system offered no advantages from a maintenance standpoint, and excess grease still ended up in the river.

(b) The chain design presented herein uses no bearing or bushing in the chain joint. The chain joint is designed as a bushing because the sidebars are made of aluminum bronze and the pins are made of stainless steel. This design will eliminate the need for greasing of the chains.

(7) Zebra Mussels. Zebra mussels have become more prevalent in the Upper Mississippi River system within the last several years. Zebra mussels attach themselves to submerged gates, intake valves, grating, concrete, etc. The submerged portions of the gate-lifting chain should be designed to reduce or eliminate zebra mussel attachment. Proper material selection is one method to reduce or eliminate zebra mussels from attaching themselves to the chain and its components. Testing and research by ERDC/CERL have shown that zinc and copper are toxic to zebra mussels. As stated, the latest designs for hoisting chains use aluminum bronze sidebars and rollers. The specific alloy used in their design was UNS No. C95500.This alloy is 78% copper. Recent field inspections of these components indicate little zebra mussel attachment to the lifting chains. Some zebra mussels were attached to the stainless

steel collars and pins, but no mussels were attached to the aluminum bronze sidebars. Refer to EM 1110-2-3401, Thermal Spraying: New Construction and Maintenance, Chapter 5, and EM 1110-2-2607, Planning and Design of Navigation Dams, Chapter 4, for discussion of applications for control of zebra mussels.

i. Shafts. Shafting should be designed in accordance with the machinery component criteria for the rated loads and increased by applicable shock and fatigue factors. A factor of safety of 5 should be applied to shafts based upon the ultimate strength of the materials, provided the stresses produced by the maximum torque of the motor do not exceed 75% of the yield point of the materials involved. The design criteria for all shafts should be the ASME Shafting Code equations with applied torsional and bending factors for heavy shock loading. The ASME Shafting Code requires additional stress reduction factors for keyways in the shaft. Stress concentration factors should be used where applicable. A combined shock-and-fatigue factor of 1.25 is recommended. Shafts should be supported at locations required to minimize bending and axial movement, yet allow for thermal expansion. The distance between bearings on shafting subject to bending, except that due to its own weight, should be such that the maximum bending moment deflection is limited to be fewer than 0.83 mm/m (0.01 in./ft) of length at the maximum rated load. Torsional shaft deflection should not exceed 0.26 deg/mr (0.08 deg/ft) of shaft length at the maximum rated load. Shafts should be fabricated from forged steel, such as ASTM A668. Where spur gears are mounted on separate shafts, the relative slope of the shafts, at the centerline of the gear mesh, should not exceed one-third (1/3) of the gear backlash divided by the smallest gear face width. The typical backlash for spur gears is 0.03/DP to 0.05/DP, where DP is the diametral pitch. Fillets shall be provided where changes in section occur. All keyways shall have fillet radii. All shafts shall have standard keyways and keys, in accordance with ASME B17.1, Class II.

j. Sheaves. Sheaves should be heat-treated cast steel, ductile iron, or manganese steel and should be of standard dimensions with grooves clad with stainless steel. Sheaves are designed to have a single lay of wire rope or to be able to layer the wire rope upon itself with each revolution. The designer should take the sheave and wire rope construction into consideration for each type of spooling arrangement. Grooved and plain drums also can be used to spool the wire rope onto a single layer or multiple layers. Drums either can be fabricated and machined assemblies or spun castings. Sheaves can be forged or fabricated. Each sheave should have a groove and pitch diameter that corresponds to the recommended factors for the mating wire rope. The hubs should be fitted with plain bearings or roller bearings with appropriate lubrication fittings. The sheave flange, rim thickness, web thickness, angle of contact, and inside diameter of the hub dimensions should be in accordance with the standard manufacturers' typical products for the appropriate size and construction of wire rope. Published ratings of sheaves should be used in determining the factor of safety. Machinery utilizing wire rope drum and sheaves are shown in Figures 2-39 and 2-41. A general machinery arrangement of hydraulic and wire rope for hoisting a vertical lift gate is shown in Plate B-58. The choice of a larger sheave diameter, for a given nominal wire rope size, improves fatigue life and reduces bending stress for the wire rope. The diameter of the sheaves may be as small as 24 times the rope diameter when used with 6 x 37 strand wire rope for an emergency-type gate. When used with a lock or dam operating gate, sheave diameter should be 30 times the rope diameter. Supporting members used for supporting the hoist drums or sheaves on the structure or gate shall be designed for the actual load plus 100% impact. The designer must provide specific instruction to ensure new and repainted sheaves are not painted within the wire rope grooves. Groove profile can be affected by paint accumulation and can result in accelerated wire rope wear or damage. Figure 2-40 demonstrates the loss of groove profile on a painted drum for proper seating of the wire rope.

Figure 2-39. Wire rope drum and sheaves

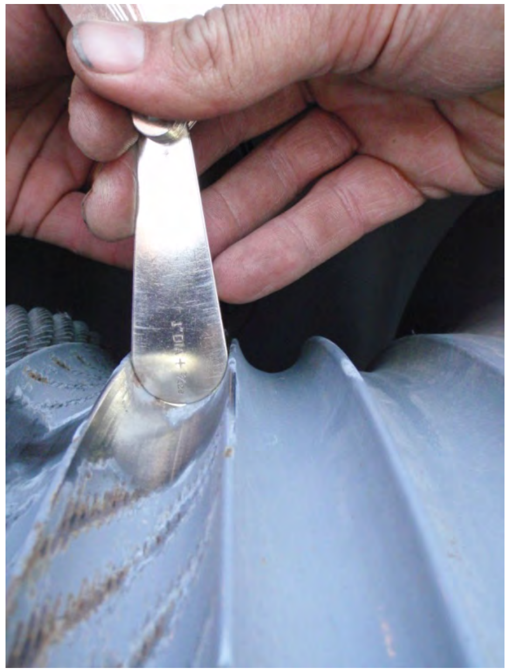


Figure 2-40. Checking wire rope drum groove profile

Figure 2-41. Wire rope sheave

k. Speed Reducers. Speed reducers should be helical, herringbone, spiral bevel, or worm type, in accordance with the applicable American Gear Manufacturer (AGMA) standards. Applicable standards include AGMA 2001, AGMA 2003, AGMA 2015, AGMA 6013, AGMA 6113, and AGMA 9005. See Figure 2-42 for a miter gate speed reducer installation. Plate B-4 shows a typical worm gear-type speed reducer. Speed reducers should be selected based upon the manufacturers' published ratings, including service factors, for the required operating conditions. Special shaft diameters or lengths are available from most major manufacturers. Gearing should be made from high-strength alloy steel, carburized, hardened, and ground after gear cutting to ANSI/AGMA Quality 11 or better, in accordance with AGMA 2015/915-1. Pinions gears should be cut integrally on the pinion shaft. Spiral bevel gears should be made from high-strength alloy steel with case-hardened teeth, crown lapped for quality and smooth operation. Reducer shafts should be made from high-strength alloy steel and should be of sufficient size to ensure rigid alignment. All keyways should have fillet radii. Keys should be provided for all shafts and be of standard keyway design, in accordance with ASME B17.1, Class II. All fabricated dimensions of the keyways and keys should be included in submittal requirements for government review. All speed reducers should be equipped with anti-friction bearings. Overhung loads on speed reducer shafts should be discouraged, unless available space is severely limited by design circumstances. Speed-reducer lubricants, for the bearings and the gear sets, should be chosen for operation in the existing ambient conditions. Where ambient temperatures fall below the normal lubricant recommendations for the type of speed reducer required, a thermostatically controlled unit heater, or heaters, should be provided in the reducer enclosure. Well-type, non-immersion heaters are recommended to avoid localized overheating, or cooking, of the oil and to facilitate replacement without removing the gearbox from service. Heating elements should have a maximum watt density of 1.5 W/sq cm (10 W/sq in.). Synthetic hydrocarbon lubricants with a higher viscosity index are an acceptable oil alternative, as approved by the speed reducer manufacturer for the normal loads and speeds encountered in service. A separate lubricating oil-pumping system, which sprays all gears and non-greased bearings before start-up and during operation, should be provided for speed reducers that operate infrequently, start under loaded conditions, or will be placed in extended storage. Speed reducers should be specified with a lubricant thermometer, a level gauge, and a hygroscopic oil breather. Condensation and moisture accumulation within gearboxes pose a significant challenge for the designer. Water bleed-off ports, desiccant breathers, and water-separating filtration are some of the different methods the designer might elect to incorporate. Connection ports for portable filtering should be furnished on the gearbox to aid in routine filtering of the oil and moisture removal.

Figure 2-42. Miter gate speed reducer

l. Wire Rope. Wire rope is a common product used for gate-operating machinery. Round wire rope typically is defined as multi-wire strands laid helically around a center core. The individual wires are woven together to form strands of wire. The strands are wound about the center core to form the finished wire rope. The wire material, method by which the wire is woven, the core construction, and any wire rope coatings all influence the properties and strength of the wire rope. It also determines the appropriate application for different wire rope types and sizes offered. Each round wire rope is constructed to close dimensional tolerances. The round wire ropes generally have a higher resistance to wear and mechanical damage than flat wire rope (woven wire rope with a rectangular cross-section). A minimum factor of safety of 5, based on the nominal breaking strength of the rope, should be applied to the maximum working load. No rope should exceed 70% of the breaking strength at the locked rotor torque of the motor. Wire rope constructions are available that are resistant to rotation, abrasion, crush, corrosion, and fatigue. Wire rope systems generally are considered more amenable to longer periods of inactivity than systems operated by hydraulically driven gates, and are easier to maintain. It is the role of the engineer to determine the most likely failure mode for which to design the wire rope application. More thorough wire rope selection criteria are in EM 1110-2-3200 and the most recent edition of the Wire Rope User's Manual. A multiple wire rope and vertical gate pin plate is shown in Figure 2-43.

Figure 2-43. Multiple wire rope and vertical gate pin plate

(1) Flattened strand wire rope often is recommended in gate hoist applications, particularly where the rope wraps on a disk-layered drum. Flattened strand wire ropes have circular cross sections that are constructed with some inner wires triangular in shape to provide a greater cross-sectional area of metal for higher load-carrying capacity. They also provide greater wire rope contact with the sheave for increased bearing surface area when compared to standard wire rope. Consequently, flattened strand has increased abrasion resistance, crushing resistance, and strength. It can be difficult to locate manufacturers of flattened strand wire rope. With this in mind, the designer must create wire rope hoisting systems with wire ropes that are readily available at the time of construction and available for a foreseeable period of time to allow for future replacement.

(2) The use of wire rope in gate hoist applications offers some advantages over other types of hoists. Wire rope hoists have been used to replace chain hoists in some applications. This is largely due to the advances in wire rope technology and necessitated from the maintenance problems associated with older-style chain hoists. The designer should provide the installer and maintenance staff adequate clearance between the wire rope/sockets and all other machinery, anchorages, and tension adjusters to allow room for movement and assembly of the components.

(3) Corrosion can be an issue with wire rope, but stainless steel and galvanized wire rope offer desirable alternatives when used in the correct applications. Galvanized wire rope is recommended for static wire conditions and should not be used for submersed or bending applications. Galvanization in submerged applications reacts similar to a sacrificial anode until the coating is gone and corrosion begins to deteriorate the steel wires. Galvanized wire rope also is not recommended for bending applications over sheaves, due to the galvanizing material filling the voids between the wires and reducing the ability of the individual wires to slide against each other as bending occurs.

(4) Stainless steel wire rope often provides the best characteristics for the longevity of the design. Stainless steel ropes have a lower strength than available carbon steel ropes, so their use often will result in larger diameter ropes or greater numbers of ropes. Fatigue or abrasion will be the dominant failure mechanism of stainless steel wire ropes, and they must be inspected regularly for broken or damaged wires and replaced when the damage becomes significant. Stainless steel is the material of choice for corrosion resistance, particularly on gate hoist applications that are subject to submersion of the ropes or splash/spray, or where inspection or lubrication is difficult. The use of sacrificial anodes is recommended for submerged stainless steel wire rope/socket applications where the stainless steel is connected to carbon steel components.

(5) Extra Improved Plow Steel wire rope provides the highest load-carrying capacity of wire rope when compared to similar size wire rope of other material, but is not protected from corrosion without added lubrication.

(6) Plastic coating or impregnation of carbon steel wire rope also is used for corrosion resistance. It generally is not recommended. Although the plastic filling helps prevent abrasion, concentrated corrosion cells can form where the plastic coating is damaged or not present. Plastic coated wire rope also is difficult to inspect.

(7) Kevlar rope is an option where corrosion has been found to be an extreme problem. Kevlar rope stretches more than steel and crushes when wrapped on a drum. Because of its stretching characteristics, it stores more energy, and additional personnel protection precautions must be taken to guard against rope breakage. The rope being crushed out of a round shape does not seem to affect its strength.

(8) Wire rope connections to dam gates have been a problem due to the environment and corrosion of mating surfaces (see Figure 2-44). Corrosion tends to bind these surfaces and prevent the movement that is required as the gate is raised. Stainless steel should be used for pins and washers, and consideration should be given to the use of self-lubricating bushing material. Sacrificial anodes also might be required in applications where dissimilar metals are used.

Broken Cables

Note the broken cables.

Cable 10" Above Socket

Note the broken wires and the abraded surfaces of the wire strands.

Figure 2-44. Broken wire ropes at dam gate connection

(9) Socket design is an important consideration to the implementation of wire rope installations. Epoxy products provide a viable alternative for the field installation of wire

rope into sockets. The epoxy products require tighter tolerances for the openings in the sockets through which the wire rope must pass. Standard zincing can be reliable, but does face galvanic corrosion problems in submerged or wet environments. The zinc sockets must be poured in a molten state, which poses additional safety concerns. The zinc method also is becoming more difficult to procure due to environmental considerations around the zinc material. Generally, it is considered good practice that end terminations have an efficiency of at least 1.0 or can develop strength at least equal to the breaking strength of the rope. Spelter sockets have been utilized extensively on inland navigation projects with good results. Poor spelter sockets occur from not following proper procedures, poor wire rope preparation, and the joining of incompatible materials. Submittal certification of the assembler of these connections is highly recommended to avoid inexperienced field assembly. An example of this is the socketing of stainless steel wire rope with stainless steel sockets. It has been found that socketing stainless steel wire ropes with zinc spelter sockets is more sensitive to technique than either galvanized or plain carbon steel sockets. Preheating the socket to 232° C, in addition to strictly following the rest of the procedure, was found to be critical to ensure good distribution of the zinc with good contact and no voids. This is necessary to ensure that the wire does not pull out of the socket prior, at a load less than the breaking strength of the rope. Adherence to industry standards for proof testing to 200% of the working load is considered good practice. This includes testing with end connections.

(10) Any multiple wire rope system must have provisions for equalizing the tension in a group of wire ropes. A turnbuckle arrangement is shown in Figure 2-45. A wire rope tensioning device or proposed method of achieving the acceptable tension criteria should be specified. Any tensioning procedure specified should be thorough to ensure proper load sharing among the wire ropes. A typical tensioning procedure should require each wire rope of each wire rope assembly be tensioned within 5%of the mean tension for that assembly. In addition, the total load supported by the wire rope assembly on each half of the hoisted component shall be shared equally. The load sharing should not be less than 48% nor more than 52% of the total load supported by the wire rope assemblies connected to the same gate component. Adjustments should be repeated until the wire ropes and assemblies are within these specified limits. In multi-wire rope applications, the designer should require all wire rope to be supplied under the same supplier lot number to avoid variations in wire rope diameters as a result of allowable fabrication tolerances. This is especially important in retrofit or repair work where a limited number of wire ropes are requiring replacement. Unequal tensioning and overloading might occur in the individual wire ropes during operation, due to the difference in wire rope diameters, as the wire rope is collected onto the drums.

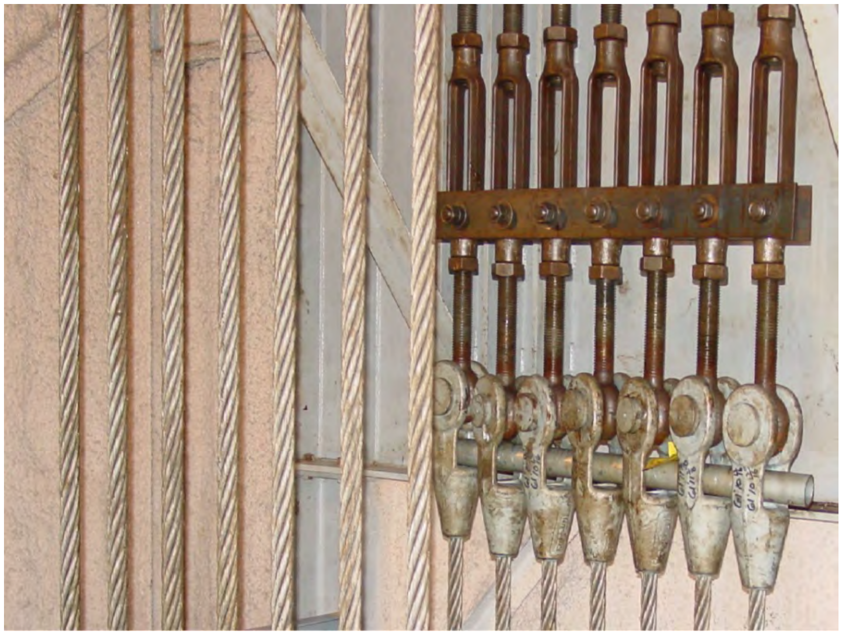


Figure 2-45. Wire rope turnbuckle arrangement

(11) Lubrication of carbon steel wire ropes is recommended for corrosion resistance. Lubrication of stainless steel wire ropes is recommended to reduce damage from abrasion and fatigue in bending.

m. Anchor Bolts.

(1) Anchor bolts should be designed for the maximum normal load and the locked rotor torque criteria. Anchor bolt groupings should be de-rated for concrete shear cone overlaps. All anchor bolts should be detailed on the contract drawings with type of material, threads, head, depth of embedment, and any special grouting or adjustment provisions.

(2) Anchor bolts, even those only for shear conditions, should have hooks, bolt heads, chairs, or body deformations designed to resist pull-out to the limits of the bolt tension rating. Anchor bolts should be installed with a weather-resistant template made from the actual device to be anchored. The specifications should have detailed requirements or tolerances for leveling the machinery on the anchor bolts, grouting, and pre-loading the bolts.

n. Fused Bolts. Several types of lock-operating machinery include devices designed to fail at a pre-determined load to prevent overloading of other machinery components. The most successful method for performing this function is usually a fused bolt or bolts. Fused bolts can be designed to fail in tension or shear fairly accurately. Bolts fail most predictably in tension. A standard manufactured bolt, of a particular manufacturing run of bolts from the same material stock, can be load tested to improve the accuracy of a design. The designer, by calculating the approximate reduction in nominal diameter of the bolt required to fracture at a specific load, can test bolts machined to the reduced diameter to failure. Similar methods can be applied to shear connections made by bolts. The designer must be careful to ensure that the maintenance personnel realize that replacement of the bolts with bolts of the same material and dimensions is critical. It is dangerous to replace these types of bolts with larger, stronger bolts, because of re-occurring failures. Regular failure indicates another problem with the machine or the original design load criteria. Larger bolts could move the failure to a more critical design component. It is important never to use these devices on a gate that can fall, causing damage to the structure. Fused bolts have been used successfully on many miter gate machine items, such as operating struts and cone brakes.

o. Keys, Pins, and Splines. Keys, pins, and splines are important connections in lock-operating machinery designed to transmit power and motion to the gate. Keys and keyways should be designed in accordance with ASME B17.1. With rare exceptions, these items should be fashioned for the general design criteria, not for failure at or near design operating loads. Any item that, by its failure, can cause a gate or machine to free-wheel to impact with the concrete or steel structures should not be allowed to be the weak link in the system.

2-3. Redundancy. Whenever possible and practicable, the design of gate-operating equipment should include redundancy in the operating systems or interchangeability of components to retain reliable operation.

2-4. Lubrication. The designer should review the guidance provided in EM 1110-2-1424 for the proper selection of oils, greases, and fluids.

CHAPTER 3

Hydraulic Drives

3-1. Description and Application. Hydraulic fluid power systems generate, transmit, control, and apply hydraulic fluid to devices that perform work. Power is generated by a hydraulic power unit (HPU) consisting of one or more pumps, valves, and controls mounted on a fluid reservoir. Pipe, tubing, hose, and manifolds transmit the fluid to the output devices. Valves control the direction, pressure, and volume of the fluid flow. Actuators, such as hydraulic cylinders and hydraulic motors, are the typical output devices. Hydraulic fluid power output devices often are used to operate lock gates, spillway gates, and culvert valves. The Unified Facilities Guide Specifications (UFGS) provide detailed assistance in the preparation of contract specifications of hydraulic fluid power systems, (Section 41 24 26 Hydraulic Fluid Power Systems and Section 41 24 27.00 10 Hydraulic Fluid Power Systems for Civil Works Structures). Other industry design standards covering hydraulic fluid power systems include NFPA and ISO. Many European-based companies primarily use the DIN (German) Standards.

3-2. Hydraulic Systems. There are two basic types of hydraulic systems: open circuit and closed circuit.

a. Open Circuit Systems. Open circuit systems generally have a high-pressure supply line from the pump to the actuator and a low-pressure return line from the actuator to the reservoir. The reservoir is vented to atmosphere and breathes as the reservoir level changes due to the volume differential from stroking a linear actuator (cylinder). Open systems usually have a directional valve and continue to circulate the fluid when in neutral. Open systems have been the most common type used on locks. They are generally less complex, have wide industry support, and one pump system can handle multiple actuators. In general, hydraulic system designs for locks are variations of open circuit systems. These variations include: centralized power unit design, local power unit design, dedicated power unit design, and integral power unit design. There are a number of other variations of these systems including adaptations for spillway gates. A typical centralized system has a single power unit location with piping and valves transmitting the fluid power to different locations. A typical local power unit design places an individual power unit near the actuators to be operated at one corner of the lock. A typical dedicated power unit design has a single power unit adjacent to each actuator. An integral power unit design has a dedicated power unit attached to each actuator to form a single, self-contained actuator.

(1) Centralized Power Unit. A typical centralized system has the power unit located in a lock control building or shelter above the flood of record. The building usually is located on the middle wall of a dual chamber lock, but can be on either wall of a single chamber lock. An extensive piping system connects the power unit to the miter gate and valve actuators. The piping system is installed in covered trenches or galleries

on each wall and in crossovers to adjacent walls. A typical arrangement for dual chambers consists of a reservoir and three electric motor-driven variable volume pumps. Two pumps are selected for service and one for backup on a monthly basis. The two-variable volume service pumps operate in tandem and can supply multiple actuators on both chambers at the same time. Proportional valves normally are used to provide variable speed control of the miter gates. A typical arrangement for a single chamber has two separate power units, with each dedicated to the actuators on one lock wall. The two power units are adjacent to each other for cross-connection. Cross-connection of the main pressure, pilot pressure, and return piping system allows the use of either system as backup for the other during malfunctions. Under normal operating conditions each power unit will operate one miter gate or one culvert valve. If one pumping system is damaged, the remaining system can operate two miter gate leaves or two culvert valves at a reduced speed by pumping through the cross-connection system to the appropriate control system. The principal advantages of centralized systems are reduced initial cost of power units, centralized maintenance, and smaller space requirements on the lock walls. The principal disadvantages are increased cost for piping, increased piping friction, cost for lock piping crossovers, high vulnerability to leakage, reduced speed/load capacity during backup operation or extremely cold weather conditions, and increased noise level when in a control building.

(2) Local Power Unit. A typical local system has a power unit at each corner of the lock walls. Each unit is used to operate the adjacent miter gate leaf and culvert valve. It often is deemed prudent to furnish each power unit with an extra main pressure pump, mounted on the same reservoir, to provide backup power. The principal advantages of the local system are a reduction in initial cost of piping, reduced piping friction, no cost for piping crossovers, reduced leakage, and lower noise levels in personnel areas. The principal disadvantages are increased initial cost, decentralized maintenance, no special provisions for flood protection, and larger total space requirements.

(3) Dedicated Power Unit. Dedicated power units take the local system approach even further. A typical dedicated system has a separate power unit at each miter gate and each culvert valve. Each power unit normally is dedicated to operate its adjacent miter gate or culvert valve cylinder, but it also can be cross-connected with another nearby power unit to provide emergency backup. The principal advantages of the dedicated system are a reduction of initial cost of piping, reduced piping friction, no cost for piping crossovers, reduced leakage, full speed/load capability during backup operation, and lower noise levels in personnel areas. The principal disadvantages are increased initial cost, decentralized maintenance, no special provisions for flood protection, and larger total space requirements.

(4) Integral Power Unit. An integral power unit features a hydraulic system that combines a hydraulic power unit with a hydraulic cylinder to form a self-contained actuator that is sealed and submersible. Instead of directional valving, integral power

units utilize a bi-rotational gear pump mounted inside a sealed reservoir and driven by a submersible electric motor attached to the reservoir. The speed and direction of the cylinder rod are controlled by a variable frequency drive in the motor control center that controls the speed and direction of the electric motor. The principal advantages of an integral system are zero initial cost of piping, negligible piping friction, no cost for piping crossovers, minimal leakage, quiet operation, low maintenance, submersible design, and a reduction in total space requirements. The principal disadvantage is a requirement to store at least one backup spare actuator of each size used on the lock. See Chapter 4 for additional information and photographs.

(5) Compact Hydraulic Drive Unit. Recently introduced in Europe, a compact hydraulic drive (CHD) unit is similar to an integral power unit, except it is not attached to the actuator it operates. The compact modular unit is mounted near the actuator and connected with a pipe or hose. A CHD also utilizes a sealed reservoir with a variable frequency drive for speed control. Quick-release hydraulic couplings and electrical plugs permit speedy replacement.

(6) Spillway Gate Power Unit. Spillway gate power unit design should be a function of the maximum normal operating requirements of the dam. A centralized system might be adequate where only one or two gates need to be moved in a single operation and the length of piping is manageable. A local system, with each power unit serving several gates, can be used for situations such as hydraulic-operated wicket gates. Dedicated systems would be the most practical solution for large dams with many simultaneous operations or remote-control capabilities.

b. Closed Circuit Systems. In a closed circuit, hydraulic fluid is returned from the actuator directly to the pump. Closed circuits generally are used only in applications with a rotary actuator or hydraulic motor. These systems have a high- and low-pressure side and a small sealed reservoir to collect pump and motor leakage. Leakage normally is returned to the system by an auxiliary pump. Closed systems are limited to one pump per actuator applications. Closed systems have been widely used for mobile hydraulics, but their application to locks that generally use linear actuators or cylinders is limited.

3-3. <u>Component Parameters</u>. Hydraulic system components generally are specified by parameters of flow rate, pressure rating, and optional features. Pressure and flow volume are intimately related to temperature and viscosity of the hydraulic fluid. The system design operating pressure can be best determined by requirements for reliability, efficiency, safety, maintainability, and life cycle cost analysis. The system flow rating generally is based upon the required gate operating times. Gate operating times should be computed on the basis of established experience and safety considerations.

a. System Operating Pressure. Design system operating pressures usually are determined by balancing the requirements of the hydraulic pump, the hydraulic actuator, the piping friction, the control valve ratings, and the potential for shock loading. Typical operating pressures for centralized power units are 14 MPa (2000 psi) or less. Typical operating pressures for local, dedicated, or integral power units are 21 Mpa (3000 psi) or less. Many modern piston pumps are capable of trouble-free operation up to 34.5 MPa (5000 psi) for the volumes needed for lock and spillway service. Increased system pressure, however, increases the risk of leakage and the size of some transmission components. Increased size translates to increased life-cycle costs. Bending and/or buckling loads often will govern when sizing the cylinders (bore and rod diameter), which sometimes reduces the required operating pressure.

b. System Component Ratings. The manufacturer's published pressure, volume, friction, temperature, and fluid compatibility ratings should be used for selection of all system components. They should have a normal minimum pressure rating equal to at least twice the maximum normal operating design pressure. Components should be specified to deliver the maximum design volume flow rate at a cumulative pressure loss of less than 1 MPa (150 psi), including main system valves, piping, hose, filters, and manifolds. The operating temperature ranges of hydraulic system components and the oil used are important considerations because parts of the system normally are located in machinery trenches and subject to ambient temperatures.

3-4. Hydraulic Cylinders. Hydraulic cylinders convert fluid power to linear motion. The three basic types of hydraulic cylinders used in typical lock-and-dam machinery applications are the tie rod type, telescoping type, and mill type.

a. Types of Cylinders

(1) Tie Rod Cylinders. Tie rod cylinders are commonly used in sizes below 10-in. bore. These cylinders are more prone to problems caused by field maintenance than other types of cylinders. They typically are designed for lower overall pressure requirements than the mill type cylinder.

(2) Telescoping Cylinders. These cylinders are commonly used in situations, similar to an elevator, where installation space is limited and loading is relatively minor but the cylinder stroke required is very long. Because each stage of the cylinder must be enclosed within another stage, the available force is limited.

(3) Mill Type Cylinders. Mill type cylinders generally are rated for the higher pressures than the other designs. The cylinder heads generally are mounted with bolts or cap screws. Most main operating systems should be designed for this type of cylinder.

b. Cylinder Features. All hydraulic cylinders should be provided with SAE four-bolt flange connections for the supply ports at the top or side of each end. All cylinders should be furnished with air bleeds and oil drains at the high and low points, respectively, at each end. All cylinders should be provided with zero-leakage sealing systems to prevent drift and environmental contamination. All piston rods should be either nickel chrome-plated steel or chrome-plated stainless steel. All cylinder mounting features, including trunnions, should be attached by the manufacturer at the factory. The cylinder stroke should be designed with sufficient overtravel to facilitate installation tolerances for proper adjustment. The cylinder rod should be designed with a minimum factor of safety of 3.0 to resist a buckling load under compression. Optional features may be obtained with the initial purchase of cylinders. The cylinder may be furnished with local control manifolds mounted directly to the cylinder, adjustable cushions to assist in deceleration when approaching the stoke limits, and stop tube or double pistons to assist in resisting side loading of the piston rod.

c. Packaging for Shipping, Handling, and Storage. All cylinders shall be packaged for the maximum storage time and conditions anticipated under the contract duration, including shipping conditions. Cylinders should be shipped with the piston rod(s) retracted and restrained from movement. The contractor shall follow the manufacturer's recommendations for cylinder storage and maintenance while stored. Unless specified otherwise by the manufacturer, cylinders stored horizontally should be rotated 90 deg every six weeks. Periodic rotation isn't required for cylinders stored vertically with the rod up. Small cylinders shipped without oil shall have a vapor phase inhibitor added to prevent corrosion, and then be flushed and filled with clean system fluid before being put into service. Large cylinders should be shipped from the manufacturer, filled with the required system fluid and remain filled with fluid while in storage. Temporary accumulators, stand pipes, or similar expansion devices should be installed by the manufacturer before filling to accommodate oil expansion due to temperature changes during shipment and storage.

3-5. Hydraulic Motors. Hydraulic motors convert fluid power into rotary motion. Pressurized fluid from the hydraulic pump turns the motor output shaft by pushing on the gears, pistons, or vanes of the hydraulic motor. Hydraulic motors can be used for direct drive, where sufficient torque capacity is available, or through gear reductions. Most hydraulic motors must operate under reversible rotation and braking conditions. Motors often are required to operate at relatively low speed and high pressure. Motors can experience wide variations in temperature and speed in normal operation. There are three types of hydraulic motors: gear, piston, and vane.

a. Gear Motors. Gear motors are compact, basic in design, provide continuous service at rated power levels with moderate efficiency, and are reliable with high dirt tolerance. There are several variations of the gear motor, including the gerotor, differential gear motor, and roller-gerotor; all produce higher torque with less friction loss.

b. Piston Motors. The most common type of motor available is the axial piston type. Axial piston motors have high volumetric efficiency, which permits steady speed under variable torque or fluid viscosity conditions. This permits the smoothest, most adaptable approach to variable loading conditions. Axial piston motors are available in two types of design, swash plate and bent axis. The swash plate design is the most commonly available hydraulic motor. The bent axis design is the most reliable and the most expensive. Radial piston motors are extremely reliable, highly efficient, and rated for relatively high torque. Radial piston motors are less commonly available, which might require extensive investigation into availability to insure adequate procurement competition. All piston motors are available in fixed and variable volume versions.

c. Vane Motors. Vane motors are compact, simple in design, reliable, and have good overall efficiency at rated conditions. Vane motors use springs or fluid pressure to extend the vanes. Vane motors generally use a two- or four-port configuration. Four-port motors generate twice the torque at approximately half the speed of two-port motors.

3-6. Hydraulic Pumps. Hydraulic pumps convert electrical energy into fluid power. The fluid power is in the form of hydraulic fluid delivered to operating devices at a pressure and volume required to perform the work of the system. Gears, pistons, or vanes are used to compress the hydraulic fluid to the conditions required by the system. Hydraulic pumps generally operate at higher speeds and pressures than hydraulic motors without significant thermal shock, speed, and load variations. While some systems use reversible pumps, most lock operating systems use a unidirectional pump with a directional control valve to reverse the operation of the actuators. Hydraulic pumps generally have three basic types: gear, piston, and vane.

a. Gear Pumps. The gear pump is a simple, rugged, positive-displacement design with a large capacity for a small size. Gear pumps have a high tolerance for fluid contamination, good overall efficiency, and are relatively quiet. While these pumps are fixed volume at a given rpm, their flow rate/rpm characteristics are linear within their efficiency ranges. Speed and direction control of an actuator therefore can be provided by driving a reversible gear pump with a variable speed electric motor, which makes them ideal for integral type power units. Gear pumps also are commonly used for pilot pressure applications. Gear pumps generally are restricted to less than 24 MPa (3500 psi) service.

b. Piston Pumps. The piston pump is the type most often recommended as the main pressure pump for hydraulic power systems. It has the highest volumetric efficiency, highest overall efficiency, highest output pressures, and longest life expectancy. This type of pump is available in variable displacement models with a large variety of control systems for pressure and capacity. It is recommended that the drive motor speed be designed for 900 to 1200 rpm, if possible, in order to reduce noise and increase pump life. Piston pumps generally are restricted to less than 42 MPa (6000 psi) service.

(1) Axial Piston Pumps. Axial piston pumps are used for high-pressure and high-volume applications. The two basic types of axial piston pump are the swash plate and the bent axis designs. The bent axis design is considered to be a higher quality pump

with less noise, vibration, and wear than the swash plate design. Swash plate pumps can be designed to drive a separate pilot pressure pump from a shaft extension, while bent axis pumps will require a separate electric motor/pump arrangement for pilot pressure.

(2) Radial Piston Pumps. Radial rolling piston pumps are an extremely reliable, simple design. A typical design includes solenoid controls for up to five discreet operating speeds. Each of the operating speeds has a variable adjustment range from zero to full volume capacity to permit field adaptation to operating conditions. The typical pumping system includes an integral pilot pump, internal pressure relief valves, and associated control devices for speed of shifting between pumping rates.

c. Vane Pumps. Variable volume vane pumps are efficient and durable, as long as a clean hydraulic system is maintained. In a simple circuit, the pressure compensation feature of the vane pump reduces the need for relief valves, unloading valves, or bypass valves. Vane pumps generally are restricted to less than 14 MPa (2000 psi) service.

3-7. Control Valves. Various types of valves are used to control pressure, volume, and direction of fluid flow in a hydraulic circuit. Typical operating elements of these valves are poppets, sliding spools, springs, stems, and metering rods. Valves can be controlled manually (i.e., with a hand wheel, lever, joystick, etc.), mechanically (with a cam, roller, toggle, etc.), hydraulically (with pilot pressure), or electrically (with a Linear Variable Differential Transformer, solenoid, etc.).

a. Circuit Types. Control valves are used in two basic types of hydraulic circuits: closed loop and open loop.

(1) Closed Loop Circuits. Closed loop circuits use a feedback system that generates input and output electrical signals to track system performance. The electronics compare the input and output signals on a continuous basis in order to automatically adjust the system to the level of performance required. Control valves for closed loop circuits are typically proportional and servo-valves. Proportional valves are also used in open loop circuits.

(2) Open Loop Circuits. Open loop circuits rely upon the performance characteristics of the individual valve components to meet the system requirements. Basic pressure control valves, flow control valves, and directional control valves are used to alter the pressure, flow, and direction of the fluid power using only simple electrical solenoids for control in an open loop circuit.

b. Valve Types. The following valve types can be used on either open or closed loop circuits, although the servo-valve is typically used in closed loop circuits.

(1) Proportional Valves. These valves can assume any position between their minimum and maximum settings in proportion to the magnitude of an electrical input signal. They can control direction, flow rate, and pressure. Because they can assume multiple positions, the directional valve spools can be designed to throttle the flow rate in each direction of motion. Actuator force or torque can be controlled by varying the pressure. Pressure is often a function of actuator speed in lock-and-dam operating equipment. Where pressure cannot be related to actuator speed, pressure control valves must be used in the circuit. Proportional valves are mass produced with interchangeable spools and valve bodies. This can lead to slight misalignment, which results in center position overlap or no flow to the outlet ports. This flow deadband, while not a problem in flow control type circuits, can cause errors and instability in closed loop feedback positioning circuits.

(2) Servo-Valves. Servo-valves are made to closer tolerances than proportional valves. These valves have superior response, repeatability, and threshold response. They are, however, considerably more expensive than proportional valves. Repeatability is a measure of the number of times a valve can produce the same flow rate with repeated signals of the same magnitude. Threshold response is the smallest variance in input signal that produces a corresponding change in the flow rate. Meticulous construction is required to produce precise alignment of the spool lands with the valve body ports. The higher cost of servo-valves usually is justified when more sophisticated performance requires a high load stiffness, good stability, precise positioning, good velocity and acceleration control, good damping, and predictable dynamic response.

(3) Pressure Control Valves. Pressure control valves are used in hydraulic systems to control power and to determine pressure levels at which various operations or actions can occur. Pressure control valves can limit the maximum pressure in a circuit, reduce pressure levels from one part of the circuit to another, provide alternate flow paths for fluid at selected pressure levels, provide resistance to fluid flow at selected pressure levels, and modulate transient pressure shock in a hydraulic circuit. Pressure control valves include pressure relief valves, sequence valves, counterbalance valves, holding valves, unloading valves, reducing valves, and shock suppressors. All these valves have some method of pressure setting adjustment. Valves operating in enclosed areas should be furnished with key-locked handles to discourage casual adjustment. Valves exposed to weather, such as in grated pits, may be furnished with lock nuts to maintain final settings.

(a) Pressure Relief Valves. A pressure relief valve limits upstream pressure to a preset value by returning part, or all, when sized correctly, of the fluid flow to the reservoir until the upstream pressure drops below the relief setting. The two principle types of pressure relief valves are the spool type and the poppet type. The poppet type has the shorter response time, but the spool type has more stability and accuracy of operation and adjustment. Most main pressure relief and pilot relief valves should be the balanced piston (spool) type with an appropriate adjustable pressure range. Pressure relief valves should be furnished for the main pressure pump, pilot pressure pump, and each actuator line between the directional valve and actuator inlet ports.

(b) Sequence Valves. Sequence valves direct flow to a circuit in a predetermined logical sequence by sensing that adequate pressure has been developed in one circuit before allowing flow in another. Sequence valves might have to actuate two or more spools or poppets to connect primary and secondary passages. Sequence valves generally are used in circuits where one actuator must complete its operation before another actuator, at a higher pressure, can begin its operation.

(c) Counterbalance Valves. A counterbalance valve is a normally closed pressure control similar to a relief valve, but has a reverse free-flow check valve. Counterbalance valves are used to control an overrunning or overhauling load. They commonly are used on culvert valve control circuits to prevent the open valve from drifting downward when main pump pressure has been blocked by the directional control valve. Counterbalance valves can be used with an internal pilot or a remote pilot actuation. Using a remote pilot can significantly reduce the power required to lower the load at a controlled rate. Selecting the proper pilot ratio for a given application is important. In general, higher pilot ratios are suitable for stable and constant loads, while lower pilot ratios are suitable for unstable and varying loads.

(d) Holding Valves. A holding valve is basically a special type of counterbalance valve, functionally similar to a pilot-operated check valve. Pilot-operated check valves trap fluid to prevent actuator movement but, during actuator travel, produce little resistance. Counterbalance valves, in addition to preventing movement, add resistance during travel, which increases the power required to operate. Holding valves avoid the objectionable features of both the counterbalance and pilot-operated check. Holding valves also provide built-in thermal-relief protection.

(e) Unloading Valves. The straight unloader and the differential unloader are two basic types of unloading valves. The straight unloading valve is a two-stage relief valve with its pilot port connected externally to a separate signal source. The differential unloading valve operates on an area differential between the control poppet seat and a pilot piston of 10 to 20%. Unloading pressure is controlled by the spring force on the control poppet. The pilot piston is actuated by external pilot pressure such that it unseats the poppet at a preselected pressure. When the poppet opens, the main valve spool shifts to the open position. The pilot piston then prevents the poppet from reseating until the pressure drops below the required differential.

(f) Reducing Valves. A reducing valve is a normally open valve that modulates, or blocks, flow at a preset pressure. They control downstream pressure by restricting flow due to the positioning of the spool, with respect to the outlet port.

(g) Shock Suppressors. Sometimes called safety valves, shock suppressors are two-way valves that snap open to relieve hydraulic shock. Hydraulic shock is an excessive pressure applied instantaneously to the circuit. When high-pressure, high flow rate events occur, the two-way valve snaps open to allow the fluid to pass from the inlet to tank. The small amount of fluid bypassed decreases the rate of pressure rise, thus, preventing the shock.

(4) Flow Control Valves. Flow control valves are used to control the rate of fluid flow from one part of the hydraulic system to another. These valves can be used to:

- Limit the maximum speed of the actuating devices;
- Limit the maximum power available to sub-circuits;
- Proportionally divide or regulate the flow to different branches of a circuit;
- Control the speed of pilot-controlled valves.

Flow control valves operate in three general configurations, meter-in, meter-out, and bleed-off. Meter-in and meter-out methods use a throttling approach to restrict the size of the fluid path, while bleed-off bypasses the flow to tank or a lower pressure area of the circuit. Flow control valves can be compensated or non-compensated. Compensated valves automatically adjust to provide uniform pressure drop across the valve to furnish constant flow rates.

(a) Meter-in Circuits. Meter-in flow controls should be used when the load might kick back, the circuit power or pressure level must be retained when the actuator pressure level falls off, and for dividing flows to multiple branch circuits.

(b) Meter-out Circuits. Meter-out flow controls should be used when the load is overhauling or overrunning, the load can decrease causing lunging, and when a back pressure is desired for rigidity in motion.

(c) Bleed-off Circuits. Bleed-off flow controls should be used when a soft circuit is desired, and when the power to be controlled is a fraction of the circuit power to the actuator.

(5) Directional Control Valves. Directional control valves, by providing a choice of flow paths, do one or more of the following:

- Control direction of actuator motion;
- Select alternate circuits;
- Perform circuit logic functions.

The check valve is the simplest of all directional controls. Other directional controls are described by the number of primary ports available for control. They usually are referred to as two-way, three-way, or four-way valves.

(a) Check Valves. Check valves can be used for a wide variety of functions in a circuit. They can be used to prevent flow in one direction, while permitting free flow or pilot-controlled flow in the other. They can be piloted externally to provide an actuator locking function. They can be used in pilot lines to provide rapid release of a pilot-operated spool.

(b) Two-Way Directional Valves. Two-way valves generally are used to perform logic functions such as AND, OR, and AND/OR decisions. These valves allow flow in one position and no flow in the other. These valves can be in the normally open or the normally closed position until actuated by levers, solenoids, pilot pressure, etc. Two-way valves can be used to perform interlock or safety functions.

(c) Three-Way Directional Valves. Three-way valves have a pressure supply port, a tank port, and one actuator port. These valves generally are used with an actuator designed with springs or other means of returning to a rest position, because they can address only one actuator inlet port. This type of valve generally is not recommended for normal lock-and-dam machinery design, because it is important to use pump flow to control operation in both directions of travel.

(d) Four-Way Directional Valves. The typical directional valve used on lock-and-dam projects is the four-way, three-position, directional valve. This valve has four main ports: main pressure, tank, actuator A, and actuator B. This permits the valve to reverse actuator direction in a controlled manner with the main pump flow. These valves can be furnished with a wide variety of spools for controlling flow. The directional control valve is usually the single greatest pressure loss point in a hydraulic circuit. Therefore, to minimize system losses, it is customary to design this valve for 1.5 to 2.0 times the maximum system flow rate.

(e) Spools. The typical spool used for modern lock-and-dam hydraulic systems is the blocked center, solenoid-controlled, pilot-operated, spring-centered type spool. This type of spool is used successfully in hydraulic systems that can be operated remotely with a series of interlocks to prevent conflicting machinery behavior. A solenoid-operated pilot pressure four-way valve applies pilot pressure to shift the pilot-operated spool in the main pressure four-way valve. The pilot pressure to each side of the spool usually is passed through a combination flow control-reverse free flow check valve to permit adjustment of main pressure spool actuation speed. The spring-centered feature is used in the pilot valve and the main pressure valve, to return the system to blocked center when the solenoids are not energized to permit machinery operation. The tandem center-type spool has been used with some success, when proper pressure control valves are included in the circuit to prevent actuator drift after main pressure pump shutdown.

(f) Controls. Solenoid-controlled, pilot operators usually are used on the more modern open loop-type systems to allow remote, or centralized, operation with appropriate electrical or electronic interlocks. Direct solenoid-operated valves are generally available in smaller flow rate capacities. All solenoids should be equipped with manual operating pins for troubleshooting and emergency operation. Lever or other manual-type operators should be used only on the most basic systems, or where human observation of the operation from the local controls is essential.

c. Mounting Systems. Control valves should be mounted on steel block-type manifolds. Manifold systems are economical, reduce leakage, minimize piping fabrication costs, and reduce space requirements. See paragraph 3-9.

3-8. Reservoirs. A general rule for the initial sizing of a hydraulic reservoir is that it should have a minimum capacity of approximately 3 times the maximum pump discharge rate. The actual capacity required will be determined by other factors such as the number and size of actuators served, long lengths of large diameter piping, or excessive thermal expansion. Because the typical hydraulic cylinder has substantially more fluid per length of stroke on the cap-end side than on the rod-end side (the rod takes up volume as it retracts into the cylinder), more fluid will be returned to the tank when a cylinder is retracted. Where multiple hydraulic cylinders are served, an analysis of all potential operating cases (number of cylinders that can be extended or retracted at the same time) should be performed to determine the maximum and minimum reservoir levels required for the complete system. The reservoir should have sufficient capacity to provide a flooded pump suction, without vortex formation, under all operating conditions. After the reservoir size is determined, its heat dissipation capacity should be checked to see if it is adequately sized to dissipate heat generated by the individual components of the system and any ambient or solar heat gain. The reservoir should be fabricated from annealed and pickled steel or stainless steel plate, designed for the loads applied by all accessories and pumping equipment. Coating the interior of reservoirs not made of stainless steel with an epoxy system is no longer recommended by many hydraulic system manufacturers. The reservoir should be internally reinforced, as required, with vertical baffles to separate the return oil from the pump suction. The baffles shall be designed to prevent turbulence at the pump suction. External pumps are recommended for ease of maintenance, and each pump/motor group should be mounted with vibration isolators and connected with flexible hoses and conduit to prevent noise and vibration transmission to the reservoir and fluid. If internal pumps are required, D-Flange motors should be used to allow easy motor removal without removing the pump from the reservoir. The reservoir should be provided with appropriate oil level gauges, low- and high-level shutoff switches, magnetic particle collector, drain valves, removable clean-out plates for suction and return sides, suction filters (if required for pumping unit design), breathers, and reservoir heaters (if required by the hydraulic fluid design).

a. Oil level gauges. Oil level gauges should be one or more sight gauges placed to show the full normal operating range of fluid levels within the middle 80% of the gauge range. The lowest sight gauge installed shall have a thermometer designed for the maximum normal hydraulic fluid operating range.

b. Shutoff Switches. Float-type shutoff switches have proven reliable. Switches can be provided to prevent low suction level and overfill, as well as issue alarms when approaching fault conditions.

c. Magnetic Particle Collectors. These devices are permanent magnets immersed in the hydraulic fluid that collect metal particles that can cause pump damage. Periodic inspection and cleaning of these devices can be essential in the early identification of pump wear problems.

d. Drain Valves and Cleanouts. Reservoir drain valves should be designed to permit easy access for draining the hydraulic fluid to the bottom of the reservoir. This includes placing the location well above the surrounding floor level sufficient to place a disposal container of modest size for collection of the fluid. Cleanouts should be properly sealed, bolted or clamped, and placed for easy access when in service.

e. Reservoir Heaters. Reservoir heat is not required for most installations. There is a large number of viscosity stable hydraulic fluids, designed for aircraft and missile service, that are usable with piston pumps at temperatures as low as –40^{o}C (–40^{o}F). Where it is determined necessary, the reservoir heating elements should not exceed a watt density of 1.5 W/sq cm (10 W/sq in.) of element in order to eliminate charring of the fluid.

f. Breathers. Hydraulic systems typically use a replaceable filter-breather device that permits atmospheric air to be drawn into the reservoir. These devices can be furnished with a desiccant-type filter for moisture control. Recent experience indicates the best method of breather protection for hydraulic systems in the damp, lock-and-dam environment is the use of a bladder-type breather system. Where sufficient space is available, a flexible bladder-type breather system, usually called a reservoir isolator, should be used. Reservoir isolators recycle the air to and from the reservoir, sealing it from dirt, water, and other contaminants. Water in the hydraulic system is one of the primary reasons for hydraulic component failures. Water usually infiltrates the system from the moisture in the air that is exchanged in the reservoir through the breathers. A bladder is an elastomeric air chamber that is connected to the reservoir. The bladder expands and contracts as the air volume changes in the reservoir, eliminating the need for a breather. Reservoirs using bladders should be pressure tested and equipped with relief valves. Where installation of a reservoir isolation bladder is not available, use of a desiccant breather may be employed, though frequent replacement of the desiccant might be necessary in humid environments. Another option is to install the reservoir in a climate controlled room with dehumidification.

g. Control Valve Manifolds. For convenience of adjustment and maintenance, it is beneficial to mount any control valves associated with the pumping system on a control manifold mounted on the reservoir. In some cases, the directional control valves, pressure controls, and flow controls can be mounted on the reservoir to conserve space.

h. Secondary Containment. Some method of secondary containment for the contents of the fluid reservoir should be provided to eliminate spills and waterway contamination. The reservoir can be specified as double-walled with leak detection electronics between the walls. The reservoir can be contained within a containment pit similar to aboveground fuel tanks with leak detection electronics within the pit. This is required for any reservoir with direct drainage features to the waterway, such as floor drains, sump pumps, or lock wall recesses. Oil/water separators may be used to treat drainage prior to discharge into the waterway, but these devices should be tested thoroughly prior to installation to ensure that the effluent meets all state and local environmental pollution criteria.

i. Reservoir Types. Separate reservoirs and sealed reservoirs are two basic types that should be considered.

(1) Separate Type Reservoirs. Separate reservoirs are the most common design in industrial, or lock-and-dam, applications. A separate reservoir can be designed for a single pump/motor group serving one or more actuators or multiple pump/motor groups serving many actuators. Some dual chamber locks with centralized power units have one reservoir supplying three pump groups (two of the three normally are used), which operate all the actuators for both chambers. The three principal versions of separate reservoirs are rectangular, L-shaped, and vertical.

(a) Rectangular Reservoirs. These types of reservoirs use a rectangular steel box to hold the fluid and house the accessories. They can be designed with the pump groups mounted on top, underneath, or inside the reservoir. A short suction line, with top mounting, is required for each pump that extends below the minimum suction submergence of the fluid. The pump groups are provided with a flooded suction, with underneath mounting, which improves pump operating conditions significantly. The pumps are submerged, with inside mounting, in the fluid and the drive motors are mounted vertically on top of the reservoir.

(b) L-Shaped Reservoirs. L-shaped reservoir packages have the pump groups mounted next to the reservoir on a common base. This arrangement provides good access to components for maintenance and repair.

(c) Vertical Reservoirs. Vertical reservoirs have the pump group in a vertical plane with the pump below a removable reservoir cover. While this arrangement is compact, with a low suction lift requirement, the size is limited by the requirement for lifting the entire pump group, controls, and reservoir cover to service any equipment.

(2) Sealed Reservoirs. Sealed reservoirs primarily are used for the integral power unit of a self-contained actuator, which consists of a power unit attached directly to the hydraulic cylinder it operates. These actuators can be configured in different ways by changing the shape of the reservoir and where or how it is attached to the cylinder. The direct connected miter gate actuators recently installed on several locks have long, slender reservoirs made from square structural tubing, bolted to brackets on the side of the cylinder. Tainter valve actuators also have been designed with shorter reservoirs made from round structural tubing and permanently welded to the rear of the cylinder tube during fabrication. This arrangement allows the actuators to fit existing recesses without modification. Sealed reservoirs have a pump mounted inside and a submersible motor mounted outside. Since these reservoirs do not have breathers or accumulators, the air pressure inside will vary with cylinder rod position and oil temperature. The actuator should be designed so the normal pressure range in the reservoir is between 3 and10 psig. Care should be taken to make sure the pressure never goes below atmospheric or above 30 psig.

3-9. Manifolds. Pre-drilled steel manifold blocks have been extremely reliable for connection of control valve assemblies in hydraulic systems. Manifolds provide short, direct flow paths between controls, which reduce friction and response time. Aluminum manifolds should not be used with steel piping or steel-bolted SAE flange connections due to the localized yielding of the aluminum threads during installation, from shock, and vibration. The specification should ask for detailed drawings of the drilled passages of the steel manifolds. Sub-plate type control valves are directly mounted or stack mounted to the manifold, and cartridge-type valves screw directly into the manifold, eliminating excess piping and reducing leakage. Maintenance costs are reduced by eliminating piping connections. Manifolds can be sensitive to filtration problems, but proper preventive maintenance should yield excellent results. Test ports should be specified where required to provide convenient gauge connections for adjustment and troubleshooting. It is essential to specify that all manifolds should be furnished with as-built fabrication drawings that document the dimensions and locations of all pre-drilled passages, including where fabricating passages have been plugged. A manifold, properly prepared for long-term storage, of each different type should be included in the spare parts.

3-10. Filters. Pressure-side and return-side filters should be provided on all hydraulic power units. Spin-on pressure line filters, rated for full maximum discharge pressure of the pump, should be furnished for the main supply and pilot supply pumps. A large, multiple cartridge, return line filter should be mounted adjacent to the reservoir. Return line filters should be the full-flow type, designed to pass all flow through the filtering elements. Return line filters should, however, include a bypass relief valve system designed to shunt flow around the filter after a pre-set pressure drop is exceeded. Return line filters should be provided with a maintenance indicator that clearly shows when the cartridges need to be replaced. All filters should be rated for the minimum required by the manufacturer for the types of valves used. Modern proportional valves can require a particulate filtration rating as low as 5 microns with a B_{10} filtration ratio of 4.0, in accord with ANSI B93.30M. The system must be designed such that all hydraulic fluid passes through one or more of these filters during installation, testing, and normal operation. A 200-mesh suction strainer should be used on hydraulic pump inlets only when required by the pump manufacturer.

3-11. Accumulators. Accumulators store hydraulic energy in a manner similar to electric storage batteries. They store potential energy by accumulating pressurized hydraulic fluid in a vessel for later release into the system. Accumulators can improve energy efficiency, absorb shocks, damp pulsation, reduce noise, prevent pump cavitation, compensate for leakage or thermal expansion, and provide emergency operation capability. Accumulators are nominally designated by their energy storage mode, either pneumatic, spring loaded, or weight loaded.

a. Pneumatic Accumulators. Pneumatic accumulators use compressed inert gas, such as nitrogen, to force hydraulic fluid back into the hydraulic system. Compressed air is not used due to the danger of explosive air-oil vapor. Accumulators should be the separated type that uses bladders, diaphragms, or pistons to separate the hydraulic fluid from the compressed gas. Bladder designs are the most versatile.

b. Spring-loaded Accumulators. These accumulators use a spring compressed by a piston to force fluid into a hydraulic circuit. They typically are used for energy storage in applications below 3.5 MPa (500 psi) and are not recommended for shock absorption.

c. Weight-loaded Accumulators. These accumulators use a heavy weight to push the piston down, forcing fluid into the circuit. They are typically very large, with installation and maintenance problems, and, therefore, not recommended.

3-12. Piping. All piping, including tubing and flexible hose assemblies, should be designed for a factor of safety of 8, based upon the maximum normal operating pressure. This should provide adequate design tolerance for shock and vibration. Proper design of hose assemblies should include adequate length, swivels, end connections and outer coverings to account for exposure to the environment, equipment movement, and adjacent hazards. Black steel pipe should be furnished in the pickled and oiled condition. Stainless steel pipe, however, has been found economically justifiable on a life cycle cost basis, with reduced maintenance and leakage due to corrosion. Hydraulic tubing can be used for diameters below approximately 40 mm (1-1/2 in). Hydraulic tube fittings should be swaged type or flare type. Bite-type tube fittings should not be used. All pipe hangers should be furnished with phenolic shock-absorbing inserts to accommodate hydraulic system shock and vibration.

a. Required Design Features. All piping systems shall have air bleed valves at the high points in the system. All piping systems shall have drain valves at low points in the system. An analysis should be performed at sufficient intervals to locate shutoff valves in the piping system and to permit localized drainage of piping for pipeline repairs. Gauge and pressure transducer connections should be furnished at appropriate locations for future system troubleshooting. Piping shall be tested to the maximum normal working pressure rating of the pipe, tubing, or hose in the system.

b. Fluid Velocity Requirements. Main pressure lines should be designed for a velocity of 3 to 4.5 m/sec (10 to 15 ft/sec). Hydraulic return lines should be designed for a velocity no greater than 3 m/sec (10 ft/sec). Pump suction lines should be designed for a velocity of 0.6 to 1.5 m/sec (2 to 5 ft/sec). Pilot and drain lines should be designed for a velocity of 3 to 4.5 m/sec (10 to 15 ft/sec).

c. Piping Layout. The piping system should be arranged to permit convenient removal of all valves, pumps, filters, actuators, and associated appurtenances. Shutoff valves should be placed around equipment that might need to be removed from the circuit for service. Piping should be sloped slightly to encourage complete drainage during servicing. Expansion and contraction should be considered in any design with long pipelines, with the inclusion of accumulators and hoses as required.

d. Hydraulic Tubing. Tubing is specified by outside diameter and wall thickness. Commercially available tubing is clean and easy to bend. Tubing provides easier installation, and fewer fittings are required. Stainless steel and carbon steel tubing is available in welded and seamless versions. Some difficulty has been encountered with the application of tube fittings to tubing above 40 mm (1.5 in.) outside diameter.

e. Hydraulic Hose. Hydraulic hose should be used to connect hydraulic components for which relative motion, or thermal expansion, must be accommodated.

Hose is specified by inside diameter and type of construction. Hose has three basic parts: the tube, which is the inner liner that carries the fluid; the reinforcement. which is the part that covers the inner liner with woven, braided, wrapped, or spirally-wound materials for strength; and the cover, which is the exterior material that protects against abrasion, chemicals, weather, and ultraviolet rays. Hydraulic hose should be specified as indicated in SAE J517. Plastic hose is lighter, smaller, and lower in electrical conductivity than synthetic rubber hose. Plastic hose is inert to most chemicals, hydraulic fluids, and ozone. Rubber hose is more resilient and flexible.

f. Piping Fittings. Most piping system leaks occur at fittings or at the connection of fittings with valves, pumps, manifolds, or actuators. Leaks generally are caused by shock, vibration, thermal expansion/contraction. or human impact at joints. Piping fittings should match the type of pipe system in use, such as butt-weld, socket welded, swaged or flare tube, and swaged or crimped hose. Swivel fittings should be used with hydraulic hose to avoid crimping and adverse bending. Quick disconnect couplers, which incorporate check valves to shut off flow, can be used for infrequent or emergency connection of equipment to the hydraulic system. Threaded fittings were once common on old locks with low system pressures. Since threaded fittings are prone to leakage, even at very low pressures, they are inappropriate for modern high-pressure hydraulic systems.

3-13. Hydraulic Fluid. Hydraulic fluid generally is selected for compatibility with the main hydraulic pumping unit. The operating range of the pump is the primary consideration for system performance. Most normal operating systems, which experience widely variable temperature and climate conditions, require the use of a petroleum-based fluid with a high viscosity index (VI). The permissible viscosity range varies with different manufacturers and types of pumps. Hydraulic fluids are covered in EM 1110-2-1424. The physical characteristics, quality requirements, use of additives, and types of hydraulic fluids are discussed in EM 1110-2-1424, Chapter 4. The EM has requirements for biodegradable hydraulic fluids that are discussed in Chapter 8. Hydraulic system cleanliness codes, oil purification, and filtration are discussed in Chapter 12.

3-14. Gauges. All systems should have properly sized pressure and temperature gauges at locations near important system operating equipment such as the pumps, pressure control valves, actuators, and directional control valves. Pressure gauges should be rated for the maximum operating pressure of the system. Gauges with smaller scale ranges have, in general, higher accuracy. Manual pressure gauges should have minimum intermediate graduations of 0.35 MPa (50 psi). Pressure gauges are essential for proper troubleshooting of system performance. All pressure gauges should be provided with pressure snubbers, to protect against shock. Shutoff valves should be used to isolate the gauges until readings are required. Glycerine-filled pressure gauges are not required, unless severe vibration is expected at the gauge location.

3-15. Special Design Considerations and Lessons Learned.

a. Hydraulic Power Units. Conventional power units should be located in areas that are not accessible to floodwaters. They can be placed in galleries, sealed pits, buildings, or on platforms that are protected from the flood of record, with appropriate freeboard. While submersible power units are available in some smaller sizes, they are a proprietary item. Secondary containment features, such as dams, pits, and piping penetration seals, should be coordinated with the structural design features. Secondary containment monitoring should be coordinated with the electrical control and power supply design.

b. Directional Control Valves. Close coordination with the electrical controls designer is necessary when using proportional or servo-valves to provide feedback control. Chapter 12 discusses power and control concerns for hydraulic systems.

c. Pressure Control Valves. Pressure control valve settings should be evaluated for the maximum setting that will cause damage to the gate. This could cause an increase in operating time, but will prevent frequent damage to expensive repair items. Structural design items generally have lower factors of safety than the mechanical equipment. Pressure relief settings should protect all equipment in the system, including the gate.

d. Pressure Transducers and Gauge Ports. Pressure transducers can be used to report, or record, pressures at key locations in the hydraulic system for troubleshooting purposes. Transducers can report to a programmable logic controller or personal computer. Gauges can be used for less sophisticated control systems for which performance evaluations will be done manually. Push-to-read gauge devices permit quick installation of portable gauges.

e. Hydraulic Cylinders. Most applications require coordination between the electrical control system and the cylinder design to provide accurate position sensing for the gate operations. The complexity of the position sensing system is dictated by the complexity of the control system. Fully automated operations require sophisticated systems. The actual length of stroke, or orientation of the cylinder, can restrict the type of system that will give accurate indication.

f. Cylinder Tubes and Rods. Guide specifications indicate cylinder tubes should be American Society for Testing and Materials (ASTM) A519 Grade 1018 heavy wall seamless tubing. However, cylinder tubes fabricated from one-piece AISI 4340 steel with one-piece ASTM A36 steel trunnions have given satisfactory service for hinged crest gates. The most common problem occurring with hydraulic cylinders has been the leakage of hydraulic fluid. This generally is caused by corrosion and scoring of piston rods. For this reason, the material and finish of the piston rods must be matched to the conditions in which the cylinder will be used. Various piston rod coatings have been developed for resistance to corrosion and abrasions. The guide specifications indicate piston rods should be either carbon steel (ASTM A108 Type C 1045 or Type CR 4140) with nickel and hard-chrome plating or stainless steel (ASTM A564/A564M or ASTM A705/A704M, Type 630, or Type XM-12), heat treated to a condition of H-1150 and hard-chrome plated. Ceramic-coated rods are no longer allowed due to corrosion failures at several locks caused by the inherent porosity of ceramics, detailed in the Engineering and Construction Bulletin No. 2009-3. Some promising new alternatives have been developed that apply a corrosion- and wear-resistant coating to a carbon steel rod by a weld overlay process or laser cladding, but experience with these coatings in a navigation environment is limited. The current rod material and coating combination of choice continues to be a stainless steel rod with chromium plating. The stainless steel provides corrosion resistance and the plating provides a smooth, hard surface for sealing while exhibiting good wear properties. In any case, the coating should be in accordance with the manufacturer's recommendations, and the manufacturer should be responsible for the selection of surface preparation of the rod, the chrome plating or nickel chrome plating process, the quality of the plating, the bonding of the coating to the base metal, and the finish.

g. Pistons. Pistons should be precision fitted to the cylinder body bore. They should be fine-grained cast iron, designed and equipped with seals and bearing rings as needed, and fabricated from materials as recommended by the contractor to provide zero leakage. The design should protect the piston seals from blowout and over-squeezing.

h. Seals. Dynamic seals should be suitable for frequent and infrequent operation and should be capable of not fewer than 500,000 cycles of operation in properly maintained systems. Cylinder tubes also should have the bore honed to a surface finish compatible with the seals being used so as to result in zero leakage past the seals. All seals should be of material suitable for use with the hydraulic fluid specified. In the past, it was common for pistons to have only alloy piston rings. This has been found to allow too much bypass of hydraulic oil around the piston, and standard practice is now for all pistons to be provided with wear bands and seals. In addition to the rod seals, a rod wiper or scraper usually is provided to exclude contaminants from the interior of the cylinder. For rod seals, there has been success using stacks of the chevron style. These are supplied as continuous rings but, in order to facilitate changing the seals without disconnecting the rod, each ring has been split and installed with each split staggered. By doing this, it has been found that cylinders have experienced no increase in leakage or decrease in service life. A recurring issue has been the compatibility of rod seals with the surface finish of hydraulic cylinder rods. This has been particularly true of rehabilitated cylinder rods for which the shop has the capability to finish the resurfaced rods with a highly polished finish but they are not as experienced with large cylinders and slow-moving applications typical of navigation locks. The rod seals can function too efficiently if the rod is too smooth, causing stick-slip due to high friction from the complete removal of the oil film on the rod. Many seal manufacturers will claim an optimal range of 3 to 12 μin Ra but, when questioned, will express a preference to stay to the upper end of that range. Above the manufacturer's stated acceptable range, accelerated seal wear and leakage become a concern.

i. Piping. Piping should be pitched a minimum of 1/2 in. per 50 ft, in order to provide high and low points, and accumulator tanks should be used in systems with long lines to minimize the effect of hydraulic surge.

j. Air Bleeds and Drains. To facilitate complete filling of the system, air bleed valves should be installed at all high points where air can be trapped. To facilitate draining of the system for maintenance, drain valves shall be installed at all low points so that a particular section can be isolated and drained and the complete system can be drained if required.

k. Design, Coordination, Testing, and Commissioning. The hydraulic system supplier should verify the design by providing detailed computations and shop drawings for review and approval. To avoid compatibility issues between the hydraulic and control systems, one supplier should be responsible for providing both the hydraulic components and the control components as an integrated system. When possible, integrated shop tests should be conducted to pretest the hydraulic/control systems and provide initial component settings before shipment. Field tests and commissioning shall be conducted to verify proper operation and document final field settings for all pumps, valves, and control components in the system.

l. Guarantees. Although an exception to DOD policy, designers of some installations have specified guarantee periods greater than one year. In some cases, the guarantee period for hydraulic cylinder parts other than the rods have been specified as two years from date of acceptance, and the hydraulic cylinder piston rod's guarantee period has been specified as five years from date of acceptance. The warranty should be against defective materials, design, chrome plating or ceramic coating of the rod, and workmanship.

m. Design/Build. When advantageous, providing the hydraulic system by means of a design/build supply contract should be considered. The design should be based upon the conditions under which they will operate, and hydraulic cylinders typically are used in exterior locations and might be exposed to hot or cold, humid, moist, and/or dusty conditions. Therefore, the conditions in which the cylinders are to be located should be specified in the contract.

n. Spare Cylinders and Parts. Spare hydraulic cylinders often are specified due to the long lead time for obtaining a replacement if damaged. If spare cylinders are specified, they must be stored and maintained properly. Some manufacturers recommend their cylinders should be stored vertically with the rod up, but this is impractical for large cylinders. Modern materials for bearings and seals have greatly reduced the seal compression problems associated with long-term horizontal storage of large cylinders. However, most manufacturers still recommend that stored cylinders should be protected from the elements and fitted with accumulators or stand pipes to ensure they are completely filled with oil. Provisions for periodically exercising stored cylinders also are recommended. Exercising usually requires extending and retracting the rod a few inches periodically to prevent the seals from sticking. Spares also should be considered for other hydraulic system components such as control valves, protection valves, hydraulic pumps, motors, etc.

o. Contamination Control During Construction. Something that will cause problems and is difficult to correct if not adequately addressed during the construction phase is protecting the hydraulic system from contamination during assembly. Modern proportional valves have stringent oil cleanliness requirements, and contamination can wreak havoc on a hydraulic system. Therefore, quality assurance and control personnel must be fully trained in the importance of contamination control and those requirements must be written into the contract specifications.

p. Other Important Design Considerations. Facilitating the ease of maintenance should be a primary consideration. This includes designing for quick and easy component replacement and, where possible, interchangeability of parts, including hydraulic cylinders. Also, designing for simplicity not only will minimize the potential failure points in the system but also will minimize the maintenance effort required in the long term. The designer should make all attempts to minimize the number of components that will be susceptible to corrosion when subject to the elements. Exposed manifolds can be fabricated from corrosion-resistant materials, but often valve bodies are carbon steel. They can be painted but often corrode in a short time. Covering these components to protect them from the elements is often the best solution not only for corrosion control but also for protecting electronic components from moisture. Finally, the designer should design the system for robustness and extreme weather/flood conditions.

3-16. Position Measuring Systems. Modern hydraulic and control systems have necessitated the accurate feedback of gate or valve position. Position sensing systems are either integral with the cylinder or external to the cylinder and sense either the cylinder stroke or the gate/valve position directly. Systems integral with the cylinder are the most popular and require little or no design effort for external mechanisms or linkages. They also offer the advantage of being included with the cylinder as a turnkey product. Integral systems include the magneto resistive systems (Figure 3-1) and magnetostrictive systems that require the rod to be drilled from the piston end for a sensor rod. Problems with the magneto resistive type have included moisture ingress of the electronics due to poor sealing and the buildup of hydraulic oil pressure between the inboard and outboard seals on the cavity within the sensor. The latter problem can lead to over pressurization of the sensor cavity and has led to failure of the sensor. Solutions have included an external drain of the cavity to a small collection tank that must be periodically checked and emptied. Limitations also include the fact that the system does not provide an absolute indication of position or, in the case of power supply failure, can lose track of its position and require reset. Magneto resistive systems used to require a non-metallic ceramic rod coating, but new laser-clad coatings that will work with these systems should be available soon. Magnetostrictive systems or linear displacement transducers (LDT) provide an absolute indication of position but are limited to a maximum stroke of approximately 25 ft because of the length of unsupported rod in the cylinder when extended.

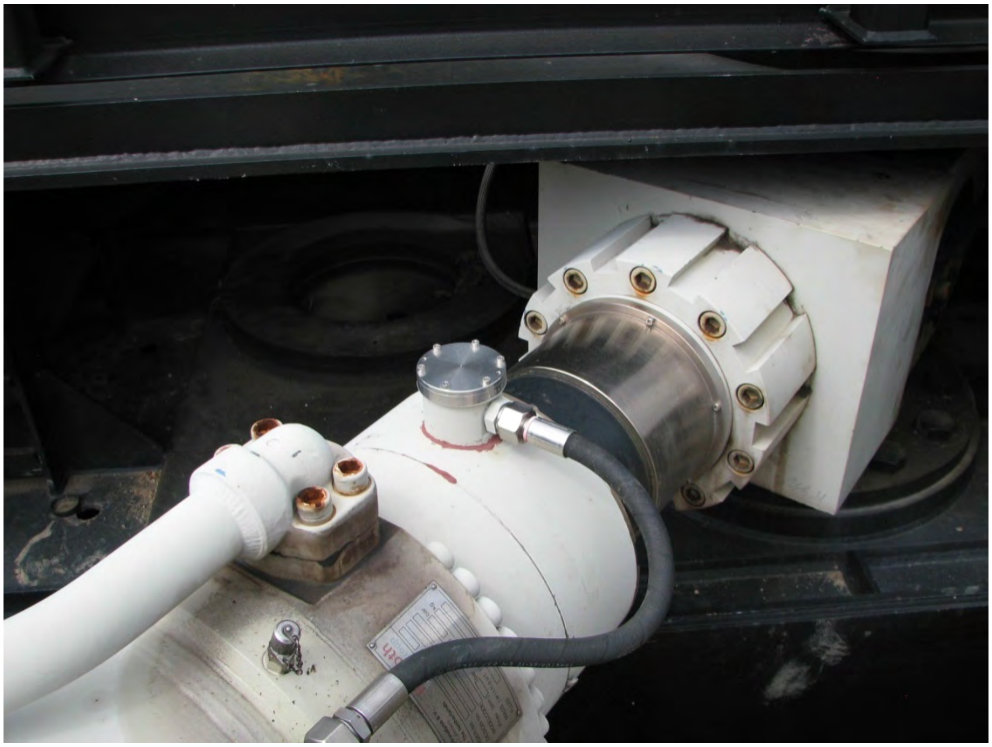


Figure 3-1. Magneto resistive position sensor on rod end gland

a. Alternatives to integral position sensors include: retrofitting the hydraulic cylinder to externally drive either a rotary encoder or LDT, driving a rotary encoder or LDT directly from the gate or valve through an external linkage, or installing a moving target to trip multiple proximity sensors. Maintenance of external systems is independent of the hydraulic cylinders, possibly allowing a simpler and faster repair and a greater likelihood of the gate or valve remaining in service while repairs are being made. Examples of external systems are shown in Figures 3-2 through 3-5. Regardless of system type, moisture ingress issues should be mitigated with appropriate sealing precautions and care to place junction boxes above anticipated flood elevations.

Figure 3-2. Miter gate cylinder with string pot for position sensing

Figure 3-3. Position encoder driven by roller chain connected to stoney gate valve

b. A problem encountered with the system shown in Figure 3-3 has been that, when temperatures are extremely cold, water spray from operation of the filling and emptying valve freezes and accumulates on the chain. This causes the chain to jump off the sprocket, and the correct position of the gate is lost.

Figure 3-4. Moving target with multiple proximity sensors at gudgeon pin

Figure 3-5. Small LDT enclosed in cylinder-type housing and attached to miter gate near the gudgeon pin

CHAPTER 4

Miter Gate Operating Machinery

4-1. Linkages and Components.

a. General Description of Linkages and Components. The miter gate is the most frequently used gate on navigation locks. The miter gate linkage provides the connection between the drive system and the gate itself. Mechanical linkages traditionally have been utilized to open and close miter gates. The majority of the lock sites built in the 1930s on the Mississippi and Ohio rivers used mechanical linkages. Recently, direct-connected hydraulic cylinders have become more prevalent and are used at both new locks and for rehabilitation of existing locks. Four different types of miter gate operating linkages have been used. The Panama Canal linkage, which has no angularity between the strut and sector arms at either the open or closed positions of the gate, is shown in Figures 4-1 and 4-2. The Ohio River linkage, which has angularity between the strut and sector arms at both the open and closed positions, is shown in Figures 4-3, 4-4, and 4-5. The Modified Ohio River linkage has angularity between the strut and sector arms at the recess or open position and no angularity at the mitered or closed position. This linkage is shown in Figures 4-6 and 4-7. A direct-connected cylinder is considered a linkage in the sense that it provides the connection between the drive system and the miter gate. A direct-connected cylinder, shown in Figures 4-8 and 4-9, consists of a hydraulic cylinder and rod connected to a pin on the gate and a pin on the lock wall. The piston force is transmitted directly from the piston rod to the gate. A self-contained actuator, shown in Figure 4-10, is a variation of the direct-connected cylinder.

(1) Mechanical drive operating machinery for miter gates usually consists of a large gear wheel, typically called a bull gear or sector gear, (this EM will use the term sector gear) and a sector arm revolving in a horizontal plane. The sector gear and sector arm are connected to the miter gate leaf by a strut. One end of the strut is connected to the sector arm, and the other end is connected to a pin on the gate leaf. The sector gear usually is driven either by an electric motor located in a recess in the lock wall or by a hydraulically operated cylinder using a toothed rack gear (Figure 4-5). The latter method is used when the locks are subject to flooding due to high river stages.

(2) The principal difference between the three mechanical linkages is the angularity of the connecting strut and sector arm at the extremities of gate travel. The modified Ohio River linkage has angularity between the strut and sector arms at the open position only. The Panama linkage has no angularity at either the open or closed position, and the Ohio River linkage has angularity at both the open and closed positions. The Panama linkage usually is driven by an electric motor. Hydraulic cylinders or electric motors have both been utilized as the driving mechanism with the Ohio River and Modified Ohio River linkages.

b. Panama Canal Linkage. The Panama Canal linkage has been used primarily where electric motor operation was feasible, that is, at locations where high water will not overtop the lock wall. The operating machinery for this linkage generally consists of a high-torque, high-slip, alternating-current motor driving the gate through two enclosed speed reducers, bull gear, sector arm, and spring-type strut.

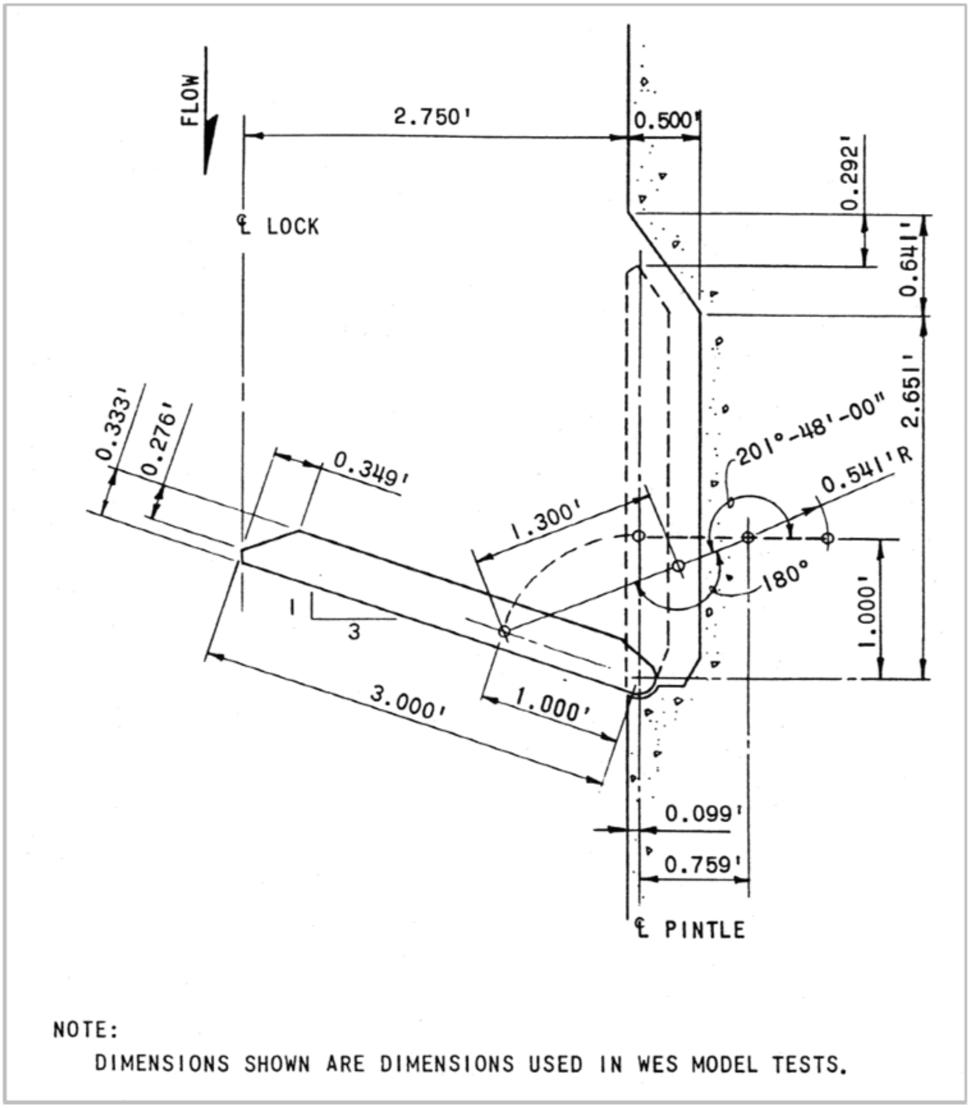


Figure 4-1. Panama Canal linkage (U.S. Army Engineer Waterways Experiment Station 1964)

(1) This linkage will permit the gate to be uniformly accelerated from rest to the midpoint of its travel, than uniformly decelerated through the remainder of its travel, thus eliminating the need for motor speed control. This is accomplished by locating the operating arm and strut on dead center when the gate leaf is in both the open and closed positions. The strut must be at a higher elevation than the sector arm to pass over the arm and become aligned for the dead center position when the gate is fully open. Spe-

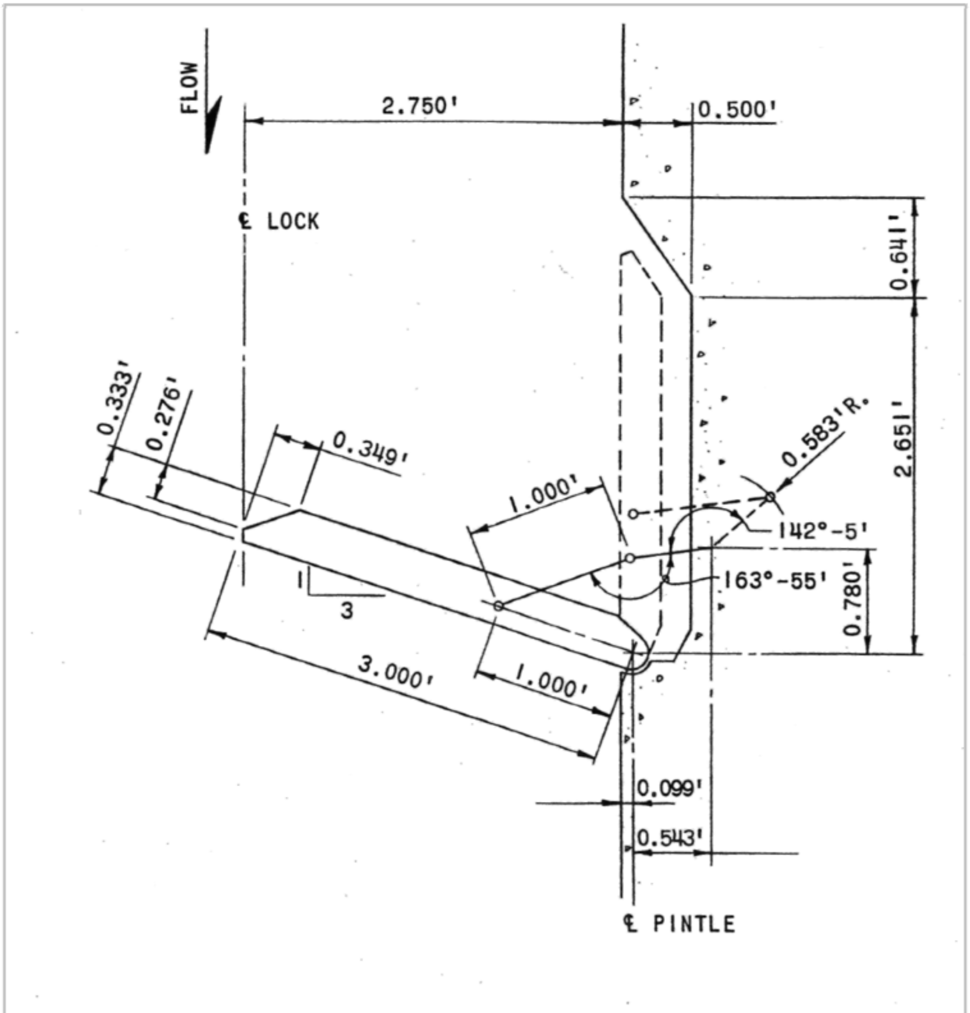
cial consideration must be given to the design of the eccentric connection between the strut and sector arm. This eccentric connection is shown in Figure 4-2 at Dresden Lock on the Illinois Waterway. The strut passes over the top of the sector gear. An assembly layout of the Panama-type linkage is shown in Plate B-13.

(2) The kinematics of the operating cycle are such that the elimination of all angularity between the strut and sector arm reduces the velocity of gate movement near the limits of gate travel for uniform rate of movement (constant travel) of the operating machinery. This in turn reduces the peak loads on the operating machinery. However, this reduction cannot be obtained at each end of the operating cycle unless the sector arm is raised above the sector gear to permit passage over the central axis.

(3) The eccentric connection is one of the primary disadvantages of the Panama Canal linkage and is one reason why the Panama linkage generally is not used anymore for new lock construction. Speed control now can be obtained through multispeed motors or variable-frequency drive systems.

Figure 4-2. Dresden Lock, Illinois River, Panama Canal linkage

c. Ohio River Linkage. The traditional Ohio River linkage consists of a hydraulic cylinder, piston rod, toothed rack meshed with a sector gear, and a sector arm. The spring-type strut is connected to the gate leaf and sector arm (see Figure 4-5). A typical machine is shown in Plate B-15. The traditional Ohio River drive system with a hydraulic cylinder driving a toothed rack gear seldom is used, being replaced primarily by a direct-connected cylinder. The exception is in a lock that is submerged a significant amount of time each year. This is discussed further below.



NOTE:
DIMENSIONS SHOWN ARE DIMENSIONS USED IN WES MODEL TESTS.

Figure 4-3. Ohio River linkage (U.S. Army Engineer Waterways Experiment Station 1964)

(1) The Ohio River linkage also can be a mechanical drive with an electric motor and gear reducer, which drives the sector gear/sector arm assembly. If the lock is prone to flooding, the mechanical drive system can be raised above the lock wall. A strut arm, which usually includes a buffer spring, connects the gate leaf and sector arm.

(2) With the traditional Ohio River drive using a hydraulic cylinder (and the direct-connected cylinder linkages), load analysis for all components is possible. Overloads due to surges or obstructions are carried through the piston and converted to oil pressure, which is released through a relief valve. In this way, all machinery component loads can be determined based on the relief valve setting. The Ohio River linkage offers

several advantages because of its unique geometric configuration relating to the acceleration and deceleration of the miter gates. The disadvantages of this system are wear, bearing forces, and mechanical inefficiencies associated with the geared rack, sector gear, sector arm, and strut.

Figure 4-4. LaGrange Lock, Illinois River, Ohio linkage

(3) For lock sites that are submerged a large amount of time each year and have water levels significantly over the lock, the traditional Ohio linkage offers some advantages. This is the case at Locks 52 and 53 on the Ohio River and several sites on the Illinois River (Waterway). The hydraulic system and cylinder driving the toothed rack gear is sealed, preventing water from getting into the hydraulic system. The cylinder is fairly well protected from debris in the river. The gear rack and sector gear can be quickly cleaned. No electrical motors are on the lock wall. The sector arm and strut arm can absorb impact loading from debris better than a direct-connected cylinder. They are less susceptible to damage than a direct-connected hydraulic cylinder.

Figure 4-5. Lock 52, Ohio River, Ohio linkage with toothed rack gear

d. Modified Ohio Linkage.

(1) The Modified Ohio linkage is similar to the Panama type except that the dead center alignment is attained only when the gate is in the mitered (fully closed) position. With the Modified Ohio linkage, the strut and sector gear are at the same elevation, thus eliminating the eccentric strut connection but preventing the linkage from attaining the dead center position with the gate recessed.

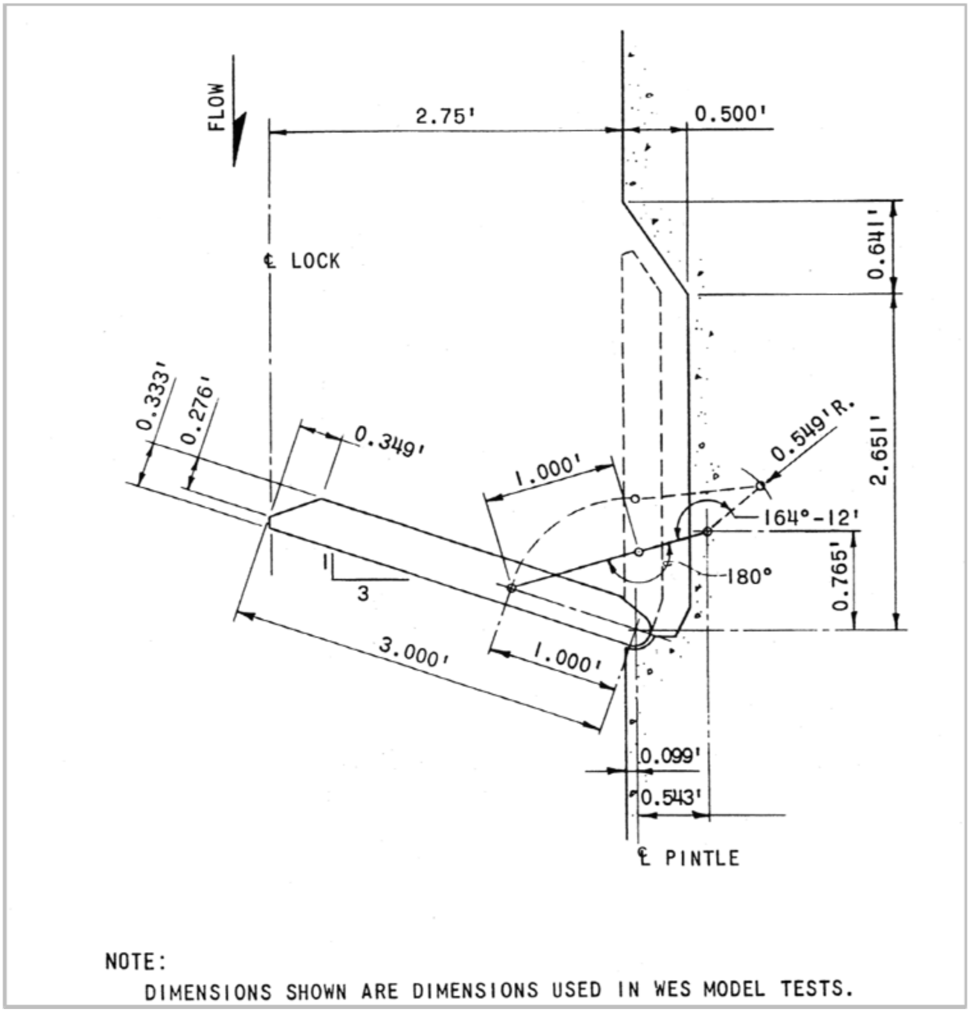


Figure 4-6. Modified Ohio River linkage (U.S. Army Engineer Waterways Experiment Station 1964)

(2) The operating machinery for this linkage has been built either for electric motor drive, as with the Panama linkage, or hydraulic operation, as with the Ohio River machine. The operating machinery also can be raised above the flood elevation, as shown in Figure 4-7. An assembly layout of the Modified Ohio linkage with electric motor drive is shown in Plate B-14. Special consideration should be given to the strut length and/or cylinder stroke, which become critical at the gate-closed position (mitered). Generally, some means of adjusting strut length should be provided to ensure that the gate leaves are fully mitered when the sector and strut arms are fully extended. If the gates do not miter completely at this position, additional travel provided by the cylinder or motor only will pull the gates farther apart. As this linkage approaches the mitered position, the sector arm and strut move near the dead center position. Should an obstruction be encountered at this time, the force in the strut becomes indeterminate. Although this linkage provides restraint against conditions of reverse head in the dead center position, it must be designed with an easily repaired weak link to limit the maximum loads that can be placed on the machinery components.

Figure 4-7. Modified Ohio River linkage with raised machinery

e. Direct-Connected Linkage.

(1) The direct-connected linkage consists of a hydraulic cylinder with its shell (or body) supported in the miter gate machinery recess by a trunnion and cardanic ring assembly (or gimbal) and its rod connected directly to the miter gate with a spherical bearing-type clevis. The layout of the cylinder and cardanic ring minimizes any damage from a barge impact. Its linkage kinematics require that the acceleration of the gate must be controlled using a variable volume pumping unit instead of relying on the mechanical advantage of the linkage. The size of the piston rod is determined by the bending and buckling load criteria. Since the piston rod is used as a strut, it is generally a little larger in diameter than the rod of the Ohio-type machine (with a toothed rack gear). This larger rod also increases the ratio of time of opening to time of closing, since the net effective cylinder volume on the rod end is smaller relative to the volume of the cap end. This variation in opening and closing times can be eliminated easily by using adjustable flow control valves or a regenerative circuit in the hydraulic system.

(2) The arrangement of the direct-connected type machine is shown in Plate B-16. The direct-connected type of machine has been used satisfactorily on both 84-ft-wide and 110-ft-wide locks in the United States and similar size locks in Europe. Experience has shown that the direct-connected cylinder design will have lower initial costs than traditional mechanical drive systems. The direct-connected cylinder linkage is becoming more widely used. Chapter 3 discusses in more detail hydraulic drive systems and hydraulic cylinder design.

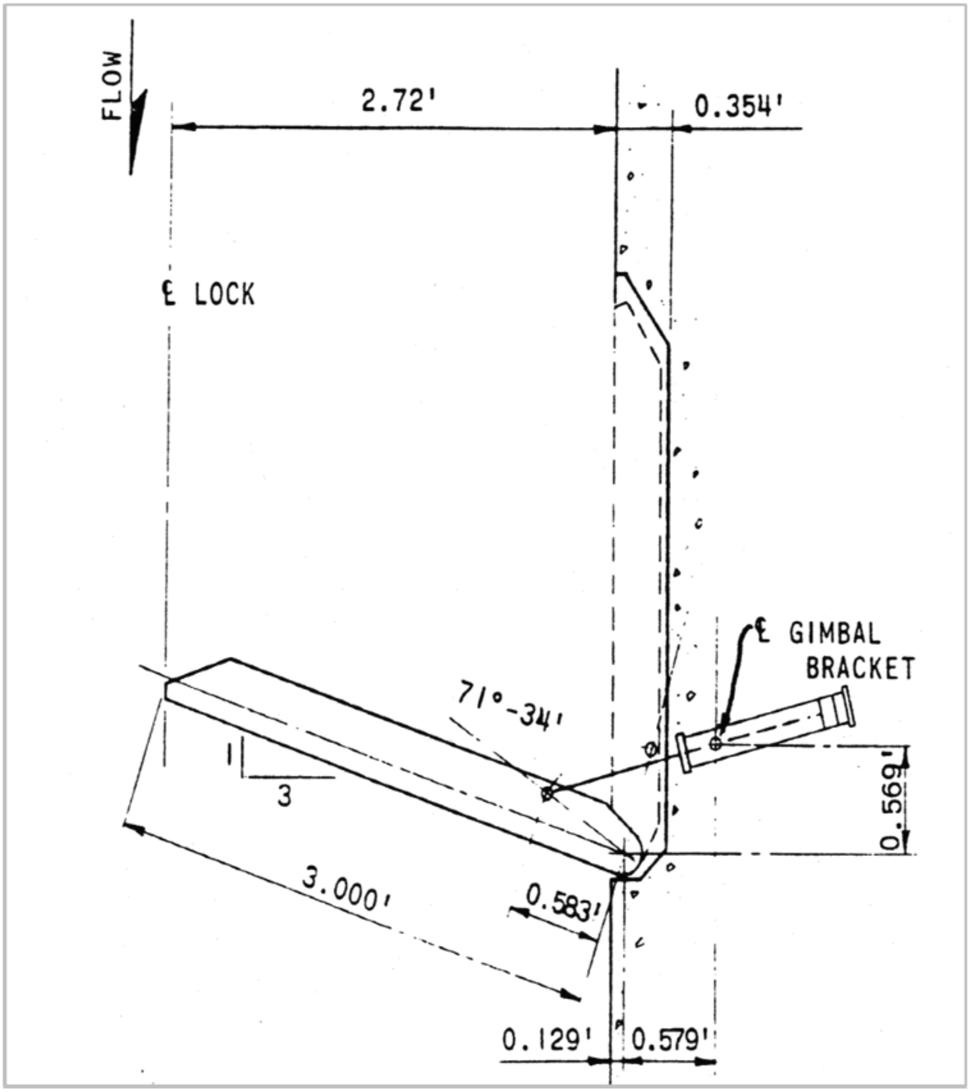


Figure 4-8. Direct-connected linkage Isometric

Figure 4-9. Direct-connected linkage for miter gate

f. Self-Contained Hydraulic Drives. A self-contained hydraulic drive system is similar to the direct-connected cylinder. These types of drives have been used at several Pittsburgh District lock sites on the Allegheny River. The drive combines a hydraulic power unit with a hydraulic cylinder to form a self-contained actuator that is sealed and submersible. The gear motor and the entire drive are designed to be submerged. More details on this system are in Chapter 3.

Figure 4-10. Self-contained actuator – US Army Corps of Engineers Pittsburgh District

(1) The self-contained actuator provides several advantages over the traditional direct connected cylinder:

- Completely self contained, meaning there are no external piping or motors or hydraulic power units, and it is sealed from dirt and moisture;
- No piping friction losses;
- Reduced total space requirements;
- Low maintenance;
- Easily replaceable (plug and play);
- Fully adjustable thrusts and speeds for each direction of travel;
- Smooth, vibration-free operation;
- Weatherproof and can be provided as an explosion-proof unit; can be installed in various mounting configurations.

(2) The primary disadvantage of the self-contained actuator is the lack of manufacturers to fabricate the unit. The units are generally custom built to allow them to be submerged.

g. Recommended Linkage. The final decision for selection of the drive system and linkage should be based on a number of factors including cost, maintainability, and availability of components. The life cycle cost of any drive system should be calculated before a final drive system is selected. When possible, rehabilitation and replacement should be done on a system level basis. Designs for a waterway system should be standardized as much as possible. The direct-connected cylinder design likely will provide the lowest initial cost. This arrangement, when properly designed, is the simplest to maintain, repair, and replace. The direct-connected hydraulic cylinder linkage is common in Europe and is becoming the drive system of choice in the United States both for new lock construction and rehabilitation. Some of the disadvantages of the direct-connected hydraulic cylinder drive system include a lack of simplified methods for position measurement and a lack of manufacturers. Hydraulic cylinders for miter gate drives generally will be custom built and expensive to construct and replace.

(1) Although mechanical drives and linkages have been in service for more than 50 years on many locks, they are becoming less common in new lock construction and rehabilitation of existing locks. One of the advantages of mechanical drives and linkages over hydraulic drives is the proven design and reliability. They are robust and generally will have a longer life span than the direct-connected hydraulic cylinder systems. They can be installed at locks subject to flooding. These systems have been in place since the 1920s and 1930s in the United States and originally were installed at the Panama Canal. Even though mechanical linkages can provide inherent speed control, new mechanical drive systems should be provided with two-speed motors or variable frequency drive systems. The drive system should be operated at a slower speed near the mitered and recessed positions.

(2) Disadvantages of mechanical drives, as compared to hydraulic drives, include a more complex operating machinery linkage. Also, the adjustment available in the strut connection is usually minimal (see h. below). There are more pivot points for wear and additional greasing requirements. Gearboxes will require maintenance and periodic replacement of oil. The mechanical drive system is labor intensive for routine maintenance and for replacement. Components can be difficult to replace and remove. Sufficient availability for spare parts such as gears is no longer assured. Operating components are generally custom built with long replacement lead time (gear boxes and open gears being an example). The alignment can be critical and, if not done properly, the life of the machinery is shortened. The mechanical drives are generally more susceptible to shock load and barge impact (although barge impact also is an issue for direct-connected hydraulic cylinders).

h. Struts. Mechanical drive systems and linkages will require a strut arm at the connection to the miter gate (Figure 4-11). Spring-type miter gate struts commonly are used with the Ohio River and Modified Ohio River linkages. Springs built into the strut assembly act as a shock absorber to soften the loads transmitted to the operating machinery. Two types of struts have been used for the above mechanical linkages. One type utilizes several nests of helical coil springs installed into a cartridge and attached to a wide flange structural steel fabricated member. The springs, when compressed, act as a shock absorber. In the case of electric motor operated machines, the compression in the springs permits the operation of a limit switch to cut off current to the motor when the gates are mitered or recessed. This switch also serves as a torque limit switch to protect the machinery against the possibility of extremely high loads that might occur if an obstruction is encountered when the strut approaches dead center in either direction. The limit switch is set to open the motor circuit at a point immediately preceding the maximum spring compression in the strut. This type of strut is shown in Plate B-17. Another type of strut utilizes a spring cartridge housing and tubular steel strut. Ring springs are used in the spring cartridge to provide the necessary deflection. Excessive maintenance and repair costs have occurred with the use of this type of strut. In addition, ring springs are available only from limited manufacturers. Use of the ring spring-type strut is not recommended. Recently, Belleville springs have been utilized in struts and appear to function satisfactorily. The Belleville spring strut is shown in Plates B-18 and B-19. However, several failures have been reported for the Belleville spring design. This design should consider the extreme loading conditions and necessity for proper lubrication and sealing.

Figure 4-11. Miter gate strut arm

i. Sector Gear Anchorage. Mechanical drives and linkages will require a sector gear or bull gear. The sector gear support and anchorage is one of the more critical items to be considered in the design of miter gate machinery. For proper machine operation and long component life, the sector gear must be maintained in rigid and proper alignment. The recommended arrangement consists of a sector base anchorage, sector base support, and a sector base. The sector base is a heavy steel casting or fabrication and contains the sector pin on which the sector gear turns. The sector gear pin should be restrained to prevent rotation in the sector base. The sector gear can also be supported (roller supports) on its outer perimeter to provide additional support. This is suggested for new installations. An important feature is the bearing choice and lubrication design for the bearings that allow the sector gear to rotate around the pin. The design is such that the final post-tension rod force is enough to resist the horizontal sector pin load by friction between the concrete and sector base support. In addition, compression blocks are welded to the bottom of the sector base support to provide additional resistance to horizontal motion. Details of this anchorage are shown in Plate B-20.

4-2. <u>Design Criteria</u>.

a. Design Loads.

(1) Miter gates are not suitable for operation under other than essentially balanced conditions or equal head on both sides of the gate. The lock chamber must be filled or emptied by means of a culvert system prior to operation of the gate leaves. Thus, the water level on each side of the gate is equalized, or almost equalized, before movement of the gate leaves is undertaken. The forces to be overcome by the gate-operating machinery are friction, wind loads, surges, hydraulic drag forces, and head differential created by the gate leaves moving through the water.

(2) Friction and wind loads can only be estimated, but are generally small in comparison with the hydraulic loads. It would appear that evaluation of operating forces caused by hydraulic drag and head differential could be made through tests on existing lock gates. In practice, however, it has been found difficult to make accurate load measurements in the field. Furthermore, it is not practical to vary field operating conditions such as gate speed and gate submergence (height of water on gate).

b. Normal Loads. Gate operating machinery normally should be designed to conform to the criteria in this section. Operating loads on the miter gate machinery should be derived by hydraulic similarity from test data obtained from model studies. The model study available for design is described in U.S. Army Corps of Engineers Waterways Experiment Station Technical Report 2-651, Operating Forces on Miter Type Lock Gates (WES 1964). An analytical means to calculate gate operating forces is provided in U.S. Army Corps of Engineers Waterways Experiment Station Technical Report 74-11, Study of the Forces Occurring During the Movement of Miter Gates of Locks (WES 1974). The WES 2-651 (1964) report includes test data on the Ohio River, Modified Ohio River, and Panama Canal linkages. This report contains necessary data for conversion to prototype torque for all three of the different types of linkages. This report is summarized as follows:

> “Tests to determine operating forces on miter-type lock gates were conducted in a 5.5-ft-wide, 66.5-ft-long, 4.25-ft-deep flume equipped with a single set of miter gate leaves located approximately in the center of the flume. Three linkages, with different kinematics of the operating machinery, were studied: modified Ohio River, Panama, and Ohio River. For each linkage, tests were conducted at gate submergences of 1 to 4 ft and at operating times of 10.1 to 40.2 sec. The effects of chamber length, bottom clearance of gates, presence of barges in the lock chamber, and non-synchronous operation of the gate leaves also were investigated. Peak hydraulic resistance to operation of the miter gate was observed as the leaves entered and left the mitered (closed) position with the maximum resistance occurring as the leaves entered the mitered position. Peak torques were actually observed as the leaves left the recesses (began closing) with the Ohio River and modified Ohio River linkages, but these torques were created by sudden application of loads to the rigid model linkages and, thus, were not considered representative of those of prototype gates that are equipped with shock absorbers. The modified Ohio River and Panama linkages resulted in peak resistances in terms of torque at

the pintles, which were approximately equal and about 40% less than the peak torques obtained with the Ohio River linkage."

c. Direct-Connected Cylinder Loads. For direct-connected hydraulic cylinders, prototype tests were made at Claiborne Locks. Results of these tests are included herein and in Appendix C for the determination of gate torque for any proposed direct-connected lock machine of similar proportions. A curve of gate torque plotted against percentage of gate closure has been included in Appendix C so that torque at any submergence or time of operation can be computed by application of Froude's law, adjusting the submergence and time to suit the new conditions.

d. Temporal Loads. In addition to the above-normal loads, the miter gate machinery should be designed to withstand the forces produced by a 0.38-m (1.25 ft) and exceeding 30-sec duration surge load acting on the submerged portion of the miter gate. For this case, the machinery must be designed to maintain control over the miter gate when the gate is in the miter position. In the recess position, the gate can be controlled by automatically latching the gate in the recess. Normal machinery operating loads govern the machinery design for the intermediate positions.

e. Operating Time. A time of operation should be selected and based on the size of gate. For smaller gates (25.6-m or 84-ft lock), an average time of 90 sec should be used. For the larger gates (33.5-m or 110-ft locks), an average time of 120 sec should be used. Any decision to increase the operating time from 1.5 to 2 min for smaller gates or 2 to 3 min for larger gates should be made only after considering the economic impact of the increased time required for navigation traffic and barges to transit the lock. Mechanical drive systems utilizing two-speed motors should be operated at slow speed when approaching miter or recess. This needs to be considered in the overall operating time.

f. Submergence. The design of the gate operating machinery should be based on the submergence of the upper or lower gate, whichever is greater. The design should be the same for all four gate machines because there would be no savings in designing and building two different size machines. The increased design cost would offset the reduced cost of the material used in constructing the smaller machine. The submergence of the gate is the difference in elevation of the tailwater on the gate and the elevation of the bottom of the lower seal protruding below the gate. A submergence selected for design of the gate machinery should be the tailwater on the gate that would not be exceeded more than 15 to 20% of the time.

g. Direct-Connected Cylinder Force. For direct-connected hydraulic cylinders, the operating cylinder size should be selected to provide a force to operate the gate utilizing approximately 6-20 MPa (900-3000 psi) effective pressure where a central pumping system is used. Where local or integral pumping units are used, an operating pressure of 10-20 MPa (1500-3000 psi) will be satisfactory. The time of gate operation automatically will be lengthened when the required gate torque exceeds the available gate torque from the machinery. This condition might occur during starting peaks or periods of higher submergence. This condition causes the pressure in the hydraulic cylinder to rise above the relief valve setting, which in turn reduces oil flow to the cylinder, slowing down the gate and reducing the required pintle torque. This increases the total time of operation; however, this slower operation will generally be experienced for only 15 to 20% of the lock's total yearly operating time.

h. Non-Synchronous Gate Operation. Peak torque can be reduced by non-synchronous operation of the gate leaves. A considerable reduction in peak torque can be obtained by having one leaf lead the other by approximately 12.5% of the operating time. The time of opening would be increased by the amount of time one gate leads the other. It has been found that, in actual practice, very few gates are operated in this manner.

i. Under-Gate Clearance. Model tests revealed an increase in gate torque values as the bottom clearance decreased, regardless of the length of operating time. When model similarity is used to compute gate loads, an adjustment should be made in accordance with model experience. Normally, 2.5-ft to 3.5-ft clearance under the gate should be satisfactory.

j. Machinery Components and Factor of Safety. General design criteria applicable to the various machine components are presented in Chapter 2. Allowable stresses may be increased by one-third for temporal loading conditions.

4-3. Load Analysis.

a. Normal Loads. Normal operating hydraulic loads on miter gates primarily are caused by submergence, speed of gate, and clearance under gate. For additional information and explanation, the designer should review WES (1964) Report 2-651. Some general observations from the report:

- An increase in submergence of the gate leaves or speed of operation resulted in increased hydraulic resistance.

- Hydraulic resistance increased as the bottom clearance of the gate leaves was decreased.

- Hydraulic resistance decreased as the length of the lock chamber was increased.

- Non-synchronous operation of the gate leaves resulted in a slight reduction in peak torque.

- Limited tests conducted with barges in the lock chamber showed no appreciable effect on torque values.

b. Ohio Linkage and Torque. WES (1964) indicates that the maximum torque recorded as the gate leaves entered the mitered position (closing) varied as the 1.5

power of the submergence; and the maximum torque recorded as the gate leaves left the mitered position (opening) varied as the 2.1 power of the submergence.

c. Modified Ohio Linkage and Torque. WES (1964) indicates that the maximum torque recorded as the gate leaves entered the mitered position (closing) varied as the 1.9 power of the submergence; and the maximum torque recorded as the gate leaves left the mitered position (opening) varied as the 2.2 power of the submergence.

d. Panama-Type Linkage and Torque. WES (1964) indicates that the maximum torque recorded as the gate leaves entered the mitered position (closing) varied as the 1.5 power of the submergence; and the maximum torque recorded as the leaves left the mitered position (opening) varied as the 1.7 power of the submergence.

e. Ohio Linkage and Operating Time. For the Ohio Linkage, the report indicates that the maximum torque recorded decreased as the 1.0 power of the operating time for both the closing and opening cycles. In other words, the required torque is directly proportional to the operating time.

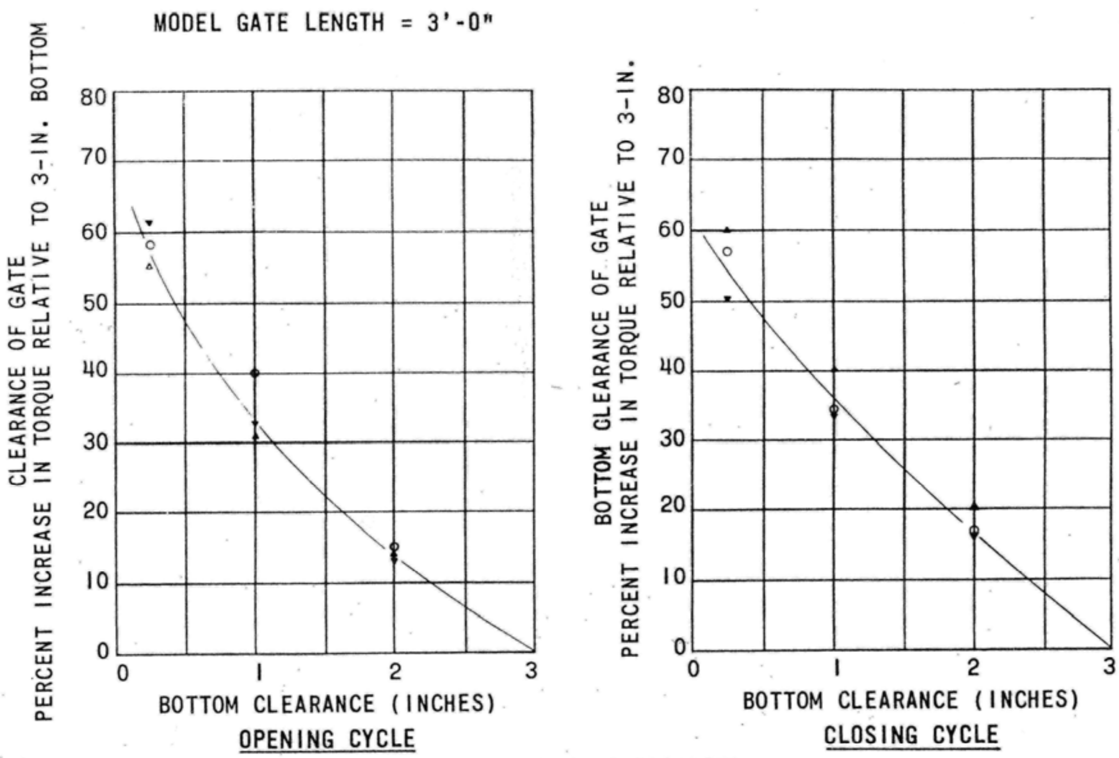
f. Modified Ohio Linkage and Operating Time. For the Modified Ohio linkage, the report indicates that the maximum torque recorded decreased as the 1.1 power of the operating time for the closing cycle and as the 1.5 power for the opening cycle.

g. Panama Linkage and Operating Time. For the Panama linkage, the report indicates that the torque decreased as the 1.1 power of the operating time for the closing cycle and as the 1.3 power for the opening cycle.

h. Torque and Under-Gate Clearance. Tests reveal that gate torque increases when the clearance under the gate leaf is decreased, regardless of the length of operating time. Data from these tests, presented in Figure 4-12, indicate the percentage increase in model torque for various bottom clearances relative to the torque observed with a 3-inch bottom clearance. These data can be used to adjust the observed torque values determined for a model bottom clearance of 3 inches when the gate length is 3 ft.

i. Non-Synchronous Operation of Miter Gates. Non-synchronous operation of miter gates results in slightly lower forces on the leading leaf. Forces on the lagging gate leaf are greater during most of the closing cycle and less during the opening cycle than similar forces recorded for synchronous operation of the gate leaves. The greatest reduction in torque appears to occur when one gate is leading the other by approximately 12.5% of the total operating time.

j. Barges. Barges in the lock chambers are found to have negligible effect on gate operating forces.



OPERATING TIME SEC.	TORQUE, LB-FT (3-IN. BOTTOM CLEARANCE) OPENING CYCLE	CLOSING CYCLE	SYMBOL
13.4	40	35	○
20.1	29	20	△
26.8	18	12	▼

Figure 4-12. Relative effect of gate bottom clearance on torque, 4.0-ft submergence

k. Chamber Length. The chamber length affects the gate torque in that the longer the chamber, the less the torque. As the length of time is increased, the less the chamber length affects the gate torque. Insufficient data are available to set up any definite adjustment factors for correcting for chamber length.

l. Pintle Friction. Torque caused by gate pintle friction is generally of small magnitude. The WES Report 2-651 (1964) incorporates pintle torque in the model tests. These are based on traditional grease-lubricated pintle bearings. The model tests do not account for any binding or seizing of the pintle bearing.

m. Direct-Connected Cylinder Torque.

(1) When operating torque for a direct-connected, hydraulic cylinder-type miter gate drive is being computed, the curves shown in Appendix C can be used. The curves are results of prototype tests made on Claiborne Lock and show gate torque plotted against percentage closed. The torque from these curves can be adjusted to suit new conditions by the application of Froude's law. Because the curves were based on the use of a three-speed pump to slow the gate travel at beginning and end of cycle, it will

be necessary to make similar assumptions on the proposed lock. Assuming a fast delivery rate of the pump at 1.0, the medium delivery rate should be 0.8 and the slow rate adjusted to 0.3 of the fast rate. A normal cycle would be to operate 10% of the gate angular travel at 0.3 capacity, 10% at 0.8 capacity, 60% at 1.0 capacity, 10% at 0.8 capacity, and 10% at 0.3 capacity. A study comparing this type of operation and the Panama-type linkage indicates that the direct-connected machine, if operated as stated above, will compare favorably with the Panama machine in angular gate velocity (degrees per second) at all positions. Assuming that the angular velocities compare with the Panama-type machine, the maximum torque will vary as the 1.5 power of the submergence (closing) and 1.7 power of the submergence (opening). The operating time should vary as the 1.1 power for closing and the 1.3 power for the opening cycle.

(2) U.S. Army Engineer District, Huntington, uses another method of sizing direct-connected miter gate machinery. They compute opening the gate against a 0.15-m (6-in.) differential head, and this usually ends up being the governing case over the torques computed using model forces.

n. Temporal Loads. Temporal hydraulic loads or surges are temporary changes in water level resulting in a differential water level on opposite sides of a lock gate. These surges or differential heads might be caused by overtravel of water in the valve culvert during filling or emptying, wind waves, ship waves, or propeller wash, for example. Depending on the circumstances, this differential has been observed to vary from 0.3 to 0.6 m (1 to 2 ft). These forces do not affect the machinery power requirements, but they do affect the design of the gate machine components when the gate is at the recess or mitered position. These forces have been known to fracture gate struts and shear sector pins.

o. Miter Gate Loading Discussion from WES (1964). The peak hydraulic resistance to operation of the miter gate occurs as the leaves enter the mitered position. This suggests that head differential on the two sides of a gate leaf is the primary cause of loads on the operating machinery. If drag forces were the predominant influence, it would be expected that the peak load would occur simultaneously with the maximum rate of angular movement. This happened for all linkages when the gate was about 45% open. Also, if inertial forces were major factors, the torque should have been greatest as the leaves moved away from the mitered position and were accelerating, while the reverse was found to be true.

4-4. Determination of machinery loads.

a. Normal Loads.

(1) Normal miter gate operating machinery loads are difficult to determine and should, whenever possible, be determined from model or prototype tests. Data compiled by the Special Engineering Division of the Panama Canal Zone, taken from tests made on the existing locks and a model for the third locks and model studies included in WES Technical Report 2-651, 1964, appear to be the most reliable sources for obtaining miter gate machinery loads. WES Report 74-11 provides an analytical means of calculat-

ing forces. When using data from the model tests, it will be necessary to adjust the data on the basis of the scalar ratio between the model and the proposed lock. The length of the gate leaf normally is used for determining the scalar ratio. From the scalar ratio, Froude's law comparing prototype to model would be:

$$\text{Scalar ratio} = \frac{\text{length of prototype leaf}}{\text{length of model leaf}} = L_R$$

Volume, weight, and force = $(L_R)^3$:1

Time and velocity = Square Root of L_R:1 or $(L_R)^{1/2}$: 1

Torque = $(L_R)^4$:1

(2) When machines having the Ohio linkage, the Modified Ohio linkage, or the Panama-type linkage are used, the forces on any size miter gate can be obtained from curves shown in Appendix C, which are plotted from the results of the WES and Panama Canal Model Tests. Readings from the curves must be factored according to Froude's law for submergence, time of operation, and clearance under gate. Curves are based on lock chamber lengths of 183 meters (600 ft) or greater. Forces for shorter lock chambers would be slightly greater. This should be considered when replacing the miter gate machinery on 17.1-meter wide (56-ft) locks, which usually have 109.8-meter (360-ft) chamber lengths.

b. Computation of Pintle Torque for Panama Canal and Ohio-Type Linkage.

(1) If the proposed lock gate is in the same scalar ratio, with respect to length of gate, and the submergence and time of operation, as shown on curves, and the type of linkage are the same, the pintle torque would equal the pintle torque at each position indicated on the curves multiplied by the ratio of gate leaf lengths to the fourth power.

$$P_1 = P(L_1/L)^4 \tag{4-1}$$

where:

P_1 = pintle torque of proposed lock gate at selected position

P = pintle torque shown on curve of model study at selected position

L_1 = leaf length, pintle to miter end, proposed lock gates

L = leaf length, pintle to miter end for curves that have been plotted on model study

(2) In the event the ratios of gate lengths (L_1/L), submergence (S_1/S), and the square of the time of operation (T_1/T) are not of the same scalar ratio, the formula should be expanded:

$P_1 = P(L_1/L)^4 \; (S_1/S_2)^x (T_2/T_1)^y$ (4-2)

where:

P_1, P, L_1 and L = same as in Equation 4-1

S_1 = submergence of proposed lock gate

S = actual submergence of model gate upon which curves are based

S_2 = adjusted submergence of model lock gate = $S(L_1/L)$

T_1 = time of operation of proposed lock gate (See arc of travel adjustment in equation 4-3)

T = actual time of operation of model gate upon which curves are based

T_2 = adjusted time of operation of model lock gate = $T\sqrt{L_1/L}$

x = power to which submergence must be raised, for particular type linkage

y = power to which time must be raised, for particular type linkage

NOTE: If only one ratio for either submergence or the square of the operating time is not of the same ratio as gate leaf length (L_1/L), then only the ratio not in agreement with L_1/L need be considered in the equation.

(3) If the arcs of gate travel differ from that shown on model curves, it will be necessary to adjust the operating time of the proposed lock (T_1) to use in Equation 4-2:

Let T_A = adjusted operating time, then:

$$T_A = T_1 \frac{\text{(arc of travel, proposed lock)}}{\text{(arc of travel, on model curves)}} \qquad 4\text{-}3$$
$$= T_1(K_1/K)$$

NOTE: T_A must be substituted for T_1.

c. Motor Slip. Use of Equations 4-1, 4-2, and 4-3 results in a pintle torque, which makes no allowance for motor slip since all the model curves were based on uniform speed of hydraulic cylinder or constant rpm of the motor. If a portion of the required gate torque curve overloads the motor, the resulting time of gate operation would be slower, which in turn would result in lower gate torque during this period. The same would occur when operating the gates with a hydraulic cylinder. Overloading the cylinder would result in some of the oil being bypassed through relief valves, which in turn would slow the gate during the overload period. When Ohio-type linkages and torque data from Technical Report 2-651 WES (1964), are used, the pintle torque should be adjusted for under-gate clearance in addition to submergence and time. The percentage increase can be obtained from curves in Figure 4-12.

d. Electric Motor Design. Electric motor operation with Panama-type, Ohio-type, or Modified-Ohio-type linkages should be designed for several factors. Motors and gearboxes should be elevated above flood levels to the extent possible. A high-torque, high-slip motor should be used and selected so that the normal full load torque available would not be exceeded by the required torque of the machine more than 15 to 20% of the time. Motors should be two-speed to allow for slower operation at the miter and recess positions. Peak torque during the overload period should not exceed 150% of full load torque. This can be determined by plotting the required torque based on curves computed from model tests described above and by plotting available motor torque curves at various degrees of slip and superimposing these curves over the required curves. Typical calculations for determining loads using the Ohio-type linkage (hydraulic operation) are shown in Appendix C. Calculations for determining loads using the Panama Canal-type linkage (electric motor operation) for the same design conditions are also shown in Appendix C.

e. Computation of Pintle Torque for Direct Connected Linkages. The kinematics of this type of machine should be developed to provide the shortest practicable piston stroke. This will require the gate pin connection to be located out from the pintle a distance of 20 to 25% of the gate length, and the center line of the cylinder gimbal bracket to be located to give the best effective operating arm about the pintle at each position throughout the entire stroke of the piston. With use of this linkage and a uniform traveling piston, gate angular velocity will be greatest at the extreme closed or open position of the gate. Uniform travel of the piston is, therefore, undesirable, and it will be necessary to slow the speed of the piston near the closed and open positions by use of a variable volume pump in the oil circuit. By slowing the travel near the open or closed position of the gate, angular travel rates will be comparable with the Panama Canal linkage. Figure 4-13 shows comparison curves for angular velocity of gate plotted against percent closed for Panama Canal Third Locks linkage and for Claiborne Lock direct-connected linkage with and without variable speed control. Time of operation should be selected for the proposed lock that will give angular gate velocities approximately equal to the velocities shown on the curve for Panama Canal. Gate pintle torque then should be taken from the prototype curves shown in Appendix C, and by means of Froude's Law of Similarity to the submergence and time requirements of the proposed lock using the same exponents as for the Panama Canal linkage. Load computations for a direct-connected machine are also shown in Appendix C.

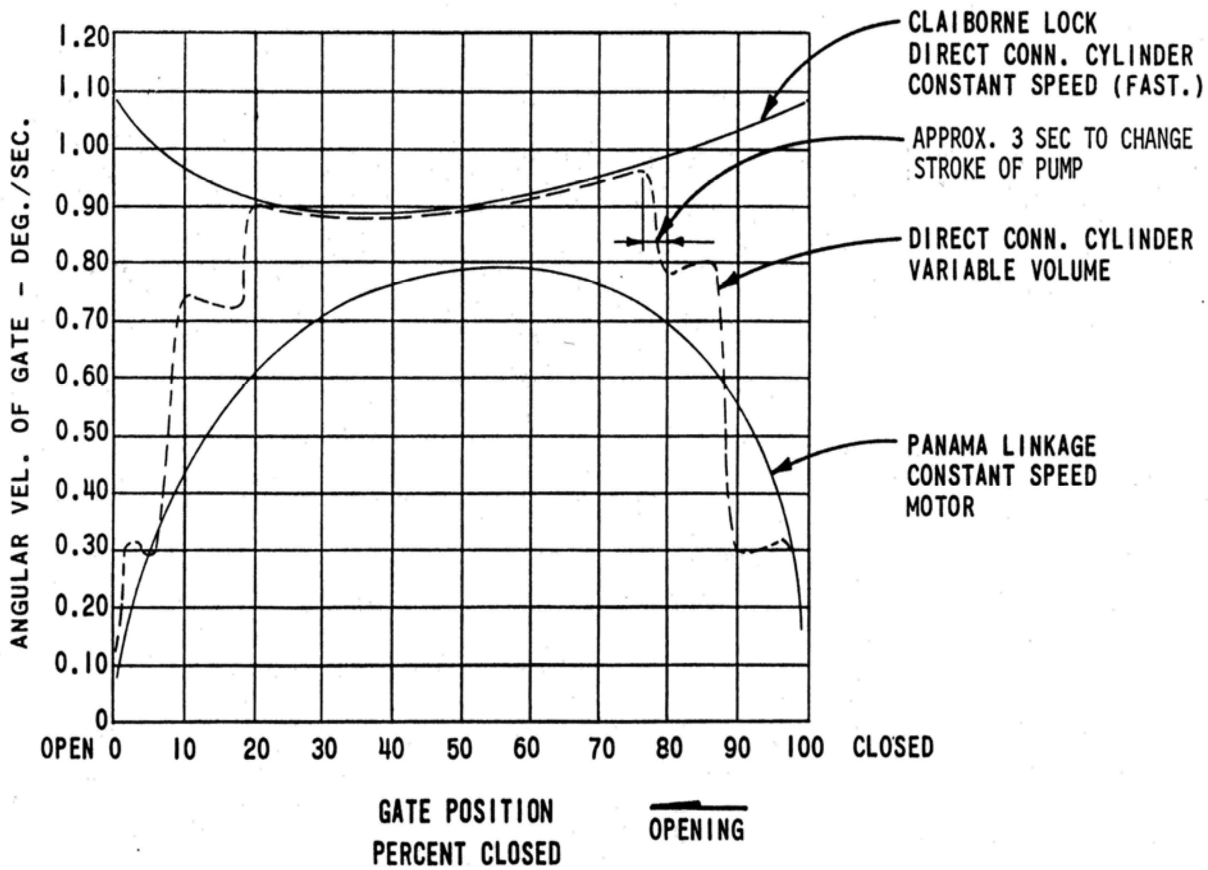


Figure 4-13. Gate velocity comparison curves

f. Temporal loads. The resulting machinery loads for the case of temporal loading are based on a 1.25-ft differential head superimposed on the normal gate submergence. These loads are considered applicable only when the gate is at either the miter or recess position. These forces are resisted by a load brake for mechanical drives or a high-pressure relief valve for hydraulic drives. For this load condition, a 33.33% overstress is allowed for component design. In the recess position, this load is resisted by automatically latching the gate. Only the sample computations for the Ohio River-type machine shown in Appendix C include the temporal load computations.

4-5. Miter Gate Operating Machinery Control.

a. Hydraulically Operated Machines. A complete description of the basic types of hydraulic systems for locks, along with pertinent hydraulic system design criteria, is presented below and in Chapter 3. Control of these systems has utilized manual, solenoid-controlled, pilot-operated, and cartridge valves. Limit switches can be incorporated into the hydraulic cylinder or installed external to the hydraulic cylinder (see Chapter 3).

(1) With manual control, a small control stand typically is located over a recess on one lock wall near the gate machinery and is equipped with control valve operating levers. A schematic piping diagram of a manually controlled central pumping system is shown in Plate B-9. This diagram includes the connections for the tainter valves and shows the complete lock operating hydraulic system.

(2) Recent control systems utilize solenoid-controlled, pilot-operated four-way and solenoid-controlled cartridge valves to control the flow of oil to cylinders. This makes the system more flexible and enables the inclusion of an electrical interlock between the miter gates and lock fill-and-empty valves so that the lock chamber water level cannot be changed before all gates are closed. Changing the water level in the lock chamber before the gates are closed creates a swell head on the partially closed gates, which could cause them to slam shut, damaging the gate and/or gate machinery. This type of control is recommended rather than the manual control. A schematic piping diagram of this control system is shown in Plate B-10.

b. Lock Control Schemes. Chapter 12 discusses further the lock control systems.

(1) Gate and valve control consoles typically are located in the control building near the upstream gate. So the operators can view the downstream gate during opening and mitering, a multi-camera, closed-circuit television system should be provided. A simplified control stand also can be provided near the downstream gate for the operation of the lower miter gates.

(2) The control system should provide two speeds for miter gates and two speeds for culvert valves. Electrical interlocks are mandatory and should be used in the control circuit to prevent inadvertent gate operation. The Inland Marine Transportation System (IMTS) Working Group has developed a Draft Interlock Standard (IMTS September 2011) that should be referred to in developing proper interlock sequences. Multiple interlock diagrams are provided in this document.

(3) Limit switches located at the miter point of the gates and in the gate machinery recesses and the culvert valve recesses are used to prevent the upstream culvert valve from being opened when the downstream gate and/or valves are open and vice versa. A miter gate position limit switch is shown in Figure 4-14. Interlocks also are used to prevent the gates from slamming or the lock chamber water level from changing when gates are mitered improperly. One miter limit switch should be located near the top of the gate. Miter gate limit switches installed on the bottom of the gate have been problematic. Switches on the bottom of the gate are highly susceptible to being damaged by the large amount of debris near and around the gates. If one of the switches is damaged, it will need to be replaced by a dive team. This is expensive, adds risk to the maintenance program, and could cause a lengthy lock outage. If the gates fail to seal along the gate sill, it usually will be obvious to the operator because a boil will be seen below the gate. If required, the operator can reverse the chamber and look into the problem at that time. Miter gate limit switch locations are shown in Plates B-89 and B-90. Since the bottom seal resistance of the gate will prevent the lower portion of the gate from closing properly, even though the top is mitered, only the top miter limit switch

and the rack-mounted, gate-mitered limit switches must be actuated before the corresponding filling or emptying sequence can be started. A logic diagram for this system is provided as a reference only and is shown in Figure 4-15. Again, the latest version of the IMTS Interlock Standards should be utilized. A manual backup system should be provided for gate and valve control, should the automatic control system fail. The manual control system should be independent of the automatic control system and bypasses all gate-valve interlocks.

Figure 4-14. Miter gate position limit switch

c. Miter Gate Control Equipment.

(1) The electrical control systems utilize either electro-mechanical relays or solid state controllers or programmable logic controllers (PLCs). This is discussed in Chapter 12. Control equipment typically consists of full voltage magnetic controllers, limit switches, and control switches. Strut limit switches are used to cut off the motor if the strut stresses in either tension or compression are beyond a preset point. This will protect the strut and machinery if an obstruction is encountered.

(2) The limit switches used in previous designs were of the traveling-nut type in NEMA 4 enclosures with heaters. Cam-operated switches now are more widely used and available and can be incorporated into the machinery drive. Limit switches also can be utilized on the miter gate to provide an additional safety cutoff and prevent over travel of the miter gates. Electrical valve-gate interlock features are mandatory and should be similar to that described above.

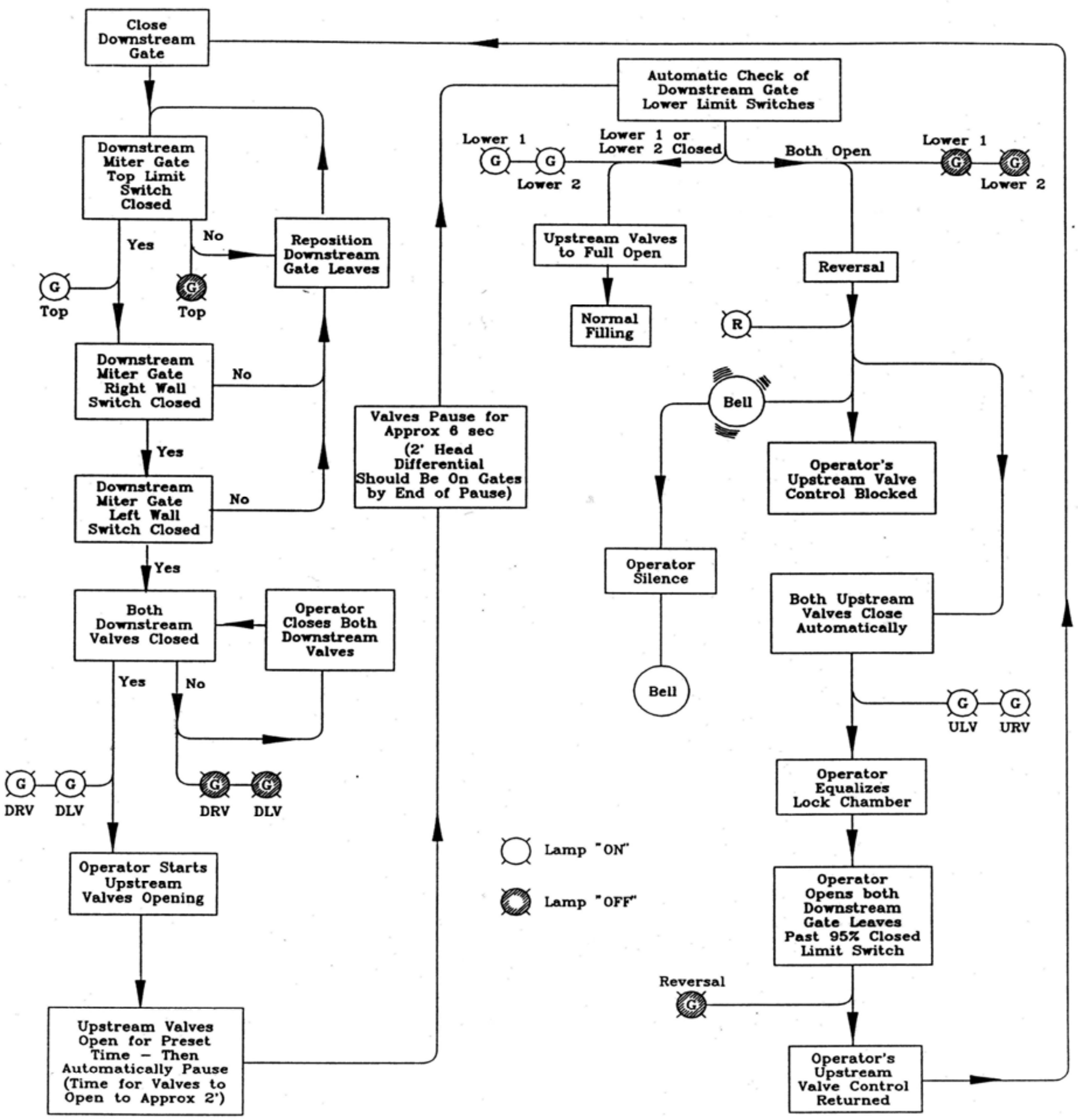


Figure 4-15. Lock filling sequence (lock emptying sequence is similar)

4-6. Miscellaneous Equipment and Systems.

a. Machinery Stops. In order to deal with ordinary construction tolerances, a means must be provided to adjust the miter gate machinery linkage at installation. For direct-connected cylinders, it is usually desirable to provide approximately 50 mm (2 in.) of overtravel at each end of the hydraulic cylinder to allow for adjustment. With the linkage connected and the miter and recess positions established for the gate, stops are installed and adjusted to limit the machinery motion to these extreme positions. For Ohio River-type machines that are operated by hydraulic cylinders, one stop is placed to stop the rack when the gate is mitered; another is placed to stop the sector arm when the gate is recessed. Details of this arrangement are shown in Plate B-7. For mechanical linkages with a sector arm and strut arm, the only adjustment available is in the strut arm and springs. The amount of adjustment, however, in the strut arm springs is typically small or negligible.

b. Lubrication.

(1) A system should be provided to grease each miter gate pintle bushing and gudgeon pin as shown in Figure 4-16. The system may be automatic or manual, depending primarily on lock operator preference. During gate movement, the automatic system should dispense a measured amount of grease to each location automatically. An automatic grease system is available with a built-in programmable controller, which will allow variations in grease cycles and quantities provided. Since the grease systems have to be field-tuned for a particular lock application, the programmable controller should be provided. The pintle bushing should be designed to permit the installation of an O-ring seal and a grease return line that can be monitored to ensure grease delivery to the pintle bushing. Special consideration should be given to the layout and sizing of the grease lines to ensure proper operation and minimum pressure loss. Grease lines should be stainless steel pipe of adequate wall thickness for the anticipated pressures. The lines should be located in areas of the gate that afford the greatest degree of protection from damage due to ice and drift. This is typically in the quoin structure of the gate. The pumping unit should be located near the gate to minimize grease line length. Provisions should be made to remove the pumping unit if flooding is likely.

(2) For pintle lubrication details, see Plates B-23 and B-24. Self-lubricated bushings can provide an alternative to greased bushings for the pintle and gudgeon pin, thus eliminating the need for greasing. The U.S. Army Engineer Construction Engineering Research Laboratory (CERL) evaluated field performance and conducted laboratory tests of commercially available self-lubricating materials used in lock and dam applications. Section 4-7 provides additional information on the use of self-lubricating pintle bushings.

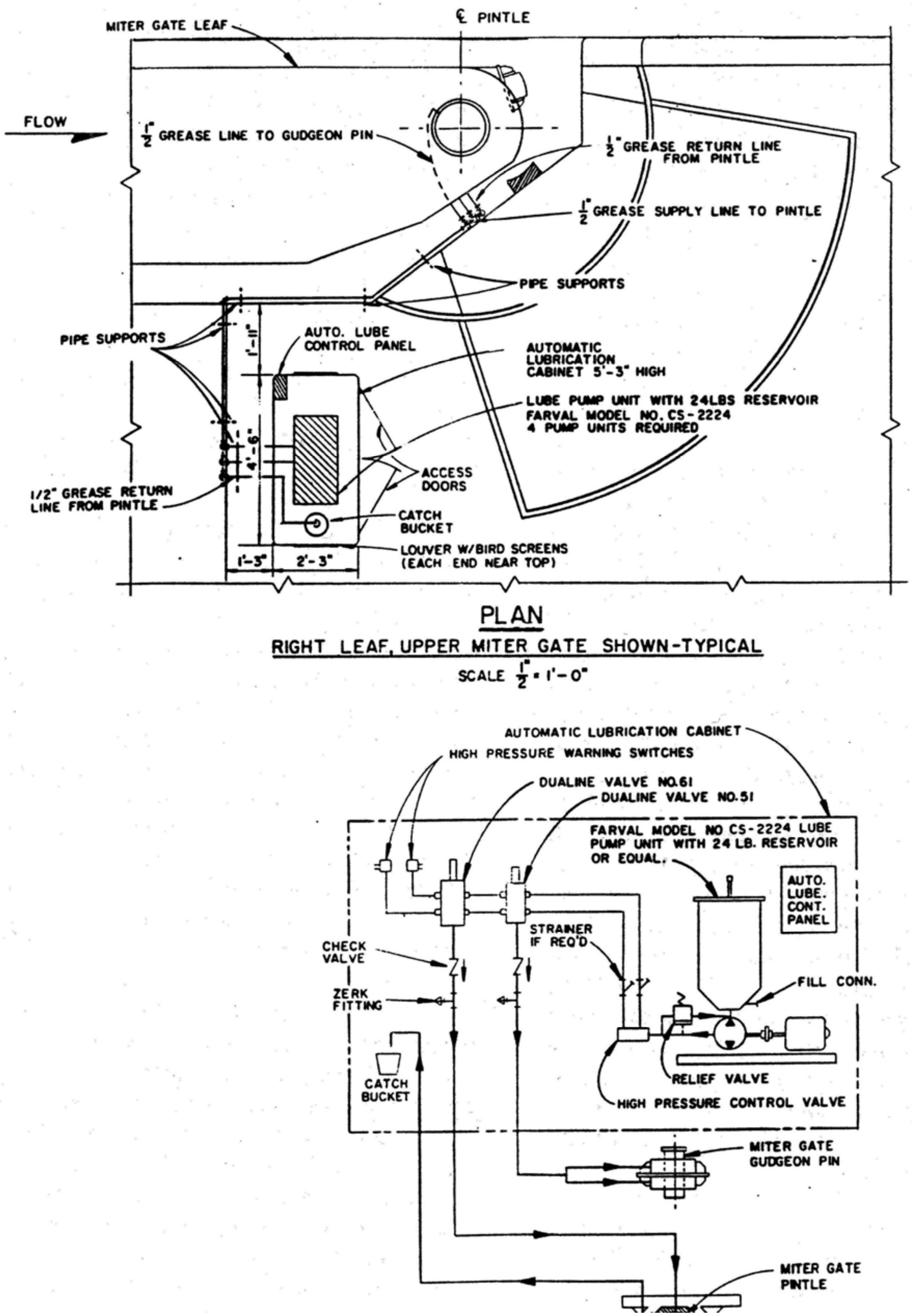


Figure 4-16. Automatic lubrication system schematic

c. Automatic Gate Latches. For miter gate drives with direct-connected cylinders, latches should be provided for holding the gates in the recess. If this is not done, the hydraulic cylinder pressure has to be adequate for holding the gate in recess. The latches should be designed to latch the gate automatically when it comes into the recess. The latches should be released automatically each time a gate-close function is initiated. The system should be provided with latched and unlatched position indication. Miter gate drives with mechanical linkages typically will not require a gate latch. This should be evaluated, however. During flood conditions and when passing ice through the lock, miter gates should be tied back manually to prevent the gate from moving. This is the case for both mechanical linkages and direct-connected cylinders.

d. Maintain-Pressure System for Direct-Connected Cylinders.

(1) A maintain-pressure system should be provided to hold miter gates closed with hydraulic pressure. Plates B-84 to B-88 show a system designed to hold the gate leaves together against wind loading or small water surges prior to changing the chamber water level. Upstream gate pressure is maintained during lock chamber emptying, and downstream gate pressure is maintained during lock chamber filling operation. The maintain-pressure system is activated by the lock operator's depressing a pushbutton on an operator console. This system can be deactivated manually by the operator or is deactivated automatically when the gate under maintain pressure is opened or after the valves are opened for a predetermined time to allow an adequate head of water on the gates to keep them mitered. The maintain-pressure system should utilize the slow valve or the lowest pumping rate available. Lock 19 on the Upper Mississippi River incorporated this system in 2006. The intent was to keep an amount of hydraulic pressure on the cap end of the cylinder to continue to push the gates together while the gates are in miter. It also worked by reapplying the pressure by turning on both miter-gate pumps, to push the gates together (add pressure because the gates are already in miter). Repressurization was prevented from automatically occurring when:

- Both riverwall tainter valves are operating, and the normal miter gate pump is in use.

- Any emergency operation function is enabled and the available pump is being used to drive a tainter valve, then initiation of repressurization shall wait until all equipment (tainter valve) motion stops.

(2) The tandem center hydraulic system is not preferred but, if used or if a tandem center system is refitted, the maintain-pressure system will provide pressure to the miter gate cylinder in the gate closed position through the use of a standard bladder-type accumulator. This accumulator, located in each miter gate machinery recess, will be charged and pressure maintained through a pilot-operated check valve installed in series with each miter gate cylinder. A pressure switch, sensing accumulator pressure, will ensure adequate pressure through a time-delay circuit. An indicator lamp on the control console will be illuminated when pressure in the maintain-pressure system is adequate. At the same time, the gate four-way valve will be shifted automatically from close to neutral position.

e. Overfill and Overempty Control System. The overfill and overempty system should be evaluated on a case-by-case basis and considered mainly on high-lift locks or locks with long, narrow approaches. A control system has been developed to eliminate overfilling and overemptying of the lock chamber. It measures water levels by sensing the back pressure of compressed air constantly bubbling through tubes extending below the surface of the water. This system compares the level of water in the lock chamber with that of the upper pool when filling and the lower pool when emptying and, at a predetermined time, begins closing the fill or empty valves, respectively. This action dissipates the energy of flowing water in the culverts, thereby eliminating lock overfill or overempty. The operators at locks who utilize the gate-mounted limit switches have developed an operating technique that eliminates or greatly reduces overfill or overemptying. As the lock fills or empties, the operator watches the indicating lights controlled by the gate-mounted limit switches. When the lights start going off, the operator opens the appropriate gate.

4-7. Pintle Design and Assembly.

a. Pintle Assembly. The pintle and related components support the dead weight of each leaf of the miter gate. Additional discussion on miter gate pintles is in Chapter 2. The unit is made up of four major components: pintle socket and bushing, pintle, pintle shoe, and pintle base. Pintle assemblies used for horizontally framed miter gates are generally two types: fixed and floating. These are described below.

b. Pintle Socket and Bushing. The pintle socket usually is made of cast steel and is connected to the bottom of the lower girder web with turned Monel™ or stainless steel bolts. The bolts are sized to carry the gate leaf reaction in shear but, as an added safety factor, a thrust plate should be welded to the underside of the bottom girder web, with a milled contact surface between the plate and pintle socket. The minimum plate size should be 31.75 mm (1-1/4 in.) in thickness and 0.3 m (12 in.) wide, with a length as required by the girder web. The socket encloses the bronze bushing, which fits over the pintle ball. An allowable bearing stress of 10 MPa (1450 psi) is desirable but might not always be practical. An automatic greasing system allows a higher bearing stress but should not exceed 17 MPa (2500 psi).

c. Pintle.

(1) The pintle generally is made of cast alloy steel with a nickel content of 3 to 5%. The pintle should conform to American Society for Testing and Materials (ASTM) A148 GR 80-40 or ASTM A27 GR 70-40. It is usually 0.25 to 0.50 m (10 to 20 in.) in diameter, with the top bearing surface in the shape of a half sphere and a cylindrically shaped bottom shaft. Pintles also have been produced with bearing surfaces of stainless steel deposited in weld passes to a thickness of not fewer than 4.8 mm (0.1875 in.) and machined to the required shape.

(2) Pintles for locations in salt or brackish water should be forged alloy steel with a stainless steel bearing surface. For use in salt or brackish water, pintles should be of forged alloy steel with bearing surfaces of corrosion-resisting steel deposited in weld

passes to a thickness of not fewer than 3.2 mm (1/8 in.) and machined to the required shape. The pintle ball and bushing are finished to a 16 micron finish where the two come in contact.

d. Fixed Pintle. This type of pintle is recommended for new construction and major gate rehabilitation. The pintle fits into the pintle shoe, which is bolted to the embedded pintle base. The degree of fixity of the pintle depends on the shear capacity of the pintle shoe bolts. The pintle should be designed so that, after the load on the pintle is relieved by jacking, the pintle assembly is easily removable. See Plates B-23 and B-24 for a typical fixed pintle. The pintle base, made of cast steel, is embedded in concrete, with the shoe fitting into a curved section of the upper segment of the base. The curved section, of the same radius as the pintle shoe, is formed so that, under normal operation, the reaction between the shoe and base is always perpendicular to a line tangent to the curve of both shoe and base at the point of reaction.

e. Floating Pintle. This type of pintle is not recommended for new construction or rehabilitations. It is discussed here because various navigation sites have utilized this design. The pintle is fitted into a cast steel shoe, with a shear key provided to prevent the pintle from turning in the shoe. The shoe is not fastened to the base, thereby allowing the gate leaf to move outward in case debris between the quoin and wall quoin prevents the leaf from seating properly. See Plates B-21 and B-22 for a typical floating pintle. Damage to the pintle bearing has occurred frequently with this type of pintle because of the relative movement between the pintle shoe and base. The movement can consist of the shoe sliding on the base during leaf operation from either the mitered or recessed position, until the leaf reaches approximately the mid-position, at which time the shoe slides back against the flange on the base. This type of movement generally is visually detectable and causes serious wear. However, an alternative to the floating circular shoe is to make the shoe three-sided, with one corner having the same radius as the circular shoe, and attach a steel keeper bar to the embedded base in front of the shoe. This would prevent the shoe from rotating on the embedded base and prevent the pintle from moving out of pocket. Again, the degree of fixity would depend on the shear capacity of the bolts in the keeper bar. This alternative will meet the requirements of the fixed pintle and provide the capacity to minimize damage in case of emergency.

f. Pintle Base. The pintle base is designed so there will be a compressive force under all parts of the base. The value of the compressive force on the concrete will vary from a maximum on one edge to a minimum on the opposite edge. Computations are based on that portion of the pintle above the point under consideration acting as a composite unit. The overturning moment can be found from the horizontal force on the pintle and will be resisted by the reaction on the section being investigated. The eccentricity of the vertical force can be determined by the angle the resultant makes with the horizontal and the distance between the horizontal force on the pintle and reaction on the pintle base.

g. Pintle Location. The center line of the pintle (vertical axis of rotation) is located eccentric (upstream) relative to the center of curvature of the bearing face of the quoin contact block. This center of curvature is on the thrust line. The center line of the pintle should be located on the point of intersection of the bisector of the angle formed by the mitered and recessed gate leaf work lines and the perpendicular line from the bisector to the quoin contact point resulting in an offset of approximately 180 mm (7 in.), as in the details shown in Plate B-25. Studies and experience show that eccentricities arrived at by this method will reduce the contact time between the fixed wall quoin and the contact block of the moving gate leaf sufficiently to minimize interference and binding between the bearing blocks. The 180-mm (7-in.) offset will be exact and constant for all gates with the same miter angle and distance from the face of the lock chamber to the recessed work line 0.37 m (1 ft, 2.5 in.), as shown in Plate B-25.

h. Pintle Bushings. Pintle bushings for lock gates traditionally have been grease-lubricated aluminum bronze. The aluminum bronze alloy typically used is C95400, meeting the requirements of ASTM B148 or ASTM B271. Plates B-21 to B-24 provide recommended grease groove and seal details. The aluminum bronze bushing is press-fit into the pintle socket and bushing and secured by bolting to the socket. The bearing surface should be finished truly hemispherical and the pintle balls fitted to the bushings by scraping or should be lapped until uniform contact is attained over the entire bearing surface. This can be determined by testing with carbon paper or a similar media transfer technique. The pintle and bushing need to be match-marked. Show finished surfaces on the drawings, in accordance with ASME B46.1. Compliance with surface requirements typically is determined by sense of feel and visual inspection of the work and comparing it to the Roughness Comparison Specimens of ASME B46.1. Grease-lubricated bronze continues to work well, but environmental issues created by pumping grease to the pintle bushing have started a shift toward considering self-lubricated pintle bushings.

i. Self-lubricated Bearings. Additional discussion on self-lubricated pintle bearings is in Chapter 2. Self-lubricated pintles and bushings are becoming more widely used. The bushings and pintle balls should be designed as a system so they work together. The self-lubricated composite materials also can be designed with much larger bearing pressures than conventional bronze for large gate loads. However, the bearing pressures for miter gate pintles are usually low and generally well suited for self-lubricated bearings and bushings. More recent designs have been completed with self-lubricating material installed onto hemispherical or near-spherical pintle sockets with matching stainless steel pintle balls. The self-lubricated material is shaped into pucks or discs recessed and secured to the socket bushing or the pintle ball. Conductivity indicator wear pins should be incorporated into the bearing surfaces to allow the project personnel to test periodically for bearing surface wear and to schedule replacement.

(1) Self-lubricated bearing material, also known as composites, has been produced for many years. CERL has conducted a number of research projects to study the performance of self-lubricating materials, first for hydropower application and more recently for navigation lock and dam application. These reports include SR-04-8 Field Evaluation of Self-Lubricated Mechanical Components for Civil Works Navigation Structures. They also include: 99/104 Greaseless Bushings for Hydropower Applications: Program, Testing, and Results.

(2) Some materials and arrangements have worked better than others. Any material selected should be tested by an independent laboratory. The CERL report SR-04-8 addresses this. The composite typically is fitted in a bronze housing through interference fit and fasteners. The pintle typically is manufactured of cast steel with bearing surfaces of stainless steel deposited in weld passes to a thickness of not fewer than 4.8 mm (0.1875 in.) and machined to the required shape.

Figure 4-17. Pintle ball with self-lubricating wear pads

(3) Selecting the correct type of self-lubricated bushing, specifying the proper design criteria (i.e., composite thickness, surface finish, interference fit, or clearance fit) for each particular application is critical to ensure a successful installation. The CERL report (2004) identifies Corps lock-and-dam projects that have used self-lubricated pintle bushings. Designers should contact the districts identified in this report to get an update on the information provided in the report.

CHAPTER 5

Sector Gate Operating Machinery

5-1. General Description. Sector gates are used on both storm protection (hurricane protection) and lock gates. Their primary advantage, as explained below, is they can operate with differential head on either side of the gate. The sector gate consists of two gate leaves, each made of a curved skin plate, with a framed structure linking the skin plate back to a point of rotation located at the skin plate's center of curvature. When open, the gate leaves are within recesses at either side of the navigation channel. Operating machinery is employed to rotate the gate leaves across the channel, meeting at a vertical seal at the center, thus affecting a closure. Hydrostatic forces on the skin plate act in a radial direction and, as such, are orthogonal to the line of action for the operating machinery, which acts tangentially to rotate the gates leaves. The design of a sector gate ensures that the hydrostatic forces across the skin plate do not directly counteract the operating machinery; however, they still do contribute to the frictional load at the point of rotation. The sector gate, thereby, is better suited for operation in the presence of differential heads than the miter gate. A general discussion and design criteria for three types of sector gate operating equipment for lock application are presented. They are: wire rope and drum, rack and pinion, and direct-acting hydraulic cylinder. Sector gate structural description and design information is in EM 1110-2-2105.

Figure 5-1. New Orleans Caernarvon Canal sector gate

5-2. Operating System Descriptions and Selection Criteria.

a. Wire Rope and Drum. Sector gates traditionally have been driven either by a wire rope and drum mechanism, shown on Plate B-37, or by rack and pinion, shown on Plate B-38. The wire rope and drum mechanism was designed to be an inexpensive method of operating infrequently used gates, such as floodgates. Wire rope systems, or similar winch or capstan systems, also may be employed as a backup to rack and pinion systems. A disadvantage of the wire rope and drum mechanism is that the wire ropes tend to lose tension with use, requiring periodic re-tensioning and replacement. Also, because the wire rope drum position does not accurately correlate to the gate position, limit switches must be located on the gate or in the gate recess, potentially exposing them to damage. The track along the gate face where a wire rope typically would lay generally requires some interruption in the vertical seals for passage of the rope, providing a leakage point near the top of the gate.

b. Rack and Pinion. The rack and pinion mechanism mainly is used on lock gates or gates that have a high frequency of use and floodgates of any substantial size. This type of system allows for simple operation and little maintenance on the major load-bearing mechanical components. Once the rack and pinion mechanism is aligned, there is no need for continual adjustments. In addition, the gate drive pinion gear accurately correlates to gate position, thereby permitting the use of limit switches that can be located to operate directly from the machinery. A disadvantage with the rack and pinion mechanism is that the vertical seal must be interrupted to accommodate passage of machinery components, providing a leakage pathway near the top of the gate. Seal arrangement and positioning should be designed to minimize the interruption of protection. Also, wear in the gate's hinge and pintle eventually results in a tightening of the gear mesh. However, by this time, it is usually wise to either replace or rotate the gate bushings. The rack and pinion gears should have a diametrical pitch of 1 or more to minimize the effects of changes in gear clearance resulting from the relative radial movement of the gate rack and pinion gears. Additionally, it is recommended that adjustment in the radial direction (with respect to the gate leaf) be built into the pinion mounting, and both radial and vertical adjustment, such as slotted bolt holes, and shim stacks, be built into the rack mounting. Such features will facilitate ease of initial alignment and accommodation for future adjustments.

Figure 5-2. Hydraulically operated pinion gear and rack

c. Direct-Acting Hydraulic Cylinder. A third design uses a direct-acting hydraulic cylinder, as shown on Plate B-39. The direct-acting hydraulic cylinder has been around for a number of years, but is not in widespread use. Direct-acting cylinders often are used on gates with the hydrodynamic feature generally referred to as ears, which complicates implementation of wire rope or rack and pinion systems. To reduce the cylinder's stroke length, the cylinder's rod end is attached to the gate's top frame near the hinge and at an operating radius that is approximately 1/5 that of either the rack and pinion or cable and drum mechanisms. The short operating radius imposes higher stresses on the gate and machinery than the previous two designs. The advantages of the direct-operating cylinder is that it includes fewer machinery components, the cylinder is self-aligning with the gate, and limit switches can be built directly into the cylinder where they are not easily damaged and all machinery is located on the protected side and does not require an interruption to the vertical seal.

Figure 5-3. Hydraulic cylinder-driven sector gate, New Orleans

d. Power Transmission. Mechanical and hydraulic are the two types of transmissions that provide power to the three gate-operating mechanisms described above. The hydraulic transmission usually consists of an electric motor-driven hydraulic pump, control valves and, except for the direct-acting hydraulic cylinder machine, a hydraulic motor. Hydraulic transmissions are inexpensive and provide flexibility in control and physical layout. The control flexibility of the hydraulic transmission is particularly suitable for lock gates or floodgates where routine operation can be ensured. Hydraulic drive systems design should include cross-over relief or a counterbalance valve to prevent an external driving load from over-pressuring the hydraulic motor or cylinder. Counterbalance and control valves should be selected carefully to ensure smooth operation without surging or pulsing of the gate. The mechanical transmission usually consists of an electric motor, motor brake, and multiple shaft speed reducer. Mechanical transmissions are dependable and require little maintenance, which makes them suitable for floodgates; however, where danger of flooding of the machinery recesses are of concern, hydraulic systems tend to be more resilient, provided HPU and control systems are protected.

5-3. Design Considerations and Criteria.

a. General. Hydraulic loading on sector gates are produced from direct heads and reverse heads. A direct head is a head differential across the gate with the highest water elevation on the convex side of the skin plate. A reverse head is a head differential across the gate with the highest water surface on the concave side of the skin plate. Under normal heads, sector gate tests have shown that the loads created by flowing water tended to close the gate but were considerably less than those observed under reverse heads. Under all reverse head conditions, loads imposed on the gate by the flowing water tended to close the gate. Loads increased with gate openings up to 5 to 7 ft, then showed a tendency for a slow decrease at greater openings. Model data for gate openings of about 6 ft can be used to predict peak torque for various lower pools and reverse heads. Model and prototype tests demonstrated that the major loads on the gate are caused by structural members in the immediate vicinity of the skin plate at the miter noses of the gate leaves and by the side seal bracket that blocks side flow at the recess edge of the skin plate. Timber fenders, which are offset from the skin plate, have a negligible effect on forces.

(1) Operating forces from direct heads are friction from the pintle and hinge, hydraulic forces on the seal bracket, and bottom seal friction.

(2) Operating forces from reverse heads are hinge and pintle friction, hydraulic forces on the seal brackets, hydraulic forces on the vertical steel members near the nose of the gate, and friction from reverse head seals. Depending on the construction of the bottom seal, bottom seal friction might not be created during reverse heads.

(3) Unpredictable forces such as those caused by silt, debris, wear, wind, and construction inaccuracies should be accounted for by applying a 1.5 application factor to the calculated loads. Ice loading should be calculated separately, than added to all other calculated loads.

b. Determination of Machinery Loads. When determining operating loads for a sector gate, Waterways Experiment Station (WES) Technical Report H‑70‑2 and Appendix A to the report (USAEWES 1970, 1971) should be used as a guide. Sample calculations for determining closing loads with a reverse head are shown in Appendix C. However, if a gate design varying considerably from the type shown in the report is used, model studies to determine the loads should be performed.

(1) Hydraulic Loads. Difficulty was experienced in the design of the first sector gates when operating under reverse heads. Prototype tests showed that hydrodynamics forces on the vertical steel member near the nose of the gate created much greater loads than anticipated during design. As a result, extensive tests were made to obtain operating hydraulic forces on sector gates and to account for the hydrodynamics forces. These tests made by WES are published in the following technical reports, which are noted in Appendix A:

- H-70-2, "Operating forces on Sector Gates Under Reverse Heads."
- H-71-4, "Calcasieu Saltwater Barrier prototype Sector Gate Tests".
- 2-309, "Filling Characteristics, Algiers Lock Intracoastal Waterway, Gulf Section, Louisiana" and Appendix. The appendix covers gate operating forces and modifications to reduce operating forces.
- CHL-TR-03-3, "Filling and Emptying System for Inner Harbor Navigation Canal Lock Replacement, Louisiana."

(2) The tests by WES resulted in the design of an improved gate with operating forces approximately 40% of those experienced in the original designs. The third and fourth reports are for tests conducted on models of modified sector gates referred to as ear sector gates. In plan view, an ear sector gate resembles a traditional sector gate with the addition of two protruding radial members at each end of the gate called ears. Ear sector gates are designed to pass water through the center of the lock and through the gates' recesses as the gates open. This enables the lock chamber to fill and empty at a faster rate and with less turbulence, because not all the water is entering or leaving the lock chamber through the center opening as is done with non-eared gates. This feature is of greater importance with increase in lock lift. The design also prevents siltation in the gates' recesses. Algiers Lock, located on the Intracoastal Waterway and the Mississippi River at New Orleans, has ear sector gates designed for a differential head of 5.6 m (18.5 ft), about 3.6 m (12 ft) higher than would be practicable with non-eared gates.

(3) After maximum operating conditions on the sector gates have been determined, the gate operating loads should be computed both for normal flow and reverse flow conditions. Loads due to reverse head conditions usually will establish the size of machine to be used; however, loads due to normal heads should be checked.

(4) Water load on the gate will be created by the projected width of miter beam, skin plate rib, and seal bracket. Figure (a), Plate 44, of Technical Report H‑70‑2, Appendix A, gives the peak closing pintle torque for the improved type gate. These torque curves are reproduced for this manual and are shown in Appendix C. This torque is based on a gate having a total projected width of miter beam, skin plate rib, and seal bracket of 30.375 in. (17.875 in. + 8 in. + 4.5 in. = 30.375 in.). The torque should be corrected, in accordance with Froude's law of similarity, to the lengths used on the proposed gate based on the scalar ratio. For gates varying considerably from the type shown in the report, initial load estimates prior to completion of a model study can be calculated assuming a linear profile between pool-to-pool water elevations across the channel side face of the total projected width of the miter beam, skin plate rib, and seal bracket and the reverse head water elevation on the recess side of the same components. As such, the estimated closing force would be half the differential reverse head acting on the projected width of those components over the average submerged height of the miter beam. Hinge friction and pintle friction torque should be added to the above water load to determine the total machinery load and a 1.5 safety factor applied for machinery sizing. Reference should be made to Miscellaneous Paper H‑71‑4, paragraph 14 (USAEWES 1971), along with establishing reasonable values of hinge and pintle friction. Typical calculations for determining loads on the improved type of sector gate are shown in Appendix C.

c. Hinge and Pintle Friction. Hinge and pintle frictional torque is the torque generated at the bearing surfaces between the stationary part of the bearing and the movable part. The bearing load is the load resulting from the gate weight, hydrostatic loads, and reaction loads generated by the operating machinery. Based on using self-aligning hinge and pintle, a bearing frictional factor of 0.25 for steel on bronze should be used. If either a cylindrical hinge or pintle is used, the designer should anticipate much higher frictional loads resulting from possible construction misalignment. WES has found that cylindrical hinge and pintle friction for Calcasieu Saltwater Barrier sector gates were 4.5 times the calculated value.

d. Bottom Seal Friction. Bottom seal friction is caused by the differential hydrostatic head across the seal and force of pre-compressing the seal 6.4 mm (0.25 in.). A coefficient of friction of 1.0 should be used, even for Teflon coated rubber seals. Initially, the seals on a sector gate are set with approximately 0 to 0.8 mm (0 to 1/32 in.) of clearance. The 6.4 mm (0.25 in.) pre-compression accounts for gate sag, hinge and pintle wear, and variations in gate temperature between submerged members and non-submerged members.

e. Contingencies. After the gate loads are calculated, an application factor of 1.5 should be applied to the combined friction and hydraulic loads. The application factor accounts for transient and unpredictable forces such as those resulting from silt, debris, hinge and pintle wear, and construction inaccuracies.

f. Machinery components. General criteria applicable to machine components are in Chapter 2.

5-4. Operating Procedures and Controls.

a. Operating machinery controls. Sector gates usually are controlled from a small control house adjacent to each pair of gate leaves. For electric motor drive, the control equipment consists of the combination of full-voltage magnetic controllers, limit switches, control pushbuttons, and switches arranged to produce the desired operating sequence. For fluid motor drive, the speed of the gate is varied by controlling the flow of oil to the fluid motor either by throttling or by use of a variable stroke piston pump. With this system, control valves can be controlled either manually or electrically.

b. Low Head Locks. Low head locks are locks that have a lift of 1.5 m (5 ft) or fewer. To fill and empty a low head sector gate lock chamber, the operator opens the filling or emptying sector gates from 0.3 to 0.9 m (1 to 3 ft). The gates then are held in this position until the differential water level across the gates is within 150 mm (0.5 ft). At this time, the gates are opened fully. A single operating speed of between 20 to 35 deg of gate rotation per minute with cushioned gate start and stop has been found satisfactory. With a hydraulic transmission, cushioned gate start and stop can be incorporated into the hydraulic system using ramp proportional valves or other flow control devices. Machinery brakes also should have cushioned movement.

c. Flood Control Gates. Single speed operation of between 5 to 7 deg of gate rotation per minute has been found satisfactory. At this low speed of operation, cushion gate start and stop are not required.

d. Medium to High Lift Gates. Medium to high lift locks are locks with lifts of more than 1.5 m (5 ft). For medium to high lift locks, where the gates are used to fill and empty the chamber, a two-speed operating system is required with a slow initial opening speed. The slower speed enables the lock operator to accurately set the gate opening to prevent excess chamber turbulence. The slow speed should be field adjustable with a range of from 1.5 to 5 deg of gate rotation per minute. A higher speed of 20 to 35 deg of gate rotation per minute can be used once the differential head across the gate is within 150 mm (0.5 ft). Starts, stops, and changes in gate speed should be cushioned.

5-5. Special Design Considerations.

a. General. The gate operating machinery is crucial to the operation of a lock or floodgate structure. Reasonable means should be made to incorporate into the design a high degree of reliability and serviceability.

b. Auxiliary Drives. For most hydraulically driven gates, an auxiliary drive has proven valuable. The auxiliary drive should be basic and provide an operating speed that is half to a quarter of that of the primary drive. The auxiliary drive should consist of a pump and motor connected permanently to the gate's hydraulic system. A portable drive system shown in Figure 5-5 can also be utilized. Other hydraulic system components such as valves, solenoids and hoses should be accessible and easily replaceable. A dual pump and motor arrangement where both pumps operate in parallel for primary operation, with each pump providing half capacity redundancy should the other fail, is an acceptable means of providing an auxiliary drive. Mechanically operated gates normally do not require an auxiliary drive. However, flood control gates with mechanical drives should have auxiliary power sources such as an auxiliary generator, hand crank, or air motor with air storage. When incorporating a hand crank mechanism, an awareness of the required time to operate such a system should be considered and possibly alleviated by providing operability with a portable actuator or PTO of some sort. The incorporation of pad eyes or bollards into the gate structure for closure by alternate means (portable winch, tow boat, etc.) might be prudent for some applications. A back-up winch system utilizing wire rope is shown in Figure 5-4. This can be used to close the sector gate in the event of a hydraulic system failure.

Figure 5-4. Backup winch system

Figure 5-5. Auxiliary power unit for Bayou Dupre Gate, New Orleans

c. Hydraulic System Contamination. Hydraulic driven sector gate drives (on storm barriers) are particular vulnerable to water contamination since they can be exposed to hurricane force winds and driving rain. This is something the designer has to mitigate. Water in the hydraulic system is one of the primary reasons for hydraulic component failures. Water usually infiltrates the system due to the moisture in the air that is exchanged in the reservoir through the breathers. To eliminate this source of contamination, the hydraulic reservoir should be located in a room with dehumidification and/or be equipped with a bladder that prevents direct exchange with outside air. Chapter 3 provides additional discussion on this issue.

d. Material Selection. When practicable, machinery components subject to damage should be constructed from field welded materials. This is especially important for items that would require a long lead time to acquire or take substantial effort to replace, such as gear racks, drive pinions, and machinery bases.

e. Generator Backup. All sector gates utilized for storm (hurricane) protection shall be provided with generator backup power. An automatic transfer switch that will transfer utility power to generator power should be provided when consistent with operational protocols and considerations specific to the installation in question. Generators should be installed in a protected building. Generators should be sized for the full operational capability of the gate, including simultaneous operation of any lighting and other ancillary systems.

Figure 5-6. Bayou Dupre, New Orleans, backup generator

5-6. Pintle and Hinge Design.

a. General. Sector gate operation is highly dependent upon a functioning bearing system at the gate hinge and pintle. It is critical the hinge and pintle bearings be designed, constructed, and maintained correctly. The hinge and pintle are the reaction points against the pool-to-pool head load when the gate is closed; they also provide support to accommodate the lateral load from the cantilevered weight of the gate. The pintle additionally receives the vertical dead load of the gate. The hinge transfers the horizontal load into the concrete wall. A high degree of reliability throughout the service life should be incorporated into their designs. Spherical bearings typically are used to prevent binding in the case of minor misalignment. Additional discussion on pintle bushings and self-lubricated bearings is below and also in Chapters 2 and 4.

b. Bronze Bushings. Traditionally, grease-lubricated bronze bushings running against stainless steel bearing surfaces have been utilized for sector gate hinge and pintle bearing systems. The bearing relies on a layer of grease to provide lubrication and, as such, bearing pressures must be at a low enough level to maintain adequate film thickness. Recommended maximum static bearing pressure should be no more than 2500 psi, with loading during operation not to exceed 5000 psi. Design of a grease-lubricated bearing system should include the proper running clearance, surface finishes and bearing material properties, the grease lubrication system, and the arrangement of any seals used.

(1) Clearance on large greased hinge and pintle bearings should be medium running fits, class RC5 to RC6. Surface finishes should be specified as 0.4 microns (16 micro-inches) or better. C95500 aluminum bronze material or similar is recommended based on its high yield strength; however, C93200 leaded tin bronze or similar material has been preferred for some applications because of its good lubricity. Surface hardness of the stainless steel bearing surface should be minimum 40 Rockwell C. Bushings and balls may be matched fit and lapped to one another prior to installation to ensure fit and finish. In addition to mechanical material properties, selection of a bushing material also should include considerations of its position on the galvanic scale relative to the pintle ball material to avoid galvanic corrosion. In some marine environments, the aluminum component of the previously recommended C95500 has been found to corrode sacrificially in contact with stainless steel.

(2) The basic grease lubrication system would include manual grease fittings located in an accessible location, with grease lines of stainless steel tubing run to ports on through the bronze bushing. The bushing should be fixed within the housing, such that a grease pathway is maintained and proper fit is ensured by eliminating the possibility of wear induced by bushing rotation within the housing. The bearing surface of the bronze bushing should incorporate grease grooves to adequately distribute the grease across the entire bearing surface. The pattern of grease grooves and number of grease ports should be designed according to the size and range of rotation specific to the bearing. In locating grease ports, priority should be given to introducing grease to the loaded side of the bearings. Automatic grease dispensers might be appropriate in gates with frequent operation. Recirculation systems incorporating grease return lines

and/or environmentally benign grease might be necessary when grease systems are used in environmentally sensitive locations. The preferred alternative in such cases would be to use a greaseless bearing system. Grease lines are typically field fit, but should be run in such a manner that they are protected by the gate structure as best as can be accommodated.

(3) Seals generally are recommended on greased systems. Seals might be required to keep dirt and debris out, to keep the grease contained, or both, and should be designed according to their intent as double acting or single acting oriented according to their purpose. It should be noted that, if return lines are not supplied, the seal should provide some pathway for old grease to exit the bearing as new grease is pumped in. Seals should be of the non-pinniped type.

(4) Greaseless Bearing Systems. Self-lubricating or greaseless bearing systems have been used for various applications including hinge and pintle bearings on sector gates and miter gates. Greaseless systems eliminate the need for grease lines, prevent the introduction of grease into the marine environment and provide a very low friction bearing system. Greaseless systems generally fall into one of three types. Plug-type systems incorporate an array of lubricating plugs in a bronze bushing, which act as a bearing surface and provide lubricant between bronze bushing and stainless bearing surface as they wear. Alternatively, liner-type systems provide a surface of synthetic material to act as the bushing material in place of lubricated bronze. A variation on the liner system is the puck-type system where, instead of a continuous liner, the same type of synthetic material is placed on an array of bearing inserts.

(5) Plug-type systems essentially are a bronze bushing with lubrication for the life of the bearing supplied internally. They are most similar to traditional greased bronze systems, and equivalent bearing pressures and friction coefficients should be used during design. The degree to which the plugs act as a bearing surface, in addition to lubricant, should be considered when determining effective bearing area. Plug-type systems might require break-in grease until the plugs wear adequately. Typically, they do not benefit from the same reduction in friction of a liner or puck-type system; however, plug-type systems do not risk liner de-lamination or grease system failure and are generally resilient regarding maintenance neglect.

(6) Liner-type systems utilize a synthetic material as the bushing material running against the stainless steel bearing surface. Materials with coefficients of friction in the range of 0.08 to 0.10 are typical, though maintaining the more conservative assumption used with traditional greased bushings is recommended when sizing operating machinery on critical flood protection applications. The practice of using a liner substrate of traditional bearing material may be incorporated to provide redundancy in case of liner failure, but shouldn’t be required for a properly designed liner system and might constrain design. Stainless steel substrate bushings on stainless steel bearings have been utilized, and allow for placement of the liner on the ball O.D., rather than bushing I.D., where beneficial. Regarding placement of the bearing on the inner or outer bearing surface, rather than trying to distribute wear across the entire bearing surface,

multiple suppliers recommend placing the liner in the location static to the loading to avoid cyclic loading that might lead to liner de-lamination. If liners are placed on the ball O.D., the treatment of wear related to vertical seams on split bushings should be discussed with the bearing supplier. Greaseless liner systems might be adequate for increased bearing pressures, and the manufacturer's recommendations should be followed. However, in this case, a conservative assumption of 50% contact area is recommended. Liner thickness, surface finishes, and running clearance and tolerances typically are provided by the liner manufacture specific to working load, cycle life, and bearing size. Liner materials should be dimensionally stable, especially with regard to swelling or shrinkage from water absorption, in order to ensure designed fits and clearances are maintained. Seals might or might not be recommended by the manufacturer, but could be appropriate depending upon site conditions.

(7) Puck systems typically use the same types of material as liner systems, except they are applied to individual inserts mounted in recesses on what otherwise would be the bushing surface. See Figure 5-7. The insert's face is raised above the mounting surface, such that the interstitial space between inserts does not contact the stainless steel bearing face. Total projected bearing area must be reduced accordingly. Puck systems facilitate fabrication on large bushings and aid in replacement, as they do not require removal of the entire bushing or pintle from the work site for recoating, as would be necessary for a liner system of machining and lapping for a greased system. For critical applications, use of traditional bearing material as liner substrate and auxiliary greasing provision might be provided for the case of liner failure; however, this typically is not necessary. Care should be taken in selecting insert fastener arrangements to ensure they fully constrain the components under the expected loading.

Figure 5-7. Pintle bushing, Caernarvon Canal, New Orleans

c. Quality Control and Quality Assurance. Hinge and pintle assemblies are comprised of large components manufactured to tight tolerances. Incorrect fits and finishes might lead to binding or vibration, causing excessive component wear and increasing the required operating force. To avoid such issues, quality control should be addressed in the design specifications, and an expectation for a reasonable amount of quality assurance effort should be anticipated to ensure radial tolerances, spherical tolerances, and surface smoothness. For bearing systems utilizing the assembly of multiple components, such as a puck system, tolerance stack-up could be an issue. Recommended is a robust quality assurance procedure utilizing coordinate measuring machines, and/or final machining of the liner material with the inserts in place. For greased bronze on steel, machinist dye or similar technique may be used to check for surface contact.

CHAPTER 6

Filling, Emptying and Water Control Valves, and Machinery

6-1. General Description. The most common type of filling and emptying system used in locks is a longitudinal culvert in the lock wall extending between the upper and lower pools. Each culvert has a streamlined intake at the upstream end and a diffusion discharge at the downstream end. Culvert flow is distributed in and out of the lock chamber by wall ports or secondary culverts in the floor of the chamber. Each culvert has two valves: one for filling and one for emptying the chamber. The filling valve allows upstream pool to fill the chamber while the emptying valve remains closed. The emptying valve allows the chamber to drain to the downstream pool while the filling valve remains closed. Guidance on hydraulic design criteria including forces due to hydraulic loading can be found in EM 1110-2-1610.

6-2. Machinery Design Criteria. The general design criteria for machinery components are in Chapter 2. Culvert valve machinery must be designed to raise the valve under flowing water conditions at the full maximum head differential. Hydraulic design engineers should provide the gate operating speeds, including any pauses, to be used at the various head conditions planned for the specific lock location. Operating speeds are based upon specific flow conditions designed to fill or empty the lock chamber without producing unsafe hawser stresses, air entrainment, or other operating conditions dangerous to the tows or their personnel. If wire rope connected, the valve should provide sufficient weight to close, even under flowing water conditions, because the wire ropes are incapable of forcing the valve to close. Closing under flowing water conditions might be required where ice or debris flushing operations are typical, especially at locks with upstream lift gates. It is customary to design all valve machines identically for economy of fabrication.

a. Culvert Valve Hoist Loads. The hoist should be designed for: (1) the gate connection load due to flowing water under the valve, (2) the buoyant (submerged) weight of the valve, (3) the weight of the operating equipment linkages or wire rope assemblies, (4) the side seal friction, (5) the trunnion bushing friction under maximum normal flowing water load, (6) bushing friction at other points in the drive, and (7) the head differentials across the top seal of the valve. Evaluation of these loads is a mandatory minimum requirement for machinery design.

b. Valve Operating Speeds. Typical operating speeds for culvert valves should permit opening in approximately 1 to 3 min. Operating times as long as 15 min have been used at the John Day Lock. Cavitation problems caused revision of the John Day Lock's opening time to fewer than 7 min, including a 5-min pause at 30% open. Discharge conditions such as scour, low water, or temporary moorings also could result in the need for slower valve operating speeds. Sequencing of valve opening or closing positions might be necessary to control lock chamber overfilling or overemptying.

c. Hoist Drive Motors. Dual speed operation can be accomplished with a two-speed electric motor. For multi-speed operation, an electric motor controlled by a

Variable Frequency Drive (VFD) would be more practical. Modern technology has resulted in the DC drive and the VFD, which provide widely variable speed and torque at a competitive cost. These devices can provide almost infinitely variable speed with constant horsepower. This system allows ice flushing at low rpm and high torque, while normal, balanced head operation can occur at high rpm and low torque. For more detailed information, see Chapter 12.

d. Hydraulic System Design. The hydraulic control circuit for culvert valve machinery should include: (1) a solenoid-controlled, pilot-operated, four-way directional control valve, (2) an adjustable pressure relief valve for opening operations, (3) an adjustable pressure relief valve for closing operations, and (4) a remote pilot operated counterbalance valve. The directional valve should be designed with a blocked center or tandem center spool providing positive pump output to the cylinder in both directions of operation. Culvert valves should not be allowed to lower through the hydraulic control circuit only by their own weight. Such operation could lead to undesirable shock and vibration within the control circuit. The pressure relief valves are provided to protect the controls and cylinders from excessive pressure, which could lead to damage of the strut, bellcrank, or associated bearings and pins. The counterbalance valve is the typical method to prevent an overrunning load while providing a positive locking of the cylinder, at any valve position, until hydraulic pump pressure is applied to the cylinder for actual planned movement.

(1) Instrumentation. The installation of pressure transducers and pressure gauges at strategic locations within the hydraulic circuit would provide useful information in the adaption of the hydraulic system to actual lock operating conditions.

(2) Directional Control Valve. There is no benefit to designing a single-acting hydraulic cylinder system that does not have a four-way directional control valve to direct positive pump delivery to the cap-end side of the hydraulic cylinder. Systems that are designed to allow the weight of the culvert valve to lower the valve do not take advantage of the speed and force controlling features of a power-down control system. Locks that have situations where closing the tainter valves against flowing water have some benefits will most certainly need double-acting control.

(3) Pressure Relief Valves. Pressure relief valves should be designed for the maximum pressure range that will not cause damage to the system. The smallest commercially available range that will meet system requirements should be used because this will yield the maximum setting sensitivity. A pressure relief valve should be provided to prevent excessive pressure upon closing the tainter valve against the sill plate in the culvert. A pressure relief valve should be provided to prevent excessive pressure upon opening the tainter valve to the full open position.

(4) Counterbalance Valve. A remote pilot-operated counterbalance valve is required to hold the tainter valve open at any position that it is stopped until positive pump pressure is applied to move the tainter valve.

e. Controls. Appropriate control devices are detailed in Chapter 12, Equipment and Machinery Controls. Overfill/overempty should be included if found to be needed through hydraulic model studies or testing of the actual lock. To limit the overfilling or emptying of the lock, this control scheme varies the valve opening.

f. Special Design Considerations. Special design considerations are areas of coordination among the structural, mechanical, and electrical designers, and are necessary to provide a proper operating system.

(1) Slack Cable Safety Devices. Slack cable safety devices are an essential safety feature for wire rope-operated culvert valve machinery. The culvert valve could seize against the valve chamber walls, or above the culvert floor, on debris or zebra mussels. The slack cable safety device will shut down the motor before too much cable is unspooled. This will prevent problems with guiding the wire rope back onto the drum properly.

(2) Positional Encoders. Positional encoders or sensors, connected to the machinery or integral to the hydraulic cylinders, are essential to the operation of the culvert valve machinery and lock electrical control system. Encoders or sensors are used to provide the elevation position of the bottom of the culvert valve, which can control the filling, emptying, and miter gate operation interlocks. Encoders or sensors can be used to indicate speed, motion, or actual angular position of various machinery components, which can be translated to culvert valve motion. For more information, see Chapter 12.

(3) Wire Rope-Connected Culvert Valve Design. The structural engineer should be aware that the operating machinery is not designed to force the valve down. Because the valve must lower due to its own weight, it is important that the structural designer compensate for any uplift hydraulic loads. EM 1110-2-1610 and ERDC TR-11-4 both provide extensive discussion of these uplift tendencies with respect to valve design and head conditions.

(4) Limit Switches. Limit switch locations must be coordinated with the structural designer to prevent overtravel in the valve opening or closing position, or to signal fully open or closed. These switches and their electrical appurtenances should be submersible.

(5) Lubrication System. Where grease-lubricated bearings or permanently lubricated bearings with grease supplementary provisions are provided for bellcranks, struts, or other connections, the supply lines should be mounted inside the structural tubes. Flexible hose connections might be required to connect piping across pivoting joints. All exposed piping and hose should be equipped with rigid structural steel guards designed to provide maximum protection against waterborne debris and ice.

6-3. Culvert Tainter Valve. The most common type of filling and emptying valve is the tainter valve. The tainter valve is constructed in a manner similar to the tainter gates typically used as spillway gates, but oriented either in the standard or reverse configuration. Additional information on culvert valves is available in EM 1110-2-1610, “Hydraulic Design of Lock Culvert Valves”, ERDC TR-11-4, ERDC Presentation on Lock Culvert Valve Design, February 2011, ERDC TR-03-3, WES TR-2-309, and WES TR-2-537.

a. Tainter Valve. Many of the navigation locks on the upper Mississippi River have conventional tainter gate-type valves. The valve is oriented with the trunnions downstream of the skin plate, causing the convex surface of the skin plate to face the flow and seal along the upstream end of the valve well.

b. Reverse Tainter Valve. Many of the navigation locks on the Ohio River, as well as some of the newer ones on the Mississippi, Red, and Arkansas rivers, have reverse tainter gate type valves. The valve is oriented with the trunnions upstream of the skin plate, causing the convex surface of the skin plate to face downstream and seal along the downstream end of the valve well. The reverse tainter valve offers an advantage over the conventional tainter valve in that its orientation prevents the introduction of air into the culverts in higher submergence applications. Figure 6-1 shows a reverse tainter valve, stored on the lock wall.

Figure 6-1. Reverse tainter valve

c. Machinery. The machinery arrangements discussed below may be considered for either tainter or reverse tainter valves. The major limitation of wire rope-operated tainter gates is they are not suitable for use where uplift forces exceed downward forces. The term tainter valve used below refers to both tainter and reverse tainter valves.

(1) Electric Motor-Driven Tainter Valve Hoist. This hoist uses two stainless steel round wire ropes, one at each end of the tainter valve. The wire rope is connected to the

convex side of the tainter valve at the lower main girder near the side strut location. The valve should provide sufficient weight to close even under flowing water conditions, because the cable system is incapable of forcing the valve to close. The cables are connected to two grooved drum assemblies, which are flanked by spherical roller-bearing pillow blocks. The drum assemblies are connected to a quadruple reduction parallel shaft reducer by geared flexible couplings. The parallel shaft reducer has dual extended input shafts to connect to the electric drive motor and hoist holding brake. A rotary limit switch assembly is connected to the brake shaft extension. The holding brake is typically a solenoid-operated shoe brake. The electric drive motor may be a custom two-speed constant torque motor or a variable frequency drive (VFD) motor system (for multi-speed operational requirements). Hard-wired overtravel limit switches also are used to supplement the rotary limit switch assembly. A slack cable limit switch assembly is provided to prevent unspooling of the cable when the gate is not moving. Plate B-40 and Figure 6-2 shows a typical design.

Figure 6-2. Electric Motor Driven Tainter Valve Hoist

(2) Hydraulic-Operated Bellcrank Type Hoist. The typical hoist for the reverse tainter valve on large capacity locks consists of: (1) a trunnion mounted hydraulic cylinder, (2) a bellcrank, (3) a gate operating strut, (4) a support base, and (5) bearings. The hydraulic cylinder has a center trunnion mounted on pillow block bearings. The cylinder rod is attached to one corner of a truss-type bellcrank made of steel pipe. The bellcrank has one corner about which it pivots, connected to a pair of pillow blocks. The other corners are connected to the hydraulic cylinder and the gate strut. The gate strut is a steel pipe assembly that contains clevis and eye end connections and a spring assembly. The gate strut connects the bellcrank to the tainter valve. All pivot connections are equipped with bushings and pins. Lubrication piping is routed to all

bushings and pillow blocks. Lubrication piping can be routed inside struts and bellcrank tubes to reduce exposure to damage. Plate B-42 shows a typical design.

(a) Trunnion-Mounted Hydraulic Cylinder. Trunnion-mounted hydraulic cylinders, used for bellcrank-type tainter valve machinery, experience a kinematic motion that places large side loads on the upper half of the rod end seals. This usually leads to premature seal wear and chatter marks on the cylinder rod. Special attention is necessary for the proper design of seals and rod material. One solution for this problem is the mounting of the cylinder in a cardanic ring to eliminate side loading.

(b) Bellcrank. The bellcrank must be specified with proper dimensional tolerances to ensure that it rotates in an accurate vertical plane. The assembly should undergo mandatory testing after fabrication to ensure that all shaft pin holes are parallel and all arms are straight within maximum standard tolerances. There should be mandatory survey requirements through its range of motion after installation. Past installations with poor quality control have caused accelerated wear of bushings, clevises, and eyes, leading to premature failure of machinery. Another important consideration is the defense of the shaft pin/bushing lubrication lines against damage by debris or ice. Lubrication piping can be placed inside the bellcrank tube arms, except at the pivot joints. Other forms of guards may be fabricated to attempt to protect the hoses used at pivot joints.

(c) Gate-Operating Strut. The gate-operating strut generally contains a spring assembly to assist in positive closure against the culvert sill. Several types of springs, including ring springs and Belleville washer-type springs, have been used. Coil springs appear to give superior performance because of their relative independence of lubrication. Because there is no easy way to verify grease effectiveness without actual disassembly of the spring, performance can be measured only by failure. A number of recent failures have been observed with the shattering of ring springs and Belleville springs in normal service. Detailed inspections show the components have not received sufficient lubricant on the essential rubbing surfaces.

(d) Support Base. The tainter valve machinery support base is designed to properly align the trunnion-mounted hydraulic cylinder and the bellcrank pivot trunnion bearings. This is essential to ensuring the cylinder, bellcrank, and strut operate in an accurate vertical plane. It is mandatory that the support base be inspected after fabrication to establish the relative positions of the machinery mounts to ensure the accurate vertical plane. The support base must be installed level, in order to allow a properly constructed bellcrank and trunnion support assembly to move in an accurate vertical plane.

(3) Hydraulic-Operated/Wire Rope-Connected. This hoist type is considered a variation on the electric motor-driven tainter valve hoist because wire ropes are used to connect the valve to the prime mover. In this case, multiple wire ropes are attached to the rod end of a horizontally mounted hydraulic cylinder, routed over a grooved drum, then connected to the valve. Plate B-41 shows a typical design.

(4) Alternative Design. Some locks use a vertically mounted hydraulic cylinder with a sealed bonnet around the cylinder rod end to exclude water from the valve well. The vertical cylinder does not pivot, but extends straight downward. The cylinder rod drives a pivoting gate-operating strut that is connected to the gate. The connection between the cylinder and the strut is guided along the wall of the recess. Plate B-41 shows a typical design.

(5) Direct-Acting Cylinder Design. A direct-acting cylinder design, which pivots about a cap end trunnion with the rod connected directly to the tainter valve, has been used successfully. This system is submerged during operation. Some evidence of water leakage mixing with the hydraulic fluid does indicate sealing problems. This system might be applicable to locations where frequent inspection and maintenance of the cylinders are feasible. Extreme measures are required to protect and maintain seals and piping/hose from debris or ice. Plate B-44 shows a typical design.

6-4. Vertical Lift Culvert Valves. Vertical lift culvert valves offer a viable alternative where the site constraints make it an economical choice or the size of the culvert is small. Advantages include that vertical gates or valves require a shorter length of lock wall compared to tainter valves and the distribution of the hydrostatic load over a greater area. Disadvantages include the need for gate slots. Also, vertical lift gates often require the use of wheels or rollers when sliding friction becomes prohibitive. These requirements contribute to the typically higher cost of vertical lift gates when compared to tainter valves. The wheels or rollers of wheeled gates also are susceptible to fouling due to their exposure to silt and debris-laden water. More information on vertical lift gates and valves can be found in EM 1110-2-2105.

a. Machinery. Vertical lift culvert valves are well suited for the use of electric rising stem valve actuators, directly connected, vertically mounted hydraulic cylinders, or electric motor/gear reduction/wire rope drum driven. If hydraulic cylinder actuators are used, a method for preventing the introduction of transverse loads to the cylinder rod should be employed to avoid premature wearing of the rod seals and rod bearing. This method could include the trunnion mounting of the hydraulic cylinder in a cardanic ring.

b. Slide Gates. A slide gate is a vertically sliding valve typically with metal-to-metal contact for end support. These surfaces also serve as the gate seal. Because of this, the bearing surfaces must be machined to tighter tolerances than wheeled or tractor type gates. Design criteria specific to the use of slide gates as culvert valves have not been developed. Nevertheless, general design criteria for culvert valves found in this manual and hydraulic criteria found in EM 1110-2-1610 may be used with slide gate design guidance available from manufacturers.

c. Stoney Gate Valves. Stoney gate valves are vertically operated gates in which the rolling load is transmitted from the face of the gate to the track through roller trains on either side of the gate. The roller trains are suspended from 2:1 reeving, with one end of the wire rope connected to the gate and the other end anchored so the roller trains move at half the speed of the gate. Plate B-45 shows a general plan and elevation of a stoney gate valve. The advantage of the stoney gate design is that the rollers and axles theoretically are subject to nominal rolling friction. Figure 6-3 shows a stoney gate valve with liner being assembled and tested in the fabrication shop. The roller train is not shown.

Figure 6-3. Stoney gate valve blade with liner

d. Caterpillar Gates. Caterpillar or tractor gates are similar in principle to stoney gate valves, except the roller train is continuous, wrapping around the both sides of the gate.

e. Fixed Wheel Gates. This type of gate uses wheels on fixed axles usually cantilevered from the body of the gate. These are generally less expensive to fabricate than tractor or stoney gates, but result in higher bearing stresses in the wheels and guides.

6-5. Butterfly Valves.

a. The closing mechanism of a butterfly valve is a circular or rectangular disc that rotates either parallel to the flow in the open position or perpendicular in the closed position. Passing through the horizontal or vertical axis of the disc is a rod or trunnion on which the disk turns. The trunnion then is connected to an actuator. Butterfly valves offer the advantage that their structure and operating mechanism is contained mostly within the area of the flow path. This arrangement lends itself to a filling and emptying system contained in the floor of the lock with submerged valve and actuator requiring no top of lock wall area. See Plate B-46 for an example.

b. Machinery. Actuators may be rotary or linear with a linkage bar. Manufacturers' torque curves should be consulted to determine torque requirements at varying heads and valve angles.

6-6. Recommended System. On the majority of modern locks, the reverse tainter valve has been the preferred choice. This is due to improved hydraulic characteristics, ease of fabrication, and lower installation and maintenance costs. For new locks, model studies should be conducted to determine the best valve configuration that will complement the hydraulic characteristics of the filling and emptying system. See EM 1110-2-1610 and ERDC TR-11-4 for further discussion.

CHAPTER 7

Vertical Lift Gate Operating Machinery

7-1. General Description and Application. There are many different types of vertical gates and hoisting arrangements used for both navigation and water control projects. Refer to EM 1110-2-2105 for vertical lift gate description and design information. The gate types consist of single or multiple leaves that either can be raised from submerged positions or lowered from overhead positions. These gates have been designed for both static hydraulic and dynamic conditions. The dynamic type designs are generally more robust in structure and hoisting requirements. This is necessary because the gates must be capable of regulating or shutting off flow during normal operation or emergency situations. The static hydraulic head gates are raised and lowered under no-head conditions for use in lock chambers. The hydraulic head then is placed on the gate by raising or lowering the lock chamber with the filling/emptying system. They are not designed to operate in flowing water conditions.

a. Hoisting arrangements and operating machinery for the vertical lift gates are as varied as the designs for the gates. The more common types of mechanical systems to mechanically raise and lower the vertical gates include wire rope, hydraulic cylinder, mechanical screw, gantry crane, engineered roller chain, and round link chain. This chapter will describe the type of vertical lift gates and provide the designer with general design criteria for the development of vertical lift gate operating systems.

b. The designer also might wish to reference the CENWP-EC-DS, April 2008 Memorandum for Record Letter Report for the John Day Navigational Lock noted in Appendix A. The independent technical review provides a discussion of vertical lift gate repairs at John Day and valuable information about the background and consequences of four incidents occurring with the upstream vertical lift gate at the facility from 1975 through 2008.

7-2. Vertical Gates for Navigation Locks.

a. Overhead gates. These types of gates use a tower with overhead cables, sheaves, and bull wheels to support the gate during its operation and use counterweights to assist hoisting machinery. The tower height is governed by the lift required to pass barge traffic. This type of gate would be used when it is not practical or feasible to use other gate types. See Figures 7-1 and 7-2 for photos of vertical overhead gates. Plate B-49 provides details.

Figure 7-1. Vertical overhead gate

Figure 7-2. Vertical overhead gate

b. Submersible gates. A submersible gate may be used as the upstream gate for a navigation lock. In this case, the gate's submersible leaf rests below the upstream sill to allow navigation to pass. Submersible gates generally are not feasible under such conditions as: high-head applications, when sufficient support cannot be provided for transferring thrust from miter gates, when the available area to place the gate monolith is limited, restricting the use of miter gates, and when the gate is used as a hurricane or tide gate and is subject to reverse hydrostatic or hydrodynamic loadings. There are two types of submersible gates: single leaf and multiple leaf. The double-leaf arrangement is most common. A multiple- leaf arrangement is shown in Figure 7-3.

Figure 7-3. Multiple-leaf vertical lift gate

(1) An emergency and service lift gate arrangement is composed of a downstream leaf used for normal lock operation and an upstream leaf used infrequently as a movable sill or as an operating leaf in an emergency. The emergency leaf is used for lock closure in the event of an accident or damage to the gate that otherwise would result in loss of the navigation pool, as shown in Figures 7-4 and 7-5. Plate B-48 shows a downstream leaf. The submersible type of gate is useful when it is necessary to skim ice and drift from the lock approaches or open the lock gates to pass flood flows. See Plate B-50 for an example of a multi-leaf gate.

Figure 7-4. Emergency gate used for lock closure

Figure 7-5. Flow over emergency gate

(a) As noted above, emergency-type gates generally consist of two leaves, one upstream and one just downstream of the other. The emergency gate leaf gate is equipped with wheels and is designed to be raised in flowing water. The normally operated service leaf is designed to be raised only in a balanced pool or when the swell head is 1 ft or less. When this type of gate is used as an operating lock gate, it normally would be operated under balanced head conditions and not through flowing water. Gate speed under balanced head conditions should be 5 ft/min to 10 ft/min.

(b) An emergency gate also can be designed for use when a catastrophic failure of a lock miter gate occurs or when it is necessary to pass ice or debris with the miter gates open and latched in the recess. When operating the gates, the gate leaves must be raised in steps and in sequence to match gate capabilities. Gate-lifting speed for both leaves should be limited to 1 ft/min to 5 ft/min, adjusted to suit the speed of the nearest standard speed motor. Operating procedures for this type of gate are shown in Plate B‑47. Figure 7-6 shows a single-leaf service gate (lowered) with a single-leaf emergency gate (raised).

Figure 7-6. Single-leaf emergency gate

(2) Common hoist arrangements include wire rope, wire rope over hydraulic, direct-connected hydraulic, and wire rope with bull wheel or friction sheave. The designer should anticipate the impact to machinery caused by high water events and provide designs that afford protection under the most extreme high water conditions anticipated. The hoist components at either side of the lock may be mounted overhead or are contained within operating galleries in the lock monoliths. The powered machinery for operation of each side of the vertical gate may be independent or commonly connected through a single drive unit and mechanically connected to drive both sides equally. Independent machinery units must have more sophisticated controls and be designed to operate as a master-and-slave unit to control gate skew during operation.

(3) An example of a single drive unit arrangement to raise emergency gates consists of a double-grooved rope drum driven by two stages of open spur gearing, a herringbone or helical gear reducer, and an electric-drive motor with a spring-set, magnet-release holding brake. The rope drum has several layers of rope. One rope from the double drum attaches to one end of the gate through a multipart reeving. The other from the drum crosses the lock through a tunnel in the gate sill and passes through a multipart reeving that is attached to the other end of the gate See Plates B‑55 and B‑56. The two drums wind both ends of a continuous cable that lifts the gate through a series of sheaves, the number of which are selected to give the mechanical advantage desired. Two of the sheaves mounted on the gate work to equalize the line pull, in the event one drum winds slightly more cable than the other. Each drum is precision grooved so that each winds the same amount of cable on each layer. Where the fleet angles of the cable approaching the drum exceed 1.5 deg, a fleet angle compensator must be provided.

(4) The hoist machine should be located adjacent to the gate and in line with the hoisting sheaves. The hoist should be enclosed in a small protective building. Plates B-53 and 54 show typical hoist arrangements for multi-leaf gates.

(5) The hoist components generally are mounted on a structural steel frame that is anchored in various ways to the lock wall or a concrete structure. Each leaf is raised by its individual hoist, mounted side by side on the lock wall. The hoist structure is of such height that the machinery will be above high water. Plate B-54 shows a typical hoist arrangement. Plate B-58 shows a wire rope over hydraulic hoisting.

7-3. Vertical Gates for Water Regulating and Protection Structures.

a. Tide/Hurricane Gate. The tide or hurricane gate is a single-leaf, vertical lift gate that is used in part of a flood protection structure or coastal lock. When they are used as hurricane gates, they are kept in the normally raised position to permit navigation traffic to pass underneath and are lowered to protect harbors from tidal storm surges. Plate B-49 shows this type of gate. Operating machinery for this style of gate must be robustly designed for the anticipated storm conditions. The designer should anticipate the machinery could remain idle for long periods without use or regular exercising operation. Reliability and simplicity are important considerations. Manual methods to lower these types of gate by gravity operation should be considered in the design. Hoisting machinery for this type of gate is normally similar to that of overhead gates described in Paragraph 7-2. A direct-connected hydraulic vertical storm gate is shown in Figure 7-7.

Figure 7-7. Direct-connected hydraulic vertical storm gate

(1) Criteria for the design of tide gate machinery are the same as those for the emergency gate machinery, except the gate must be capable of being raised or lowered

against a differential head and against a force created by wind on the exposed section of the gate. To clear traffic passing under the gate, the gate must be raised a greater distance than either of the emergency-type gate leaves. Therefore, the lifting speed should be 5 ft/ min to 10 ft/min or a speed sufficient to permit opening the gate in approximately 10 min. Wind load on the exposed section of the gate should be assumed to be 20 psf for machinery design, unless warranted by locally higher speeds or storm conditions.

(2) The machine used for raising this type of gate consists of a dual-drum cable hoist mounted adjacent to one of the lifting towers. The two drums are driven by a pinion gear between the two drums. A triple-reduction, enclosed gear unit drives the pinion. The gear unit is driven by a two-speed electric motor with a double-ended shaft. A magnet-type electric brake is provided between the motor and reducer. The motor shaft extension permits the connection of a hydraulic emergency lowering mechanism. The low speed of the motor is used when starting and stopping the gate. The gate normally is lowered by means of the electric motor; however, in the event of a power failure, the gate can be lowered by means of the hydraulic mechanism. The emergency hydraulic lowering mechanism consists of a radial piston-type hydraulic pump connected to the electric motor shaft extension, a flow control valve, oil cooler, check valve, and necessary piping, all connected and mounted on an oil storage reservoir. When lowering without electric power, the weight of the gate, acting through cables and reduction gearing, turns the hydraulic pump. Oil from the pump is circulated through a flow control valve, creating a transfer of energy to the oil in the form of heat. Excess heat in the oil is removed by a tubular type oil cooler. This is discussed further below.

b. Spillway Crest Gate. The vertical lift spillway crest gate sometimes is preferred over tainter gates because the spillway crest requires a shorter length of spillway pier and provides a more economical pier design. These gates usually are raised by using mobile gantry crane or fixed hoists for each gate located on the spillway deck or operating platform. Dogging devices are sometimes are provided to engage projections spaced at intervals on the gate to hold the gate at the proper elevation. In some cases, it might be advantageous to mount the dogs in the gate and provide a dogging ladder in the gate slot. However, the earlier arrangement is more common and preferred. Different types of spillway crest gates include the single-section and multiple-section gates.

(1) The single-section gate consists of one section that provides a variable discharge between the bottom of the gate and the sill. Single-section gates operate similarly to the multiple-section gates but are dogged off in the service slots.

(2) A multiple-section gate consists of two or more sections in the same slot with variable discharge between the sections or between the bottom section and the sill. Multiple-section gates may be equipped with a latching mechanism to allow use as a single-section gate. As the required discharge increases beyond the capacity of the largest opening between sections, top sections are removed from the service slots and dogged above the pool level in emergency slots. The latching mechanisms should be designed carefully so they do not stick or corrode. Latching mechanisms have proven to

be a maintenance problem for some projects. Figures 7-8 and 7-9 show a single-section, vertical crest gate and multiple-section, radial crest gate.

Figure 7-8. Single-section, vertical crest gate

Figure 7-9. Multiple-section, radial crest gate

c. Double-section Gate. This gate consists of two sections in adjacent slots with variable discharge over the top section or beneath the bottom section. The double-section gate is used less frequently because removing the gate from the slot is more cumbersome, sealing is more complicated, and additional length of pier is required. This type is useful for skimming ice and trash; however, that function also can be performed by shallow top sections of a multiple-section gate that are lifted clear of the pool.

d. Outlet Gate. Lift gates often are used for emergency closure of water intake systems or outlet works. They normally operate in the open position. They are not used for throttling flows, rather to stop flow under operating conditions. They rest on dogging devices during normal operation. In emergencies, they are lowered into the closure slot to stop the flow of water.

(1) Emergency gates are required for sudden closure of the turbine intakes to prevent subsequent damage to the turbines or powerhouse. The hoisting system uses either hydraulic cylinder(s) or wire ropes. The type of hoisting system will be based on economics and governing criteria for closure times under emergency conditions. The hoisting system for wire ropes may be deck mounted or placed in recesses above the high pool elevation. Cylinders for the hydraulic system are mounted below the deck in the intake gate slot. Because these gates must be capable of operating under full head and flowing water, tractor-type gates are used to reduce friction. See EM-1110-2-2105 for more on structural design parameters of this style and end supports. See Figures 7-10 and Plate B-59 for examples of a tractor-style gate and wheeled track roller-style gate.

Figure 7-10. Tractor-style gate

(2) Emergency closure gates for outlet works are similar to those used in powerhouses and often are used for service gates and flow control. Tractor gates for fully submerged outlet works are usually more advantageous for use due to the reduced

friction under full head and flow. However, wheeled vertical gates often are used where loading allows. The hoisting system might require the use of a gantry crane or its own hoisting system, either wire rope or hydraulic.

7-4. Operating Equipment for Vertical Lift Gates. The design of operating equipment for vertical lift gates requires close coordination between the structural and mechanical engineers to determine the operating equipment loading. The general design criteria for tainter gates, found in paragraph 9-2.m.(5), also may apply to vertical lift gates. The mechanical engineer will need gate deadweight, hydrodynamic loads (both horizontal and vertical), ice load, silt load, sliding or rolling friction load, and side seal friction load to determine the total gate load imposed on the operating equipment. This total gate load also should include any known inertial effects. The total load then is applied to the general machinery layout design to determine the individual component loads for sizing of the machinery. Examples of load calculations are in Appendix C. Further discussion and design details are in EM 1110-2-2105.

a. Wire Rope Hoists. As already described in this chapter, vertical lift gates commonly use hydraulic or wire rope hoist systems. Wire rope hoists are used for spillway crest, outlet, and navigation lock gates. Wire rope hoists are more suitable for gates that have deep submergence requirements, installations that do not allow portions of the hydraulic cylinders above the deck (shallow settings), or when hoisting loads are too large and economics makes hydraulic cylinders impractical. Wire rope hoists consist of drums and a system of sheaves and blocks that are driven through a motor and arrangement of shafts, speed reducers, and spur or helical gears. Motors may be electric or hydraulic driven. It is common to provide two speeds to permit lowering at approximately twice the raising rate. The hoisting equipment normally is located next to the gate or slot, with controls located in the control room, governor control cabinets, or next to the navigation lock gate, depending on the gate and its intended use.

(1) Bull wheels are used in overhead lift gates as a friction drive for hoisting the gate. The bull wheel, motor, and gearing system are located in a tower, high enough to raise the gate to its full and open position. The wire ropes wrap over the top of the bull wheel in grooves with one side of the wire ropes connected to the gate and the other end to a counterweight. The motor and gear system provide the mechanical effort required to hoist the gate. This type of drum system is advantageous when the hoisting loads are large.

(2) Counterweights are used mainly in overhead-type gates to offset the dead load of the gate to minimize the hoisting effort. The weight of the vertical lift gate will determine the mass of the counterweight required. It should be designed to compensate for adjustment of its mass to calibrate it with the weight of the gate once the system is in place. It is normal to have the gate/counterweight slightly unbalanced to allow the gate to close without power. Another method for reducing the lifting effort is with a multi-reeve system through a series of drums and sheaves. The number of sheaves and arrangement are selected to give the desired mechanical advantage for reduction in the size of wire rope and machinery components.

(3) Electric motors are the primary drives for wire rope hoist systems. Guidance for design can be obtained from sources referenced in paragraph 7-1 and from Guide Specification UFGS 35 01 42.00 10, Vertical Gate Lift Systems, and UFGS 35 20 20, Electrical Equipment for Gate Hoist.

(4) Guidance for specifying brakes is provided in Guide Specification UFGS 35 01 42.00 10, Vertical Gate Lift Systems, and UFGS 35 20 20, Electrical Equipment for Gate Hoist. See Chapter 2 for brake design guidance.

(5) Open gearing is often necessary in hoisting systems to achieve the required reduction to minimize motor torque and horsepower requirements. Open gearing is typically of the spur, herringbone, helical type as required to develop the necessary loading requirements. Design guidance for open spur gears is included in Chapter 2.

(6) The speed reducers shall be designed, rated, and manufactured in accordance with applicable AGMA standards noted in Chapter 2 and listed in Appendix A. In all cases where the standards might conflict with one another, the designer shall select the more conservative design standard. See Chapter 2 for speed reducer design guidance.

(7) Bearings are necessary for all load-carrying rotational movement and to allow for additional degrees of freedom to prevent binding or unwanted lateral loads. Bearings may be of the roller, self-aligning spherical roller, ball, sleeve, or greaseless type as necessary. See Chapter 2 for design guidance on bearings.

(8) Shaft couplings for vertical gate systems are recommended to be of the flanged exposed bolt, double engagement, gear type made of forged steel. Couplings of this style may be installed in either the vertical or horizontal orientation, depending on the shaft on which they are mounted. See Chapter 2 for further guidance on the description and selection of shaft couplings that might be applicable for use in vertical lift gate systems.

(9) Torque limiting couplings may be used in designs to prevent motor over-torque in vertical gate-hoisting arrangements. The designer may refer to UFGS 35 01 42.00 10, Vertical Gate Lift Systems for specification guidance and Chapter 2 for design guidance on torque-limiting couplings.

(10) Wire ropes for vertical lift gates are commonly specified to be 6 x 37, pre-formed, lang lay, independent wire rope core, 18‑8 chrome nickel corrosion resisting steel. A 6x19 wire rope construction in the arrangements of 6x25FW and 6x26WS also are specified for vertical lift gates. They provide increased abrasion resistance compared to the 6x37 construction that has superior flexibility for bending fatigue. The type of wire rope lay should be selected by the designer to accommodate the reeving arrangement and style of sheave or drum the wire rope will engage. The factors that must be considered when selecting a wire rope include: wire rope breaking strength; resistance to bending of vibration fatigue; resistance to abrasion and crushing; reserve

strength; and factor of safety for anticipated shock, acceleration/deceleration, corrosion, and environment.

(a) To stop the gate drive motor and to engage the holding brakes, limit switches should be installed in the event the wire rope cables become slack.

(b) The designer is strongly encouraged to utilize EM 1110-2-3200, Wire Rope Selection Criteria for Gate Operating Devices, for wire rope selection and design of hoist operated systems for vertical lift gates.

(c) Additional useful literature related to an actual case study of a wire rope failure within the Corps is provided in the John Day Navigation Lock Upstream Lift Gate Wire Rope Failure Investigation Report, DACW57-01-D-0009, February 2003, Portland District.

b. Hydraulic Hoists. Hydraulic hoists normally consist of a single acting cylinder, pumps, reservoir, controls, and piping. More recent applications use telescoping cylinders to accommodate deep submergence gates. One or two cylinders may be used. The number of cylinders is determined by the hoisting requirement and economics. The arrangement may include the cylinder supported above the gate, with the gate and cylinder rod hanging from the piston, or the cylinder recessed within the gate. When the cylinder is above the gate, both the gate and cylinder bottom may be designed to have removable dogging pins or beams. When the vertical gate is dogged off, the cylinder can be extended to raise the cylinder bottom connection to a higher elevation to perform a repeat lift of the gate. This arrangement allows for higher gate-lifting capabilities and helps to minimize cylinder stroke length. Further guidance for the selection, installation, maintenance, and inspection requirements of hydraulic cylinders and components are in Chapter 3.

c. Roller chain hoist arrangements consist of the lifting chain, drive and idler sprockets, drive machinery, and counterweight. The roller chains are located in recesses in the lock wall. Roller chains are flexible about an axis parallel to the lock center line and rigid about an axis perpendicular to the lock center line. Near the top of each recess, the lifting chain is redirected to the drive sprocket by an idler sprocket. The drive sprocket is located in a recess below the top of the lock wall. Beyond the drive sprocket, the lifting chain continues to a second idler sprocket at the top of a counterweight chase. From the second idler sprocket, the lifting chain extends vertically to the counterweight. The chain connection to the gate leaf is a 3D gimbal. The gimbal allows rotation about the axes both parallel and perpendicular to the lock center line. Rotation of the connection point is allowed to prevent the lifting chain from being bent about its rigid axis when the gate leaf rotates. The connection points on the gate should be located at the end portions to coincide with the approximate center of gravity of the gate. The drive machinery, located in a watertight recess at the top of the lock wall, consists of an electric motor, open gear sets, and reducers. The main advantage of the roller chain drive arrangement is the positive drive connection over the drive sprocket. This arrangement also does not require the larger space of a cable drum.

Disadvantages include relative high cost of chains, frequent maintenance for lubrication, corrosion, and critical alignment required between sprockets.

7-5. Lift Gate Design Components.

a. Dogging Devices. Dogging devices (dogs) are recommended for vertical lift gate designs. The dogging devices have mechanical components that either manually or automatically can be engaged, depending on the application and frequency of gate operation. The dogging devices provide positive reassurance that gates are in a secured position before other operations are allowed. They also provide a means to remove tension in the load-carrying components of the hoisting machinery. This allows maintenance to be performed while the vertical gates are in the dogged off positions. The designer should coordinate with the structural engineers for method and location of gate engagement or support.

(1) Dogging mechanisms usually are mounted on grillages in the piers recesses at opposite end posts of the gate. They generally pivot or slide to permit insertion and retraction to provide clearance of the operated gate. Two or more dogging positions at each end of the gate slot might be required. The number and location of the dogs are determined by the operating requirements for discharge regulation and gate storage. The gate sections require dogging seats fabricated with structural or cast steel, welded or bolted on the end posts. Welded fabrications for dogging devices should be coordinated with structural engineers for design and welded, in accordance with AWS D1.5.

(2) One method of dogging consists of a horizontal pin that moves into pin plates attached to the top of the gate. The pin should be so arranged that it can be operated from the control station of the gate. Instrumentation should be provided to show when the dogging pin is fully engaged or fully released. See Plate B‑51 for suggested details.

(3) Another type of dogging device consists of a cantilevered, mild-steel H-beam that retracts inside the gate at each end between the top and second girder web. The beam is located at the center of gravity of the gate in the upstream/downstream direction and runs through the end post to a reaction point at an interior diaphragm. The dogging beam is extended and retracted by using a bar as a manual lever extending through a hole in the top web and into a row of holes in the top of the dogging beam. The cantilevered end of the beam rests on bearing pads recessed in the piers. This type of dogging device is preferred for powerhouse gates and bulkheads because they also can be dogged at the intake or draft tube deck level and there are no mechanical devices to be lubricated or maintained.

(4) Where the gate is lifted above the lock wall, the designer should design all components for dogging devices on both the gate and the structure to support two times the calculated full gate load, to allow for impact loading. This should apply to all related pins, bolts, and anchor bolts.

(5) In dogging arrangements where the gate is not lifted above the lock walls, the beams should be designed for shear and moment, using 50% impact for the applied loading. Stiffener plates should be used on each side of the support beam web under the support brackets of the gate and at the reaction points of the support beam.

b. Lifting Beams. Lifting beams normally are provided for outlet gates and maintenance bulkheads. Because these gates usually are stored in a submerged condition, the lifting beam provides a latching and unlatching mechanism to lift the gate from the slot. Design guidance for lifting beams is in EM 1110-2-4205.

c. Guide Tracks. Tracks for vertical lift gates usually are incorporated into the guide system, with the track itself consisting of a corrosion-resisting plate. Corrosion-resisting plate use is especially important in brackish or saltwater environments. In freshwater applications, the bearing plate or track may be of structural steel with a cladding of corrosion-resisting material on the exposed surface. The guide system for a vertical lift gate on each end of the gate consists of two bearing or track plates (upstream and downstream) and an end guide plate. The bearing plates are so arranged that the wheels or bearing plates of the gate react against the bearing plates of the guide system. The system is arranged so that the gate can be loaded from either side and the bearing plates will remain effective. The end bearing plates are similar to the reaction bearing plates, but are placed so that bumpers on the end of the gate will strike the end bearing plate and prevent excessive lateral movement of the gate in relation to the lock or structure slot. The end guide or bearing plate should be of the same material as the bearing or track plates, using the same criteria to determine the use of corrosion-resisting steel, clad steel, or standard structural steel. To minimize the effects of the guide system on the support towers, the system should be connected to steel towers only at panel points of the structure.

(1) Mechanical designers must be aware that some guide systems for hydraulic cylinders and gates have close tolerances between plates to ensure proper alignment of the system. In some applications, the mechanical engineer must provide recommendations for machined plate surface finishes and running clearance fits for the system. The normal clearances for a gate should allow for not more than 1 in. of total movement between the gate and bearing plate and not more than 0.5 in. between the gate and end bearing plate. See Plate B-49 for suggested details of a guide showing the recommended clearances. In systems with guided cylinder movement, much tighter tolerances might be required to ensure no eccentric loading is imposed on the cylinder rod, for protection of the hydraulic cylinder seals. It is recommended to further mitigate the risk of eccentric loading by mounting the cylinders in cardanic rings or other such arrangements which allow rotation of the cylinder as bearing plates and rollers wear.

(2) The bearing plate or track is attached to a suitable support member. The support members are normally standard rolled beams or sole plate with embedment straps. The support member is embedded and anchored in the concrete wall or attached to the tower at tower panel points. Consideration should be given to make the bearing plate or track easily replaceable to compensate for service life wear in the

bearing surfaces. The allowable surface variation between non-continuous plates must be specified to ensure the gate or cylinders shall pass by without catching.

7-6. Lift Gate Design Considerations and Criteria. Machinery components' general criteria, applicable to all types of operating machinery covered in this chapter, are presented below.

a. Motor Torque. The required torque of the vertical lift gate hoist motor should be the root mean square value of torque versus time curve for operation of the gate with the motor selected having a 1.15 service factor. The peak torque required should not exceed the service factor of the motor. The normal hoist load for the gate leaf will be the loads resulting from the required torque of the motor. The hoist motor should have torque characteristics conforming to Guide Specification UFGS 35 01 42.00 10, Vertical Gate Lift Systems, and UFGS 35 20 20, Electrical Equipment for Gate Hoist. In some applications, it might be desirable to consider variable speed (AC or DC) hoist motors with a ramping function adjustable through the drive controllers.

b. Motor Hoist Load. The factors of safety from Chapter 2 shall apply in determining the vertical gate hoist loads. Both a normal hoist load and a maximum overload condition shall be calculated by the designer. The normal hoist load shall be considered as equally divided between the two drives of the hoist. The magnitude of the maximum overload condition will depend on the type of hoisting system. The factors of safety from Chapter 2 shall apply to the maximum overload condition. Specifically, for the overload condition, unit stresses shall not exceed 75% of the yield stress of the material, and wire rope loads shall not exceed 70% of the nominal breaking strength.

(1) For hydraulic cylinder hoists, the system should be designed to support and lower the entire gate from one side. The maximum overload condition would be limited by the hydraulic system relief valve setting.

(2) For systems consisting of synchronized but independent electric motor hoists for each side of the gate, the maximum overload condition would be the forces created by the locked rotor torque of each motor applied to each side of the gate.

(3) For single motor non-equalizing hoist arrangements, the required loading for the maximum overload condition will be the forces created by the locked rotor torque of the motor applied one side of the gate.

(4) For vertical gates with a single motor and a means to equalize the hoist loading, the locked rotor torque of the motor will be considered as equally divided between the two drives of the hoist.

(5) For emergency-type gate machinery, force control switches may be used to limit the rope pull under stalled conditions and, thus, reduce the loads on the machinery components. However, the machinery must still be designed to accommodate the design overload conditions noted above.

c. Roller Wheels. Wheels for the underwater gates are a critical item and should be designed for individual conditions. A gate being raised with a considerable horizontal load caused by flowing water would have considerable deflection at the ends. To avoid point contact of the wheels on the flat plate track caused by gate deflection, the wheels should be constructed with a crowned, hardened tread. A method for designing a wheel subject to gate deflection is shown in Appendix C. This method was developed utilizing formulas from Roark and Young (1975). The formulas in the fifth edition also may be used, except the formulas for computing maximum compressive stress are wrong. The formulas give the maximum compressive stresses occurring at the center of the surface of contact, not the maximum shear stresses. The maximum shear stresses occur in the interiors of the compressed parts. The formulas do not provide the maximum tensile stress, which occurs at the boundary of the contact area and is normal thereto. Due to the flexure in the gate, it is difficult to determine accurately the distribution of load on the gate wheels. However, it is considered satisfactory to design the wheel tread for a maximum compressive stress equal range of 2.0 to 2.5 times the yield strength of the material involved. This calculation is based on the maximum wheel load from the gate. A slight misalignment of the track surfaces will prevent a wheel of the gate from bearing on the track for short distances of travel. This can cause an overload on some of the adjacent load-carrying wheels. This condition should be taken into consideration when determining maximum wheel load. An option for the crowned wheel, to compensate for gate deflection at the ends, would be to use flat wheels with self-lubricating, self-aligning spherical bushings. These are available in many bearing and lubricant combinations to suit a variety of applications. Self-lubricating, self-aligning spherical bushings have been used successfully in nuclear offshore, industrial, structural, and dam applications.

d. Hydraulic Fluid Brake. A hydraulic lowering brake should be used in the design, to lower vertical tide or hurricane gates when reliable utility power or backup standby power is not available. The vertical tide gates normally are lowered by an electric-drive motor on the hoist, with a diesel electric generator set standing by in the event of power failure. In cases where this is not reliable or available, the gate may be lowered by coupling a hydraulic motor to the shaft extension of the electric-drive motor. This fluid motor is connected in an oil circuit, which permits free flow of the oil in the raising position but restricts flow in the lowering position. A typical circuit required for this operation is shown in Figure 7-11. The flow control valve used in this circuit should be designed and adjusted in the field to limit the speed of the electric motor to about 140% of its synchronous speed, so not to damage its windings or rotor. The flow control valve and fluid motor shall be sized so that the pressure of the oil leaving the motor shall not exceed the normal working pressure rating of the fluid motor. When lowering the gate, approximately 10 min might be required. During this time, the braking energy will be transformed into heat in the oil as it passes through the flow control valve. A shell-and-tube-type heat exchanger must be provided in the circuit to prevent the temperature of the oil in the tank from exceeding 120°F. A cool, clean potable water or raw water source may be used in the heat exchanger to cool the oil. A thermostatically controlled valve may be used to automatically control the flow of water through the heat exchanger. The water can be exhausted to drain after use and should be capable of being winterized during periods of non-use.

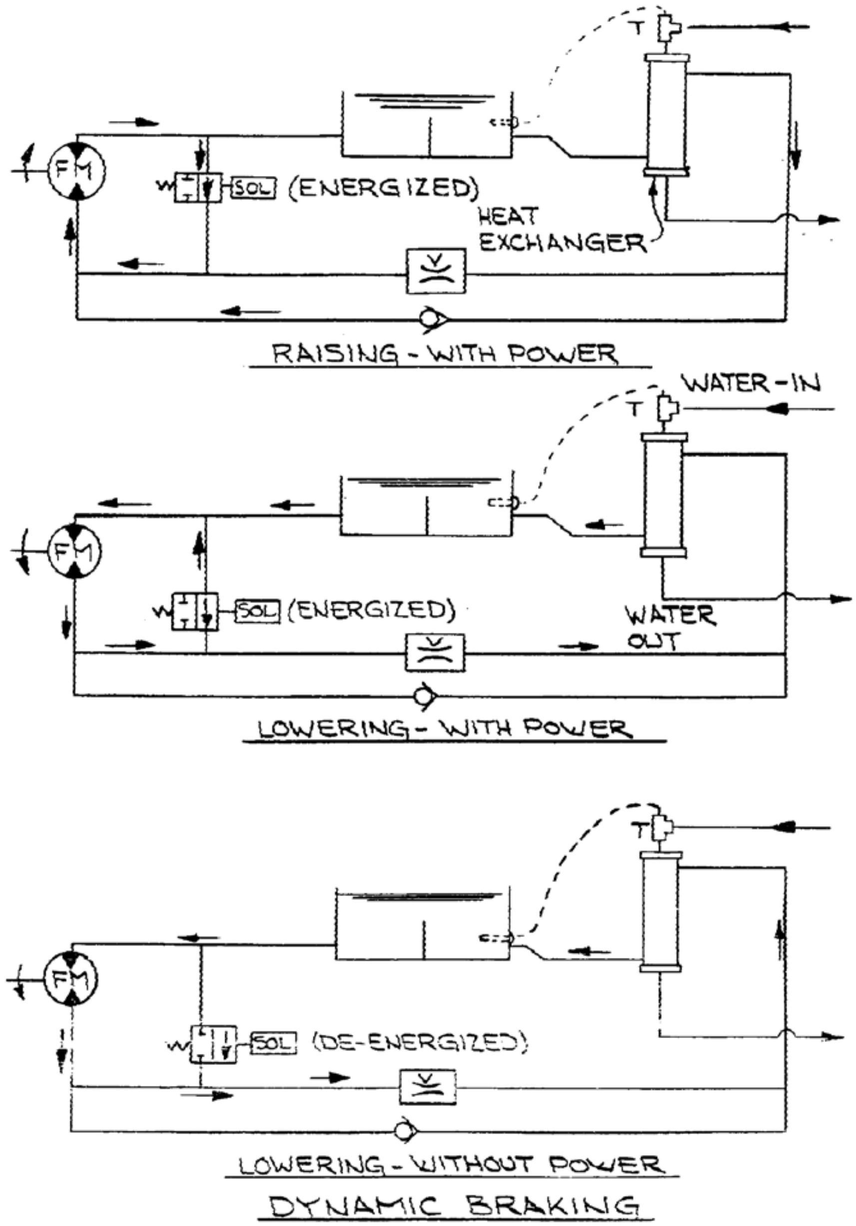


Figure 7-11. Typical circuit requirements for raising and lowering a vertical tide gate

e. Gate Hoist Loads. Typical calculations for determining loads for design of emergency gates are shown in Appendix C. The hoist design load for balanced head condition gates, or when the lower pool is 1 ft or less below the upper pool, will be the dead weight of the gate leaf in air, side seal preset force, weight of trash screens, weight of silt load (when raised to maintenance position), or the summation of the following, whichever is larger:

- Weight of gate leaf minus weight of water displaced;
- Silt load amount trapped by flanges less weight of water displaced;

- Sliding friction due to horizontal force caused by 1.0-ft swell head. The coefficient of friction for this condition should be assumed as 0.40 for steel on steel;
- Downward hydrostatic load due to 1.0-ft swell head;
- Weight of recess protection and trash screen minus weight of water displaced.

(1) Loads used for design of vertical tide gates are similar to the loads used for vertical emergency gates, except the wind load is a more critical factor. The gate is hoisted high above the structure, permitting barge traffic to pass underneath. This exposes the gate to a considerable wind load, which must be included. To find the hoist capacity, the following two conditions should be considered, with the one creating the greater load being used for design of the hoist.

(a) Condition I. Weight of gate leaf in water consisting of the skin plate, framing, sheaves and brackets, wheels, etc., and the weight of silt (125 pcf) trapped by the flanges of the gate girders less the weight of water displaced. Include the rolling friction load calculated to be 5% of the horizontal load on the gate caused by the largest combination of differential head. Finally, incorporate the wind load, calculated at 20 psf, acting normal to the gate’s surface for the exposed portion of the gate.

(b) Condition II. Weight of the gate leaf in air consisting of the skin plate, framing, sheaves and brackets, wheels, etc., and the weight of silt (125 pcf) trapped by the flanges of the gate girders. Rolling friction of 5% of the horizontal load on the gate caused by the wind load, at 20 psf, for the exposed surface of the gate.

(2) Silt Load. Additional weight due to accumulated silt should be added to the overall load of the hoisting machinery, if not furnished by the structural engineers. The silt can become trapped above the web of the girders to the height of the downstream flange. Calculate the weight from the possible storage volume on the gate, using a silt density of 125 pcf.

(3) Sliding Friction Load. Additional load on the hoisting machinery also is caused by sliding friction of the wear surfaces and seals against the stationary mating structure surface. A coefficient of friction of 1.0 should be used for seals where the rubber is against steel. The coefficient of friction for steel on steel in dry contact condition is 0.40. The designer should be aware that coefficients of friction vary widely, depending upon the materials in contact and the amount of lubrication between the surfaces. Lubrication can be any medium in contact on the surface of the material (i.e., water, grease, oil, moss, zebra mussels). The designer should perform his own research to decide the best coefficient of friction to use in calculations. This might require independent testing to determine the applied coefficient of friction in applications that are critical. Since frictional loads can be a substantial part of the overall gate load, the coefficient of friction testing to replicate in field conditions will help to make accurate calculations and avoid gross oversizing of the hoisting machinery.

(4) Roller Friction Load. The total friction due to the gate reaction rollers running against steel tracks and the friction of the bearings in the reaction rollers is much lower

than sliding friction surfaces, and shall be taken as 5% of the load normal to the gate leaf. Where self-aligning, spherical, greaseless-type bearing reaction rollers are used, the designer should consult with the greaseless bearing manufacturer to help determine the reaction roller-bearing friction.

(5) Hydrodynamic Load. A downward hydrodynamic force exists for all gates that must be raised through flowing water. This force can be obtained from the curve showing results of studies conducted by WES. This curve is provided in Appendix C.

7-7. Control System Considerations.

a. Control stations for vertical lift gates are usually adjacent to the gate, along with the hoist machinery. The control equipment consists of the combination of full-voltage magnetic controllers, limit switches, and control switches arranged to produce the desired operating sequence. Many of the older designs still used today incorporated traveling nut-type limit switches in heavy cast iron National Electric Manufacturers Association's NEMA 4-type enclosures. This style is no longer commercially available, and designers now are choosing rotary cam-style limit switches.

b. In applications with PLC-based systems, the designer might wish to incorporate electronic encoders or a linear variable differential transformer (LVDT). The designer must give caution to position feedback systems in hoisting systems that are not mechanically coupled to give 100% reliability in gate position. All electronic feedback instruments must have absolute positioning, in case power is lost. This is to ensure the gate's position and skew always are known by the control system. Slack cable limit switches or pressure switches and skew control should be used on vertical lift gates to prevent them from becoming racked or jammed in the slots. Gate leaf control should be coordinated with the electrical engineers and can be performed as indicated on the typical electrical schematic diagram for an emergency gate hoist shown in Plate B‑96.

CHAPTER 8

Buoyant and Other Gate Operating Machinery

8-1. General Description.

a. Introduction. Gate types other than those discussed elsewhere in this document have been utilized at navigation structures and storm barriers. Though less common and, therefore, lacking as thorough a design history and standardization, specific circumstances might prompt consideration of such gates for a particular project.

(1) The required gate driving and operating forces can be reduced substantially by letting the gate float during closing and opening. Buoyancy reduces the hinge and/or bearing forces. Buoyancy-based systems are utilized on a variety of gates including sector gates, rolling (pocket) gates, sliding gates, and swing-type gates. One significant design consideration is decreased control of a floating gate behavior during motion. Various gates of this type have been constructed and operate well today. The main application field of buoyancy-based gates is in storm surge and flood barriers. The primary reasons are long closing and opening times are available and high leakage is acceptable in the gate closed position. However, there are multiple navigation lock gates that employ buoyancy to help reduce operating forces. These are typically either sector gates or rolling gates. Sector gates are discussed in Chapter 5.

(2) Buoyancy can be accomplished using water tanks or air tanks. There are advantages and disadvantages of both systems, as discussed below. Water-filled tanks require a pumping system to fill and empty the tanks. Air-filled tanks allow the gate always to be buoyant in either the open or closed position. The tanks, however, can be prone to leakage, at which time buoyancy will be lost.

(3) Another example of a buoyancy-based sector gate is the drive system of the world largest storm surge gate of the Maeslant Barrier in Hoek van Holland, Netherlands (Figure 8-1). The gate is of a sector type with a ball hinge. It is fully buoyant during opening and closing. In the nearly closed position, the buoyancy tanks fill with water. As this happens, the gate sags, then settles on its bottom sill. The drive itself consists of a locomotive winch system and a drive track on the top edge of the gate. To avoid collision due to long translate waves, a vertical gap of up to 1 m (3 ft) must be left open. However, such openings (unacceptable in lock gates) usually do not present a problem for storm surge barriers.

Figure 8-1. Buoyant sector gates of the Maeslant Barrier, Netherlands, in closed position (courtesy of R.A. Daniel, Rijkswaterstaat, Division of Infrastructure, Netherlands)

b. Barge Gates. Floating leaf swing gates or barge gates have been used, with some success, as storm surge barriers across navigable waterways. They typically are constructed from concrete or steel. The gate consists of a floating structure swung into position across the channel way and sunk into place, sealing along the bottom and against receiving structures on either side. The hinge assembly might be little more than a simple mooring point. Operating machinery, in theory, could consist of any means employed to manipulate barges; however, it typically is some form of rope and capstan or winch and cable/chain system to pull the barge into position. Ancillary equipment includes a valve-and-pump system to scuttle and re-float the barge. Barge gates can be quickly fabricated to provide economical closures, though the operation tends to be somewhat slow and cumbersome due to the limited control of the gate when buoyant, with imprecise positioning contributing to difficulty in producing complete sealing. As such, they are not considered appropriate for lock closures.

c. Rolling (Pocket) Gates. The rolling, or pocket, gate is a rectangular structure stored in a recess along the lock wall. The gate is extended and moves transverse to the lock across the waterway to a recess on the opposite side. These types of gates are used extensively in Europe. Rolling gates are partially buoyant and supported by wagons that then ride on a railroad track. Operating machinery may consist of a cable and pulley system, hydraulic cylinder, or rack and pinion or similar wheel-and-track type system. All or portions of the rail systems require submerged mechanical components.

d. Drawbridge Gates. Mechanized vertical flap gates or drawbridge gates are made of a rectangular gate leaf hinged at the bottom that is stored flat in a submerged recess on the channel bottom, allowing navigation. The gate is rotated into the vertical position by its operating machinery closing off the channel. At least one example of this type of gate, Empire Floodgate, designed in 1973, is in operation. This gate design provides an economical option for a navigable channel closure, though does suffer some drawbacks, specifically in operation and maintenance. This gate is discussed further below.

e. Drop Leaf Swing Gates. Drop leaf swing gates are essentially a multiple-leaf, vertical lift gate, where the top leaf doubles as a rigid derrick from which the other leafs are lowered, which can be rotated out of the waterway to pass navigation with no vertical clearance issues. The top leaf typically would extend from the top of flood protection to a point slightly above the water surface level expected when the gate is intended to be operated. This type of gate was proposed by New Orleans District as an alternative floodgate design for smaller navigable channels requiring significant vertical clearance and having operating water levels that are substantially below expected flood levels. This description is characteristic of the many small waterways passing through south Louisiana hurricane and storm surge protection, where shrimp boats with their tall booms are prevalent. This gate is discussed further below.

8-2. <u>Barge Gates</u>. Swinging leaf barge gates have been incorporated into hurricane protection projects in the New Orleans area and by non-federal entities in protection rings around the areas near Houma, Louisiana. Figure 8-2 shows a barge gate for the New Orleans District Inner Harbor Navigation Canal (IHNC) surge barrier. Discussion in this section will focus on the swing leaf arrangement. Barge gates generally are chosen as an economical alternative when other traditional gate designs become restrictive due to cost of construction or required schedule. Also, for very large structures the benefits of positive buoyancy become more attractive; however, the inherent control and operational issues also become more apparent. Operation of this type of gate might be somewhat more involved than the typical push-button scenario common to most modern automated lock-and-dam equipment. Barge gates might require an operating staff of eight to 10 people, and might take several hours to close and scuttle in the proper position depending on the nuances of the installation. Barge gates and their receiving structures can be constructed quickly, often in the wet with no cofferdam, reducing costs and facilitating short schedules.

Figure 8-2. New Orleans IHNC surge barrier barge gate

a. Design considerations. Barge gates are essentially a floating vessel (ship). Barge gates can be operated only with equal water head on both sides of the gate. Hydraulic loading during operation is primarily an issue of current rather than head differential across the gate. Similar to a miter gate operating under a differential head, the large surface area exposed to this force would make the gate difficult to control. As such, these gates are practical only when a nominal head difference is expected. Even in this case, the hydrodynamic loading induced by such a head difference is the primary driving force on the sizing of the operating equipment.

(1) The principal operating equipment for barge gates tends to be a line pull system from the barge to either its closed or its stored location. Both chain and cable may be utilized. For smaller gates that have means of transporting a line across the channel, a cable or rope may be used with a winch or capstan. For larger applications, where the size of the line is prohibitive or it is impractical to run it across the channel, the line may be left permanently along the channel bottom. Typically, chain or corrosion-resistant cabling is preferred in this case. This approach also exposes the cable to damage or snagging by passing vessels. Modern high-strength, lightweight synthetic hawser lines might provide a more manageable alternative to steel cable or heavy chain and allow for onboard storage of the operating line. Capstan systems should employ low-stretch ropes, and might need to employ a reversing mechanism to back off the line

if it binds. Winch systems should employ a level wind system or be run to a sheave, preserving the necessary fleet angle for proper cable winding. As chains become corroded and lose cross-sectional area, this will affect the ideal spacing for chain wheel or wildcat systems, and could lead to jumping or popping of chain as it nears replacement.

(2) Scuttling and pump systems must be versatile enough to maintain proper trim while raising or sinking the gate structure in a controlled manner without inducing instability or upending the barge gate. Typically, this is done by dividing the buoyancy area into multiple independent chambers, with individual control over the filling or emptying monitored by operators or an automated system. Re-floating of the structure typically is accomplished by evacuating the buoyancy chambers by pump. Sinking typically would be achieved by opening scuttle valves and flooding the chambers. Scuttles placed flat to the bottom might induce the accumulation of sediment into the buoyancy chambers. In extreme conditions, pumps might need to be utilized to avoid excess sedimentation of the barge chambers by taking in surface water. For floodgates, where the water level can be expected to rise on one or both sides of the gate after the gate is closed, scuttling by means of valves alone may not be sufficient. Operational conditions may require flooding the ballast chambers to a greater depth than can be achieved by valves alone, since valves alone can only flood the ballast chambers to the same depth as the water outside the barge. For this condition, pumps can also be used to fill ballast chambers when surrounding water levels are too low.

Figure 8-3. New Orleans IHNC barge gate set in position

(3) Large barge gates might employ a substantial amount of piping, valves, and pumps. It is recommended that the barge be treated as a vessel and American Bureau of Shipping standards be followed for items such as below-deck piping. Adhering to these standards facilitates transport of the structure to marine repair facilities for maintenance.

(4) Seals are typically block or bulb rubber, usually on the barge itself, with the sturdier mating surface exposed to channel traffic. The seal arrangement may be completely on the vertical protected side face of the gate, or may extend along the bottom of the gate. Benefits of extending the seal on the bottom, and usually toward the flood side, are that the gate weight aids in compressing the seal, and the relative uplift on the gate are based on landside water levels across the sealed portion of the gate bottom. Vertical face seals typically require a raised face across the channel bottom to seal against, and rely on the buildup of differential head, load-binding, ratcheting-type devices to achieve seal compression. Bottom seals, however, are much more susceptible to debris or siltation impeding the seal, thereby negating their advantages.

(5) Hinge design typically consists of some type of mooring point that roughly positions the gate in the proper location as it swings closed. Hinge design must accommodate changes in elevation of the barge gate as it is scuttled and in the tides while open. The hinge may be as elaborate as a steel collar with internal rubber bumpers around a piling, or as simple as a chain to a pad eye or rope to a bollard. In either case, the hinge system must accommodate the required movement of the gate, as well as enough repeatability in positioning to mate the seals to their respective seal plates, while resisting the loads imposed by currents, operating equipment, and even the inertia of the barge gate itself.

8-3. Rolling (Pocket) Gates.

a. Rolling gates have been used in Europe for more than 100 years and were common in the United States in the early 20th century. These gates generally are called rolling gates in Europe. In the United States, the term pocket wheel is more commonly used. Modern gates of this type typically are employed for large lock structures in the range of 33- to 55-m-wide (110- to 180-ft-wide) closures. Currently, the largest lock in the world is at Berendrecht Lock, Antwerp Belgium, with rolling gates operating in a 68-m-wide (220-ft-wide) lock. Based on the history of this design type on structures of similar size, rolling gates were selected for use on the new 55-m-wide (180-ft-wide) Panama Canal lock gates. Rolling gates typically are designed for operations under minimal differential head conditions in the range of 10 cm (4 in.), which is adequate for lock gate installations. Rolling gates are designed as buoyant to reduce the operating loads. The use of rollers and buoyancy chambers on these types of gates necessitates the submergence of some of the mechanical components necessary for operation. Maintenance of these items, with the exception of the cross-channel track, is facilitated by the ease of dewatering the recess bay, granting dry access to the gate structure. Pumping ballast water out of the structure's buoyancy tanks allows it to be floated and supported before dewatering. Due to the weight of the larger single gate and rolling friction, operating machinery is typically larger for an equivalent miter gate or sector gate.

Figure 8-4. Zandvliet Lock, Belgium, rolling gate in closed position

b. Design considerations. In addition to required forces that will be applied by the operating machinery, track loading, wear, accommodations for lateral gate movement, and debris management are important in the design. Hydrodynamic loading is typically from small tidal fluctuations or current from leakage through the opposite lock gate, as well as wave- or wake-induced loading. Typically, some lateral movement is allowed of the structure by its support system, permitting it to shift into its sealed position. The support type must accommodate this, and the effects on track loading must be taken into consideration.

(1) Principal operating equipment is usually a mechanically driven cable system. A system would consist of a pair of cable drums located on the recess structure, at the end away from the channel, with a continuous cable winding and unwinding off the top and bottom of each drum (Figure 8-5). The cables would be looped through sheaves on either side of the channel end of the recess and fixed to the upper gate carriage. Winding the drums in either direction would open or close the gate. Rollers or guide pads are used to support and guide the cables. Rack-and-pinion drive systems also are used. For smaller gates of this type, it is conceivable that a hydraulic cylinder would be practical; however, the large range of motion required for typical applications generally would preclude this.

Figure 8-5. Rolling gate machinery, Van Cauwelaert Lock, Antwerp, Belgium

(2) Rolling gates typically are supported on two sets of wagons, of four wheels each, though eight wheels may be used on larger gates. Arrangement of the wagons follows one of two types. The classical or regular system places both wagons on the gate bottom, as shown in Figure 8-6a. The wheelbarrow system utilizes wheels riding on tracks at either side of the top of the gate recess. The port of Antwerp locks all utilize the wheelbarrow type design. These are mounted to a cantilevered portion of the gate structure that remains in the recess when the gate is fully closed, as in Figure 8-6b. The regular system provides simpler design of supports and recess and has performed adequately at many locations. The wheelbarrow system, however, improves overall stability, as the gate center of gravity and hydraulic loading center of effort generally are located near the line of action between the supports. Also, the wheelbarrow system places fewer moving parts below water. Wheel sets generally employ balance beams or some type of suspension system to equalize wheel pressures. Submerged wheel assemblies would be an appropriate application for greaseless bearing components to reduce maintenance. Replacement cycles will vary with design and water conditions; however, in northern European applications, replacement of submerged wheels typically is done at five-year intervals and, for wheelbarrow-placed supports, 15 years is expected.

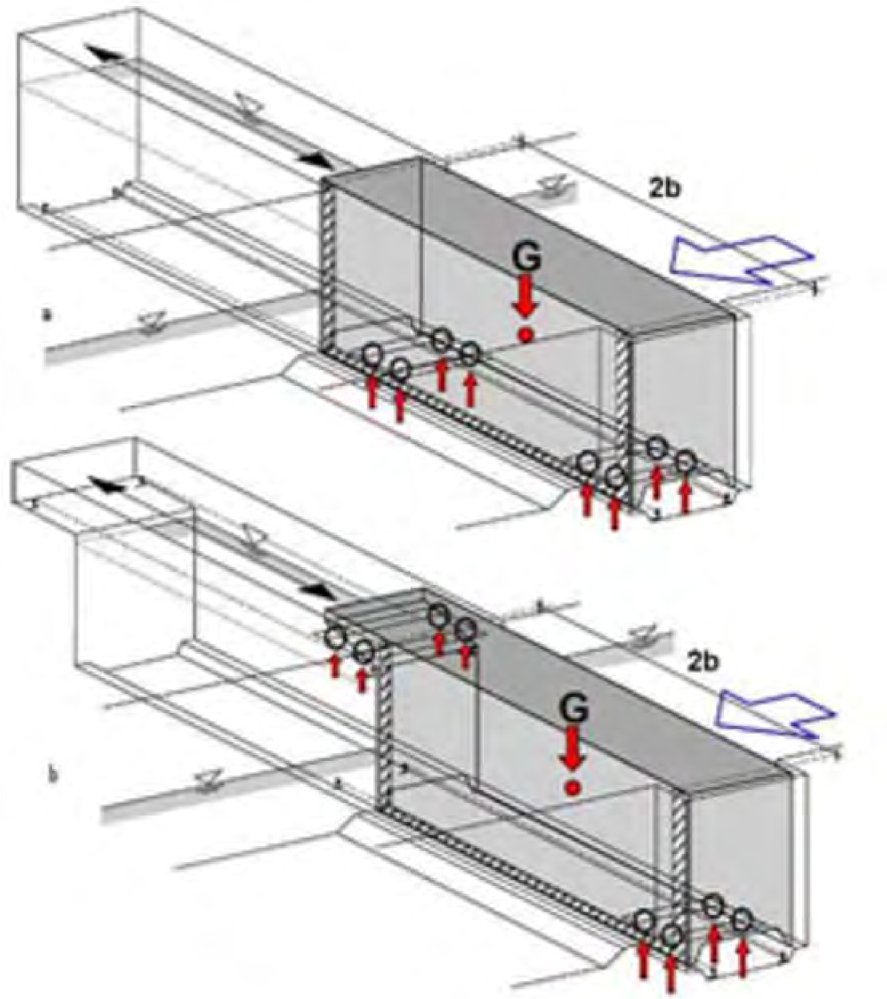


Figure 8-6a and 8-6b. Rolling gate support arrangements (courtesy of R.A. Daniel, Rijkswaterstaat, Division of Infrastructure, Netherlands)

(3) Support of the gate carriage on the wagons follows two different styles in Europe. The carriage bottom may be fitted with mitered plates in the shape of shallow inverted V's, resting on rollers mounted to the wagon, with the roller axis parallel to the direction of gate travel. This arrangement allows the gate to shift laterally under loading, with a tendency to shift back to the centered position under the gate's weight. The alternative style employs a rubber block centrally mounted to the carriage and wagon centerline, to absorb shock and spread the load to both rail tracks. Still, it allows for enough lateral movement for the seals to clear the gate's supporting structure. Devices at the supports employed to minimize gate lateral movement during the course of travel, while allowing the gate to shift to the sealed position once closed, also are found to greatly reduce track wear.

Figure 8-7. Wheelbarrow gate, Van Cauwelaert Lock, Antwerp, Belgium

(4) Rolling gates need to be designed to minimize maintenance. A cofferdam system allows for inspection and access to the supports and wagons. In Belgium, both rollers and wear pads have been used for the wire rope guide system (winch system for moving the gate). The rollers have tended to seize, and these are slowly being replaced with wear pads. A debris-removal system needs to be incorporated into the design. The gate and the roller carriages should be equipped with track clearers to prevent them from getting stuck. A waste grid should be positioned at the bottom of the gate, in front of the roller carriages. Sediment and debris need to be controlled as the gate moves across the lock chamber. The Belgian locks in Antwerp utilize mixers and a venturi system to keep debris from settling as the gate is moved. See Figure 8-8.

Figure 8-8. Zandvliet Lock, Belgium, mixer for sediment control

8-4. Drawbridge Gates.

a. Drawbridge gates are not common. The design of the floodgate at Empire, Louisiana, was conceived as a cost-savings measure over traditional gates. Some of the drawbacks of the design are specific to that installation, while others are inherent to the design. However, the gate had performed more or less successfully, despite maintenance issues and adjustments, for nearly 40 years.

b. Design considerations. Similar to a miter gate rotated to operate with a lateral hinge axis, the drawbridge gate presents a large surface area to any head difference present. For shallow channel applications, however, the aspect ratio of the gate reduces the effective moment arm on the hydraulic loading somewhat, and the force acts against the inertia of both the gate weight and that of the counterbalances. Normal differential heads would act to close the gate, where a reverse head would act to open it.

(1) The installation at Empire Floodgate has been found to operate adequately with small head differentials in the range of 1 ft or so. Alignment of the multiple gate hinges along the channel bottom proved difficult during construction. Otherwise, operating issues generally have been with chain maintenance and dredging.

(2) The principal operating equipment is a lifting chain operated by the sprocket, also called a wildcat. An additional chain run through idler sprockets is fed to a counterbalance system to greatly reduce machinery load. Providing adequate means of synchronizing the operating mechanism on either side of the channel is necessary to promote smooth operation and even load distribution. The wildcat sprockets must be machined correctly to the size chain utilized. As chain segments corrode and lose

material, the dimension between links changes, possibly resulting in load popping as the chain binds slightly while it is run through the sprockets. This is not a desired situation. Chain breakage or jumping off the wildcat is a dangerous event that might happen if chains are not properly maintained and binding is allowed to continue.

(3) When open, the floodgate rests in a recess along the channel bottom. As siltation of this recess occurs, the gate might not be able to fully open flush with the channel bottom. This or a vessel entering the channel prior to complete opening or closing greatly increases the likelihood of a collision of the gate and a passing vessel. This has occurred at Empire Floodgate, resulting in damage to the gate and, on one occasion, the sinking of a shrimp boat. Frequent dredging of the recess and strict adherence to navigational instructions are necessary to prevent this type of occurrence.

8-5. Drop Leaf Swing Gate.

a. This gate design was conceived for the multiple, small floodgate closures of the hurricane protection system across the bayous and canals in the New Orleans area. Typical closures would be in the range of 56 ft (18 m) wide and require an elevation of protection as far above the normal water level in the same range as the channel depth. As such, the large, above-water, top-leaf component capable of supporting the rest of the gate is justified by the height of protection requirements. Typically, the closures would be made with small sector gates when budgets permitted or vertical lift gates when clearance issues could be accommodated. Benefits of this system are the elimination of the large recess bays of sector gates or towers of lift gates, and all working components are stored completely in the dry. The gates would require multiple mechanical actions to complete closure, complicating design somewhat, though the individual mechanisms are simple. Though this type of gate lacks design history specific to itself, the operating principles are similar in combination with roadway swing gates and vertical lift gates, both of which have been well developed.

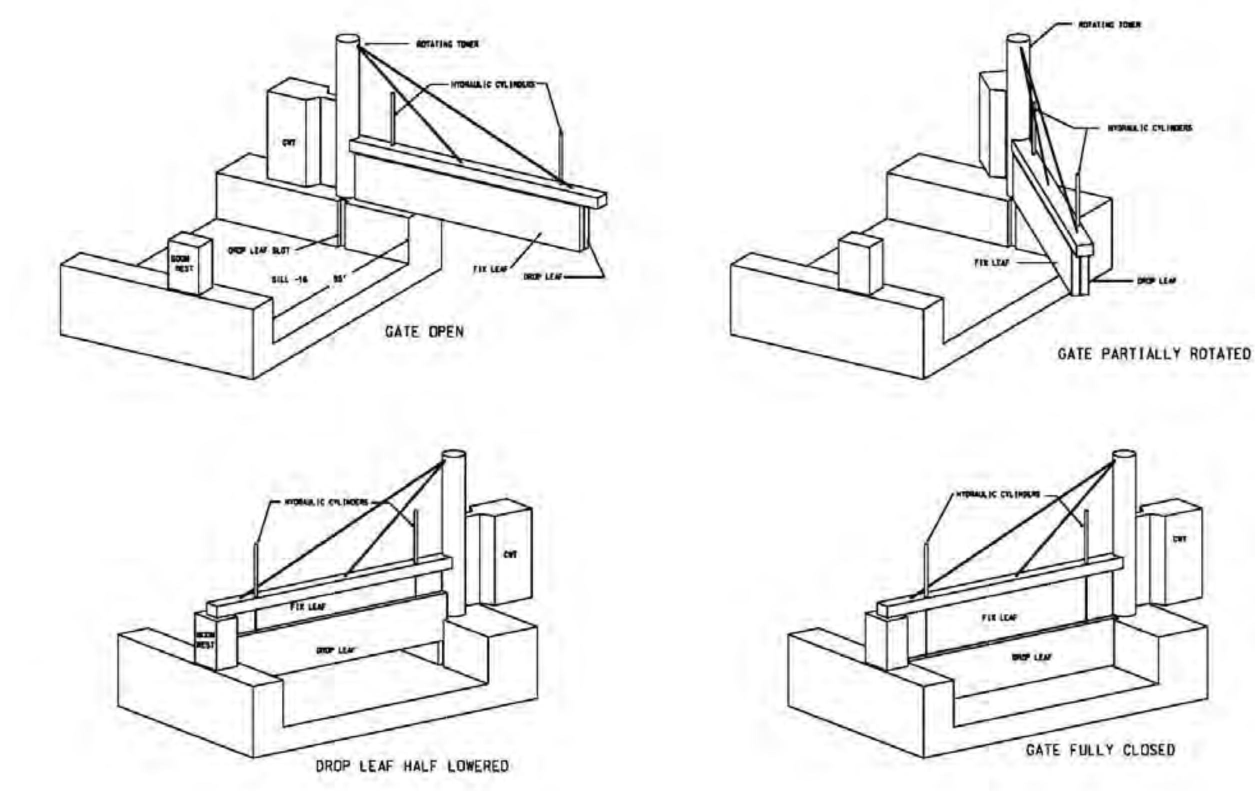


Figure 8-9. Conceptual drawing for drop leaf swing gate

b. Design Considerations. This design is premised on avoiding the bulk of the hydraulic loading until such time as the gate is supported on both sides of the channel and in the phase of operation where it is acting as a vertical lift gate described in Chapter 7. During the initial phase, primary loads would be gate-bearing friction and wind loading. Pivot machinery would be similar to that of miter gates, as described in Chapter 4, or borrow from rotating crane design. Lowering machinery would be similar to a hydraulic cylinder or cable system utilized with vertical lift gates, as described in Chapter 7.

CHAPTER 9

Dam Gate Operating Machinery

9-1. General Description. Dam gates are used to regulate water flow over a spillway. This chapter will focus on the operating machinery used for the most common USACE dam gates. It is intended to give a designer's perspective on the major considerations and best practices relevant to the design of dam gate operating systems. Consult Chapters 2 and 3 for information on selecting individual machinery components for these gate systems.

9-2. Tainter Gates. Tainter gates typically are considered the most economical and suitable type of gate for controlled spillways. Compared to other types, tainter gates are lighter in weight and have smaller hoist requirements. A major advantage of the tainter gate is its curved skin plate, which is concentric with the gate trunnion. This design feature works to focus the resultant hydrostatic loads acting on the surfaces of the gate skin plate through the trunnion. This results in no moment arm between the resultant hydrostatic load (acting on the skin plate) and fulcrum of the gate. This prevents moments caused by the hydrostatic loads, applied to the gate, which otherwise would have to be resisted by the hoist machinery. Tainter gate design also provides lifting points for the hoist machinery at a greater radius from the trunnion than that of most other operating loads such as the gate center of gravity and trunnion friction. The larger radius for the lifting points provides mechanical advantage. This allows the hoist equipment to apply less force to hoist a gate. Overall, these advantages help to reduce the size of the operating machinery compared to other common gate types.

Figure 9-1. Conventional spillway tainter gate

a. Operating Machinery. Tainter gate operating machinery is used to hoist a gate through its operating range and hold it at any desired position in its travel. While a dogging device usually is provided to hold a tainter gate in its fully open position, they typically are not used for any other operating position. Hoist systems used for tainter gates are similar to hoist systems used for other gate types. The primary difference for tainter gate operating machinery is that it needs to accommodate the rotational gate travel around the trunnion. It is most advantageous to locate tainter gate machinery above the gate and clear of the maximum water elevation. Typically, the machinery is located on structural steel framework on the piers adjacent to the gate or on a structural bridge extending between piers.

b. Service Factors. Tainter gate hoist systems rarely experience anything but constant and uniform hoisting loads. It is usually valid to assume that shock, impact, and vibration factors will have a negligible effect on the hoist system. Ice loading may need to be considered in northern climates. Service factors applied to the components of the hoist machinery system should be as discussed in Chapter 2. However, service factors should be selected based on the anticipated conditions for each individual application. If shock, impact, or vibration loading is anticipated, engineering judgment needs to be used to determine appropriate service factors.

c. Lifting Point Locations. Hoist lifting points for a tainter gate vary with the type of gate (submersible/non-submersible) and hoist machinery system used. To maximize mechanical advantage, lifting points typically are located on a structural member as close to the skin plate as possible. To maximize the gate travel, the lifting points typically are located on a structural member near the bottom of the gate. There are typically two lifting points, one on each side of the gate. Locating them requires balancing the load sharing between lifting points. The balance of lifting loads directly affects the amount of structural distortion and deflection a tainter gate will experience as it is being hoisted. Hoist system designers will need to coordinate lifting point locations with the structural engineer.

d. Hoisting Speed. A hoisting speed of 1 ft/min has been found satisfactory for most installations. Generally, this speed is used as a guideline to determine the power requirements of the hoist motor and reduction requirements for the overall hoist gear ratio. The mechanical designer always should coordinate the operating speed parameters or requirements with the hydraulic engineer.

e. Mechanical Hoist Components. Reference Chapters 2 and 3 for information and requirements for specific mechanical components. Component features specific to tainter gate hoists are discussed below.

(1) Motors. Motors used to hoist tainter gates shall be squirrel-cage induction, high-torque, high-slip, NEMA Design D motors. They shall be rated for continuous duty and sized to drive the gate without overload during any portion of the operating cycle. For the protection of the motor windings, means shall be taken to provide winding heaters or encapsulation.

(2) Worm Reducers. Worm reducers often are used with other reducers to establish the reduction requirements for a hoist system. Worm reducers also have an advantage in that they can be used to prevent back-driving of the hoist system. This can provide a secondary means (besides the brake) of preventing back-driving the drive train. However, designers should be aware that not all worm reducers prevent back-driving. Often, only worm reducers with gear ratios larger than a 30:1 prevent back-driving. The back-driving properties of worm reducers must be coordinated with the gearbox manufacturer.

(3) Brakes. Tainter gate brakes shall be drum brakes of the holding type and be specified with the features discussed in Chapter 2.

(4) Couplings. Tainter gate hoist systems commonly use a combination of rigid and flexible couplings to transmit torque between shaft sections of the drive train. Couplings used in the drive train of tainter gate hoist equipment are highly recommended to be a type that is considered to have high torque-carrying capabilities. The most common types of flexible couplings used for tainter gate hoists are gear couplings and grid couplings.

(5) Lifting Point Connections. Tainter gate lifting points shall be designed to accommodate the rotational motion of the gate travel.

f. Position Indication. Different devices have been used successfully for position indication on spillway gates. A gate's operational use needs to be considered when determining the type, location, style, redundancy, etc., of position indication devices.

Often, a combination of devices will be used to meet the position indication requirements for a gate. Sometimes, multiple devices are needed to provide indication for both local and remote operation.

(1) Local Indication. Local position indication devices can be used only for local gate operation because they do not provide a feedback signal.

(a) Dial Indicators. Dial indicators are one of the most popular means of local position indication. Dial indicators are driven off a shaft in the hoist machinery drive train. They use a gear reduction so that full gate travel results in no more than one revolution of the dial. The dial face is marked with desired openings for reference.

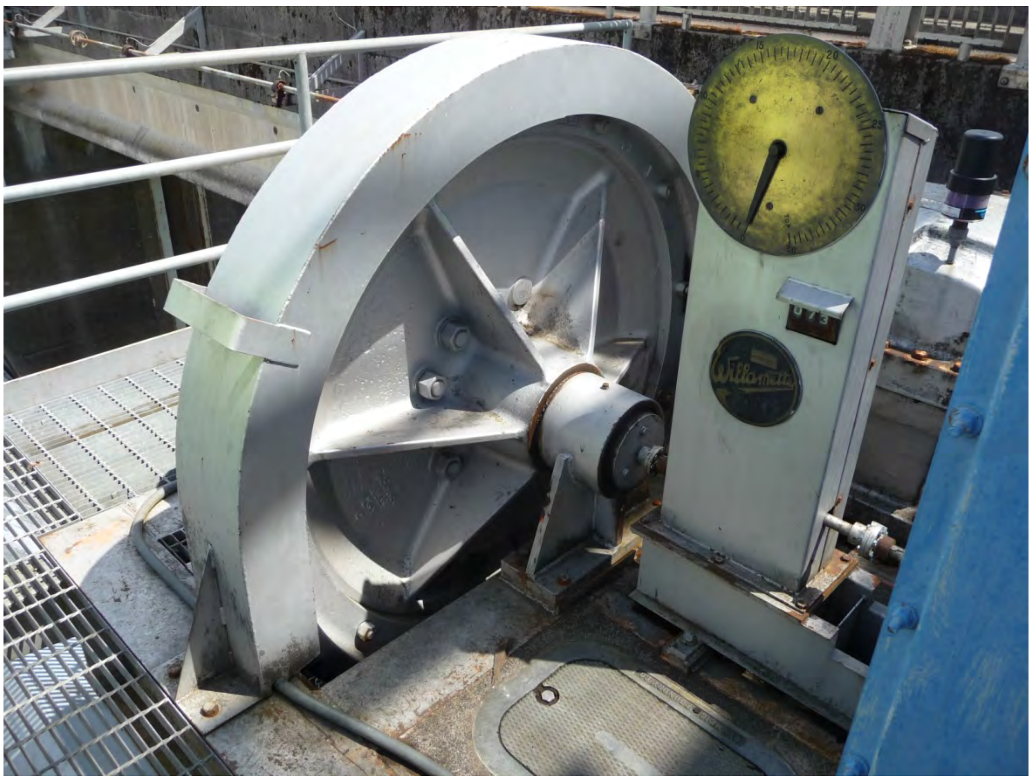


Figure 9-2. Dial Indicator

(b) Staff Gauge. A gate staff gauge is a simple method of providing local position indication. A gauge is mounted to the gate’s upstream skin plate or other convenient location. As the gate is operated, the gauge moves past a reference marker mounted to the pier face. The gauge has markings for the relevant gate openings. Designers should consider the visibility of the gauge from the gate operator’s location and the effects of ice or debris to which the gauge will be exposed.

(2) Remote Indication. The need for remote gate operation has increased the use of electro-mechanical position indication devices. Selection of such devices requires

coordination between the mechanical and electrical engineers. Designers shall comply with the current applicable requirements for remote control gate operation (see EC 1110-2-6071, Remote Control and Operation of Water Control Systems).

(a) Rotary Encoders. Rotary encoders are one of the most common electro-mechanical position indication devices used. They are driven off a shaft or operating cylinder in the hoist machinery and are used to count rotations of the encoder device. The count of shaft rotations is calibrated to the gate position. Encoders can be driven off any shaft for electric motor hoists. However, more accuracy usually can be obtained from the higher resolution (more counts) of a high-speed shaft. Rotary encoders typically are regarded as one of the most accurate electro-mechanical position indication devices.

(b) Inclinometers. Inclinometers are mounted directly to a gate and used to measure a gate’s opening angle relative to the direction of gravity. They are not used as commonly as encoders because they typically must be installed at locations on the gate where accessibility for maintenance can be an issue.

g. Limit Switches. Tainter gate limit switch assemblies are typically the rotary cam or traveling nut style. Limit switches are driven from the hoist machinery or hydraulic operating cylinders. Redundant limit switches shall be provided for the fully open and fully closed gate positions (two upper and two lower switches within the limit switch assembly). Often, the operational uses of a gate require additional switches for intermediate positions.

h. Trunnion Bearings. Trunnion bearings are the main bearings of a tainter gate. They support the major loads, including hydrostatic and gravity, to which a gate is exposed. The design of the trunnion often is shared by the mechanical and structural engineers, with the mechanical engineer focusing more on the bearing design. The most common types of bearings are discussed below.

Figure 9-3. Tainter gate trunnion bearing

(1) Straight Sleeve. Straight sleeve bearings are the most common bearing style used for tainter gate trunnions. They are typically interference fit into the trunnion hub and use a thrust washer to support thrust loading. Of the different common bearing types, straight sleeve bearings are the most economical and simplest to design and construct. They also provide the most bearing area for the required size envelope. While they have a smaller allowance for misalignment, it is typically enough to accommodate the amount inherent from reasonable structural tolerances.

Figure 9-4. Straight sleeve-style trunnion bearing

(2) Spherical. Spherical bearings can accommodate the largest amount of misalignment. Use of spherical bearings typically trades bearing area for misalignment capabilities. They are most common on wide tainter gates where the standard structural tolerances can create large misalignment at the trunnions. Spherical bearings typically add significant complexity and cost to the design of a trunnion bearing system.

(3) Lubrication. Trunnion bearings have relatively low surface speeds and rely on boundary layer lubrication. Grease lubrication systems typically are used and consist of grease lines that connect to grease grooves in the bearings. An automatic (Farval style) grease pump or manual grease pumps are used to supply the pressurized grease. Exercising and lubricating a gate without hydrostatic load is highly recommended on a regular basis because it helps ensure the lubricant is spread between the bearing contact surfaces. For regular operation, it is highly recommended that lubrication is started before the start of gate travel and is continued for the extent of gate travel. Trunnion bearing lubricant properties shall comply with the recommendations of Engineering and Construction Bulletin No. 2006-11.

(4) Self-Lubricated Trunnion Bearings. Self-lubricated bearing materials have been used successfully as the main trunnion bearings for tainter gates. In general, self-lubricated bearing materials are good substitutions for greased bronze bearings that move slow enough to establish only boundary layer lubrication. The primary advantage of a self-lubricated trunnion bearing system is the reduction of labor to grease the

bearings and maintain a grease lubrication system. Self-lubricated bearings also provide a more reliable lubrication system because the primary lubrication is provided by the bearing material and not supplied grease. Despite the more reliable primary lubrication system of self-lubricated materials, a minimum coefficient of friction of 0.3 shall be used for design purposes (see discussion of trunnion bearing coefficients of friction later in this chapter). This coefficient of friction is based partially on a degraded effectiveness of the lubrication system due to the intrusion of water, dirt, debris, etc., between the bearing surfaces. This remains a valid assumption for self-lubricated materials. Designers should consider working into a self-lubricated trunnion bearing design features that help seal the bearing from the intrusion of water, dirt, debris, etc. Even though most self-lubricated bearing materials are durable enough to function in dirty environments, the coefficient of friction can be impacted negatively. Sealing of a self-lubricated bearing can be performed with a physical bearing seal. Another common sealing method is to use supplied grease. Because the supplied grease is intended as a seal and not the as the primary lubricant for the bearing, the lubrication frequency can be minimized. Also, the consequences of failure of a supplied grease lubrication system (such as a clogged grease line) intended for sealing are much less severe than that used as the primary lubricant. Selection and design of self-lubricated materials shall follow the guidance established earlier in Chapter 2.

i. Wire Rope Electric Hoists. Wire rope electric hoists are one of the most common tainter gate hoist types. They've been used extensively at USACE dam sites and have a long, successful operating history. Wire rope electric motor hoists usually consist of two similar, but opposite hand hoist units mounted on piers, connected and synchronized by a lineshaft or torque tube, and arranged to lift each end of the gate. The machinery drive train typically consists of an electric motor that drives one or more gearboxes. Typically, a pinion gear is mounted on the output shaft of the final gearbox, which is used to drive a bull gear attached to a wire rope drum. Brakes for these systems are spring-set drum brakes and typically are located on the input or output shaft of the first reducer. Plates B-62, B-65, B-66, and B-67 show common machinery layouts for wire rope electric hoist systems.

Figure 9-5. Wire rope electric motor hoist

(1) Wire Rope Drums. Wire rope drums for tainter gate hoists typically are designed to spiral wrap each rope directly on itself. This is necessary because multiple wire ropes typically are needed for hoisting a gate. The direction the rope wraps on the drum, with respect to the gate, usually doesn't matter from a structural loading perspective. However, designers should consider that the direction the rope is wrapped on a drum affects the direction of the open gear mesh contact forces and resulting loading of the gearboxes. The most common method of driving a hoist drum is to mount a pinion gear on the output shaft of the final reducer, which drives a bull gear integral to the hoist drum. The direction of rope wrap on a hoist drum determines if the contact force on the pinion loads the final reducer housing in compression or tension.

(2) Wire Rope. The selection of wire rope for gate operation should be in accordance with EM 1110-2-3200, Wire Rope Selection Criteria for Gate Operating Devices. Settlement of the wire ropes and gate system components can cause individual wire rope tensions to change over time, resulting in uneven load sharing between individual ropes. Tainter gate wire rope hoists systems shall be designed to allow tension adjustments to be made to each of the individual wire ropes. Designers also should consider developing requirements for the allowable deviations of tension values to determine when re-tensioning is necessary. This should include tension values on each side of the gate, which can affect the distortion/deflection of the gate structure. Figure 9-6 shows one of the most common methods for allowing individual rope tension adjustments. This design uses nuts on the threaded ends of U-bolts to adjust tension for each wire rope. The U-bolts wrap around a pin spanning between the gate connection ears. This allows the U-bolts and hardware to pivot around the pin as

the gate is lifted and the angle between the ropes and gate changes. See Plates B-63 and B-64 for typical sketches of this gate connection style.

Figure 9-6. Wire rope gate connection

j. Chain Electric Hoists. Chain electric hoists usually consist of two similar, but opposite, hand hoist units mounted on piers and arranged to lift each end of the gate. The machinery drive train typically consists of an electric motor that drives one or more gearboxes. Usually, a pinion gear is mounted on the output shaft of the final gearbox, which is used to drive a bull gear attached to the chain lifting feature. The chain-lifting feature depends on the number and type of chains. The features commonly used are pocket wheels, grooved drums, or sprockets. Brakes for these systems are spring-set drum brakes and typically are located on the input or output shaft of the first reducer. Plates B-70, B-71, and B-72 show typical pocket wheel and chain gate connection details. Designers should reference Chapter 2 for types of chain-lifting features (pocket wheels, grooved drums, and sprockets) and chain designs. Tainter gate chain hoists systems shall be designed to allow for tension adjustments to each of the individual hoist chains.

Figure 9-7. Hoist chain gate connection

k. Hydraulic Cylinder Hoists.

(1) Hydraulic cylinder hoists usually consist of two hydraulic cylinders, one mounted on each pier and arranged to lift each end of the gate. Typically, the cylinders are trunnion-type, mounted in cardanic rings that are supported by hoist frames cantilevered over the side of the pier. Piston rods usually are connected through a spherical bearing to a lower framing member on the downstream side of the gate. Plate B-68 shows a general arrangement of a direct-connected hydraulic cylinder-type tainter gate hoist. Plate B-69 shows details of the mounting arrangement. Individual hydraulic power units usually are mounted in rooms at the top of each pier, although an arrangement with a single power unit is possible. As much valving as possible is mounted in manifolds connected directly to the cylinder ports. This includes a pilot-operated check valve on the rod end port used to hold the gate in a raised position. This arrangement minimizes interconnecting piping and the potential for leakage or failure.

(2) Cylinder Synchronism. Hoist cylinders are kept in synchronism by the hydraulic controls. Usually, position indicators mounted internal to each cylinder provide a signal, relative to cylinder stroke, to the control system. The system generates an error signal that is used to control a small proportional valve. The valve is used to bleed oil from the rod side of the lead cylinder, when raising, and from the rod side of the lag cylinder, when lowering. For small gates or gates that are infrequently operated, such as on flood control spillways, a simpler system utilizing a flow divider might provide sufficient synchronization.

l. Rack-and-Pinion Hoists. Some older tainter gate installations are operated with a rack mounted to the skin plate of the gate, driven by a pinion on the final reduction of the electric motor-driven machinery. These rack-and-pinion systems are not used for new installations, but still are being used at some USACE sites.

m. Mechanical System Analysis and Design Criteria

(1) Determining Hoist Size. The designer also should reference EM 1110-2-2105. The normal load required to lift a tainter gate is a function of the external loads applied to the gate (hydrostatic forces, gravitational forces, friction forces, etc). To hoist a gate, the motor or hydraulic cylinders must overcome the various forces acting on the gate. Calculating required motor or cylinder sizes is performed by creating a free-body diagram and applying operating loads and reactions. The diagram is created as a snapshot of the operating loads acting on the gate at a particular instant of time. With the snapshot approach, it is valid to use dynamic or static coefficients of friction. Use of a dynamic coefficient of friction would represent a steady-state moving gate. Use of a static coefficient of friction would represent the instant of time just before incipient motion and transition to dynamic friction. Because both will be experienced by a gate during operation, the most conservative (largest) coefficient of friction, between static and dynamic, shall be used. The friction forces applied to the free-body diagram always will act in a direction that opposes the motion of the gate. After applying the operating loads to the diagram, a summation of moments and forces can be used to solve for the reaction forces, which then can be used to determine required motor or cylinder sizes.

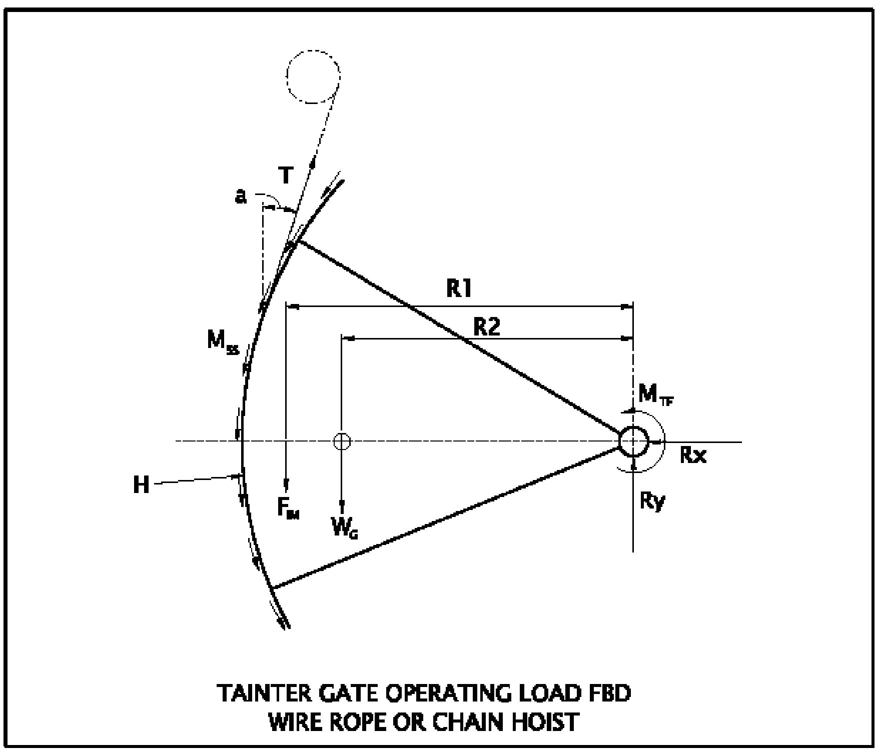


Figure 9-8. Wire rope hoist example, free-body diagram

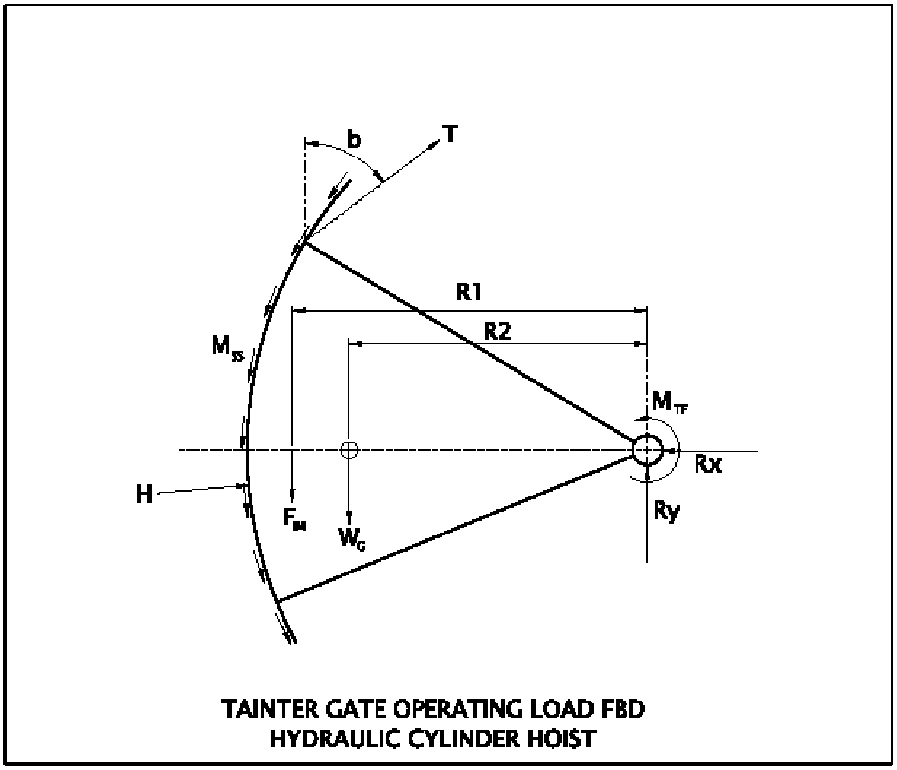


Figure 9-9. Hydraulic cylinder hoist example, free-body diagram

(2) Operating Loads. The maximum normal load the operating machinery will experience is typically at the gate position at which the moment arm between the gate center of gravity and trunnion is maximized.

(3) Hydrostatic Load (H). As discussed at the beginning of this chapter, tainter gates are designed so that the curvature of the skin plate is concentric with the gate trunnion. This serves to focus the resultant hydrostatic load acting on the skin plate through the trunnion. This eliminates direct moment contributions from the hydrostatic load that otherwise would affect the required hoist loads. The assumption that the hydrostatic load acts through the trunnion is valid for most cases. However, this assumption is based on the geometry specific to a gate. Some gates are designed with surfaces that hydrostatic loads act on that are not concentric with the trunnion. The applicability of this assumption needs to be assessed on a gate-by-gate basis. While in most cases the hydrostatic load does not directly apply a moment to a tainter gate, in almost all cases it contributes largely to normal forces that generate friction that does apply a moment. The hydrostatic loads typically contribute greatly to the total reaction force at the trunnion. The trunnion reaction forces (reactions on the main bearing and any thrust surfaces) are the normal forces used to calculate the trunnion friction. Also, the hydrostatic load typically contributes to the normal force caused by the gate side seals contacting the pier faces.

(4) Gravity Loads.

(a) Gate Weight (W_G). Calculations of gate weight shall include estimates for components that have significant weight contributions. Components to consider include the gate coating system, weld material, and fasteners. Engineering judgment shall be used when determining the components needed.

(b) Ice and Mud (F_{IM}). Reasonable estimates shall be made for other factors such as ice, mud, or debris that can accumulate on gate surfaces. The potential for ice and mud buildup varies widely among climates and locations. Engineering judgment shall be used to make reasonable, site-specific estimates for ice, mud, and debris weights.

(c) Trunnion Friction (MTF). The trunnion friction load shall be calculated based on the reaction forces at the trunnion. This typically involves two separate reaction forces: one acting on the trunnion main bearing and the other acting on the trunnion thrust surface. The trunnion bearing coefficient of friction is dependent largely on the condition of the bearing and how well it is maintained. Realistic coefficients of friction for a well-maintained, lubricated bronze bearing are typically within a coefficient of friction range of 0.1 and 0.2. A minimum coefficient of friction value of 0.3 shall be used for design purposes. The 0.3 value can be experienced in cases where greasing and other necessary bearing maintenance are not performed at the designed lubricating intervals, or in cases where water, dirt, debris, etc., get between the bearing surfaces. Maintenance of a trunnion bearing performed at time periods longer than the designed maintenance interval, or the intrusion of water, dirt, or debris are conservative yet reasonable operating conditions that shall be accounted for in the design coefficient of friction. Some tainter gate bearings are believed to have experienced higher coefficients of friction than 0.3. In almost all cases, maintenance was stopped for extensive periods or the lubrication system experienced a failure, such as a clogged grease line, that prevented the lubricant from reaching the bearing. Bearing designers should use engineering judgment to determine acceptable coefficient of friction for which these conditions are anticipated. Trunnion bearings shall be designed to comply with the bearing pressure requirements in Chapter 2.

(d) Seal Friction (M_{SS}). The side seal friction is a function of the preset force in the seal and the hydrostatic pressure on the seal surface. See EM 1110-2-2105 for the method used to calculate the side seal friction.

(5) Design Criteria. The components of a tainter gate hoist system shall be designed to comply with the criteria in Chapter 2. The designer should evaluate two load conditions: the maximum normal working load divided equally on both sides and the maximum overload condition. The magnitude of the maximum overload condition will depend on the type of hoisting system. The factors of safety from Chapter 2 shall apply to the maximum overload condition. Specifically, for the overload condition, unit stresses shall not exceed 75% of the yield stress of the material, and wire rope loads shall not exceed 70% of the nominal breaking strength.

(a) For hydraulic cylinder hoists, the system should be designed to support and lower the entire gate from one side. The maximum overload condition would be limited by the hydraulic system relief valve setting.

(b) For systems consisting of synchronized but independent electric motor hoists for each side of the gate, the maximum overload condition would be the forces created by the locked rotor torque of each motor applied to each side of the gate.

(c) For systems consisting of a single hoist motor with a drum for each side of the gate, the maximum overload condition shall also be calculated from the locked rotor torque of the motor. However, the loads from this overload condition shall be divided between the two sides of the hoist by a percentage of 70/30. This percentage split represents a conservative but reasonable estimate of the maximum uneven loading. Also, with the 70/30 split, even if the full locked rotor torque was applied to one side of the gate, components will likely not exceed 90% of the ultimate tensile strength. The designer shall calculate and verify this. Previous design criteria have required loads resulting from the maximum torque of the motor to be split equally between sides of a hoist. While this design criteria has been successfully used for the life span of most tainter gate wire rope electric hoist systems, it is recognized that this criteria may lead to component failure under the most extreme loading conditions such as the locked rotor torque being applied to one side of a hoist. It is also recognized that designing for the locked rotor torque being applied completely to one side of a hoist can be cost prohibitive for gates that have low consequences of a major hoist system failure. If and when the PDT determines that designing to less than the 70/30 requirement is appropriate and reasonable, a cover letter along with the supporting cost and risk analysis documentation shall be submitted to USACE-HQ requesting exemption from the 70/30 requirement. The lowest allowable limit for the cost and risk analysis shall be the loads from the locked rotor motor torque divided 50/50 between each hoist side.

(d) Manufactured items should be selected based on the published catalog ratings, without the application of additional safety factors or based on published overload ratings if available. Coordination with the structural engineers on these loading conditions for structural items is required.

n. Tainter Gate Variations.

(1) Submersible Tainter Gates. Submersible tainter gates are similar to conventional tainter gates, but are designed to be lowered to a submerged position that allows unrestricted flow through the gate channel or the passage of vessel traffic. Submersible tainter gates are used at USACE sites for both spillway and navigation lock applications. A general arrangement of a submersible tainter gate is shown in Plates B-60 and B-61.

(a) Hoist Types. Although any hoist type could be used for a submersible tainter gate, greater distance between the machinery (located on the top of the piers above the gate) and lowest gate elevation (submerged position) tends to make hydraulic cylinder and screw stem hoists less feasible hoisting options. Wire rope-and-chain hoists are the most common choices because they can accommodate the greater distance between hoist machinery and gate.

(b) Gate Connections. Gate connections for submersible tainter gates most commonly are located near the skin plate on the top horizontal girder or other convenient structural member on the concave side of the gate. This location most often is chosen because it usually maintains an unobstructed path to the machinery for the full range of gate motion. Figure 9-10 shows this gate connection location and a common gate connection style for a submersible navigation lock gate.

Figure 9-10. Common gate connection style and location for submersible tainter gates

(c) Submersible tainter gates are commonly and successfully used as the primary operating gates in navigation locks. While the hoist machinery usually has many similarities to non-submersible tainter gates, there are typically a few key differences. Navigation lock tainter gates are required to span the width of the navigation lock, which tends to create a different aspect ratio from spillway tainter gates (navigation lock tainter gates tend to be much wider than they are tall). The width of the gates and need for clear space across the lock (to pass vessel traffic) directly affect the feasibility of using a mechanical means, such as a torque tube or line shaft, to synchronize sides of a hoist. Synchronization typically is done by power selsyn motors. Typical hoist systems consist of a rope drum, open gear set, speed reducer, brake, hoist motor, and power selsyn. The hoist drum typically is mounted on a cantilevered shaft of a size adequate to prevent excessive error in the mesh of the final drive pinion and gear due to shaft deflection.

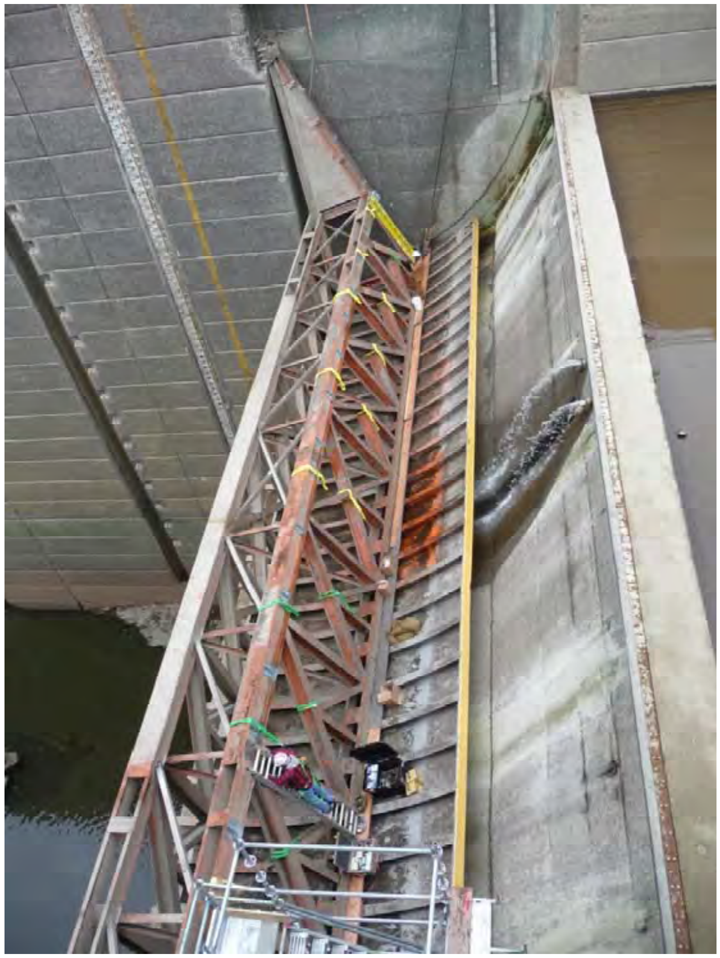


Figure 9-11. Navigation lock submersible tainter gate with lock dewatered

(d) Hoisting Speed. The need to accommodate vessel traffic efficiently might require hoisting speeds other than the 1 ft/min guideline recommended earlier in this chapter. Hoisting speeds should be coordinated to make sure the necessary time per lockage parameters are being met. Most often, navigation lock submersible tainter gate hoists are sized to operate a gate from full open to closed, and vice versa, in 2 to 3 min.

(e) Spillway Applications. Submersible tainter gates are used on spillways as a method of providing unrestricted flow (gate in the fully lowered position). They also can allow the passage of vessel traffic during times when restricted flow is not needed.

(2) Piggyback Tainter Gates. A piggyback tainter gate also could be described as a tainter gate sitting on top of another tainter gate. Piggyback tainter gates are used as a means of releasing water from two different elevations (not simultaneously). The bottom gate seals against the spillway concrete ogee. The top of the bottom gate is designed with its own ogee shape to accommodate overtopping flow when the top gate is hoisted. The top unit resembles a short conventional tainter gate that seals against the ogee of the lower gate and is held in the closed position only under its own weight. When the bottom gate is hoisted, the top gate is lifted with it. Piggyback tainter gates

are chain hoisted with standard chain hoist systems. One end of the chain is connected to the bottom gate. The other end of the same chain is connected to the top gate. The top gate is hoisted by first lowering the bottom gate to the fully closed position. The machinery then is operated in the same direction that lowers the bottom gate. This first pulls the slack out of the section of chain between the machinery and top gate, then starts to hoist the top gate. The Corps' Pittsburgh District owns and operates piggyback tainter gates and can be contacted for further information.

(3) Sydney Tainter Gates. A Sydney tainter gate is a conventional tainter gate that can be hoisted by its dedicated hoist equipment vertically after the gate is lifted to its highest point of rotational travel. The typical operation of a Sydney gate is the same as a conventional tainter gate. Sydney tainter gates are designed with vertical guide slots in the concrete piers that the trunnion pin can slide through. If extremely high flows are encountered, the entire gate can be lifted vertically, past its typical operating range, to move it out of the flow. As the gate is lifted vertically, the trunnion pin slides through the vertical guide slots in the pier. Sydney gates are advantageous when high river flows, requiring unrestricted water passage, are at a much higher elevation than typical river flows. The Corps' Pittsburgh District owns and operates Sydney tainter gates and can be contacted for further information.

9-3. Vertical Lift Gates. Vertical lift gates are a common type of dam gate. They are used in many different applications including spillways, control towers, and regulating outlets. Machinery typically is located on a structural feature above the gate. The design criteria discussion for tainter gates in Paragraph 9-2.m.(5) also can apply to vertical lift gates. For most dam applications, vertical lift gates must be operated under differential head conditions. The differential hydrostatic pressure can create large transverse forces, creating large friction forces as a gate is being operated. Rollers or other features almost always are needed on the downstream side of the gate to reduce friction between the gate and guides to allow hoisting of the gate. Chapter 7 is dedicated to vertical lift gates for all applications and should be referenced for further information. This discussion of vertical lift gates is provided for comparison to the other dam gate types covered in this chapter. The design of vertical lift gates is covered in EM 1110-2-2105.

Figure 9-12. Spillway vertical lift gates

a. Advantages. Vertical lift gates can be built to different sizes and aspect ratios. They've been used successfully in high head applications and their simple construction also makes them cost effective for low head applications.

b. Disadvantages. Friction has the largest operational affect on vertical lift gates than on any other dam gate type. The large transverse hydrostatic forces supported by vertical lift gates can create large friction forces for which the hoist machinery must be designed. Unlike tainter gates, the location of the gate's lifting points gives almost no mechanical advantage with respect to friction. Features to reduce friction, such as rollers, are usually necessary for reasonable hoisting loads. Roller maintenance is critical to ensure operation of a vertical lift gate. In addition to large friction loads, vertical lift gate hoist machinery also must lift the weight of the gate through the full operating range. This is not the case for tainter gates hoists, which share load with the trunnion bearings to support the weight of the gate.

9-4. Wicket Gates. There are different types of hydraulic gates, used in different applications, that are called wicket gates. In this manual, only the application of a wicket gate used to create a dam will be covered. A wicket gate, or wicket, as shown in Plates B-73 and B-74, is a structurally framed gate attached to the sill of a dam. Wicket gates are installed side by side and, when raised, create a wicket dam that restricts flow to increase the depth of the upstream pool. Wicket dams are used most commonly at USACE facilities to maintain required river navigation depths during periods of low river flow. Wicket gates are installed at several sites on the Illinois Waterway, Locks 52 and 53 on the Ohio River, and will be utilized at the new Olmstead dam on the Ohio River. When river flows are high enough to maintain required pool levels, without restricting flow, wicket dams are lowered so the wickets lay flat on the sill of the dam. When wickets are in the lowered position, vessel traffic can pass freely over the top. Most commonly, wickets gates are designed with a flat skin plate reinforced with structural members. Curved skin plate wickets also have been designed. However, curved skin plates are less common because they are more likely to develop low-pressure areas as flow passes around the wicket during raising and lowering, which can increase operating loads. In their raised position, wickets sit at a fixed inclined angle.

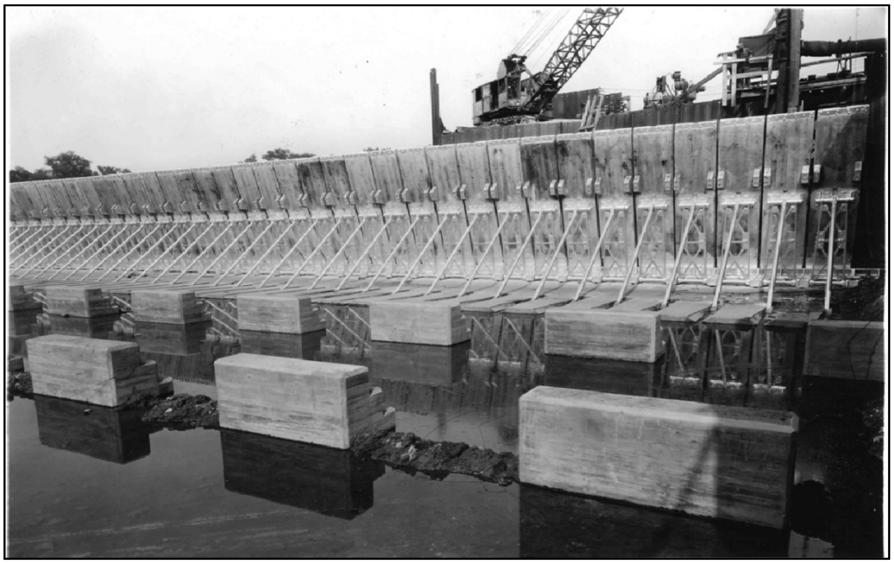


Figure 9-13. Manually operated wicket gates in the raised position (viewed from the downstream side)

a. General Considerations. Manually setting wicket gates is extremely dangerous for lock personnel. The setting, maintenance, and operation of these gates have life safety implications. Any new wicket gate designs should be done to minimize life safety risks and minimize operation and maintenance. It typically takes a crew of 10 or more personnel to set wicket gates.

b. Operating Machinery. Wicket gates are raised and lowered by mechanical means. There are multiple operating machinery types that are discussed in more detail below. The function of each machinery type is to raise a wicket gate against flow (no dewatering devices are installed for typical raising and lowering functions) and support the gate for the duration it is raised.

c. Bearings. All bearings associated with wicket gates shall be of the self-lubricated type and designed in accordance with the criteria in Chapter 2.

d. Operating System Design. Specific design guidance for wicket gates is provided in the Fort Belvoir Engineer School Design Manual – Canalization of the Upper Mississippi River and Ohio River. Wicket gate designers also should reference USACE Waterways Experiment Station Technical Report SL-97-12, Hydraulic Forces on a Wicket Gate Under Upstream Quasi-Laminar Flow, for the hydraulic forces on a wicket gate. Wicket gates are considered a Type A hydraulic steel structure, per EM 1110-2-

2105. The design of a wicket gate's mechanical operating system should be coordinated with the appropriate structural designers.

e. Manually Operated Wickets. Manually operated wickets are raised and lowered using equipment mounted on a floating vessel. The equipment, usually a crane or excavator with a custom-designed hook, raises each wicket individually as the vessel is maneuvered across the river. Manually operated wickets are outfitted with a frame, called a horse, and prop that support the wicket in the raised position.

Figure 9-14. Modified excavator operating manual wicket gates

(1) Raising. To raise a manually operated wicket gate, the floating vessel-mounted lifting equipment is manually hooked to an attachment point on the upstream end of the wicket. A pivoting frame, called a horse, is connected to the sill on one end and to the midsection of the wicket on the other. A prop also is mounted on the bottom of the wicket at the horse, to the wicket pivot point. As the wicket is lifted, the horse rotates forward, pulling the prop through a track assembly, called a hurter, mounted to the downstream sill. As the wicket is lifted from the sill, the flowing water passes under the wicket and aids in lifting (the wicket stays relatively horizontal during lifting). When the horse has reached a designated angle with the sill, the prop is designed to fall into a notch in the hurter. After the prop drops into the notch, the upstream lifting connection to the wicket is released and the current holds the wicket elevated horizontally from the sill. The wicket then is rotated into position by pushing the upstream end of the wicket down. As the upstream end of the wicket lowers, the force from the water current flowing past the wicket increases below the pivot point (fulcrum) at the midspan of the wicket. The wicket is pulled into contact with the sill as the moment from the forces below the fulcrum of the wicket overcomes the moment from the forces above. When

the wicket is set in its raised position, the hydrodynamic and/or hydrostatic pressures (depends on the differential pool and flow past the gates) on the upstream face of the gate hold the wicket steady against the sill and wicket prop. Plate B-73 shows the different stages of raising a manual wicket.

Figure 9-15. Raising manual wicket gates

(2) Lowering. To lower a manually operated wicket gate, a connection point on the top of the inclined wicket is used. The vessel-mounted equipment is used to pull the top of the wicket upstream, rotating the wicket about the sill into the flow of the river. As the wicket is rotated, the movement of the wicket pulls the prop out of the notch in the hurter. After the prop clears the notch, it is lifted manually to prevent it resetting into the notch as the wicket is lowered. Once the prop is clear, the wicket is released from the equipment and falls under the influence of gravity and back to a flat position on the sill. The water under the wicket is used to cushion the impact of the falling wicket. The hurter is designed to realign the prop for the next lifting operation of the wicket as the wicket moves to its lowered position.

(3) Hurter Design. Hurters are typically a fabricated steel weldment that is embedded in the concrete under the wicket gate (Figures 9-16 and 9-17). When designing a wicket dam that uses a hurter and prop, the hurter should be designed so the movement of the prop through the hurter return guide is assisted by the river flow around the lowered wickets. For example, if the wicket dam is designed to be lowered from right to left looking downstream, the hurter prop return guide should be located on

the left looking downstream. If the flow around lowered wickets is not taken into consideration when designing the hurter, lowering operations could be inhibited due to water flow preventing the prop from moving to the return guide in the hurter.

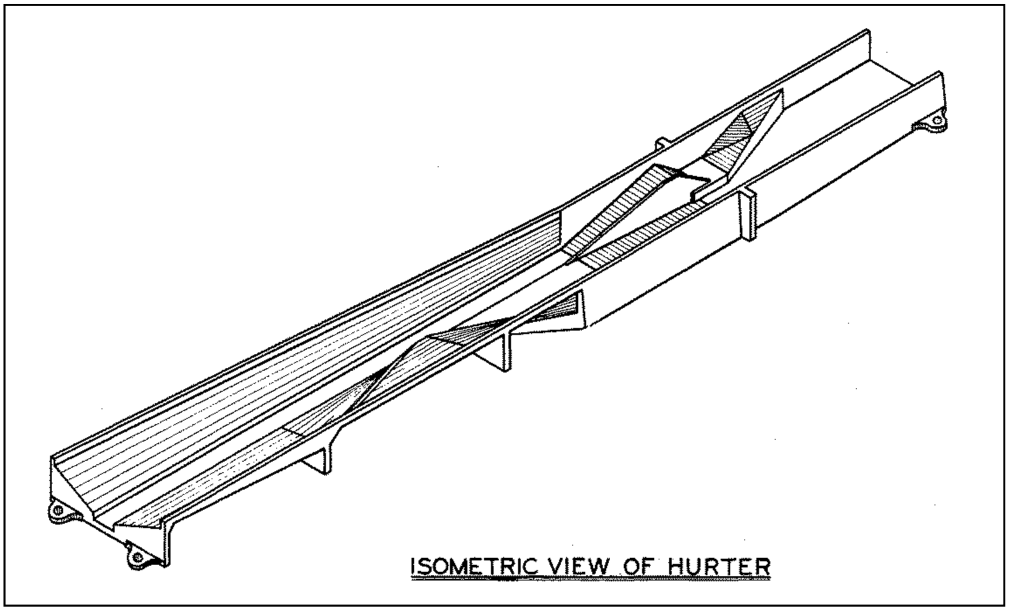


Figure 9-16. Hurter isometric sketch

Figure 9-17. Wicket gate hurter

(4) Advantages. Manually operated wickets minimize the amount of submerged mechanical equipment. Operation of the gates does not rely on a dedicated hydraulic system that otherwise would have to operate in a submerged environment. The submerged mechanical equipment is limited to the horse-prop frame and hurter, which are relatively easy to design for a submerged environment.

(5) Disadvantages. Manually operated wickets are the most labor intensive because raising and lowering a wicket dam require a team of people to operate the floating vessel and wicket-lifting equipment. At Lock 52 on the Ohio River, it takes a crew of about 20 to set the wicket gates. Lifting a wicket requires connecting the lifting equipment to the wicket lifting point. The wicket lifting point cannot be seen from the floating vessel. The connection to the lifting point must be made by feel. This requires a skilled operator to perform the lifting operations.

f. Dedicated Operating System Wickets. Two styles of dedicated wicket gate operating systems have been tested by USACE. While neither have been selected for use at a USACE facility (to date only manual wickets are used), full-scale testing on each has proven the operating systems to be successful. The two types are retractable hydraulic cylinder and dedicated or direct connected hydraulic cylinder. These systems, along with manual wicket gates, were considered for installation during the planning stages of Olmstead Dam. The primary advantages of dedicated operating system wickets are there are significantly fewer operational safety risks and they take less time and labor to operate. However, manual wickets ultimately were chosen. Olmstead Dam is on a wide stretch of river and required approximately 140 wicket gates. The dedicated operating systems have higher maintenance costs. Overall, the higher maintenance costs for so many wickets outweighed the lower amount of labor required to operate the wickets. While the dedicated operating system wickets were not feasible for such a large application, they are still a viable choice for narrower rivers.

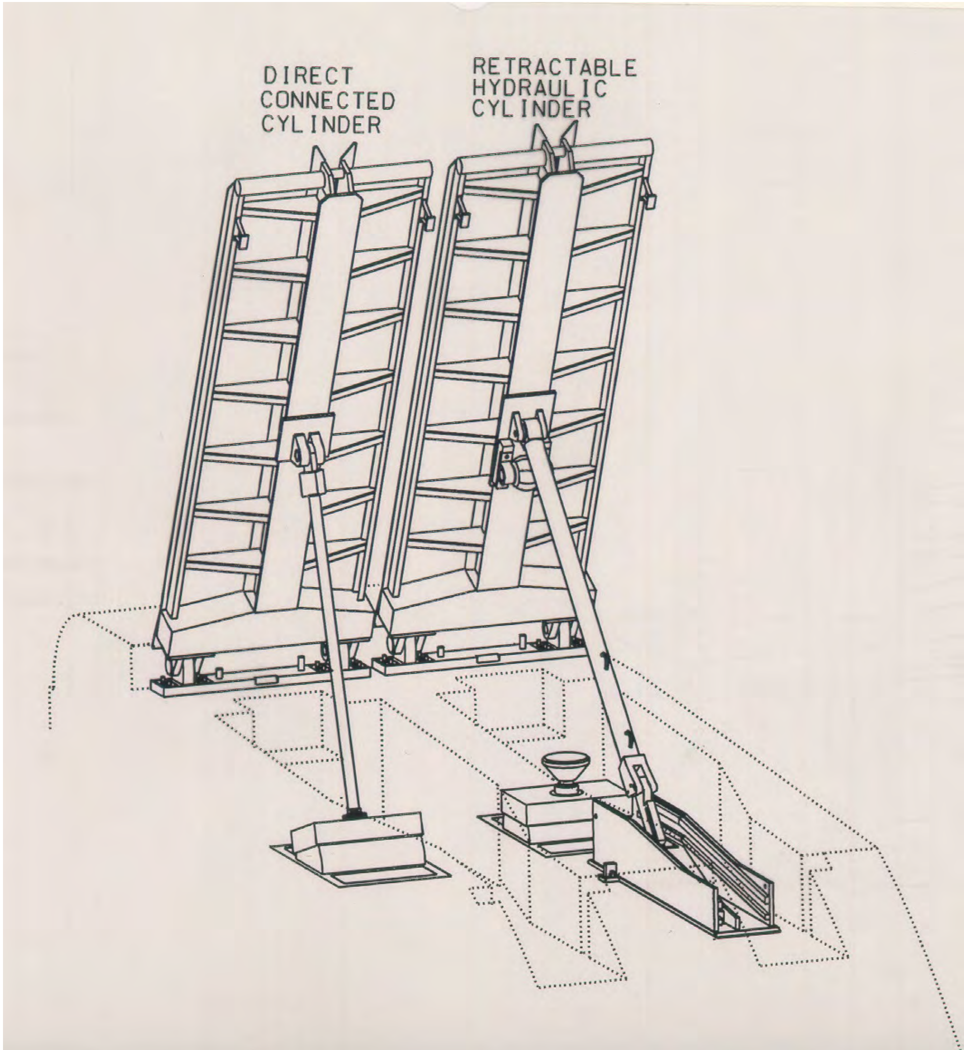


Figure 9-18. Retractable hydraulic cylinder and direct-connect cylinder wicket gates

Figure 9-19. Full-scale Olmstead Dam wicket gate prototypes (retractable hydraulic cylinder and direct-connect cylinder styles shown)

(1) Retractable Hydraulic Cylinder Wickets. The retractable hydraulic cylinder wicket, shown in Plate B-74, uses the same principles as the manual wicket with a couple modifications. Instead of using a vessel-mounted lifting device to operate the gate, a hydraulic cylinder with a hydraulic power unit is used. The cylinder is under the lowered gate. The gate is hinged at the crest, and it pivots about the hinge as it operates. The retractable cylinder lifts the wicket until the prop engages the hurter to support the gate (similar to the manual wicket). The hydraulic power unit can be located on shore, above flood stage, or in a gallery beneath the wicket, depending on the size of the dam being built. The wicket is connected directly to the sill on the upstream end and does not use a horse frame.

(a) Advantages. The advantage of the retractable hydraulic cylinder over the direct-connect cylinder is the use of a prop to support the gate while it is in the raised position. Leakage past the cylinder piston will not affect the incline position of the wicket over time.

(b) Disadvantages. The retractable hydraulic cylinder system is more complex. A secondary cylinder is required to position the main lifting cylinder to perform the lowering operation.

(c) Operating Speed: A raising time of 12 min, lowering time of 3 min, and operating time of the secondary cylinder of 10 sec was found to be successful for the Olmstead prototype testing.

(d) Raising. The wicket is raised into position by rotating it from downstream to upstream about the hinge of the gate. The retractable design is made of two cylinders, one for raising and lowering the wicket and one for aligning the raising and lowering cylinder. The hydraulic cylinders are mounted under the wicket. The lifting cylinders piston rod is mounted with a cup that engages a ball mounted on the downstream side of the wicket. The wicket is raised by extending the lifting cylinder, which engages the ball and rotates the wicket to a fixed angle where a prop engages a notch in the hurter in the same manner as the manually operated wicket design. Once the prop is set in the notch, the combination of current and gravity of the inclined wicket keep the prop securely fixed in the hurter. The piston rod is retracted to remove it from potential damage caused by debris.

(e) Lowering. To lower the wicket, a second smaller alignment cylinder is used to align the larger lifting cylinder to the proper angle to contact the cup with the ball and the gate. The piston rod rotates the wicket forward until the prop clears the notch in the hurter and the flow of fluid out of the cylinder controls the speed at which the wicket lowers.

(2) Direct-Connected Hydraulic Cylinder Wicket. The direct-connected cylinder design is very basic. One cylinder is connected directly to the backside of the wicket, as shown in Figure 9-19. The connection is made at the same location the prop is connected to in the retractable cylinder design. To raise the gate, the cylinder is pressurized and the piston rod extends, rotating the wicket to the incline angle. The hydraulic cylinder valves hold the wicket in the raised position until the wicket is lowered.

(a) Advantages. The main advantage of the direct-connect cylinder system is simplicity. It relies on only one cylinder for operation, eliminating the need for a prop-and-hurter system.

(b) Disadvantages. The main disadvantage is that this system relies on the cylinder to support the wicket gate during the entire time the gate is raised. The cylinder valves must maintain hydraulic pressure in the cylinder to support the wicket. Leakage past the cylinder piston seal also could result in unintended lowering of the raised wicket over time.

(c) Operating Speed. A raising time of 12 min and lowering time of 3 min was found to be successful for the Olmstead prototype testing.

9-5. <u>Hinged Crest Gates</u>. Hinged crest gates are mounted with a hinged connection at the crest of a spillway. Hinged crest gates are raised to achieve a closed position and lowered to achieve an open position. They can be designed to allow overtopping flow at any height through their range of travel. This functionality is most commonly used as USACE sites to provide a method of maintaining required navigation depths during periods of low river flow. Like wicket gates, hinged crest gates can provide unrestricted navigation when in the lowered (open) position. Hinged crest gates also have the capability to completely stop river flow if the upstream pool is held below the top elevation of the gate. This function can be used to increase upstream storage capacity.

Figure 9-20. Hinged crest gate, torque tube-style (viewed from the downstream side)

a. Operating Machinery. Hinged crest gate operating machinery opens and closes the gate by rotating it about the hinged pivot point. In addition to lifting or lowering the gate, the operating machinery might need to be designed to hold the gate at any position between open and closed.

b. Bearings. All bearings associated with hinged crest gates shall be of the self-lubricated type and designed in accordance with the criteria in Chapter 2.

c. Dogging Devices. Dogging devices also can be utilized to hold the gate at fully closed (raised) or any intermediate position. It also is recommended to provide dogging devices to hold the gate in the open (lowered) position to prevent flow vibration and gate movement if the hydraulic operating cylinder(s) are removed. This allows a method to perform maintenance, repair, or replacement of the mechanical operating system.

d. Pressure Relief Systems. Pressure relief systems should be considered for the hydraulic operating systems of hinged crest gates located on navigable waterways where vessel impacts could cause unacceptable damage to a gate. These systems utilize a pressure relief valve to rapidly release hydraulic fluid from the cylinder bore into an auxiliary reservoir sized for the maximum volume of hydraulic fluid that can be contained in the cylinder bore. This system is activated only by a spike in the cylinder pressure and relies on no electrical devices for its operation or activation. After impact release, the hydraulic fluid in the auxiliary reservoir must be drained into a suitable container and manually returned to the hydraulic power unit's (HPU) main reservoir.

e. Redundancy. A backup HPU should be provided to allow operation of the gate if the primary HPU experiences a failure. The gate cylinders for multiple gates should be supplied with hydraulic power from at least two main HPUs, as well as from two separate accumulator HPUs. Typically, all HPUs are located in a control tower. The main hydraulic system raises and lowers the dam gates one at a time.

f. Debris Detection Systems.

(1) Debris buildup behind closed hinged crest gates can inhibit operation. Some installations have incorporated detection systems to notify the operator of buildup.

(2) Operating System Design. In addition to supporting hydraulic and gravitational loads, hinged crest gate operating systems shall be designed to operate a gate with a minimum of 5000 lbf·ft of ice and debris impact load applied to the top of a gate.

g. Cylinder-Operated Torque Tube Gates. Cylinder-operated, torque tube hinged crest gates (Figure 9-20) use one or more hydraulic cylinders located at the end(s) of the torque tube to apply torque to lift or lower the gate. The torque tube extends between piers and is supported by intermediate bearings. The ends of the torque tube extend into a gallery in the piers, which house the operating hydraulic cylinder(s) and HPU. The torque tube penetration into the piers is sealed to prevent water from entering the gallery when the pool elevation is above the torque tube. The gate leaf is cantilevered off a rigid connection to the torque tube. To operate the gate, the torque generated by the hydraulic operating cylinders must overcome the torque created by loads acting on the gate leaf that are not acting through the center axis of the torque tube.

(1) Advantages. The primary advantage of a cylinder-operated, torque tube hinged crest gate is that the operating machinery can be enclosed in the pier gallery. Doing so provides maximum protection for the operating machinery that otherwise would be exposed to weather, water, debris, etc. Locating the machinery in an enclosed gallery also can provide better containment for hydraulic fluid leaks.

(2) Disadvantages. The primary disadvantage to the operating machinery of a cylinder-operated torque tube gate is the larger cylinder size requirements. The effective moment arm the cylinder uses to apply torque to the torque tube changes as the gate is rotated through its motion. The cylinder must be sized to operate the gate when its effective moment arm is shortest. Torque tube gates also have practical size limits for the gate leaf. The gate size tends to be limited by the strength of the rigid connection to the torque tube, which supports the cantilevered gate leaf. Torque tube hinged crest gates also require packing gland seals at the pier penetrations, which periodically require replacement.

h. Pierless Cylinder-Operated Torque Tube. A different style of torque tube hinged crest gate (described above) that locates the operating machinery in a dry gallery below the gate has been developed. This eliminates the need for piers that otherwise would create additional navigation obstacles. Similar to the torque tube gate described above, a pierless gate would require shaft seals to prevent leakage into the dry gallery. See Plates B-75 and B-76 for a typical system layout. These plates show the torque tube supported between two bearings that are the torque tube penetrations into the dry machinery gallery. Split seals and chevron packing is provided around the shaft at the bearing supports to prevent water leakage into the machinery gallery.

(1) Advantages. The primary advantage of this system is there is no need for piers to house the operating machinery, therefore, eliminating navigation obstacles.

(2) Disadvantages. The primary disadvantage of this system is the need for a dry gallery below the hinged crest gate to access the machinery.

i. Crest-Mounted Lifting Cylinders. Crest-mounted lifting cylinder hinged crest gates use one or more hydraulic cylinders mounted on the spillway crest below the gate. The hydraulic cylinders are attached to the back of the gate and are loaded in compression as they support the weight of the gate and other operating loads. Typically, a recess in the concrete foundation below the gate is needed to accommodate the required length of hydraulic cylinder.

(1) Advantages. Crest-mounted lifting cylinders eliminate the need for piers. Gates can be built directly adjacent to one another. These gates can provide unrestricted vessel navigation when the gates are fully opened.

(2) Disadvantages. Installations that have a spillway crest submerged by the downstream pool require the crest-mounted cylinders to be designed for operation in a submerged environment. Hydraulic system leaks end up directly in the waterway.

j. Pier Mounted Tension Cylinders. Pier-mounted cylinder hinged crest gates use to operate the gates cylinders that attach to the pier and upstream side of the gates. The cylinders are loaded in tension as they support the weight of the gate and other operating loads.

(1) Advantages. The operating hydraulic cylinders are loaded in tension and do not need to be designed for a buckling, therefore, reducing the required cylinder size.

(2) Disadvantages. A pier is required at each end of a gate to mount the operating hydraulic cylinders.

k. Inflatable Bladder Operation. Inflatable bladder-operated hinged crest gates, shown in Figures 9-21 and 9-22, use an inflatable bladder secured below the gate to raise and lower the gate. Bladders are inflated with air or water from a supply piping embedded in the spillway. Straps are attached to the sill and backside of the gate to provide a physical gate stop to prevent over-rotating the gates. Inflatable bladder systems are discussed further in Section 9-9 below.

(1) Advantages. Inflatable bladder systems typically have a lower initial cost than hydraulic cylinder-operated systems. The gates tend to be much lighter and have reduced foundation requirements than hydraulic cylinder systems.

(2) Disadvantages. The air bladders are susceptible to vandalism and other operational damage. Gunshots easily can puncture the air bladders, inhibiting operation of the gates. Ice expansion and heavy winds have been known to overload the bladder or restraining strap. The potential for damage from operational factors can be minimized if the bladder gate system is designed properly. The bladders also require replacement on a 10-to-20-year cycle, which can contribute to the life cycle cost.

Figure 9-21. Air bladder-operated hinged crest gate

Figure 9-22. Inflated air bladder below closed hinged crest gate

9-6. Roller Gates. Roller gates are large cylindrical gates suspended between spillway piers that are raised or lowered through diagonal slots in the piers. Roller gates are used to regulate water flow and have been designed so flow can pass over or under the gate. Roller gates typically are designed so their maximum hoisted elevation is above flood conditions on the river. This allows for unobstructed flow in flood situations. Roller gates were a popular gate choice worldwide in the early 1900s. Numerous roller gates were installed in the 1930s in USACE districts including the St. Paul, Rock Island, St. Louis, and Huntington. Most of these installations are in service today. However, roller gates are not typically used in new installations. The lower cost and other operational advantages of tainter gates have made new installations of roller gates obsolete. The transition from roller gates to tainter gates is captured in Chapter VII of Gateways to Commerce: The U.S. Army Corps of Engineers' 9-Foot Channel Project on the Upper Mississippi River. Because roller gates usually are not used for new installations, the content of this manual will be limited to a short description of their advantages, disadvantages, and operation. Specific design guidance for roller gates is provided in the Fort Belvoir Engineer School Design Manual Canalization of the Upper Mississippi River and Ohio River.

Figure 9-23. Roller gate

a. Operation. Roller gates are chain hoisted from one end of the gate through angled slots in the gate's supporting piers. The angled slots contain inclined racks that interface with cogs on the ends of the gates. The rack-and-cog design allows the gates to move or climb up the racks as the chain hoists the gate. The chain hoists are mounted permanently on the piers above the gates and are operated with electric motor hoist equipment.

b. Advantages. Roller gates are some of the widest available. Most USACE roller gate installed are between 60 and 110 ft wide. This allows the use of fewer gates to regulate flow. Another main advantage is that roller gates can pass flow over or under the gate. The ability to pass flow over the gate allows for superior ice and debris passing capabilities.

c. Disadvantages. Roller gates are known to produce dangerous undercurrents, which can be hazardous to river users. Anything drawn into the dam most likely is pulled under the gates by the strong undercurrents. This has happened at Lock 3 on the Mississippi River. The safe clearance distance at Lock and Dam 15, on the upper Mississippi River in the Corps' Rock Island District, is 600 ft upstream and 150 ft downstream of the gates.

d. Existing Installations. One of the most well-known uses of roller gates is at Lock and Dam 15. The dam there has 11 roller gates used to maintain minimum a 9-ft. river depth required for tows to navigate the river safely. St. Paul District typically has five or six roller gates at each lock and dam site. The St. Paul, Rock Island, or other districts mentioned above may be contacted for more information on roller gates.

9-7. Drum Gates. Drum gates are buoyant gates of hollow steel construction that are installed between spillway piers and attached to the spillway by a hinged connection at their upstream edge. In the lowered (open) position, the gates sit in a recess/chamber built into the concrete spillway. When fully lowered in the concrete recess, the top surface of a drum gate forms the crest of the spillway ogee. In the open/lowered position, a drum gate forms an overflow-type spillway. To raise a drum gate, a system of pipes and valves are used to fill the concrete recess, usually with water from the upstream reservoir. The buoyant gate floats on top of the water as the recess is filled. In a raised position, floating on top of its recess filled with water, a drum gate forms a barrier to regulate or stop water passage. Similar to hinged crest gates, drum gates spill only from overtopping flow. In the raised position, side seals are utilized to seal against the adjacent spillway piers and prevent water passage around the sides of the gate. To lower a gate, a system of pipes and valves, separate from the pipe/valve system used to fill the recess, is used to drain water from the concrete recess. This typically is done by a gravity flow system on the downstream side of the gate. Drum gates are not common at USACE sites, but have been used successfully on privately owned dams. Because they are rare at USACE sites, the content of this chapter will be limited to a short discussion of major considerations, advantages, and disadvantages.

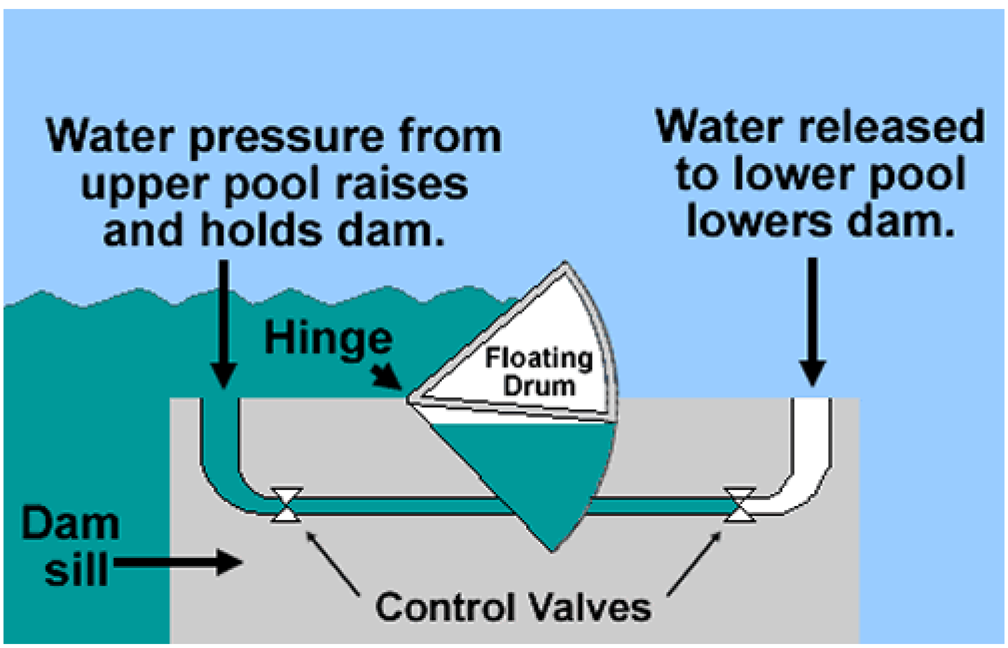


Figure 9-24. Drum gate

a. Major Considerations. While well-designed drum gates can operate reliably, they do have vulnerabilities that need to be understood and monitored for successful operation. Proper function of the concrete recess, filling and draining system, and gate features critical to the buoyancy of the gate is necessary for reliable operation. The main failure mode of a drum gate, unintentional lowering, usually is caused by one of these systems. An operational failure of a drum gate could result in the uncontrolled release of water and loss of the upstream reservoir water elevation.

(1) Float Recess. Maintaining water level in the float recess is critical to the operation of a drum gate. Leakage of the drain system could result in unintended lowering of a gate. Often, the supply system is sized to provide water at a faster rate than the drain system. This allows for a method of holding a gate in the raised position in the event of a failure of the concrete float recess draining system.

(2) Gate Buoyancy. Maintaining buoyancy of a gate is critical to successful gate operation. Drum gates are typically of welded steel construction and rely on welds for structural integrity and to seal the structure from water penetration. The structure must be inspected regularly to ensure adequate performance of the water seal welds. Access to the inside of the gate needs to be considered for inspection and maintenance. Typically, a gravity drain system is used inside the gate to help maintain gate buoyancy if there is water leaking into the hollow gate. The drain systems use a flexible hose to drain water that collects inside the gate to the downstream side of the dam or a drain line in the concrete section of the spillway.

b. Advantages. A major advantage of drum gates is the elimination of an expensive electric motor or hydraulic operating system. While the operating system for drum gates is critical, it also is simple and relatively inexpensive. Using the buoyant system to operate the gates also minimizes the size and strength requirements for the piers relative to other common spillway gates. Another advantage is the minimal power requirement to operate the gates. The main energy used to operate the gates is hydrostatic pressure provided by the forebay.

c. Disadvantages. The need to seal the hollow interior of a drum gate can make fabrication more complicated than other common spillway gate types. Also, the design and construction of the concrete spillway is more complicated. The interior of a drum gate typically is classified as a confided space that requires training for maintenance and inspection activities. Depending on the application, the tendency for the main operational failure modes to cause a gate to fail in the open position can be a major disadvantage.

9-8. <u>Bear Trap Gates</u>. Bear trap gates operate with a similar concept to drum gates, but utilize two gate leaves to form the barrier against water passage. The downstream gate leaf is a buoyant leaf and works with the float chamber to close (raise) the gate system. Like drum gates, bear trap gates use a float chamber formed by a recess in the spillway concrete to operate the gate leaves. The float chamber is filled and drained with a system of pipes and valves similar to that of a drum gate. However, sometimes the operation of bear trap gates is assisted by air or hydraulic cylinders. To lower bear trap gates, the float chamber is drained. When in the fully open (lowered) position, both gate leaves lay flat on the spillway with the downstream leaf tucked under the upstream leaf. The operation of a bear trap gate system is more complicated than most other gates. Bear trap gates are not common at USACE facilities, so the content of this chapter will be limited to a brief description and a discussion of major advantages and disadvantages. Specific design guidance for bear trap gates is provided in the Fort Belvoir Engineer School Design Manual Canalization of the Upper Mississippi River and Ohio River.

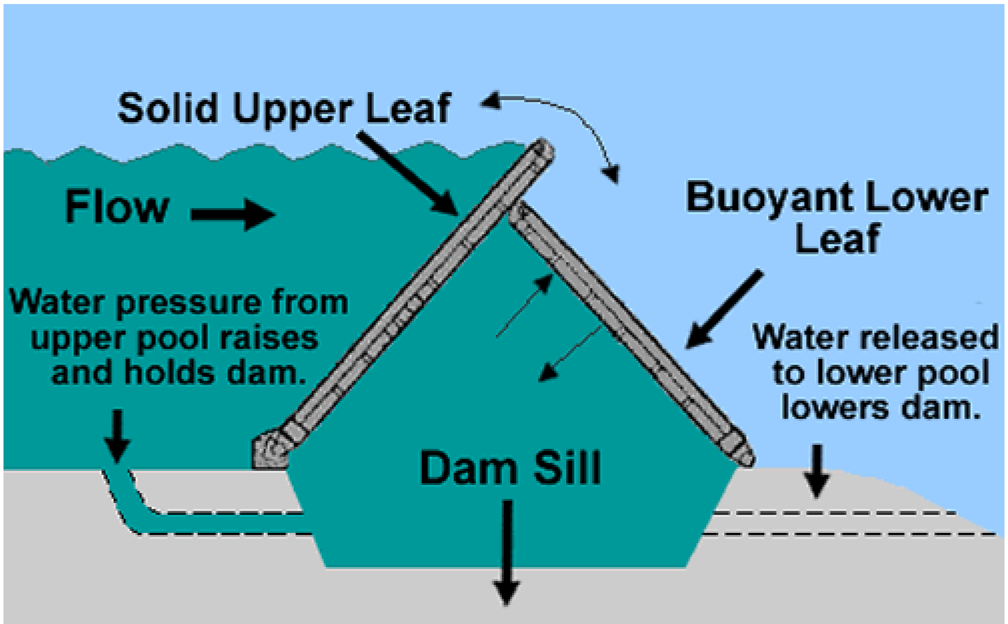


Figure 9-25. Bear trap gate

a. Advantages. The advantage of bear trap gates over drum gates is simpler design and fabrication of the gate leaves. Like a drum gate, a major advantage of bear trap gates is the elimination or minimization of expensive electric motor or hydraulic operating systems. While the operating system for bear trap gates is more complicated than a drum gate, they are still relatively simple, inexpensive, and have low power requirements than most other spillway gates. Using the buoyant system to operate the gates also minimizes the size and strength requirements for the piers relative to other common spillway gates.

b. Disadvantages. Bear trap gates share the same disadvantages as drum gates. In addition, the dual gate leaves of bear trap gates create the need for a more intricate and complicated sealing system for the float chamber.

9-9. Bladder Dams. Bladder dams consist of an inflatable rubber bladder anchored to a spillway crest and walls. The bladder dam is inflated with air or water to create a barrier to stop water flow. Supply piping is provided to inflate the dam and typically is embedded in the foundation upon which the bladder is installed. When deflated, the bladder is on the bottom of the river and provides unrestricted flow. Bladder dams have been used primarily at USACE sites to increase reservoir storage capacity. However, in other countries, they also are used as storm surge flood barriers. Bladder dams have been used at many sites around the world and are becoming more popular in the United States. Bladder dams provide a low initial cost alternative to traditional steel gates. The design of bladder dams is a specialized trade. A strong background with bladder dam systems is needed. Bladder dam installations at USACE sites likely would be designed by the bladder dam manufacturer to comply with a USACE-written performance specification. Because of the specialized knowledge required to design bladder dam systems, specific design details will not be discussed in this manual. The discussion will be limited to the advantages, disadvantages, and major considerations.

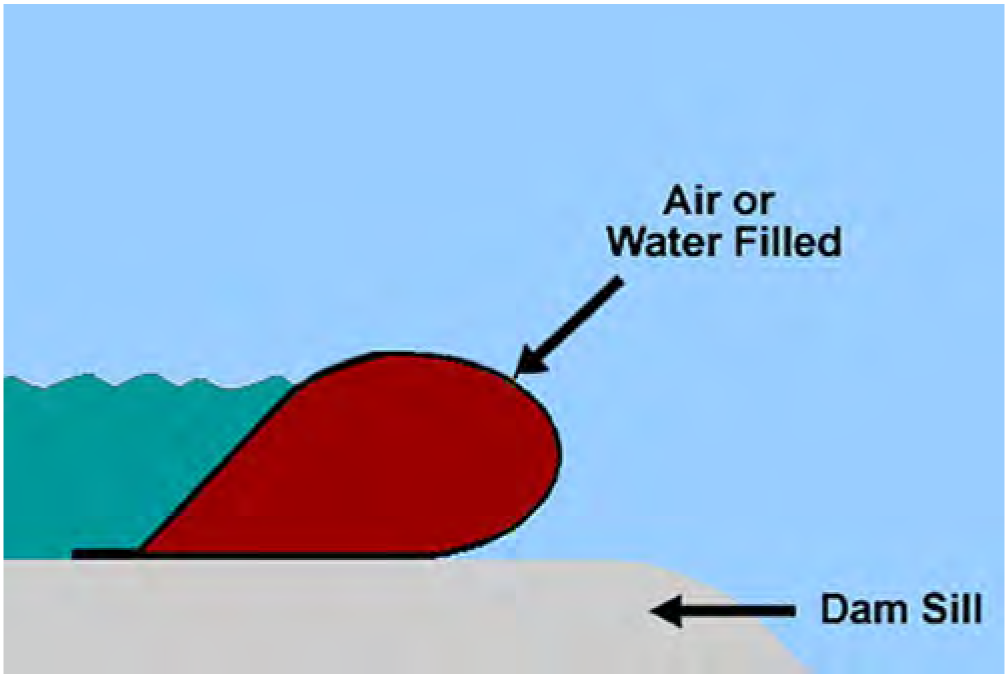


Figure 9-26. Bladder dam

a. Operation. Bladder dams are inflated through supply pipes by an air compressor or water pump. The air or water pipe system is typically is embedded in the bladder dam foundation. Operating systems consist of a compressor, piping, valves, and a control system. The bladder systems typically operate with air pressures around 10 psig. Some installations fill the bladder with a combination of air and water, which can improve the stability of the dam. However, using water to inflate a bladder dam is not recommended for situations in which the water could freeze.

b. Storage. When deflated, the bladder systems lay flat on the river bottom. The Ramspol storm surge flood barrier in Holland is the world's largest bladder dam. When deflated, the Ramspol bladder dam system remains in a standby position in a recess in the bottom of the waterway. Storing the bladder in the recess helps to protect the bladder from debris punctures and exposure to ultraviolet light.

Figure 9-27. Ramspol storm surge protection bladder dam (courtesy of R.A. Daniel, Rijkswaterstaat, Division of Infrastructure, Netherlands)

c. Advantages. Low initial cost is the primary advantage of bladder dam systems. Installation can be significantly less expensive than a traditional steel gate system requiring a hydraulic cylinder or other mechanical operating system. Compared to a steel gate system, bladder dams are also easier to maintain. Some bladders, mainly those exposed to ultraviolet light, require periodic lubrication. A bladder dam's simple operating system requires little maintenance, compared to a traditional steel gate system.

d. Disadvantages. The main disadvantage of a bladder dam, compared to steel gates, is the potential for puncture damage. Bladders dams typically are made up of one inflatable section. This means punctures to the bladder can jeopardize the effectiveness of the whole bladder system.

e. Considerations.

(1) Punctures. Puncture damage is the biggest vulnerability of a bladder dam system. Unlike bladder-operated hinged crest gates, bladder dams do not have steel barriers to shield the rubber bladder from debris impacts or vandalism such as gunshots. To minimize puncture potential, some bladders are reinforced with Kevlar or steel materials.

(2) Piping System. The air pipe systems have a risk of filling with water. The air pipe systems typically are embedded in a submerged concrete foundation. This creates a potential for the low-pressure air pipe system to be exposed to hydrostatic water pressure. In some cases, water has flooded the air piping system. Design of the air pipe systems should allow for a method to drain water. Piping galleries have been used to minimize the potential for water entering into the piping system. However, adding piping galleries will increase cost substantially.

(3) Maintenance. Some rubber dams require periodic lubrication of the rubber bladder.

CHAPTER 10

Other Systems and Ancillary Equipment

10-1. Introduction. This chapter discusses and provides engineering design guidance on ancillary equipment and other mechanical and electrical systems for navigation structures. This includes tow haulage systems, ice and debris control, floating mooring bitts, ship arrestors, firefighting systems, lock dewatering systems, and cathodic protection systems.

10-2. Winch or Tow Haulage Systems. Winch systems or tow haulage systems at navigation locks provide the capability of moving commercial vessels through the lock chamber. In the United States, many commercial tows are longer than the lock chamber. A commercial tow consists of a towboat pushing multiple barges tied together. For example, most of the locks on the Mississippi River are 182 m (600 ft) in length, and tows can be 364 m (1200 ft) in length. The tow (barge sections) needs to be split in half to lock through the chamber. Once the barges are split, the winch or tow haulage system is utilized to pull the first barge section through the chamber while the tow boat remains with the second barge section. The winch (and travelling kevel discussed below) typically pulls the first barge section to the end of guide wall and past the miter gates. This allows the towboat and second barge section to lock through.

Figure 10-1. USACE, commercial tow split apart

a. Types of Tow Haulage Units. EM 2602 provides additional design guidance for tow haulage units. The simplest tow haulage installation is a pair of single-drum hoists, usually electric or hydraulic. One hoist is on the top of the lock guide wall upstream from the upper gate bay. The second usually is downstream from the lower lock gate on the lower guide wall. Many sites on the Mississippi River utilize this system.

Often, the downstream unit is not used. Rather, the tow is flushed out of the chamber by cracking open a culvert valve. The downstream winch unit is used when the tailwater and pool elevations are nearly the same, or during flood conditions. The tow haulage unit always should be on the guide wall side, and upstream and downstream approach walls (guide walls) must be located on the same side of the lock in order for the tow haulage unit to function properly.

Figure 10-2. Mechanical winch unit with level wind system

(1) The free end of the winch line is spooled off the drum and goes through a fairlead mounted on the lock wall. Many winch units also utilize a level wind system (Figure 10-2) to insure the wire rope is evenly spooled off and wound back on the drum. The line then is fastened to a bitt on the back barge of the first section. The barge’s sections are pulled out of the chamber and attached to a travelling kevel. Once the barges reach the end of the guide wall, they are attached to check posts or line hooks until the second section is locked through. The travelling kevel rides on a rail system along the guide wall. Further discussion is below.

Figure 10-3. Fairlead for the tow haulage winch

Figure 10-4. Hydraulic-operated tow haulage unit with level wind system

(2) Another type of winch unit consists of a single reversible hoist located near the center of the lock. An endless cable is utilized that runs along the face of the lock wall and around sheaves near the gate recesses. The wire rope comes with a flexible fiber line long enough to reach the pool level and is fastened to the back barge either directly or with an intermediate hawser (line). The single-hoist layout always should be used because the double-hoist arrangement with a single line is extremely dangerous to operating personnel. The hoist drum is designed so, as the cable (wire rope) is paid out at one end at the bottom of the grooved drum, it returns to the top of the drum at the

same end. The length of the cable on the drum equals total travel plus two wraps. The most sophisticated system uses a reversible hoist and endless cable that pulls a wheeled towing bitt on the top of the wall between the gate bays. The bitt may travel in a recess provided for it, or it may be mounted on a rail fastened to the concrete. The latter is used for locks already constructed or in cold climates where snow and ice would clog the recessed type. A hawser furnished by the tow is slipped over the traveling bitt and fastened to the back barge as before. It is important that the lock operator be able to see the barges that are being pulled during the entire operation. Therefore, the lock operator should operate the tow haulage unit from the lock wall opposite the tow haulage unit.

(3) Within Chamber Tow Haulage System.

(a) The Corps' Pittsburgh District, utilizes another type of tow haulage system. It is mounted within the chamber because the upper and lower guide walls are too narrow. Two hydraulically driven winches pull against each other in a counter-torque (pull-retard) fashion. This eliminates the slack rope problems of an endless system, so the wire rope tracks properly over all sheaves and stacks correctly in even layers on each drum. The alignment and distance from a winch drum to the first (lead) sheave is set to provide the proper fleet angle. Using two smaller winches located at each end of the chamber also allows better integration of the system into lock.

(b) The control system is designed to vary infinitely the pulling speed and control the maximum line pull while controlling the trailing winch hold-back tension in a regenerative mode. The hold back automatically regenerates approximately 90% of the hold-back power to minimize power loss and heat buildup in the hydraulic system. The control system permits each winch to be operated in either the hauling or hold-back mode, depending on the direction of the joy-stick movement from neutral. Electric drive must have a variable speed control, as the barges will accelerate slowly.

b. Travelling kevels. All tow haulage units should be furnished with travelling kevels installed on a rail system. The purpose of these unpowered travelling kevels is to hold the head of the tow into the guide walls as the haulage unit pulls the tow out of the lock chamber. The kevels and tow haulage winch work as a system. The tow haulage winch will pull and accelerate the first barge sections out of the lock chamber. Once the barges are accelerated, the mooring lines are then are attached to the travelling kevel. The kevel guides the barge sections along the guide wall. On the Mississippi River locks, the kevels are sometimes called mules or travelling mules.

Figure 10-5. Travelling kevel

(1) One of the kevels should be located on the upstream guide wall and another on the downstream guide wall. Typically, the kevel will travel the full length of the guide wall. This allows the first barge section to clear the lock completely. Ideally, the first barge section and kevel should slow and nearly come to rest once the end of the guide wall is reached. Once the barge section and kevel reach the end of the guide wall, the barge section is tied off and moored. The first barge section must completely clear the lock gates in order for the second barge section to lock through.

(2) The minimum length of travel of each of these kevels should be equal to the travel of the tow haulage unit. The length of travel for the tow haulage unit should be equal to the clear inside length of the lock chamber (the distance between downstream miter gate recess and upstream miter sill). A power-retrieved kevel can be provided. This will eliminate the need for the lock operator to walk the length of the guide wall and return (retrieve) the kevel manually. One issue with travelling kevels is the rail system and the gaps between the rails. Kevel rails should be continuous to avoid gaps in the rail. Any gaps or misalignment of the rails can cause the kevel to stop. This poses a safety risk to lock personnel and barge operators because the mooring lines can snap and the kevel can break and launch into the air.

Figure 10-6. Gaps in rail section of travelling kevel

c. Lessons Learned. General design guidance from EM 2602, plus some lessons learned:

- Maximum line force on winch unit should not exceed 10,000 to 12,000 lbf or 44,480 to 53,376 N.
- Provide level wind feature on all winch units.
- Speed should be 100 ft/min, or 0.5 m/sec, maximum on winch units. Barge sections should be accelerated up to this value.
- For electric drives, provide variable frequency drive for speed control.
- Hydraulic drives should provide for inherent speed control.
- Hydraulic drives should have removable (plug-and-play) power units. These can be disconnected and stored during flood conditions.
- All controls should be installed above flood levels.
- Misalignment of travelling kevel rails has been an issue. This can cause the travelling kevel to stop while the barges are moving. At one site on the Mississippi River, this caused the mooring line and kevel to break. Routine inspection is necessary.
- Inspect travelling kevel rails and anchors for corrosion. Failure of anchors could cause the rail to pull out of the concrete.
- New lock construction should be designed for a kevel rail system to minimize rail misalignment and provide for a continuous rail system. The Corps' Nashville District has designed one such system.
- New lock construction should be designed to embed the tow haulage winch line. This makes for a safer operation.
- Provide spare travelling kevels.

d. Additional Design References. Available design guidance include:

- United States Army Corps of Engineers, Engineering Manual EM 1110-2-2602, 30 September 1995, Planning and Design of Navigation Locks, Chapter 5, Paragraph 5-11, Tow Haulage Units, and Chapter 10, Paragraph 10-4, Tow Haulage Unit and Movable Kevel;
- PIANC Report WG 106 Innovations in Navigation Lock Design, 2009, Paragraph 5.7.1.4, Bollards on Tracks (Tow Haulage Unit and Movable Kevel).

10-3. Ice and Debris Control. Ice and debris control systems utilizing compressed air are used on many USACE locks and navigation structures. These systems also are used on the St. Lawrence Seaway and European navigation structures. There are other means to address ice conditions at navigation structures, including mixers and heaters. PIANC report WG 106 Innovations in Navigation Lock Design discusses various other means for ice control in lock structures. USACE EM 1110-2-1612 (2002 and 1999) discusses other means of ice control, including air bubbler designs and heaters. Other design references are noted below. The U.S. Army Cold Regions Research and Engineering Laboratory also can provide design guidance. The discussion in this section will focus on compressed air bubbler and air screen systems. Some design considerations, advantages, disadvantages, and lessons learned will be presented.

Figure 10-7. Soo Locks on the Great Lakes, typical ice conditions in winter

a. Summary and Overview. The basic system utilizes an air plant or compressor, supply piping with manifolds and control valves, and diffuser piping (installed on the gates, lock floor, and in the gate recess) to aerate or bubble the water in front of lock gates and across the gate sills or lock floor. The system is effective for clearing ice and debris from the miter gate recess and to prevent slush ice from entering the lock. Some sites have used a central air plant, while others have used smaller systems or localized plants spread out on the lock. For existing locks, the installation of high-volume bubbler systems should be included as part of lock and dam major maintenance contracts. This allows the installation of the submerged pipe and accessories while the lock is dewatered, and reduces the overall cost to install submerged high-volume bubbler piping.

Figure 10-8. Localized compressor installed on miter gate machinery

Figure 10-9. Central air compressor plant

b. Advantages and Disadvantages of Air Bubbler and Air Screen Systems. Air systems provide several advantages over mixers and heaters. They are less labor intensive than a mixer system. The operational cost is generally lower then electric heaters. A heater system is ineffective for moving large ice floes and for pushing debris out of the gate recess and quoin area. Below is a summary of some of the advantages and disadvantages.

Advantages:

- Controls ice and debris in front of miter gates and in gate quoin area.
- Controls accumulation of ice in front of miter gates.
- Reduces manpower and time required to remove ice and debris.
- Clears ice and debris from miter gate recess.
- Provides a screen to prevent ice from being pushed into the lock.
- System can be designed to modulate air flows and to distribute air flows as required.
- Lower operational cost than utilizing a heating system on lock gates.

Disadvantages:

- Operation cost of running air compressor.
- Initial capital cost of system.
- Maintenance to underwater piping and components requires a dive crew or a lock dewatering for repair and replacement.
- Towboat prop wash can damage underwater piping and components.

c. Lessons Learned. Air bubbler systems have been used successfully at a number of lock sites. Without an effective air bubbler system, ice or debris can build in the miter gate recess. This affects the ability of the gate to fully recess. Slush ice also can build in front of barges entering the lock chamber. Bubbler diffusers installed on the gate sill or across the lock floor can prevent much of this ice from being pushed into the lock. The air flow rate directly impacts the size of the air plant and should be optimized. This generally will be site specific and depends on the amount of debris normally passed through the lock. Locks in northern climates have reduced barge traffic during the winter or are completely shut down. Reduced air flows can be used to keep ice off the miter gates and out of the gate recesses. Low-flow systems can bring warmer water from the bottom of the lock.

(1) The USACE Soo Locks on the Great Lakes have an extensive air diffuser system and extensive ice conditions during late season navigation. These locks generally are shut down in mid-January and open again in mid-March. Ship traffic needs to pass through the locks until closure. At the Soo Locks, 3-m-thick (10-ft-thick) brash ice can build in front of the miter gates. They utilize air curtains across the lock to move ice for setting bulkheads and to move miter gates. They also utilize point source bubblers (550 ft^3/min or 15.4 m^3/min) for controlling ice at multiple other locations. The system utilizes individual control of each point source bubbler and utilizes a central plant with three rotary screw drive compressors.

Figure 10-10. Soo Locks, air curtain system in operation

(2) Holes at the bottom of the pipe are important in order to allow water and debris out when air is forced through and for proper air distribution. The hole diameters should vary between 6.35 mm to 15.8 mm (1/4 in. and 5/8 in.), depending on the air flow and spacing. Utilize a welded outlet over diffuser holes, and a threaded nipple to take standard rubber check valve or nozzle configurations, therefore, facilitating exchange and replacement. Proper attachment to the bottom of the lock is critical because ships create turbulence that might move improperly fastened piping. Piping diameter must be designed properly to avoid choke points causing loss of efficiencies. Pipe connections are a common place for air loss; therefore, grooved type couplings or similar are recommended. Ideally, a cleanout port should be located on the end of air curtains that can be opened by a diver to flush out debris that forms inside the pipe (provide a larger diameter than holes). Consideration should be given to installing a smaller diameter line at the opposite end of the feed line that can be used as a backwash or purge line. The smaller line needs to be away from navigation or other contact points to prevent failure.

(3) Some other lessons learned:

- Both central plant and localized plant systems have been successful. The selection of the type of plant is site specific. There are many sites that use either. For sites with localized plants, utilize a compressor on each gate leaf.
- Rotary screw-type compressors have been used successfully and are preferred. They can run for long periods with little maintenance.
- Specify low-ambient temperature compressor enclosures to permit operation at ambient air temperatures as low as -28.8°C (-20° F).

- Environmentally friendly or environmentally acceptable oil has been used successfully on both the large central compressors and air blowers.
- Size back-up generators to accommodate both the electrical load from the lock and dam and the electrical load of the compressor.
- High-volume systems work well for moving ice and reducing brash ice.
- Low-volume systems can be utilized to transfer warmer water at the bottom of the lock to top water surface. Low-volume systems can run continuously and be used during the winter in northern climates to keep ice off lock gates.
- Rubber pinch-type check valves have worked successfully in keeping silt, debris, and zebra mussels out of the nozzles and piping system. Always provide a diffuser system that has a means for preventing backflow into the pipe.
- Heat tape has been used successfully on air pipe at the water surface to prevent ice buildup inside the supply pipes.
- Check valves within the vertical piping have not been 100% reliable, and freezing in the pipes has occurred. It might be better to install isolation valves, cross fittings, and pipe plugs to allow lock personnel to either charge the vertical piping with air or fill them with environmental RV antifreeze. Charging the piping with air forces the static water level below the freezing surface, and is the preferred method.
- Stand pipes with isolation valves are used successfully to allow for addition of alcohol or antifreeze to clear ice buildup.
- Space check valves and nozzle diffusers 0.912 to 1.216 m (3 to 4 ft) apart. Install orifices and check valves pointing down. This reduces the chance of the diffusers being damaged by barge tows and debris. Orifices installed pointing down also trap the air remaining in the screen piping after the screen is turned off. The trapped air substantially reduces the amount of water in the pipes the next time the system is used. As a result, the orifices begin bubbling sooner.
- Use all stainless steel pipe, fittings, and components for below-water diffuser and supply piping. Do not use any galvanized components below water or mix and match galvanized components and stainless steel components. Provide dielectric couplings between galvanized and stainless steel components.
- Air blowers or air compressors can be utilized. However, air blowers are limited to around 89.6 kPa or 0.896 Bar (13 psig). The advantage of an air blower is that it will provide more air volume. Sites with long supply lines and deep drafts will need to use air compressors.
- For air blower systems, minimize air supply friction losses to 6.895 kPa to 13.790 kPa (1 to 2 psig).
- Orifices in supply piping should be 6.35 to 9.5 mm in diameter (1/4 to 3/8 in.). Higher volume systems can utilize up to 15.8 mm diameter orifices (5/8 in.).
- Ball valves or positional butterfly valves with 90-deg full open to full close operation are best suited to deliver the air to the bubbler screens.
- Provide point source air bubblers for control of ice in multiple locations.
- Provide programmable logic controllers (PLCs) and variable frequency drives for control of air compressors. This will allow modulation of air flows.

- Ultraviolet protection is required for all exposed compressor controls to prevent deterioration.
- Piping should be attached using stainless steel offset clamps.

d. Piping, Manifolds, and Diffusers.

(1) Air manifolds should be provided at each gate corner. Run the supply line from the air compressor or blower directly to the manifold and not directly into the supply piping. The manifold will allow proper distribution and control of air.

Figure 10-11. Typical diffuser supply system and manifold

(2) Use butterfly valves on all supply lines to allow for closure and throttling. Electric solenoids also can be used to actuate the valves. Piping should extend from each manifold down to the lock floor and lock gates. Diffusers should be provided for distribution of air at the bottom of the lock. The diffuser consists of an orifice drilled into the pipe. A welded outlet is placed over the orifice and a pipe nipple then is threaded into the welded outlet. The rubber check valve then is placed over the pipe nipple. Rubber check valves have been successful at multiple locations on the Mississippi River in the United States. These diffusers can generally be replaced by divers.

Figure 10-12. Typical manifold with control valves

Figure 10-13. Rubber pinch-type check valve

e. High-Volume Air Systems. High-volume bubbler systems provide lock personnel a means to control debris and ice formation, and ice movement. The Soo Lock system was noted above. This has proven to be highly effective. High-volume systems also are installed on the locks on the Ohio, Mississippi, and Illinois rivers. Technical information available to designers considering installing high-volume bubbler systems are in EM 1110-2-1612, EM 1110-8-1(FR) Chapter 6, REMR Bulletin Vol.12, No. 2, May 1995, and the Cold Regions Technical Digest, No. 83-1. These documents provide valuable guidance in designing high-volume bubbler systems and the theories involved with using air to melt ice. Controlling the formation and movement of brash ice improves the efficiency of lockages. The benefits are:

- Fewer lock personnel are required to assist with the lockage.
- Less physical work from lock personnel is required to push ice with long pick poles. This promotes a safer working environment.
- The time required to perform a lockage during winter ice conditions can be reduced.
- Controlling ice against the lock gates reduces gate operating machinery wear and tear. Stresses imposed upon the gate's structural members are lower. Machinery life and structural components life, and time between periods of major maintenance are extended.
- Adhesion of ice to the lock structure and gates can be minimized by the melting action associated with the use of high-volume bubblers. Ice of varying thickness can be melted in areas contacted by the released air bubbles.

(1) The Pittsburgh District has installed high-volume air systems, and has learned some lessons. The district noted that a quoin flusher does a good job of clearing floating debris before a miter gate is opened, while consuming much less air than a gate recess screen. These flushers consist of a single orifice located near the pintle of each miter gate. Each flusher is supplied by a smaller 19-mm (0.75-in.) line and is solenoid operated from the control station. Standard procedure is to operate the quoin flushers briefly each time the gates are opened. The gate recess screens still are needed for ice and heavy debris. The St. Paul District also utilizes quoin bubblers with a 2-in. supply line. They have also proven effective for clearing ice and debris in the gate quoin area. A 2-in. pipe cap is utilized with three, 5/16-in. diameter holes drilled in the cap.

(2) The Pittsburgh District also did an experiment with orifices facing up and down. Bubbler systems were installed on two identical locks with the exact same arrangement, except the orientation of the orifices. The orifices for one system were installed pointing up, while the orifices for the other were pointed down. This was done to see if the response time of these systems could be improved. Each system had quoin flushers and flushing screens for the miter gate recesses, upper bulkhead seal, and upstream approach. As expected, the system with the orifices pointing down had a significantly faster response time (time required for all orifices in a screen to begin bubbling). Orifices installed pointing up allow all of the air remaining in the screen piping after it is shut off to escape. As a result, the pipes are full of water the next time the screen is needed. It takes time for the incoming air to displace the water in the piping through the orifices. The orifice closest to the supply line starts bubbling first, and each successive orifice follows until the last one in the screen begins to bubble. Orifices installed pointing down trap the air remaining in the screen piping after the screen is turned off. The trapped air substantially reduces the amount of water in the pipes the next time the screen is needed. As a result, the orifices in the screen begin bubbling sooner, with many starting about the same time. Once all the orifices in a screen were bubbling fully, there was no observable difference in the bubbling action between the two systems.

f. High-Volume System at Starved Rock and Peoria Lock. A high-volume system was designed and installed at Starved Rock and Peoria Lock. The major components of high-volume air systems are modeled from the research and design calculations conducted by CRREL. The findings of the research laboratory are from a prototype installation at Starved Rock and Peoria Lock. This research should form the basis of design for high-volume air systems to control ice at locks. The components of the system described below are particular to systems installed on the Mississippi River.

(1) The compressors are 150 HP, electric motor driven, positive displacement rotary screw-type. Each compressor is capable of delivering 1275 m^3/hr (750 cfm) of free air at 690 kPa (100 psig) full flow and is designed for continuous operation. One compressor serves each bubbler system. The compressor delivers flow to the upstream and downstream gates. Compressor sizing is determined by an iterative air system analysis, which determines air discharge rates from orifices in the piping, assuming a dead-end pressure. A computer program (Bub-300) developed CRREL is capable of making this simulation to achieve a 1% difference between the calculated and specified compressor outputs.

(2) Supply pipes traditionally have been 3-in., schedule-40 galvanized steel piping. Galvanized steel piping is acceptable for piping installed above the water line. The piping is routed from a centrally located compressor to each end of the lock chamber. Valve manifolds are installed near the gate recesses to control the delivery of air to each submerged flushing screen. The control valves typically have been 3-in. butterfly valves with manual control. Electric control valves were installed at Starved Rock and are well liked by the operators.

(3) The submerged piping varies in size from 76 to 32 mm (3 to 1.25 in.). The varying size is dependent upon the flushing screen being served and the proximity to the dead-end of the pipe. Galvanized pipe is not recommended for submerged piping. Galvanized piping typically has needed to be replaced in fewer than 10 years. Stainless steel piping is recommended for all submerged pipe. The chamber screen is maintained at 76 mm (3 in.), due to the volume of air being delivered and the distance across the lock chamber. This screen is 29 m (96 ft) long for a 33.5-m-wide (110-ft-wide) chamber and is designed with 2.4-m (8-ft) orifice spacing. Gate recess screens are supplied with 76-mm (3-in.) piping and reduced accordingly to meet the requirements established by CRREL. The gate recess screens have varying orifice spacing to provide more air near the quoin end of the gate. The orifice spacing follows the recommendations of EM 1110-8-1(FR). Nine orifices are installed along each gate recess flushing screen.

(4) Drilled pipe plugs provide the desired quantity of air to the water. The pipe plugs are installed in vertical tee fittings along the horizontal pipe runs. Holes that are 9.5 mm (3/8 in.) in diameter have been determined to deliver the desired quantity of air from the prototype installations. A design flow rate of 51 m^3/hr (30 cfm) per orifice is desired.

g. Design Procedures. The USACE EM 1110-2-1612, 30 October 2002 and 30 April 1999, both provide detailed design guidance and calculations for sizing supply lines and air diffuser lines, and are references to this engineering manual. The detailed equations will not be repeated in this manual. CRREL also is a source for design guidance.

(1) The numerical analyses for air discharge rates are determined by an iterative procedure starting with a trial dead-end pressure at the end orifice. Working toward the supply source, the air flow and pressure at each orifice and in the supply line are calculated to obtain a calculated compressor pressure. The trial dead-end pressure then is adjusted and the procedure repeated until the calculated and the true compressor pressures agree. The sum of the nozzle flows gives the required compressor capacity.

(2) Output pressure must be high enough to overcome hydrostatic pressure at the submergence depth, and frictional losses in the supply and distribution lines, and still provide a pressure differential at the last orifice to drive the air out at the desired rate. Supply and distribution line diameters should be large enough so that frictional pressure losses along the line are small. A small increase in line diameters often results in significant reduction in frictional losses and more uniform discharge rates along the line. Orifice diameter and spacing should be selected to maximize rates. Too large an orifice diameter can result in all the air being discharged at one end. Submergence depth will be dictated by operational limitations but should be lower than the expected depth of trash pile-up. Typical installation depths for low head locks are 10 to 15 ft. For high head locks, the submergence depth and the hydrostatic pressure likely will be the controlling factor in the required compressor pressure. A typical detail for a recess air diffuser is shown below.

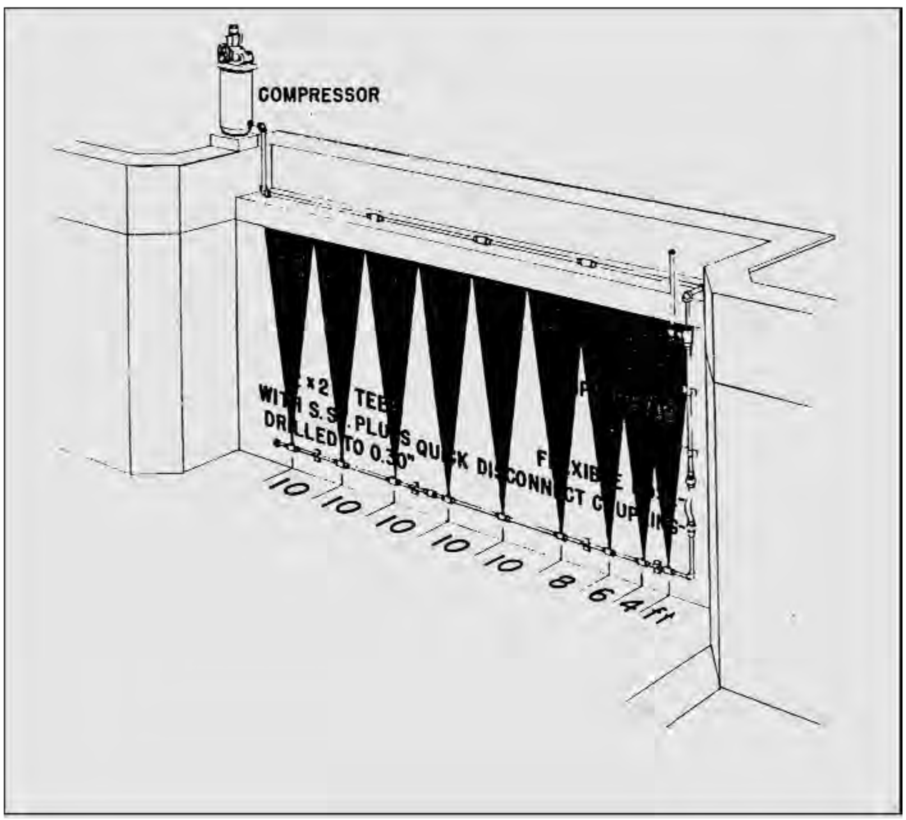


Figure 10-14. Emsworth Lock and Dam, Ohio River, air screen gate recess flusher

h. Additional References and Design Documents**.** Available design guidance includes:

- Compressed Air and Gas Handbook by Compressed Air and Gas Institute;
- Compressed Air and Gas Data by Ingersoll Rand Corporation;
- United States Army Corps of Engineers Cold Region Research Laboratory, Melting Ice with Air Bubblers, Cold Regions Technical Digest, No.83-1, March 1983, Kevin L. Carey;
- United States Army Corps of Engineers Cold Regions Laboratory, CRREL Report 79-12, Point Source Bubbler Systems to Suppress Ice, George D. Ashton, May 1979;
- United States Army Corps of Engineers REMR Bulletin Volume 12, No 2, 1995, Use of Radiant Heaters to Prevent Icing and Air Bubbler Systems;
- United States Army Corps of Engineers, Engineering Manual EM 1110-2-1612, 30 October 2002, Ice Engineering, Chapter 18 Ice Control for Navigation, Chapter 20 Control of Icing on Hydraulic Structures;
- United States Army Corps of Engineers, Engineering Manual EM 1110-2-1612, 30 April 1999, Ice Engineering, Chapter 3: Ice Control
- United States Army Corps of Engineers, Engineering Manual EM-1110-8-1(FR), Chapter 6, Section II, Winter Navigation on Inland Waterways;
- United States Army Corps of Engineers, Engineering Manual EM-1110-2-2602, Planning and Design of Navigation Locks, Chapter 11, Ice Control Measures;
- PIANC Report WG 106 Innovations in Navigation Lock Design, 2009, Paragraph 4.6.2, Ice Control in Locks;
- PIANC Report “Final Report of the International Commission for the Study of Locks”, Chapter 10, Ice Control at Locks;
- Vankan, L.J.: Ice Fighting by Hydraulic Structures (in Dutch: IJsbestrijding bij Kunstwerken), Bouwdienst Rijkswaterstaat – Centrum for Ice Fighting, Uitgeverij Matrijs, Utrecht, 2000.

10-4. <u>Floating Mooring Bitts</u>. In the United States, the term floating mooring bitt is widely used. In Europe, these are called floating bollards. However, the terms floating bollards and floating mooring bitts are used interchangeably throughout the world. Floating mooring bitts provide a means to secure vessels inside the lock chamber during a lockage. This is opposed to securing vessels on top of the lock chamber to fixed bollards or posts that are imbedded in the lock, or having lock personnel handle lines.

a. Description and Design Data. The mooring bitts raise and lower with the water elevation in the lock chamber. Because of that, they typically provide a more stable means for securing vessels as the lock is filled and emptied. Many improvements have been made in filling-and-emptying system designs in recent years. This includes reduction of turbulence in the lock chamber and elimination of overfill and overempty situations by improving culvert valve operation. However, it is still necessary to use floating mooring bitts to keep barges and pleasure craft from drifting into the lock gates and bumping into each other, and to compensate for any human error in the filling and emptying process. It might also be necessary to use only one culvert for lock filling and emptying, in which case the turbulence in the lock chamber could result in greater forces on the vessels than that normally experienced.

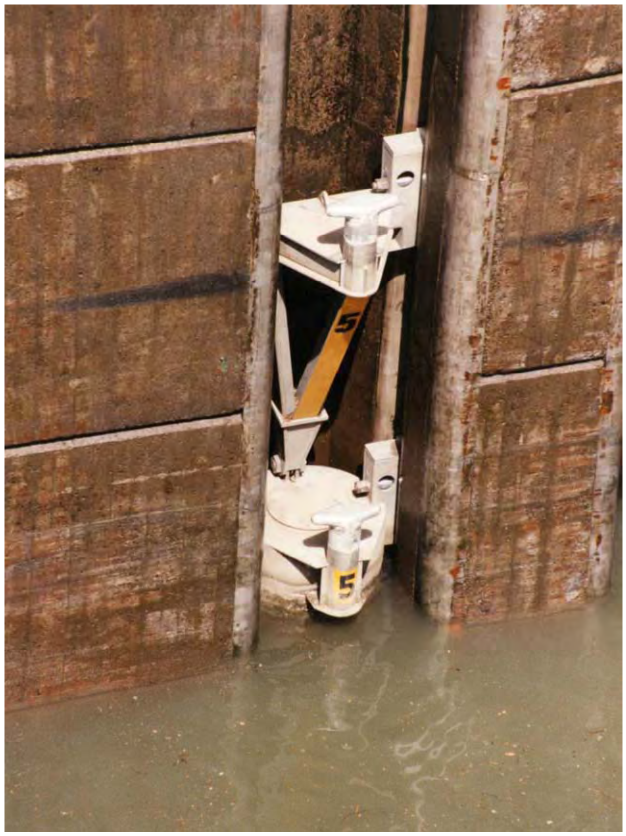


Figure 10-15. Bonneville Lock, floating mooring double-bitt system

(1) Locations. Four to eight floating mooring bitts are usually are provided in each chamber wall, depending on the length of the lock chamber, with a variable spacing to fit tows of different size barges. Generally, floating mooring bitts should be no closer than 30 ft from the upper gate or 75 ft from the lower gate in the mitered position, to help protect the gates from barge overtravel.

(2) A floating cylinder is an integral part of the system. Floating mooring bitts consist of a watertight floating cylinder or tank that rises and falls as the lock chamber water level raises and lowers. This floating tank is mounted with wheels that ride inside steel guides in the mooring bitt recesses. Guide and reaction rollers are used to resist mooring rope loads and reduce frictional losses as the bitt travels up and down.

(3) It is desirable to provide two mooring posts or bitts on each tank, spaced at levels to accommodate the height above water level of either loaded or empty barges. Usually, a vertical spacing of about 6 to 8 ft for the two posts will be required. A double

post or bitt system provides an upper bitt for larger commercial vessels and a lower bitt for smaller recreational vessels. Also, a double-bitt system allows for variance in barge height, depending on whether they are loaded or unloaded. The barge heights above the water level can vary up to 2.4 m (8 ft). Floating mooring bitts are also useful in high head locks where there are large variations during the filling and emptying process.

Figure 10-16. Mooring bitt cylinder

(4) The wheels should be as large in diameter as practical with a minimum of 12 in. to resist high stresses from overloading and impact. Axles should be made of corrosion-resistant steel and should be as large in diameter as is practical to resist bending. Stops (or feet) should be provided at the bottom of the floating mooring bitt to prevent the bottom wheel from striking the floor of the recess and bending the axles. Stops should incorporate rubber bumpers to cushion the impact.

Figure 10-17. Single Bitt System

b. Design Loads.

(1) One trend in navigation is for the towing industry to utilize stronger lines including synthetic lines. Synthetic lines now are used for checking and tying up the larger tows. This, in turn, has created the need for larger and stronger floating mooring bitts. The floating mooring bitts sometimes are used to help decelerate tows in the lock chamber. This puts additional stresses and loads on the entire floating mooring bitt assembly. The 2-in. synthetic lines now in use have a breaking strength up to 107,000 lb, compared to the 31,000-lb breaking strength of new 2-in. manila lines that were the criteria used for the original development of the floating mooring bitts.

(2) USACE EM 2602 provides some additional guidance. All affected components of the floating mooring bitt should be designed for 160,000-lb load or 712kN at normal working stresses. This force is derived from the parting strength of a doubled line reduced by a factor of 0.75 to account for unusual loading. Design the mooring bitts for deceleration of the tows in the lock chamber. PIANC report WG 106 also discusses extensively the mooring line forces and hawser forces for ships and tows. The PIANC report and EM 2602 are provided as references to this engineering manual.

Figure 10-18. Bonneville Lock, floating mooring bitt assembly

c. Stick-Slip Operation.

(1) One significant issue with floating mooring bitts is the tendency for them to stick during travel. This also is called stick-slip operation. Stick-slip operations result in unsafe conditions for the vessels using the lock. At Bonneville Lock, the mooring bitts started to stick almost immediately after the lock became operational in 1993. Several design changes were made, but the problem persisted until the mooring bitts were redesigned with greaseless bearings. Further discussion on utilizing greaseless bearings is in Chapter 2. It is recommended to utilize greaseless bushings and bearings on the wheels and axles.

(2) If greased bearings are utilized, aluminum bronze is recommended for wheel bushings because it will perform better under heavy axle loads than most materials readily available. Grease fittings should be provided for the lubrication of the wheel bushings and axles. Submerged wheels will require grease lines extending to the top of the tank. All grease lines should be in the interior of the tank to prevent damage to the grease lines from debris and ice.

d. Advantages and Disadvantages. Some of the advantages and disadvantages of floating mooring bitts:

(1) Advantages:

- Minimizes and reduces labor for site staff. Vessels do not have to tie off to bollards or check posts on the lock wall. Reduces line handling for site staff.
- Floating mooring bitts are at the same elevation as the water level in lock. This provides a more stable tie off as the lock is emptied or filled.
- Floating bollards provide a level of safety to the personnel of the vessel and the lock, because the bollard floats with the vessel.
- Can provide quicker lockage by reducing line handling.
- For high head locks, it is generally safer to tie off to the floating mooring bitts than to the top of the lock wall.

(2) Disadvantages:

- Maintenance and upkeep of the floating mooring bitt system. The system requires more inspection and maintenance then the fixed bollards because of the mechanical parts.
- Initial capital cost. A floating mooring bitt system retrofitted into an existing lock will require structural modifications to the lock wall.
- The floating mooring bitts can get stuck in the wall recesses when moving with the water level during filling and emptying of the lock chamber. When a ship is moored to such a floating bollard, this has potential life safety consequences and could capsize smaller recreational vessels. Mooring lines could snap. Also, the floating mooring bitt and rails can be damaged.
- The wear rate of the guide wheels has been an issue at various locks, specifically the lower wheels. They tend to diminish in diameter during use, especially if the material of the wheels is too soft.
- Although the mooring hooks are not designed or meant to decelerate the ships' movements, they often are used as such. As a result, the hooks can break from the top of the assembly.
- The lubrication of the bronze bushings in the wheels has been an issue at some sites. It is a time-consuming and cumbersome job to lubricate all wheels regularly. Lubrication also can contaminate the waterway.
- Floating bollards and operation in ice conditions can be problematic. The bollard can be frozen in place.

e. Lessons Learned. A summary:

- Optimize location and quantity of mooring bitts in the lock chamber. Provide a minimum of four floating bitts in chamber. Provide floating mooring bitts on both lock walls. The floating mooring bitts also need to be located near the lock gates for commercial tows.
- Provide floating mooring bitts for high-head new lock construction. Floating mooring bitts also should be considered for low-head new lock construction.
- Do not use single-post floating mooring bitts. All floating bitts should be provided with double posts or double bitts.

- Recess all components within the lock wall. A recess opening from 1.5 to 2.5 ft is recommended.
- The lower guide bars and embedded guides should be composed of stainless steel plate. The bitts should be stainless steel.
- The top of the guides should be terminated in a yoke arrangement to assure that the floating mooring bitt cannot accidentally be propelled out of its recess by buoyancy. This arrangement also can provide a means of hoisting and storing the bitt out of the water when it is not in use. Locks with the top of their walls near the upper operating pool should have the guides extended above the wall to accommodate the floating mooring bitt.
- The location of guide wheels and rail anchors must be designed in order to prevent contact between wheels and bolts of the guiding structure. New wheels should be installed with enough resistance to prevent rapid wear.
- To decrease the maintenance on the rolling elements of the system, the wheels should be fitted with self-lubricating bushings. The material of the wheel shafts should be a hard stainless steel material.
- Provide sufficient fabrication and construction clearances to eliminate or minimize stick-slip operation.
- Install self-aligning and self-lubricating guide and reaction rollers.
- Minimize contact stresses on guide and reaction rollers. Keep stresses below manufacturer guidelines.
- Install self-lubricating thrust washers.
- Install self-lubricating guide plates.
- To prevent the bitt from being suddenly propelled upward because of sudden release after freezing or being jammed by debris, an automatic sinking device should be provided to permit the tank to fill with water. The water can be forced out later by using compressed air. The compressed air inlet connection must be extended above the lower pool when the floating mooring bitt is resting on the bottom of its recess.

f. Many European locks also utilize floating mooring bitts. At the Born Locks, Netherlands, multiple floating bitts are used. There are similar lessons learned at this project. The travel of the bollards was not equal to the travel of the water level. The bollard hit the end-stop, while the water level rose an additional meter. The tow captain always had to be aware of this because he had to lengthen his line. When he forgot, serious damage was caused to the floating bollard or to the line. This problem was controlled by introducing a controlled breaking mechanism, so the vessel’s line will not break and the damage can be repaired easily. The best practice is to let the bollards float free the entire range of water levels with enough clearance for deviations in water levels.

Figure 10-19. Born Locks, bollard is travelling with the water level

Figure 10-20. Born Locks, bollard is pressed against the end stop and the water level is raised to highest point

g. Additional Reference and Design Documents. Available design guidance includes:

- United States Army Corps of Engineers, Engineering Manual EM 1110-2-2602, 30 September 1995, Planning and Design of Navigation Locks, Chapter

5, Paragraph 5-10, Floating Mooring Bitts, and Chapter 10, Paragraph 10-2, Floating Mooring Bitts Design Guidance;

- PIANC Report "Final Report of the International Commission for the Study of Locks", Chapter 5, Floating Mooring Bitts, Paragraph 3.1.3;
- PIANC Report WG 106 Innovations in Navigation Lock Design, 2009, Paragraph 5.2.3 Mooring Line/Hawser forces during filling and emptying of the lock chamber and Paragraph 5.7.1 Mooring Lines;
- Netherlands (Rijkswaterstaat) Waterway Guidelines 2011, Paragraph 4.3.7 Floating Bollards.

10-5. Dewatering Systems. A permanent dewatering system provides a means to quickly dewater a lock in lieu of utilizing portable pumping and should be provided for new lock construction. The common method of dewatering is using vertical lift pumps or submersible centrifugal pumps. The primary components of vertical lift pumps consist of a motor, pump casing, shafting, and the impeller. Submersible pumps should be rail mounted for removal. All components of the pump, including the motor, are submersible. Regardless of whether submersible pumps or vertical pumps are utilized, the system should be accessible for operation and maintenance.

Figure 10-21. Vertical pumps, St. Lawrence Seaway

Figure 10-22. Vertical pump motors, St. Lawrence Seaway

a. Design standards for pumping systems are noted below. USACE utilizes EM 3105, Mechanical and Electrical Design of Pumping Stations. The majority of locks in the United States rely on portable pumping systems for dewatering, rather than a fixed system.

Figure 10-23. Setting a vertical pump, St. Lawrence Seaway

b. Chittendon Locks in Seattle utilize two, 223 kw (300 hp) vertical pumps and one, 56kw (75 hp) vertical pump. The dewatering system has been operational since the lock construction in the 1920s. However, the system and valving need rehabilitation.

Figure 10-24. Dewatering pumps and valves, Chittendon Locks, USACE

c. At Soo Locks, there are three, 30-in., double-suction pumps. There is a planned rehabilitation of this site. As part of the rehabilitation, four guide rail-mounted, submersible, centrifugal pumps will be provided. Three of the pumps will have 343 kw (460 hp) motors and will be capable of delivering a combined total of approximately 378,500 L/min (100,000 gpm) at minimum static head to pump the lock down in approximately 9 hr. The 9-hr dewatering time has been determined to be acceptable by the Soo Area Office as fast enough to minimize ice formation and personnel effort. A fourth pump will be used to remove leakage from the bulkheads and to dewater the pump well. All four pumps will be connected to a manifold discharging below the lower pool. Each pump discharge pipe will have a check valve to prevent water from the lower pool from entering the pump well.

d. Advantages and Disadvantages.

(1) Having a pump in place avoids the necessary mobilization work to bring in portable pumping equipment. The system can be used easily, whether for planned or emergency work. Pumps can be large, and maintenance requires specialized equipment. Pumps require initial capital cost and are maintenance intensive.

(2) Pumping sizes will require an increase in the overall electric service size to the lock site. It is likely the sizes of the pumps will dictate the size of the electric service to the site.

e. Lessons Learned. They include:

- Design system for 8 to 10 hr or fewer dewatering time. This will minimize ice formation in locks in northern climates.
- Utilize at least three pumps, preferably four. Providing additional pumps minimizes the size of each pump. This reduces the weight of the pump and affects the crane size for removal. Multiple pumps also provide redundancy in the event of a pump failure.
- Provide a dewatering pump chamber or pump well.
- Provide variable frequency drive control to modulate flow rates.
- Provide vertical pumps with reverse ratchet mechanism to prevent rotation of impeller in opposite direction. This could damage pump and could cause the impeller to unscrew. Provide means for impeller adjustment at top of the motor for vertical pumps.
- Utilize stainless steel impeller shafting.
- Periodically exercise and rotate pump shaft every six months or less. This prevents bearings from seizing on the pump.
- Use grease-lubricated pump and line-shaft bearings.
- Conduct vibration analysis on the intermediate bearings on a regular basis. This can prevent catastrophic failures.
- Select appropriate pump materials. Material selection is critical. In a salt water or brackish or corrosive environments, appropriate stainless steel components should be selected. Work with pump manufacturer to select correct components.
- Vertical pumps or submersible centrifugal pumps can be utilized.
- Submersible pumps should be designed for removal with a crane and mounted on guide rails.
- Include a pump-removal system as part of any design. Pumps need to be accessible for operation, maintenance, and removal.

f. Reference and Design Documents.

- United States Army Corps of Engineers, Engineering Manual EM 1110-2-3105, Mechanical and Electrical Design of Pumping Stations, 30 March, 1994.

10-6. <u>Ship Arrestor or Collision Protection Systems</u>. Ship arrestors or collision protection systems are devices that prevent barges and ships from impacting the lock operating gates. These barriers usually are composed of large steel cables or chains with counterweights on either end. Steel booms also are used. When a ship strikes a barrier, the kinetic energy is changed to potential energy by lifting a counterweight or compressing a shock absorber. Emergency barriers have been provided at locations where damage to or loss of gates would result in catastrophic consequences.

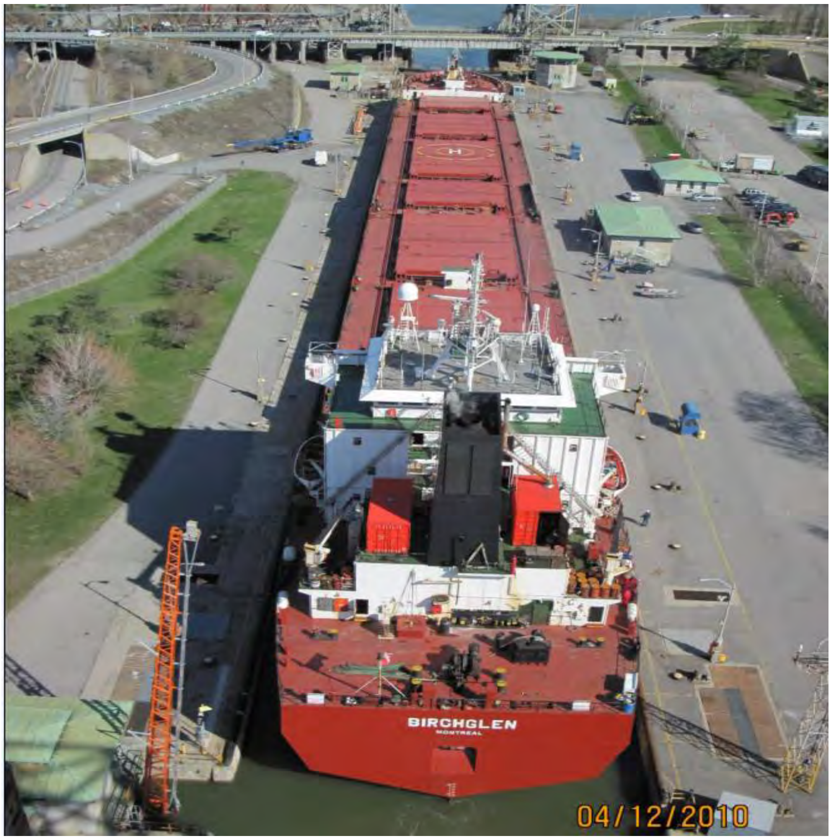


Figure 10-25. Overview of boom and cable system, St. Lawrence Seaway

a. General.

(1) The systems are used primarily for deep draft locks such as the St. Lawrence Seaway and the Soo Locks on the Great Lakes, at Sault Ste. Marie, Michigan. In the United States, no feasible arrestor system has been developed for shallow draft locks typical on inland waterways. However, in the Netherlands at Born Locks, a fixed arrestor system has been developed for an inland lock. This lock has a 12-m lift, and a fixed arrestor system has been installed at the downstream gate. At the lower tailwater elevation, the ships pass under the arrestor and under the downstream gate beam. The ship arrestor system at Born Lock is discussed in detail later in this section.

Figure 10-26. Born Locks near Maastrich, Netherlands, fixed arrestor boom

(2) The most common types of arrestors include cables, booms, and a combination of cables and booms. Booms should be designed to be sacrificial. The St. Lawrence Seaway has a combination of boom and cable and two types of absorption mechanisms. The Soo Locks also utilize a similar boom-and-cable system.

Figure 10-27. Boom and cable, St. Lawrence Seaway

b. Design. Ship arrestors have three main sections: the arresting mechanism, the barrier (boom and cable, for example), and the drive system for the barrier. A typical system consists of a boom that transfers a steel wire rope across the lock and locks onto the opposite side of the lock. The boom then can be retracted, leaving only the wire rope in place, or the boom also can remain lowered across the lock. The St. Lawrence Seaway has both types. The wire rope configuration can have one pass across the channel or several passes, depending on the force required to stop the vessel.

(1) The fender boom on arrestors usually is constructed of welded plate girder made of aluminum or steel. The ship arrestor boom typically is driven via a mechanical drive or hydraulic drive. The mechanical drive uses a driver with shafting that connects worm gears to a bascule portion of the base of the boom and drives the boom up and down. The hydraulic drive uses cylinders and consists of the boom, hydraulic cylinder, wire rope brake, and receiving socket cylinders and various associated machinery components. The St. Lawrence Seaway recently converted its booms to hydraulic drives, as shown in Figure 10-28.

Figure 10-28. Hydraulic-operated boom, St Lawrence Seaway

(2) Elastomeric buffers are used in Germany and consist of a cylinder and tank and a cable deployed across the lock. The basis of this system is a highly compressible elastomeric material. Silicon oil typically is used in Germany. The fluid inside the tank has a set pre-load pressure. After the piston is pushed by an outside force, the fluid in the tank is compressed. The pressure increases to a maximum value when the piston

rod is completely inside the tank. After unloading, the piston returns automatically to the starting position. The difference in dimensions between the piston and the tank allows the element to be a spring or a buffer. A big difference means low friction and it works as spring, and a small difference means high friction and it works as buffer.

Figure 10-29. Elastomeric ship arrestor system, Lock Eibach, Germany

(3) An elastomeric system is highly available because there are no motorized elements integrated. It is a self-adjusting and self-restoring system. One disadvantage is that the system can be used only at locks with a high working level or lift, so that ships can pass the always-fixed wire rope in the low-water position of the lock.

(4) The ship arrestor system at the Born Locks in the Netherlands operates similar to an elastomeric system. The protection system has a rigid steel bar with wood at the vessel impact location. The bar is connected to shock absorbers on both outer ends, which are mounted to the concrete of the lock chamber walls.

Figure 10-30. Ship arrestor, Born Locks

(5) At Born Locks, the shock absorbers are filled with hydraulic oil and nitrogen gas. During the deceleration of a ship, the piston rod is driven into the shock absorber. The hydraulic oil in front of the piston is forced through a multitude of metering orifices. The number of metering orifices in action decreases proportionally to the distance travelled through the stroke. The internal pressure and, thus, the reaction force remain constant throughout the complete stroke length. The oil, forced back by the piston rod, is accumulated by the gas accumulator and the compressed gas provides the return force to reset the rod to its extended position.

(6) The system at the Born Locks is robust and takes up little space on the lock deck. It has few movable parts. The system is possible, however, only for high-head locks where vessels can navigate below the crash protection. The bar is a large steel construction that needs painting and maintenance. The replacement of the bar, in case of destruction, is a large and expensive operation.

(7) At the Soo Locks and St. Lawrence Seaway, mechanical systems and hydraulic systems typically are used for the ship arrestors. Germany also utilizes hydraulic type ship arrestors. Hydraulic ship arrestors employ a hydraulic cylinder fixed at the end of the piston rod with wire rope. The energy absorption portion can be a hydraulic sealed cylinder that increases the restrictive force on the cable as the speed of impact increases. This rope may be positioned across the lock by a jib or boom. The rope locks into place on the opposite side of the lock to some fixed element. In case of ship arresting, the rope pulls the piston rod outside and the hydraulic fluid flows from the front side of the cylinder to the backside, which is combined with a small, flexible high-pressure tank. To manage a large amount of fluid in a short time requires large pipes between the front and the backside of cylinder. With the right choice of diameter for those pipes, a specified delay can be provided. Integrated between the end of the piston

rod and the rope is an element with a security break, in case the strength of the system is reached.

Figure 10-31. Hydraulic-driven arrestor system, Germany

Figure 10-32. Cable set in position, St. Lawrence Seaway

(8) The St. Lawrence Seaway also utilizes a mechanical system. A mechanical drum absorbs the energy of the cable by using brakes that engage at specific points of cable unwinding. The system also can employ a series of clutches that are adjusted to engage at different points.

Figure 10-33. Energy absorption cable drum, St. Lawrence Seaway

c. Net System.

(1) At the Djerdap Lock in Serbia, another type of ship arrestor is used. In front of the mitre gates, a movable fender net is present. Hydraulic cylinders keep the net under tension, and ships hitting the net will be retained by dissipation of their energy with a hydraulic system using separate cylinders at both sides of the lock chamber. The maximum oil pressure is set at 150 bars. The net is lowered hydraulically when the service miter gates are closed, and is lifted when the vessels have to pass and the miter gates are opened.

Figure 10-34. Fender net, Djerdap Lock

(2) Some issues with the system at Djerdap Lock include:

- Sunlight has deteriorated the hydraulic high-pressure hoses, and they must be replaced.
- The steel wire rope net is kept under tension using a pressure control. Because there is no accumulator in the system, the pressure drops rapidly and the hydraulic pumps have to start too often to maintain the prescribed pressure. The Djerdap engineers, therefore, changed the pressure control to a displacement control, which reduced the number of starts of the hydraulic pumps considerably. The plan is to add a hydraulic accumulator to the system.
- The end switch of the arms in the form of a micro-switch is too tiny for this structure and, as a result, it is sensitive to ice and snow.
- The hydraulic pump units for lifting and lowering of the net and for the tensioning of the net are housed in a separate engine room at both sides of the lock chamber. Two electrically driven pump units (one active and one standby) service both the lifting and lowering and the tensioning of the net. Two other small pumps offer the opportunity to operate at different speeds.

Figure 10-35. Gate protection structure: arms and net lifted to let the ships pass

Figure 10-36. Gate protection structure: steel wire ropes across the lock chamber

d. Advantages and Disadvantages. The ship arrestors prevent catastrophic damage to gates, therefore minimizing downtime. They are relatively simple in concept and operation. Newer systems can be designed to minimize maintenance. Some older systems are maintenance intensive (e.g., setting of clutches and brakes). The replacement of boom and cable is labor intensive and time consuming. The design for setting of the boom and cable automatically is complicated and requires periodic maintenance. Ice loading of the fender boom is an issue at locks in northern climates, such as the Soo Locks.

e. Lessons Learned. They include:

- Boom should be sacrificial. The intent is to save and prevent damage to the lock gate.
- Hydraulic drive system is less maintenance intensive and preferred.
- Elastomeric buffers and springs should be considered for fixed arrestors.
- Boom should preferably place cable, then retract,
- Absorption mechanism should be of hydraulic type, therefore minimizing maintenance,
- Marine-rated LED warning light fixtures should be provided at each fender boom. These boom warning light fixtures should be illuminated at all times.
- Greasing of the various fender boom bearing components should be done with an automated system.
- The operation time of the fender boom machinery should be approximately 90 sec.
- Position sensing of the fender boom piston rod should be provided by a magnetostrictive, continuous, linear position-indicating system on the cylinder. This system provides appropriate slow-down points during opening and closing of the boom.
- Account for ice loading on fender boom. Assume fender boom is loaded with ice across the full width of lock chamber. Waves from ships also will create ice on the boom.
- Operator errors can bring arrestor boom down on top of the ship. This has happened several times at the Soo Locks. Incorporate interlocks and electric eye.
- Existing limit switches on fender booms at Soo Locks have been problamatic. These could be improved by using an electric eye system.
- A vessel self-spotting system is used on the St. Lawrence Seaway. This could be combined with an arrestor system.
- The drive system should match the system used to operate miter gates and culvert valves. In Germany, the hydraulic drive arrestor utilizes a compact drive system similar to the miter gate drives.
- The arrestor system needs to be tested periodically to verify the operational capability. This is done in Germany and discussed below.

f. Testing of the Arrestor System. Each new or reconstructed ship arrestor system should be tested against a consistent and verifiable method. This can be accomplished by a real ship pushing against the arrestor system. In Germany, a testing program was undertaken to prove the capability of the country's ship arresting systems. For a hydraulic arrestor system, once a ship pushes against a wire rope, the pressure inside the cylinder increases and a valve opens the connection to a variable high pressure tank. The piston rod can run out, and the ship gets arrested. If the pressure level is lower than the opening pressure, the system stops and the ship is arrested. The system should be tested at a low enough force to avoid damage to the arresting system.

Figure 10-37. Test of ship arrestor, Germany

g. Additional Design References.

- United States Army Corps of Engineers, Engineering Manual EM-1110-2-2602, Planning and Design of Navigation Locks, Chapter 11, Paragraph 11-3, 30 September, 1995;
- PIANC Report "Final Report of the International Commission for the Study of Locks", Chapter 6, Protection of Lock Structures from Ship Impacts, Paragraph 3.

10-7. Crane Systems. There are many types of cranes on Corps civil works facilities, including fixed and mobile boom cranes, gantry cranes, jib cranes, and bridge cranes. A number of cranes installed on barges are used for maintenance and repair at navigation facilities. Cranes for setting bulkheads and emergency bulkheads are discussed in the next section of this chapter. Some cranes have a unique configuration or application (like bulkheads) where the Corps defines the complete crane specification. In some cases, it is not clear what crane or hoist category applies. No specific USACE crane design manual has been developed. However, there are multiple industry standards and design guidelines.

a. Most cranes are considered commercially available products, with multiple nearly-identical copies being produced. Commercially available products have reasonable product development with associated costs spread over the production run. Competition among multiple manufacturers also drives down the cost. But crane manufacturers, like other commercial product manufacturers, do not disclose their design criteria, which likely varies. This makes writing specifications for new cranes challenging.

Figure 10-38. Bridge crane installed on an undercarriage

b. Specifications. Typical USACE procurement documents for commercially available cranes often are produced on a district level, and will contain performance criteria and lists of features but sometimes not define the structural design criteria. It is critical that that the designers and builders of the crane and any manufacturers must have experience in building cranes. This needs to be written into the specification. A request for proposal-type procurement is recommended for any crane purchases. The stability calculations and counterweight requirements for the crane should be done with the crane manufacturer.

(1) The crane design and operation involves life safety, and this needs to be the overriding requirement in any design and procurement. The crane must be stable for all operations. The requirements in the USACE Safety and Health Requirements Manual EM 385-1 must be followed, specifically Sections 15 and 16. The manual is available on the home page of the USACE Headquarters website. The manual requires that all cranes and hoists be classified by an ASME B30 standard, which helps with inspection and stability criteria but not design.

(2) All cranes should be provided with safety features per EM 385-1-1, including load indication, load control, boom angle, anti-two block indication, and load-limiting

devices. Other safety requirements are provided in ASME B30 and NAVFAC documents noted below.

Figure 10-39. Barge-mounted crane for maintenance and repair

(3) U.S. government centralized specialty groups can serve as a resource for crane planning, procurement, and ownership. These agencies or centers of expertise should have a large archive of previous procurements to help formulate the procurement documents. The Army Corps of Engineers Marine Design Center is one crane resource. Its website is:

http://www.nap.usace.army.mil/mdc/capabilities.htm.

(4) The Naval Facilities Engineering Command (NAVFAC) Navy Crane Center is another crane resource with several design documents that can be used. The NAVFAC website is:

https://portal.navfac.navy.mil/portal/page/portal/navfac/.

(5) Two NAVFAC documents should be considered required reading for crane specialists and crane design. They can be downloaded for free, and are provided as a reference to this engineering manual:

- NAVFAC P-307 – Management of Weight Handling Equipment;
- UFC 3-320-07N - Weight Handling Equipment.

(6) A number of guide specifications are available for cranes. Also, in addition to the EM 385-1-1 Safety Manual, the OSHA manual for overhead and gantry cranes should be referenced as applicable. Available guide specifications include:

- OCCUPATIONAL SAFETY & HEALTH ADMINISTRATION
- OSHA 1910.179 Overhead and Gantry Cranes
- UNIFIED FACILITIES GUIDE SPECIFICATIONS
- UFGS 41 22 13.13 Bridge Cranes
- UFGS 41 22 13.14 Bridge Cranes, Overhead Electric, Top Running
- UFGS 41 22 13.15 Bridge Cranes, Overhead Electric, Under Running
- UFGS 41 22 13.16 Gantry Cranes
- UFGS 41 22 13.33 Portal Crane Track Installation
- UFGS 41 22 23.19 Monorail Hoists

c. Industry Design Documents. American Society of Mechanical Engineers (ASME) Standards (B30) provides specific design guidance for a number of crane types including mobile cranes, portal cranes, and locomotive cranes. The ASME standards should be considered a starting point for specific crane design. Other industry crane publications include:

AMERICAN PETROLEUM INSTITUTE

- API-2C Specification for Offshore Pedestal Mounted Cranes

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

- ASME B30.2 Overhead and Gantry Cranes (Top Running Bridge, Single or Multiple Girder, Top Running Trolley Hoist)
- ASME B30.4 Portal, Tower and Pillar Cranes
- ASME B30.5 Mobile and Locomotive Cranes
- ASME B30.6 Derricks
- ASME B30.8 Floating Cranes and Floating Derricks
- ASME B30.9 Slings
- ASME B30.22 Articulating Boom Cranes

CRANE MANUFACTURERS ASSOCIATION OF AMERICA
http://www.mhia.org/industrygroups/cmaa

- CMAA 70 Multiple Girder Cranes
- CMAA 78 Single Girder Cranes

10-8. Maintenance Bulkheads and Emergency Bulkhead Systems. Maintenance gates and the associated crane systems for setting these have been used extensively on USACE navigation structures. Maintenance gates for locks and dams include bulkheads, stoplogs, and needle girder systems. Maintenance gates allow dewatering of a lock chamber or a dam gate bay to enable maintenance and inspection. Emergency gates or bulkheads provide a means for emergency closure of a lock or dam gate under differential head or conditions of flowing water. These are deployed in the event of a failure of the primary lock or dam gate.

a. Maintenance Bulkheads. Maintenance bulkheads generally are installed and removed under balanced heads only and usually are handled by cranes with the help of lifting beams. Often at USACE navigation sites, maintenance bulkheads are set using barge-mounted cranes. Lifting beam hooks should be designed for automatic or semiautomatic connection and release of the bulkhead. This eliminates the need for personnel to climb onto the bulkheads and manually release the lifting hook. Bulkheads may be made from steel, concrete, or timber. The most common type of bulkheads at navigation projects are made of steel. Steel bulkheads typically consist of a skin plate supported by multiple steel girders and/or stiffeners. Structural steel guides are provided to limit the movement of the bulkhead horizontally, either in the direction of the flow or at right angles to the flow.

Figure 10-40. Maintenance bulkheads installed by floating plant

(1) Bulkheads can be provided in a single section or in multiple sections, depending upon clearance restrictions and handling equipment capacity limitations. For most navigation project sites, bulkheads are provided in multiple sections. Multiple-section bulkheads are sometimes called stoplogs. For multiple-section bulkheads, additional rubber seals should be provided between sections to seal the horizontal joint.

(2) After completing maintenance works, the dry chamber section must be filled again before removing the bulkheads. The conventional way is to utilize pumps for filling the chamber to equalize loading on the bulkheads. An alternative is to use a simple, manually operated water inlet (Figure 10-41).

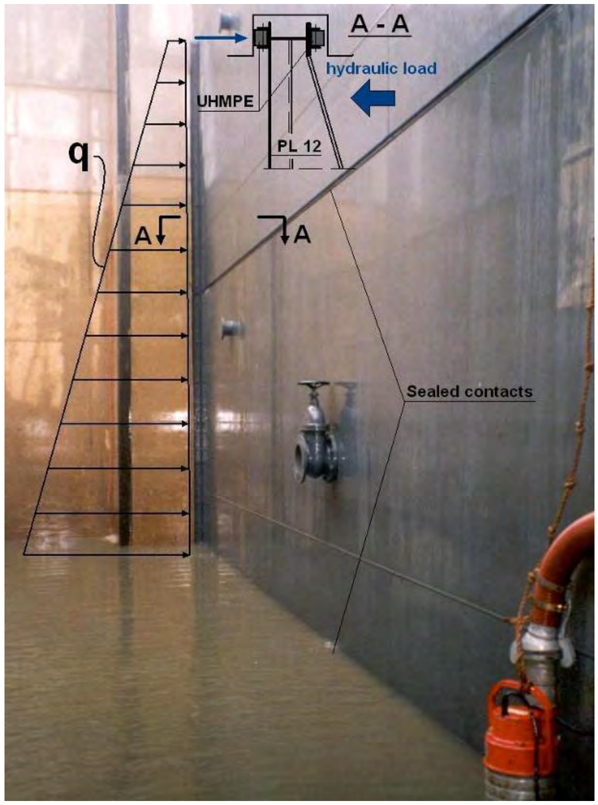


Figure 10-41. Maintenance bulkhead, Naviduct Lock chamber, Netherlands, inlet valve in the lowest section allows easy filling of the chamber

(3) Typically on dam structures, cranes are mounted on an undercarriage assembly. The undercarriage then is installed on a rail system. This allows the crane to travel across the length of the dam. The design of this type of system should allow the undercarriage of the crane to incorporate the bulkhead lifting equipment rather than the crane. Rail grabbers should be utilized on the undercarriage to provide additional safety. The design of this type of system should follow ASME B30.5 for a locomotive-type crane. Stability calculations need to be done for all load conditions and all stability calculations should follow ASME B30.5. Stability calculations need to be done for bulkhead placement and removal using the undercarriage. Counterweights should be added to the crane, rather than the undercarriage assembly. Cranes often utilize multipart wire rope hoists. Fouling of the sheaves or ropes can lead to overloading of the ropes in either the raising or lowering direction. Load-monitoring equipment, load limit switches, and force control switches should be provided to avoid overload of wire ropes and the crane assembly.

Figure 10-42. Maintenance crane installed on undercarriage

Figure 10-43. Bulkhead crane with auxiliary piggyback crane, Racine Dam, Ohio River, one bulkhead section is stored in each gate bay

(4) Floating cranes mounted on barges should be designed per ASME B30.8. Stability calculations need to be done for all load conditions, and all stability calculations should follow ASME B30.8. Load-monitoring, load limit, and force control switches should be provided on the crane.

b. Emergency Bulkheads. Some lock sites have emergency bulkhead systems designed to be set under differential head and flowing water conditions. These generally are provided on the upstream side of the lock. The system provides emergency closure so the navigation pool is not lost. They also are used for maintenance dewatering of the lock chamber. At Chittendon Lock, the entire ship canal would be lost if the miter gate closure was lost. At Old River Lock in New Orleans, the entire Mississippi River could be diverted. The bulkheads utilize rollers and/or lowering carriages to overcome the differential head of water. Port Allen Lock in New Orleans also uses an emergency bulkhead system. The emergency systems usually are confined to the lock at which the crane can be safely anchored. Typically, emergency bulkheads are not utilized on dam structures because of safety issues, and the crane system usually cannot be secured safely to the dam.

Figure 10-44. Emergency bulkhead crane system, Chittendon Lock, Lake Washington Ship Canal

(1) In lieu of emergency bulkheads, other locks utilize an additional gate such as a vertical lift gate to close off flow under emergency conditions. These types of systems are used on the Ohio and Illinois rivers. Emergency gates are discussed in Chapter 7.

Figure 10-45. Emergency bulkhead system, Old River Lock between Mississippi and Atchafalaya rivers

(2) Emergency bulkheads are designed to be set under differential head and flowing water and are utilized as maintenance bulkheads. This eliminates the need to provide a floating plant for setting bulkheads.

(3) Emergency bulkheads typically require a lowering carriage or rollers or some combination. It is also a labor intensive process. At Chittendon Lock, it takes a crew of six to eight personnel approximately four hours to set all the bulkheads. The lowering carriage system is essentially a wire rope hoist that forces the bulkheads down into the lock. The hoist machinery typically consists of an electric motor, gearbox, and open gearing. The carriage assembly and hoist machinery at Port Allen Lock is shown in Figure 10-46.

(4) Without a carriage assembly, the emergency bulkheads will require rollers to help overcome the differential water head. The weight of each bulkhead will need to overcome any head difference plus the friction of the roller assemblies. It is recommended that rollers be designed with greaseless bearings. This will reduce the coefficient of friction further. It also will eliminate the requirement to manually grease all the roller units.

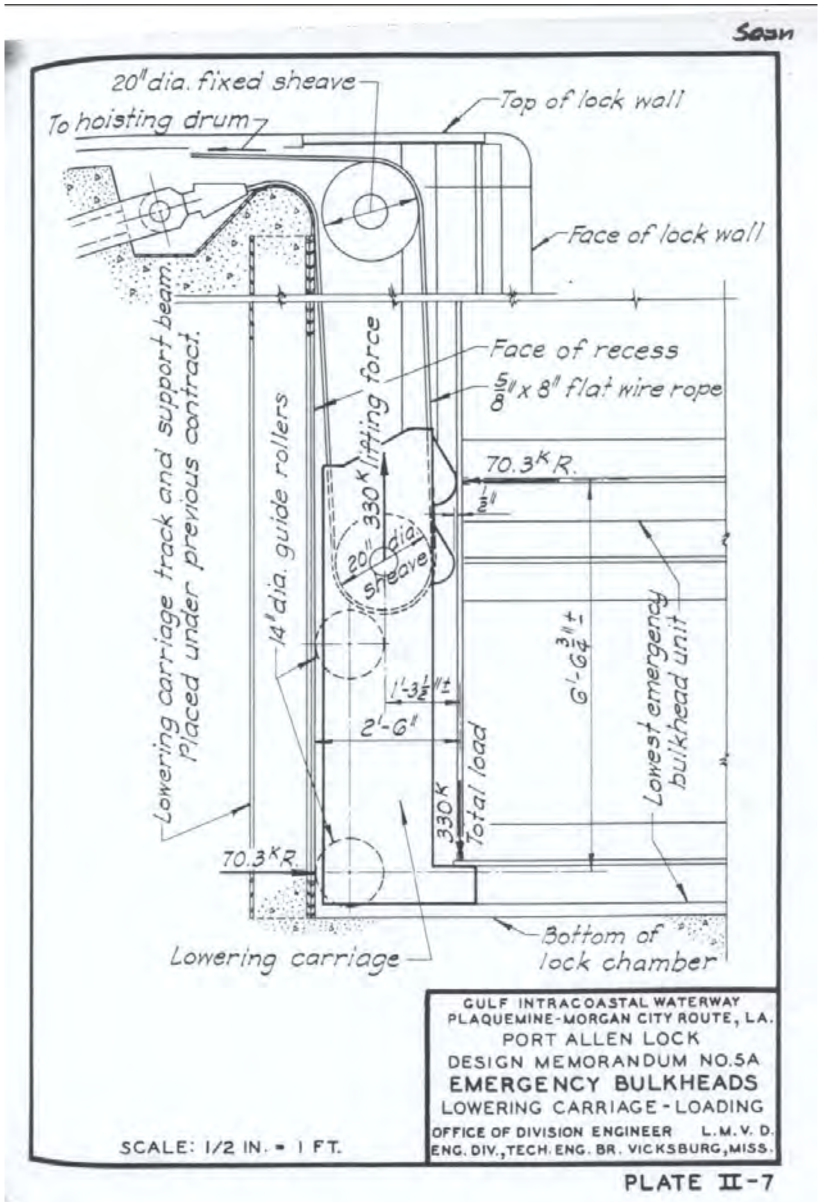


Figure 10-46. Lowering carriage assembly, Port Allen Lock

Figure 10-47. Lowering carriage machinery, Port Allen Lock

Figure 10-48. Emergency bulkheads, Chittendon Lock, Lake Washington Ship Canal

c. Advantages and Disadvantages of Emergency Bulkheads. There are a number of advantages of emergency bulkheads. This type of bulkhead eliminates the need for a floating plant for maintenance dewatering. However, a floating plant still might be required to set bulkheads on the downstream gate. The emergency bulkheads require a fixed crane that, in turn, requires qualified operators.

d. Lessons Learned. New lock construction should incorporate either an emergency bulkhead system or emergency gates. Maintenance gates (bulkheads) on a waterway should be standardized. In that case, one set of bulkheads can be used to perform both scheduled and emergency maintenance on multiple locks across multiple Corps districts.

- For new navigation dam construction, a bulkhead and crane system should be designed to be set under flow conditions. This situation could occur after a barge impact to the dam gates.
- For most lock-and-dam sites, water maintained on spillway gates year-round severely restricts gate inspection and maintenance. All gated spillway installations on dams should have bulkhead or stoplog capability. For existing projects, these typically will be maintenance bulkheads, not emergency bulkhead systems.
- Emergency bulkheads are designed for setting under flow conditions and differential head. For new lock construction, provide a lowering carriage system in addition to rollers and bearings on the bulkheads. This will help ensure the bulkheads can be set properly.
- Lowering carriages need to be inspected periodically. Also, inspect the wire rope system periodically and replace as necessary.

e. Reference and Design Documents.

- United States Army Corps of Engineers Engineering Manual 1110-2-2602: Engineering and Design, Design of Navigation Locks, Chapter 7;
- United States Army Corps of Engineers Engineering Manual 1110-2-2607: Engineering and Design, Design of Navigation Dams, Chapter 6.

10-9. Firefighting Systems. Firefighting or fire protection systems provide the capability to protect vital components such as miter gates. Firefighting systems also can provide capability to fight a fire within the lock chamber (e.g., a fire on a barge or tow). It should be noted that the operating staff at many lock sites do not have the capability or training to fight fires in the lock chamber or on a tow or barge. The information provided in this manual is simply to provide awareness of the various systems that are in use and to provide lessons learned. USACE EM 1110-2-2608 establishes fire protection guidelines for navigation locks. This is included as a reference document.

a. Miter Gate Fire Protection. A miter gate spray system will help to protect the miter gates from heat damage during a fire. Hose stations for firefighting also can be utilized and placed on the lock walls. These are spaced along both the land and intermediate walls. They can provide the capability for fighting a fire within the lock chamber. A typical system will include pumps installed upstream of the upstream miter gates and possibly downstream of the lower gates. Pumps can be installed within culvert valve recesses. Distribution piping and spray nozzles can be designed to spray the downstream face of the upstream miter gates and the upstream face of the downstream miter gates.

(1) Per EM 1110-2-2608, a fire protection system may be provided for miter gates. In operation, this system provides a dense spray of water on the miter gate surface between the gate and barges that might be on fire in the lock chamber. This spray would keep the gates cool and minimize distortion during a fire. The system consists of a series of water spray nozzles along the top of each miter gate leaf, discharging into the lock chamber. These spray nozzles are fed by high-capacity raw water pumps. One pump is provided for each lock chamber. Control stations are located near each gate, with controls for starting and stopping the raw water pump and for opening and closing the motorized valve located in the supply line to each set of gate nozzles. The decision to include the gate spray system should be evaluated on a case-by-case basis, depending upon the consequences of the loss of the gate. Many of the fire protection provisions in EM 1110-2-2608 apply to both miter and sector gate locks.

Figure 10-49. Fire pump system mounted on the lock wall

(2) Sprinkler systems should be considered at high-head locks. Miter gates at high-head locks have a large surface area that is exposed. A fire on these gates could damage the gate severely. Spray nozzles should be the open type, with a flat spray pattern for uniform distribution. The nozzles should be constructed of brass or stainless steel and should be sized and spaced to provide complete coverage. The sprinkler piping should be connected to the supply piping with a galvanized steel swivel joint, with

stainless steel ball bearings and grease fitting for lubrication. The piping should be designed so the water can be easily purged from the nozzles and piping to avoid freezing problems.

Figure 10-50. Spray nozzle mounted on lock wall

b. Design Requirements.

(1) All components should be rated for fire protection, and the fire pumps should be in accordance with National Fire Protection Association (NFPA) 20. NFPA has several codes that relate to fire protection systems.

(2) Aboveground and embedded pipe should be corrosion resistant. Stainless steel piping generally is not necessary. All fittings should have a minimum rated working pressure of 1206 kPa (175psi). Underground piping should be ductile iron with a 1034 kPa (150 psi) working pressure. In the United States, all aspects of the fire water-piping systems should comply with appropriate NFPA and American Water Works Association (AWWA) standards and all local fire codes.

c. Zebra mussels. Zebra mussels are an increasing problem on the navigation structures in the United States. They can clog pump intakes. In areas contaminated by zebra mussels, water supplies for fire pumps should not be drawn from the river unless suitable zebra mussel control strategies are implemented.

d. Hose Stations and Hydrants.

(1) Hose stations should be provided on all new locks and can be retrofitted to existing locks. Hose stations can be used for cleaning off the lock walls in addition to fire protection. Hydrants should be provided, in addition to hose stations, so a local fire department can respond to a fire. Location of the hydrants and connections must be compatible with the fire department’s equipment. Hose diameters should be a minimum

of 1.5 in. or 38.1 mm. Hydrants should be the base valve (dry barrel) type, with the valve and feed piping protected from freezing.

Figure 10-51. Hose station mounted on lock

(2) Hose stations should be spaced 45.6 to 91.44 m (150 to 300 ft.) apart along the walls on both sides of the lock. Provide hose reels with sufficient length to reach all points of the lock.

Figure 10-52. Hose reel mounted inside cabinet

e. Lessons Learned.

- On new lock construction, a high-volume, raw water pumping and piping system should be provided to all parts of the lock and dam. The system should be designed for fire protection and provide water to extinguish small fires and water for spray systems to cool the lock gates in case of fire in the lock chamber.
- Work with local fire protection officials in the design of fire protection system.
- Provide hydrants at site, in coordination with local fire protection officials.
- Provide backup fire pumps for redundancy. Fire pumps can be designed to draw water from the lock culvert or chamber.
- The pumps should have a pressure rating of not less than 690 kPa or 100 psi with a capacity of 946 L/m or 250 gpm. In addition they should have sufficient capability to provide a flow of 189 L/m or 50 gpm to five hoses simultaneously, with an effective pressure at the nozzle of not less than 414 kPa or 60 psi.
- Many sites utilize lock wash-down systems to help clean lock walls. The fire system can be combined with the wash-down system. Many of the components such as the pumps and hose stations are nearly identical.

f. Reference and Design Documents. The following are design and reference documents for fire protection systems:

- United States Army Corps of Engineers, Engineering Manual EM 1110-2-2608, 28 February 1994, Navigation Locks – Fire Protection Provisions.

10-10. <u>Corrosion and Cathodic Protection</u>. Lock gates usually are located in river water or brackish water, which are submerged corrosive environments. Corrosion causes different degrees of structural and metallic deterioration of the gates that, in turn, affects operation and maintenance of the gates.

a. Corrosion protection is a combination of the proper selection of materials and proper selection of the coating system for the application. In some instances, it also might include the use of cathodic protection systems. It is not the intent of this engineering manual to provide specific design guidance for coatings and cathodic protection. The designer should become familiar with the guidance provided in the CERL Technical Report 02-7, "Material Selection Guide for Mechanical Components Used in Civil Works Projects," for the proper selection of materials. This report is provided as a reference to this engineering manual.

b. EM 1110-2-3400, Painting: New Construction and Maintenance, provides guidance on paints and coatings. The guide specification UFGS-09 90 00.00 40 should be used for the proper selection of coating systems. EM 1110-2-2704, Cathodic Protection Systems, provides design guidance for the different types of cathodic protection.

c. The National Association of Corrosion Engineers (NACE) also provides design guidance. NACE SP0169, Control of External Corrosion on Underground or Submerged Metallic Piping Systems, says:

"This standard presents procedures and practices for achieving effective control of external corrosion on buried or submerged metallic piping systems. These recommendations are also applicable to many other buried or submerged metallic structures. It is intended for use by corrosion control personnel concerned with the corrosion of buried or submerged piping systems, including oil, gas, water, and similar structures. This standard describes the use of electrically insulating coatings, electrical isolation, and cathodic protection (CP) as external corrosion control methods. It contains specific provisions for the application of CP to existing bare, existing coated, and new piping systems. Also included are procedures for control of interference currents on pipelines."

d. This excerpt from NACE SP0169 is a good summary for the most effective corrosion control and prevention system available for USACE hydraulic steel structures in immersion service: "When cost and all other factors are considered, dielectric coating systems supplemented with cathodic protection (either sacrificial or impressed current, whichever is required by the specific design conditions in order to meet the minimum NACE SP0169 protective potentials) is still the most effective corrosion control system available for buried and submerged steel structures."

e. Corrosion and Corrosion Control. Lock gates are vital components and require periodic maintenance in areas such as the quoin and miter blocks, diagonals, and the submerged areas of the gate. Since the gates are key operating elements, failure causes disruption of river traffic. Repairs and/or replacement are expensive. Properly designed and installed coating systems and/or cathodic protection systems are critical for extending the life of lock gates.

(1) Corrosion occurs primarily as a result of an electro-chemical process. For corrosion to take place, there are four required elements: an anode, a cathode, an electrolytic path, and a metallic or electron-conducting path.

(2) Dielectric coatings are the primary means of protecting steel from corrosion. For corrosion to take place, four factors must be present: anode, cathode, electrical path between anode and cathode, and an electrolytic path between cathode and anode. Coatings form a barrier that separates the steel from the electrolyte. Coatings work to stop corrosion by forming a barrier. They should be considered the primary corrosion control measure for submerged steel structures. Cathodic protection supplements the coating to make corrosion control complete and effective.

(3) A significant number of the maintenance problems can be delayed, even prevented, with effective coating and cathodic protection. It can be established that lock gates can be protected from the effects of corrosion with proper coating and cathodic protection. A properly maintained and adjusted cathodic protection system will decrease the need for many lock repairs and prevent the lock facility from becoming obsolete by

deterioration. The initial expense of installing cathodic protection and the expense of regular maintenance of the systems can provide a high cost-benefit ratio.

f. Painting and Coatings**.** The primary corrosion control system for lock gates is painting. Vinyl paint systems traditionally have been used in freshwater applications. Vinyl paint systems are durable and have a proven track record. They should not be used in brackish water or saltwater applications. Coal tar epoxy systems and high solid epoxy systems have been used successfully in saltwater. ERDC's Construction Engineering Research Laboratory (CERL) can be consulted for specific design guidance and paint applications. When paint is scratched and cut with waterborne impact, metal surfaces or holidays are exposed to the corrosive electrolyte. The base surface areas become anodes and will corrode in a concentrated area. This affects the limit states of strength and serviceability of the steel in the gates.

(1) To improve the painting system and ensure its durability:

- Determine that the surfaces to be painted have rounded edges and corners, and smooth joints;
- Install cathodic protection and a coating system to protect the exposed areas of the steel.

(2) Environmental concerns have caused some restrictions on metal preparation and application of quality paints used in the past. The environmental concerns have resulted in poorer quality of paint on the market. With the advent of poorer quality paint, the proper installation and maintenance of cathodic protection will become more critical. Periodic inspection reports on civil works structures should include a detailed coating inspection and the most recent dated electrolyte potential survey, condition of cathodic protection system, and plans for cathodic protection system repair and modification if required.

(3) Future emphasis on limiting dewatering of locks is another factor that increases the importance of properly designed and applied coating systems and operational cathodic protection. The length of time between lock dewatering likely will increase in the future.

g. Cathodic Protection. Protective coatings alone generally cannot offer complete corrosion protection because they usually contain some pinholes, scratches, and connected porosity, and over time these imperfections become increasingly permeable. As coatings degrade with time, these imperfections, called holidays, have a profound effect on overall coating integrity because of underfilm corrosion. Cathodic protection systems (CPS), when used with protective coatings, have been effective in controlling corrosion. CPSs consist of anodes that pass a protective current to the structure through the electrolyte environment. CPSs can be one of two types, sacrificial anode or impressed current anode. Hybrid CPSs installed on structures can include both types of anodes to provide protective current.

h. Impressed Current. Impressed current is required to protect the large areas between the horizontal girders and the skin plate of the gate. The anodes of the system are placed between the horizontal girders, with the vertical wiring passing through holes in the girder webs. This makes the system much less susceptible to damage from traffic or debris. In most cases, metallic conduit and some angle iron are required to protect the cathodic protection anodes. Impressed current systems are generally maintenance intensive. After installation, they will require constant upkeep and adjustment. Without adjustment, the proper current will not be provided, accelerating corrosion.

(1) Cathodic protection for lock gates should be provided using Guide Specification 26 42 19.10. Impressed current systems employ anodes that are made of durable materials that resist electrochemical wear or dissolution. The impressed current is supplied by a power source such as a rectifier. All impressed current systems require periodic maintenance because they employ a power supply and are more complex than sacrificial systems. However, impressed current CPSs can be used effectively with bare or poorly coated structures because these systems include much flexibility in terms of the amount of protective current delivered and the ability to adjust it as conditions change.

(2) In an impressed current CPS, an artificial anode is attached to a **POSITIVE** DC voltage source and the protected structure is connected to the **NEGATIVE** DC voltage source.

(3) Each system consists of a rectifier supplying protective voltages to anodes that will distribute uniform protective voltages through the river water to the submerged gate structure. Cathodic protection should be installed on those portions of the gates submerged at normal pool levels. The faces of the gates are protected to upper pool stages, except that the downstream face of the lower gates shall be protected to the lower pool. Meters are provided as part of the rectifier for monitoring of the CPSs. Surveillance of the rectifier output (voltage and current) is required to ensure that the rectifier unit operates on a continuous basis at the desired output levels. Voltage and current indications with the lock chambers filled provide surveillance of the rectifier and cathodic protection operation. The CPSs will encounter flooding and floating debris, and will require impact protection to prevent damage to the cables providing voltage to the anodes.

i. Sacrificial Anodes.

(1) Sacrificial CPSs, also called galvanic CPSs, employ sacrificial anodes such as specific magnesium- or zinc-based alloys, which are anodic relative to the ferrous structure they are installed to protect. This inherent material property enables sacrificial anodes to function without an external power source, so they generally need little maintenance after installation. However, by design, sacrificial anodes are consumed by corrosion during their service life and must be replaced periodically to ensure continuing protection of the structure. Therefore, these anodes should be installed in accessible locations on the structure. Sacrificial anode CPSs generally are recommended for use with a well-coated structure that is expected to be well maintained or subjected to a minimum of damaging wear during its design life.

Figure 10-53. Sacrificial anodes installed on a sector gate, New Orleans

(2) One of two basic concepts for providing cathodic protection should be used. The first uses cast iron button anodes on the skin plate and string-type (sausage) anodes in the compartment areas. The sausage anodes and cables are protected from impact by placing them in perforated plastic pipes and in areas subject to ice and debris and by installing channel and angle iron in front of each anode siring. The button anode cables are protected from impact with conduit. The cast iron button anodes are more durable and can withstand impact. The second method uses cast iron button anodes in all areas.

j. New techniques. Guide Specification UFGS 26 42 19.10 provides new techniques for impact protection and for achieving operational CPSs, and addresses methods of providing protection against ice and debris, steps to increase quality construction, and training on design, installation, operation, and maintenance of CPSs.

k. Cathodic protection testing, evaluation, and restoration. All testing, evaluation, restoration, or new installation should be supervised by a professional engineer registered in corrosion engineering, or an individual who has satisfied the requirements for accreditation as a corrosion technologist or specialist by the National Association of Corrosion Engineers (NACE). Installation of the CPSs also should be witnessed by appropriate government representatives qualified in cathodic protection.

l. Restoration of CPSs. Existing inoperable CPSs at many navigation structures can be restored. This approach is less expensive than installing new systems and, therefore, should be considered first. When graphite anode strings are exhausted, they should be replaced with cast iron anode strings. In many cases, anode strings can be replaced and CPSs repaired without dewatering.

m. Complete replacement of CPSs. Guide Specification UFGS 26 42 19.10 should be followed when designing and installing new systems or when complete replacement of systems at navigation structures is necessary.

n. Cathodic Protection Tests, Adjustments and Reports.

(1) Tests should be performed and data tabulated showing structure to reference cell potentials at a number of different points. Test data should include rectifier voltages and currents. There is no prescribed time interval for testing new systems. But, as a general rule, measurements should be made monthly until steady-state conditions are obtained, at about six-month intervals for the first year or two, and thereafter at least yearly, depending on the judgment of the corrosion engineer responsible for the tests. Based upon the measurements taken, the current and voltage of the rectifier should be adjusted, as required, to produce a minimum of -850 mV instant off potential between the structure being tested and the reference cell. This potential should be obtained over 90% of each face of each gate leaf. This must be achieved without the instant off potential exceeding 1,200 mV. Acceptance criteria of the CPSs are defined in the National Association of Corrosion Engineers Publication NACE SP 0l69.

(2) Reports in a format similar to that illustrated in NACE SP 0l69 (see Table 10-1) of a miter gate showing the measurements made and data obtained should be prepared and evaluated.

o. Measurement of Existing CPSs. Performance should be measured annually and appropriate actions taken.

(1) One structure to electrolyte potential survey (using reference cell) should be performed annually. Any system found not to be operating in accordance with established criteria should be optimized (adjusted).

(2) Any CPS found in need of repair should be repaired.

(3) A report showing the condition of the CPS and/or plans to repair the systems should be submitted each year.

p. Cathodic Protection for Miter and Quoin Blocks. One of the most expensive maintenance problems that occurs on lock gates is corrosion of the miter and quoin blocks. This can be prevented with impressed current anode strings in the vicinity of the miter and quoin blocks. CERL Technical Report 95/05, Detailed Cathodic Protection Design Procedures for Pike Island Auxiliary Lock, provides design drawings and guidance showing anode locations that can provide voltage sufficient to protect the miter and quoin blocks. The report also provides general design guidance CPSs. The report is included in Appendix B of EM 2704 included as a reference document. All areas of the miter block and push-pull rods should be painted. The area where the blocks seal against each other does not have to be painted.

Table 10-1

Steel to Reference Cad Potentials

Rectifier No. 1
UPPER GATE - LAND LEAF - UPSTREAM SIDE
(Impressed Current Installation)
REPORTS CONTROL SYMBOL ENGW-E-7
DATE OF TEST: 1 Oct. 1991

Current Off	Pre-Pro-tection	Current On	(Instant Off)						
Depth Below	**Quoin**	**Middle**	**Miter**	**Quoin**	**Middle**	**Miter**	**Quoin**	**Middle**	**Miter**
Water Surface	**End**		**End**	**End**		**End**	**END**		**End**
0'-6	-0.500	-0.505	-0.495	-1.050	-1.000	-1.055	-0.655	-0.700	-0.650**
2'	-0.500	-0.500	-0.500	-1.040	-1.030	-1.035	-0.700	-0.735	-0.705
4'	-0.500	-0.500	-0.500	-1.050	1.085	-1.050	-0.825	-0.755	-0.815
6'	-0.500	-0.495	-0.495	-1.050	-1.100	-1.055	-0.855	-0.765	-0.850
8'	-0.495	-0.490	-0.490	-1.050	-1.085	-1.050	0.865	-0.770	-0.850
10'	-0.490	-0.480	-0.485	-1.080	-1.110	-1.070	-0.880	-0.880	-0.850*
12'	-0.490	-0.480	-0.480	-1.070	-1.080	-1.060	-0.885	-0.880	-0.880
14'	-0.480	-0.479	-0.470	-1.070	-1.070	-1.065	-0.880	-0.885	-0.980
16'	-0.470	-0.464	-0.460	-1.000	-1.020	-1.030	-0.885	-0.890	-0.980
18'	-0.465	-0.455	-0.450	-1.000	-0.979	-1.050	-0.880	-0.885	-0.985
20'	-0.460	-0.445	-0.440	-0.950	-0.930	-1.000	-0.870	-0.875	-0.1075

Rectifier voltage = 2.10 volts
Rectifier current = 0.50 amps
Coarse tap position = L
Fine tap position = 2
Meter used 5 meg ohms/volt 2 volt scale
Half-cell 0'-3 or less from lock steel
Resistance of circuit: E = IR therefore 2.10= .5R and
R = 2.10/.5 = 4 ohms

* Acceptable reading
** Unacceptable reading

NOTE: Include as many 2’ increments as necessary to cover submerged depth of gate.

CHAPTER 11

Power Distribution

11-1. Power System Arrangement. Electrical distribution will vary for each project. Two possible configurations for a navigation lock's power supply system generally are described as either a radial distribution system or network distribution system.

a. Radial System. A radial distribution system has only one simultaneous power flow path from the power source to downstream distribution points or to the loads. A radial distribution system has a lower cost. It provides relatively lower reliability because a single point failure can result in loss of electrical power to the facility. In general, the use of a radial power distribution system should be provided for projects where either the economics or the characteristics of the protected property do not justify or require power feeder redundancy and the higher cost associated with a network power distribution system.

(1) A typical radial power distribution system for a navigation lock has either a main switchboard, a main panelboard, or a main motor control center (MCC) located in a central control building. The central control building typically houses the main electrical equipment and is located at the facility to provide the best operating views and control accessibility of the lock chamber. A navigation lock can be described as having four operating corners, each of which is near the lock gates and the filling or emptying valve machinery, where they are locally operated. Electrical power feeders are routed extending from the main switchboard location to these operating corners of the lock to supply power to the local power panelboards or to the machinery loads themselves.

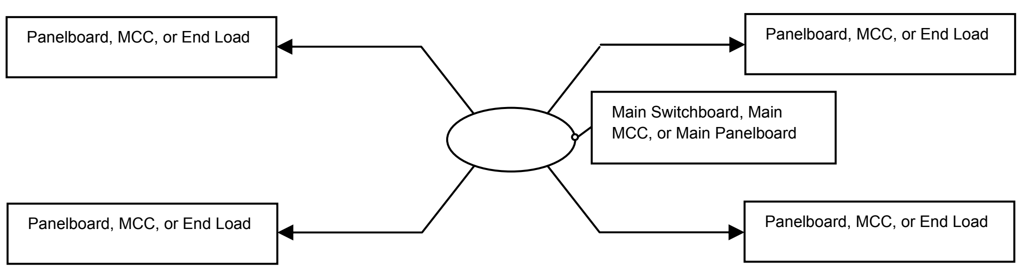


Figure 11-1. General example of a radial power system

(2) A typical radial power distribution system for a navigation dam may have a configuration by which power is supplied by single or redundant power feeders from a separately-derived commercial utility service or from redundant dam power feeder cables supplied from the adjacent navigation lock. The dam power source is provided with a main disconnect switch or a main distribution panelboard, switchboard, or MCC that is located in a dam control house, near the first dam pier, or housed elsewhere on the dam. The power distribution is routed laterally along the dam to feed the loads along the length of the dam. If provided, the redundant dam feeders may extend either partially or the full length across the dam, utilizing manual transfer switches to provide redundant power sources to the dam gates. See Plates B-81 and B-82 for an example of a one-line diagram for a navigation lock and dam.

(3) A typical radial power distribution system for a water control structure may have a configuration by which power is supplied by a single power feeder or by redundant power feeders supplied from a separately-derived commercial utility service. The facility power source is provided within a protective central control building that houses a main disconnect switch or a main distribution panelboard, switchboard, or MCC that is located in a control house on or near the structure. The power distribution is routed laterally along the structure to feed the loads along the length of the structure. The gate controllers may be either housed in the control building equipment, or they may be provided in weatherproof cabinets located adjacent to each set of gate machinery for which it controls. A standby generator set-and-transfer switch is necessary as a backup power source to the normal commercial utility power source.

b. Network (Loop) System. A network, or loop-type, distribution system provides more than one simultaneous power cable path from the power source to specifically selected downstream power distribution equipment. These multiple paths enable the power source to come from an alternate direction to the important downstream panelboards or MCCs located in the electrical system, in the event the normal path of power fails or otherwise is not available, providing power redundancy. There are several configuration variations for network distribution systems, each specific to the critical power needs of the facility. The additional power paths provided by a network distribution system will be more relatively expensive than the cost of a radial distribution system. Also, a network distribution system might be more complicated to operate than a radial distribution system because it is necessary to be knowledgeable regarding which feeders are energized, which feeders are de-energized, and how switching must be operated to obtain power.

(1) A typical network power distribution system for a navigation lock is provided with a main switchboard, a main panelboard, or a main MCC located in a central control building. The central control building typically houses the main electrical equipment and is located at the facility to provide the best operating views and control accessibility of the lock chamber. As noted above, a navigation lock has four operating corners and each is located near the lock gates on both sides of the lock chamber from where the gates are locally operated. Also, filling and emptying valves are typically in close physical proximity to these locations. Electrical power feeders are routed extending from the main switchboard location around the entire lock to these operating corners of the lock in a loop to supply power to the local power panelboards or to the machinery loads themselves.

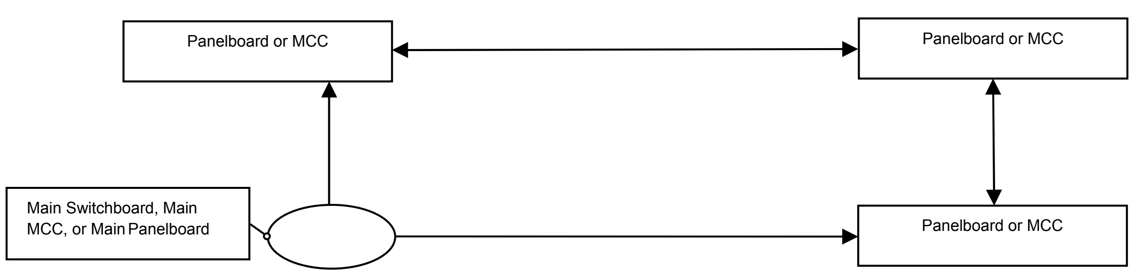


Figure 11-2. General example of a network (loop) power system

(2) A network power distribution system should be used when it is necessary and economically reasonable to provide redundant power paths.

(3) A network distribution system should be considered when a project has power redundancy requirements such that its electrical power system must not experience any power outages.

(4) A fully redundant network power distribution, such as a system with two full-capacity transformers, two station service busses, and a tiebreaker between the busses, offers the best system. Even though each transformer is capable of carrying the lock's entire load, the distribution system should be configured for normal operation with approximately half the load on each bus. In the event that one supply transformer fails, an automatic transfer of load via the bus tiebreaker will restore electrical service quickly to the affected bus. The system should have a minimum of two, redundant power sources with a bus tiebreaker between them. This duo-end with bus tiebreaker power configuration dramatically improves power supply availability over a simple, radial distribution system.

11-2. Power Sources. An evaluation should be done to determine if the navigation lock, navigation dam, or water control structure should be allowed to be without electrical power as a result of the loss of the electric power service source. The effect of the power outage should be considered, as well as the acceptable duration of any power outage that might be experienced. An independent redundant power service may be provided if a second commercial utility service is available and can be reasonably provided. Otherwise, it is typical to provide a standby generator as a backup power source to the normal main power service. Some facilities generate hydro-turbine power, which can be used as either the main power source or as the alternate power source for the facility. The electrical power supply quality might affect the facility's reliability and efficiency. As for most industrial applications, the electrical power supply typically should be rated 480/277-V, three-phase, 60 Hz, with an ampacity rating not less than the estimated maximum demand load, plus not less than 20% extra capacity for future growth, as stated by UFC 3-520-01, Interior Electrical Systems. Additional power system criteria and guidance for interior electrical system components and arrangements also can be found in UFC 3-520-01, Interior Electrical Systems. Wye-connected systems allow the ability to identify and locate quickly a faulted circuit in a widely dispersed area. Further guidance for rights-of-way, ownership, operation, etc., of the transmission line and substation can be found in UFC 3-550-01, Exterior Electric Power Distribution. Each power component must be properly rated to withstand the maximum fault current rating at its location in the electrical system determined by a calculation of the fault current study.

a. Estimating Electrical Service Load.

(1) The facility service load rating size should be developed from the expected maximum demand. The expected demand may be determined from a total of the feeder loads to which an appropriate diversity factor is applied, or by listing all connected loads and applying corresponding demand loads in kilo-volt-amperes (kVA). The size of the

service transformer often is determined by the local serving commercial electric power utility company, based on load information provided by the designing engineer. The service main disconnecting means must be a circuit breaker, a switch, or a fused switch, and its size should be based on the service size including the aforementioned capacity considering future load growth.

(2) Demand factors used for developing the facility's service equipment capacities can vary widely due to the type of facility (i.e., high-head, stand-alone power plant versus low-head power plant integrated with a dam structure and a navigation lock). Demand factors for special loads can and should be developed based on trends for similar special loads at similar existing facilities. For example, shore power receptacles often are installed as needed to connect floating plants and USACE marine vessels to the facility's power system. The shore power receptacle power circuits must have sufficient capacity to start and run general machinery and tool loads, as well as critical power loads such as HVAC and refrigeration loads. These receptacles may be designated a demand factor as high as 100% and connected to the standby generator system because of their importance, location, and frequency of use.

(3) The facility power service system should be designed to anticipate load growth. Anticipated growth will depend on a number of factors, including the size of the facility and the location. Examples of additional loads that should be considered are the development of extensive maintenance, warehouse, or visitor center facilities constructed at the facility, or electrical requirements resulting from environmental protection issues such as fish bypass equipment. The electrical service design should have provisions for unanticipated load growth for multipurpose projects.

b. Types of Electrical Power Sources. Normally, the source of electrical power will be the commercial power utility company. The purchasing, metering, and characteristics of the electrical supply will be coordinated with the power company. Under special circumstances, such as lack of a commercial power source, national security requirements, or where justified by cost analysis, prime power may be supplied by an alternative power source. Capacity of the power source should be sized for 100% of the demand load, but must be coordinated with the power utility company. Spare capacity should be coordinated with the power company as required. The design of the power supply will ensure maximum continuity of operation, especially for primary mission requirements. Reliability and availability requirements of the power system will be analyzed and design tradeoffs made to determine the optimum equipment configuration. In determining if this approach would be cost effective and meet mission requirements, consider using dual utility feeds rather than a single utility feed with standby generator. The benefit of the local power company sizing, owning, operating, and maintaining the service transformer is that the project or facility is not responsible for maintaining, servicing, or repairing the medium-voltage equipment.

(1) Commercial Power. The designer should consider requiring the commercial utility company to provide, own, and maintain the main power service transformers and should consider using standard secondary metering. This is the most common electric power utility arrangement. Electrical utility companies might offer primary metering,

which means a lower cost rate per kilowatt-hour if the service power is provided to the owner at distribution medium-voltage levels (i.e., 2400, 4160, 12,470, or 13,800 V) instead of at utilization low voltages (120/240, 480, or 480/277 V). Primary metering is beneficial if the facility has a large load, if it is in a remote area, and if the facility has electricians who can maintain medium-voltage system equipment. A lower cost rate might be available for primary metering because most metering is applied at the utilization low-voltage level and includes the service transformer losses for which the commercial utility company adjusts its rates. It passes these costs to the customer, so the electric utility company has no need to adjust for transformer losses if the metering is on the primary voltage side of the service transformer. It also should be considered that, if the navigation lock and dam, flood control dam, or water control structure facility owns the service transformer, the facility will be responsible for all maintenance and repairs to the service transformer and pay any costs incurred. Additionally, service maintenance and repair for medium-voltage equipment will require specialized tools, materials, electrician knowledge, training, and skills that otherwise would likely not be required at the facility. The commercial utility company is prepared to perform these maintenance activities. Also, some navigation locks and dams, flood control dams, or water control structures own, operate, and maintain medium-voltage distribution systems and might have the resources to support facility-owned service transformers.

(a) Two commercial electrical utility service entrances from different geographical locations might be beneficial at some facilities, if the need and cost are deemed necessary. Two electrical service entrances provide a redundant power supply so, if one area of the electrical power grid fails from one geographical direction, the second service entrance fed from the second geographical direction can be still be energized and used. Each power service should be fed from different geographical directions of the power grid for this to be effective. It is likely that the electric power of both service entrances would be connected to and supplied from the same commercial power utility. If considering multiple service entrances, the designer should consider coordinating with the local power utility company to perform a power reliability study.

(b) Two commercial utility service entrances should be considered only for large critical loads. Normally, one utility feed with a correctly sized standby generator is sufficient for a navigation lock and dam. Two service entrances could be designed for a large critical load navigation lock and dam in a similar configuration as some large flood control pumping stations are fed by two commercial utility service entrances.

(c) The facility service switchgear considered should be metal-enclosed switchgear with 600-V, draw-out power air circuit breakers used for each supply and as a bus tie breaker for high-current services. The equipment must be sized appropriately for the power and energy required by the facility. The switchgear should be located near the station service transformers. A complete station service supply and distribution system should be provided to furnish power for the station, lock auxiliary equipment, lighting, and other adjacent features of the project.

(d) Each supply circuit breaker and the bus tie breaker should be electrically operated for remote operation from the control room in attended stations. Normally, the supply breakers should be interlocked with the bus tie breaker so only two breakers can be closed at any one time. This interlocking scheme is to prevent paralleling the two supply transformers. If the two supply transformers are paralleled, the result is a higher available ground fault current. This could increase the maximum interrupter rating for the switchgear, therefore, raising the cost.

(e) As a minimum, bus voltage indication for each bus section should be provided at the remote location where remote plant operation is provided. Transfer between the two normal sources should be automatic. Transfer to the emergency power sources also should be automatic when both normal power sources fail. Feeder switching is performed manually, except for specific applications.

(f) Duplicate feeders (one feeder from each station's electrical service supply bus) should be provided to feed important distribution center locations. Appropriate controls and interlocking should be incorporated in the design to ensure critical load sources are not supplied from the same bus. Feeder interlock arrangements and source transfer should be made at the feeder source, not at the distribution centers. All the auxiliary equipment for a main unit usually is fed from the MCC reserved for that unit. Feeders should be sized based on maximum expected load, with proper allowance made for voltage drop, motor starting inrush, and to withstand short circuit currents. Feeders that terminate in exposed locations subject to lightning should be equipped with surge arresters outside the building.

(g) The distribution system control must be evaluated thoroughly to ensure that foreseeable contingencies are covered. The distribution centers should be at accessible points for convenience of plant operation and accessibility for servicing equipment. Allowance should be made for additional future loads. Protective and control devices for station auxiliary equipment should be grouped and installed in distribution centers or MCCs. The motor starters, circuit breakers, control switches, transfer switches, etc., should be located in MCCs.

(h) An overlapping protection zone should be provided around circuit breakers. The protective system should work to remove the minimum amount of equipment from service. Over-current relays on the power supply and the bus tie breaker should be set to trip feeder breakers on a feeder fault without tripping the source breakers. Ground over-current relays should be provided for wye-connected station service systems. The adjustable tripping device built into the feeder breaker is usually adequate for feeder protection on station service systems using 480-V, low-voltage switchgear.

(2) Additional power sources include gasoline, natural gas, biodiesel and diesel-fueled, engine-driven electric power standby generators, hydro-turbine power, batteries, and supplemental renewable energy sources. The type of source selected should be based on the economics, feasibility, and requirements of the application. Loads served by alternate power sources should be limited to those required to directly support essential or mission-critical equipment, lighting, environmental control, personnel and equipment safety, alarm systems, and shutdown and startup equipment necessary for mission accomplishment.

(3) Hydro-Turbine Generator. Some facilities have hydropower turbine power generation capability. This is a sustainable power source. The hydro-generated power typically is connected to provide a power source for the facility. Excess generated power that is not used by the facility is provided to the national electrical power grid, measured for revenue, and the revenue is paid for by the local power provider. Synchronizing or paralleling switchgear and controls are required to connect the hydropower turbine generator to the power grid.

(4) Standby Engine-Driven Generator Set. An engine-driven generator set may be provided when backup power is deemed necessary. Unless otherwise defined, an engine-driven generator standby power system should be considered a non-emergency, optional (not legally required) standby power system. The common terminology is emergency generator, but the term should be considered carefully. Backup power sources may be considered either a legally required standby system or an optional standby system, as appropriate, and therefore are subject to the respectively applicable Article 700, Article 701, or Article 702 of NFPA 70, National Electrical Code. Navigation locks and dams, and water control structures should be equipped with an engine-driven generator standby service with sufficient size capacity to operate the navigation lock, the spillway gate motors, the navigation lock bubbler air compressors, shore power receptacles, navigation lock chamber dewatering pumps, as determined necessary, and all essential auxiliary loads on the structure. It is also important to provide local control at the standby generator for testing and exercising the generator system (also refer to the Paragraph 11-2 c., Automatic and Manual Transfer Switches, included below). A load-shedding plan might be required for generator system operation if the generator capacity is limited. For example, lock dewatering pumps may be powered by temporary generators at the time of their use instead of providing permanent capacity as part of the standby generator.

(a) Configuration. Standby diesel generator sets consist of an internal combustion engine directly coupled with an electrical alternator (synchronous or asynchronous). The 60-Hz frequency of the electrical generator is regulated by the driving speed of the combustion engine and electronic controls. The diesel engine and generator are mounted on a common frame. Smaller units (up to 5 kW) are portable and able to be moved by hand. Larger portable units sized up to 150 KW and perhaps larger are towable installed on a trailer. Permanent generator sets can be installed in a facility building structure or they can be provided integrally enclosed, housed inside prefabricated enclosures often offered by generator set manufacturers. These enclosures or generator houses integrate the engine-generator set and fuel tank into the enclosure design. Otherwise, the generator must be installed in a suitable facility building room that has the capacity to provide fresh air, duct-heated air from the radiator end of the generator set, suitable combustion exhaust pipes, a critical silence exhaust muffler, fuel-piping system or tank. Generators for navigation locks normally will be larger units and generate three-phase electrical power.

(b) Engine Cooling System. The cooling system normally is a fan-operated water cooling system that uses a radiator. For small units, the cooling system is mounted with the motor generator unit on the same frame. Bigger units need separate cooling systems and, sometimes, separately mounted radiators.

(c) Electrical Control. The electrical start and stop control is either mounted directly on the generator or in a separate control cubicle. There might be manual start and stop controls, or the generator set might be programmed to automatically start after a power failure. The generator set then can be programmed to automatically switch back to the normal public utility power supply after power is restored. A re-synchronizing switching feature provides an uninterrupted transfer of the power supply to the public grid after power return.

(d) Fuel Considerations. Although it is common that diesel fuel engines are used to produce the necessary power for large power systems, there are also applications where biodiesel fuel, gasoline, natural gas, and liquid propane gas engine generators are used. Selection of the fuel type should be based on the power generation capacity, fuel availability, and convenience for the delivery system. The diesel fuel tank usually is mounted within the frame of the generator set unit or within the base of the integral housing enclosure. The fuel tank, called a sub-base fuel tank, typically is located between the frame skid beams or between the structural members of the generator set’s integral housing. The capacity of this tank can be custom sized to range from a small, 1-hr capacity to a 24-hr operation capacity or more, depending on the operational requirement. To operate the unit for longer periods, needed are separate fuel tanks in a separate room or bulk tanks installed outside (buried or above ground). To protect the environment from leaked fuel, the entire fuel system must be installed according to codes. The Environmental Protection Agency (EPA) regulates fuel tank installation. Typically, double-walled tanks and piping, and containment design are required. Also, multiple standards and codes produced by the National Fire Protection Agency (NFPA) and Underwriter Laboratories (UL) regulate fuel system design.

(e) Engine Exhaust System. The engine exhaust system must be designed according to the prevailing environmental requirements respecting noise and exhaust regulations. Also, the ventilation system to provide air for combustion and cooling must be designed according to the prevailing environmental requirements. Generators installed indoors will require a ventilation system capable of both exhausting air and drawing air into the room (make up air). Larger generators will exhaust large amounts of air through the cooling system radiator. This air must be made up, or else negative pressures will result in the room.

(f) Starting and Stopping Consideration. Starters normally are battery operated by either 12-V or 24-V DC. Batteries typically are provided with a 2-A trickle charger for regulated charging and continuous readiness. For large engine-generator systems, starters using compressed air are possible. Genset enclosure heaters or jacket water heaters can be used in cold environments to help engine starting. The diesel generator is started after a power cutoff, with a time delay of a few seconds up to some minutes. Automatic transfer switches are used to transfer power from the utility grid to the generator system. A generator needs some time to cool down before the engine can be stopped, even after the power on the public utility grid is restored and the electrical system is switched back to normal powers.

(g) Advantages. Diesel generators are available in sizes starting from a few kilowatts up to more than a megawatt and, therefore, can be used to feed nearly all kinds of machinery at navigation locks. In areas or regions where no commercial utility power supply is available, it is also possible to use this equipment for a permanent power supply. A large fuel supply will be required.

(h) Disadvantages. Generators might not be efficient or usable for systems that can endure short interruptions in power supply without significant consequences. The time delay setting, which allows the generator set engine to reach steady state before transferring, might be a longer delay than the duration of the power outage. A generator should be used only in combination with an uninterruptible power supply (UPS) system for the power supply of computer systems. Diesel generators need frequent maintenance, and it is necessary to operate them at least once a month, to exercise them, to assure that they are maintained in an operable condition. A generator should be exercised so that it is started and operated with an attached electrical load. According to the generator manufacturer’s requirements, the equipment must be operated with at least 50% electrical load for at least half an hour. This makes it necessary either to operate some machinery with the generator or load resistors, or load banks that charge the generator during the exercise period must be provided. Both measures require permanent personnel effort. It is possible with many permanent diesel generators to program the generator to start automatically once a month and run for a specified length of time so that it exercises itself automatically. This capability is provided as a control feature and is offered by multiple generator manufacturers. Take special care of the electrical protection system because generators are not able to deliver high short-circuit capacity in the event of a failure in the electrical network. Circuit breakers and other protection systems, therefore, must be selected in accordance with the short-circuit capacity of the generators to provide a safe operation. If needed, the generator part of the system should be oversized, to be able to deliver a larger short-circuit power, guaranteeing the selectivity in the electrical circuit. Also, the starting currents need to be considered when sizing a generator to ensure the generator will start and run all critical loads connected to it. A generator should be load tested periodically by the facility or by a generator service representative to ensure the standby generator system operates at its characteristic ratings and settings.

(5) A UPS is important for navigation locks, navigation dams, and water control structures that utilize computer (PC) systems or programmable logic control (PLC) system equipment to provide clean wave form power with suitably sized battery backup capacity. Clean, reliable power is perhaps the most important factor in assuring long life of a computer system, and an effective method to achieve this is through the use of a UPS. Careful consideration should be given to designing the power supply for the computer system. UPS systems also should be used for short-time power supply of computerized systems either for controlled shutdown of the computer systems or for bridging the transfer time until the backup power standby diesel generator is in operation. Large, harmonic currents caused by nonlinear loads should be reduced as

applicable with larger, grounded (neutral) conductors and with K-rated transformers suitable for nonlinear electronic loads. Note that a UPS does not eliminate harmonics and that multiple harmonic generating devices connected to a UPS output could affect each other. The power system design should take into account the harmonic distortion caused by the UPS on its line side. UPS should be separately circuited. For PLC or computers centrally located, a large, floor-mount UPS might be suitable for providing backup power and surge protection. Several smaller floor-mounted UPSs also can be used. For remote standalone PLC and computers, a small UPS should be provided. Units should be sized carefully to handle the power loads for all the computer components, including the PLC chassis and all internal devices, as well as the video display monitor. Peripheral components such as printers, modems, and network hubs might or might not need continuous power. Consideration should be given to connecting these devices to the UPS, however, to provide them with lightning and surge protection. A rack-mounted UPS can be used for rack-mounted computer equipment. Depending on the application, several rack-mounted units might be required because of their limited power capacity. It might be beneficial to use a UPS that includes communications capabilities so the UPS can be monitored and controlled with software through a communication link to the PLC, or to another PC.

(6) Renewable energy sources should be considered as supplemental energy sources at navigation locks, navigation dams, and water control structures. Wind turbines and photovoltaic solar panels should be considered sources, in addition to the aforementioned hydroelectric turbines. This equipment can be incorporated into a facility's power distribution system to provide supplemental power to help reduce the total power consumption of the facility. Other energy efficiency measures are use of efficient light sources, such as light-emitting diode light fixtures, and use of light fixture controls to use lights only when needed. The designer can refer to the U.S. Department of Energy website, http://www.eere.energy.gov/, for more about renewable energy and energy efficiency.

c. Automatic and Manual Transfer Switches.

(1) An automatic transfer switch (ATS) is used to automatically transfer a facility's critical loads from the normal power source to the standby power source whenever the normal source fails. Most navigation locks, navigation dams, and water control structures possess diesel-driven standby generators to provide standby backup power during utility outages. The generator is usually is tied to the facility's main distribution bus with a transfer switch or with manual, interlocked (i.e., Kirk Key Interlocked) circuit breakers. ATSs are the most common equipment used for transferring power from the normal utility source to the standby generator unit and for re-transferring when utility power is restored because they are convenient, smart, reliable, and fast. ATSs can be furnished as open-type construction for installation into a MCC or switchboard-type construction with digital communication ports for incorporation into a PLC control system for alerting the processor if the facility is on normal or standby power. ATSs also can be provided in separate enclosure with control and instrumentation provided on the door of the ATS enclosure. An ATS can be susceptible to lightning. A surge protection

device (SPD) might be provided to protect an ATS. The SPD might be offered by the manufacturer either as an option integral to the ATS or as an SPD installed upstream of the ATS in the power distribution system.

(a) A PLC can be coordinated to control the ATS to prevent the ATS from switching back to the normal source until the navigation lock operator has completed a navigation lockage and is ready for the power to be interrupted again. This can save unnecessary wear and strain on the machinery. Also, a PLC can perform orderly shutdowns of facility equipment, and then communicate with the ATS before the ATS transfers power back to the utility.

(b) Standby generators often are not sized the same or sized as large as the utility service. A PLC can be programmed to stagger start motors, turn off noncritical lights, limit the use of air compressors, and perform other load-shedding procedures, as deemed appropriate, to reduce the connected load before allowing the ATS to transfer the load to the standby generator.

(c) Operators for remotely operated locks and dams might not know that power has been interrupted or that the remote navigation lock and dam is operating on standby generator. Such information, available from ATS communication to a PLC or an alarm annunciator panel, is useful for identifying the power outage and for coordinating corrective actions necessary for repair and restoration of the incoming electrical power service.

(2) A manual transfer switch (MTS) may be used for facilities when its operating personnel are available and it either is not required or not desired to immediately transfer loads to the standby power source. Consider how the facility will be used and whether an MTS is sufficient, or if an ATS is better suited for transferring power from the normal utility source to the standby generator unit and retransferred when utility power is restored. An MTS can be furnished as a separately enclosed, stand-alone unit or integrated into a switchboard or MCC. An MTS can be provided with auxiliary output dry contacts for signal or control. An MTS also can be provided with optional digital communication modules for connection for remote monitoring.

11-3. Power Distribution Equipment. Power distribution equipment such as a main switchboard, main MCC, or main panelboard must be selected for use based on the specific design application for each facility. Redundancy and spare capacity in electrical power distribution equipment comes with a higher cost; however, the addition of spare vertical structures and/or spare empty buckets allows for future expansion if the electrical loads increase. A four-corner MCC arrangement supplied from either a main distribution MCC or a main switchboard located at a centralized location is typical for traditional large-size navigation locks of 1200-by-110 ft (366-by-33.5 m) size chambers. This local distribution method reduces the quantity of long power cables. Medium-size navigation locks that have lock chambers with a size of 600-by-110 ft (183-by-33.5 m) or navigation locks of similar size might consider using a four-corner MCC arrangement. But, in many cases using only one main distribution MCC or one main switchboard placed at a centralized location (e.g., the central control station building or an oversized control stand) will serve the project well. Smaller navigation locks and most dams need only one centrally located MCC or switchboard to distribute power and provide controls for the facility.

a. Electrical Loads. Group all electrical loads according to their distribution center (i.e., switchboard, MCC, or power panelboard location). Determine the ampacity, including the duty cycle, and diversity factor, and perform connected and demand load calculations to determine the rating of the main circuit breaker and horizontal bus at each distribution center. Always consider future expansion when making final determination of distribution center ampacity ratings. Consider and calculate the voltage drop to determine the size of electrical wires and cables that have long lengths. Voltage drop shall be not greater than recommended by NFPA 70. The designer then should perform short-circuit, motor start, and coordination calculations to determine current interrupting ratings (AIC), circuit breaker types, and circuit breaker short-circuit trip settings.

b. Distribution Equipment Locations. The project design team should provide a sheltered, protected electrical room or other suitably protected area in which to house the electrical distribution center equipment. These areas should have easy access and clearances, in accordance with NFPA 70, National Electrical Code, and efficient means to route cables to electrical raceways, such as gallery cable tray systems, lock wall manhole and conduit duct banks, surface conduits, bus ducts, or surface cable trenches. Electrical power equipment must be located and installed either above the design flood elevation, or the equipment must otherwise be protected during a flood. Electrical rooms should provide sufficient access for maintenance and, if possible, the electrical equipment should be installed at an elevation that is safely located above the design flood elevation. It is prudent also to provide no fewer than 18 in. of additional freeboard elevation above the flood design elevation to protect from floodwater wave activity when installing electrical equipment inside or on top of elevated structures. If the existing switchboard, MCC, or power panelboard is located in a central control building, a control house, or similar structure, it is prudent to locate the new main electrical power and control equipment in the same protected areas when rehabilitating a project.

c. Metering. The designer should consider required and supplemental power measurement and meters. The utility revenue meter for new or changing facilities must be coordinated with the local power utility company. Typically, the facility provides the meter socket and the power company provides the revenue meter and meter instrumentation transformers. In addition to the utility revenue meter, the designer should consider incorporating a separate, facility-owned digital power panel meter to measure the main electrical power characteristics in the main switchboard, main MCC, or main panelboard. Identification of major differences in electrical characteristics, such as power usage, current draw, and low voltage, provides a significant benefit to monitor and troubleshoot the electrical system. Digital power panel meters can be surface mounted, mounted flush through cabinet doors, or separately housed in their own enclosure. Smart power meters are microprocessor-based, available with different levels of measured parameters, and programmable for connection to a power-monitoring system.

d. Switchboard. A switchboard provides the benefit of combining and centralizing several electrical features within a single, multi-section, electrical equipment assembly. A main switchboard used as the central electrical system's hub for a facility may be configured with either standard, factory-assembled sections or custom-fabricated sections assembled to include each of or some combinations of the following electrical features: a utility service entrance and metering section, a main service disconnect circuit breaker section, a transfer switch section, a circuit breaker distribution or panelboard section, a motor combination starter and controller section, a lighting transformer and lighting and receptacle panelboard section, a general controls section (either PLC or relay-based controls), and other special sections if required. The main switchboard typically is installed in the central control building on a concrete housekeeping pad, with the electrical cables entering through the open bottom of the enclosure. Note that motor controllers that are out of sight of the motor and motors that are out of sight of its controller must comply with NFPA 70, National Electrical Code, for lockout and tagout safety requirements. Switchboards normally are rated 480-V, three-phase, with standard horizontal main bus ampere ratings (e.g., 400 A, 600 A, 800 A, 1200 A, 2000 A, and 2500 A). The switchboard fault current rating must be no less than the calculated results of the power system short-circuit fault current analysis. The switchboard must be provided with a grounding bus. Higher voltage and amperage units also are available.

e. Motor Control Center (MCC). Consider using standard, off-the-shelf MCC construction for new and rehabilitation lock-and-dam electrical distribution and control systems. Standard construction allows for easier future additions or changes to the distribution system. A smart MCC allows monitoring and control of the power distribution equipment that it contains. Programmable logic controllers, intelligent circuit breakers, network lighting panels, automatic transfer switches, variable speed drives, relays, and motor starters are some of the control features available with standard MCC construction. An MCC provides the benefit of combining and centralizing several electrical features within a single, multi-section, multi-compartment electrical equipment assembly with an emphasis on combination motor-starting compartments (also called buckets). An MCC used as the electrical system central hub for a facility may be configured with standard-fabricated or custom-fabricated buckets assembled in factory-assembled sections to include all or some of the following electrical feature combinations: a service entrance circuit breaker section, a metering bucket, a transfer switch section, a circuit breaker distribution section, combination motor starter and controller buckets, a lighting transformer section, panelboard section, a general controls section (either PLC or relay-based controls), and other special buckets if required. A main MCC typically is installed in the central control building on a concrete housekeeping pad, with the electrical cables entering through the open bottom of the enclosure. Additional MCCs can be installed throughout the facility in a four-corner arrangement in which they are located closer to the loads to facilitate power distribution to allow the motor controllers closer to the motor locations. Note that motor controllers that are out of sight of the motor, and motors that are out of sight of its controller must comply with NFPA 70, National Electrical Code, for lockout and tagout safety requirements. MCCs normally are rated 480-V, three-phase, with standard horizontal main bus ampere ratings of 400 A, 600 A, 800 A, 1200 A, 2000 A, and 2500 A. The MCC fault current rating must be no less than the calculated results of the power system short-circuit fault current analysis. The switchboard must be provided with a grounding bus.

(1) New Construction Considerations. When designing MCC systems for new lock construction, a designer first should consider a four-corner arrangement. Consider the quantity and the location at which the electrical power distribution equipment will be installed. Locate electrical equipment to prevent flooding by installing the equipment above the project flood elevation. Also, avoid the need for the electrical equipment to be protected by flood-fighting activities, or at least keep the quantity of electrical equipment required to be protected by flood fighting to a minimum. It is best to place equipment strategically, to prevent the need for additional flood-fighting activities. Most new facilities can afford a four-corner, MCC arrangement, and most loads can be fed from one of four corners without significant over-sizing of conductors to account for voltage drop. When designing MCC systems for a new dam, power loads usually can be served from stand-alone, 480-V gate controllers tapped from the main dam feeder (or redundant feeders with transfer switch capability) or otherwise supplied from a main distribution panelboard. It is preferable to install each of the distributed MCCs inside control buildings that can provide shelter from the environment.

(2) Rehabilitation Construction Considerations. Facility loads often change from the loads installed as part of the facility's original construction; therefore, the facility's loads must be verified for most new additions to the electrical system or for a rehabilitation project. There might be benefits to installing new electrical equipment in the same places as the existing electrical distribution equipment intended to be removed, in order to take advantage of existing raceways or other features of the project when rehabilitating an existing navigation lock, navigation dam, or water control structure. Temporary power distribution equipment and temporary cables may be installed, as required, to keep the navigation lock and dam functioning while the existing power equipment is being removed and the new equipment is being installed. Equally, there might be benefits to installing new equipment on a new footprint, which allows new equipment to be installed and facilitates pulling in new cables prior to the facility's closure period, allowing existing equipment to provide power for operation during construction or facilitate power system changeovers.

f. Power Distribution Panelboard. A power distribution panelboard can be used as the power distribution center for a facility. Metering, motor, lighting, or other equipment controls cannot be provided integrally as part of this equipment, but must be provided separately. The panelboard can be the service entrance equipment, if so rated, and the main circuit breaker can act as the main service disconnecting means. The power distribution panelboard can be fed by or can feed redundant feeders by using Kirk Key interlocked circuit breakers in a manual transfer switching arrangement. A main power distribution panelboard used as the central electrical systems hub for a facility may be configured with standard panelboard components custom arranged, as required by each specific application. The main power panelboard typically is installed in a central control building or a central protected location (e.g., dam electrical room, dam pier house, or a control room) on a concrete housekeeping pad, with the electrical cables entering through the bottom of the enclosure. Place power panelboards considering the design flood elevation to protect the panelboards from flooding. The panelboard must be provided with sufficient gutter space for the feeder conductors or multi-conductor cables routed to it. Consider that many lock-and-dam feeder circuit and branch circuit cables might be oversized to compensate for voltage drop at the load. The conductor termination lugs in the panelboard at its bus connection and, for each of the circuit breakers, conductor terminals must be verified and sized to ensure they accept each feeder cable size routed to and from the panelboard because of the aforementioned conductor oversize consideration. Provide properly sized terminal blocks to facilitate connection of oversized field conductors, if necessary. Power distribution panelboards normally are rated either 480-V, three-phase, three-wire, grounded or 480/277-V, three-phase, four-wire, grounded, and typically have standard main bus ampere ratings (e.g., 225 A, 400 A, 600 A, 800 A, and 1200 A). The fault current rating and its circuit breakers' fault current rating must be no less than the calculated results of the power system short-circuit fault current analysis. A power distribution panelboard must be provided with a grounding bus. It must be provided with a neutral bus, as required and as applicable.

g. Low-Voltage Circuit Breakers. Standard low-voltage (0 to 600 V), molded-case circuit breakers should be installed to provide typical feeder and equipment short-circuit protection. Although not required, the designer should consider requiring bolt-on type circuit breakers to help assure the circuit breakers do not loosen from the bus over time. Instantaneous trip or motor circuit protector-type (MCP), thermal-magnetic trip-type, or electronic-type, bolt-on, molded-case circuit breakers must be provided with cable terminals properly sized to accept the intended conductor, considering as applicable both standard circuit conductor size and that conductors might have been oversized to offset voltage drop. Auxiliary contacts and other accessories such as under-voltage trip, shunt trip coil, and motorized reset mechanism are available. After conducting a power coordination study, the circuit breakers should be sized and trip settings set accordingly to coordinate the short-circuit protection system. The short-circuit fault current rating of the circuit breaker must be no less than the calculated results of the power system short-circuit fault current analysis.

h. Power Receptacles. Consider providing 480-V, three-phase, grounded receptacles in locations determined useful for the facility. The design team should coordinate with facility operations and maintenance personnel to determine the various receptacle locations around the navigation lock where 480-V power receptacles are needed. For example, a 480-V receptacle might be installed adjacent to a crossover manhole to facilitate dewatering pumping activities. Similarly, the specific locations of 480-V receptacles intended for dedicated applications should be coordinated to ensure that such receptacles are placed where they will be needed. For example, 480-V receptacles with an integral interlocked safety switch might be provided in a location accessed by USACE marine vessels to allow connection to shore power. All power receptacles must be grounded and rated for the application in which they will be installed. The designer must consider voltage drop when sizing these receptacle feeder circuits. Similar considerations for 120-V applications may be applied to 120-V ground fault circuit interrupter (GFCI), weatherproof (while-in-use), convenience receptacles that can be installed at needed and convenient locations around the facility.

11-4. Low-Voltage Electrical Power Cables.

a. Cable Ratings and Types. Low-voltage cables are suitable for applications rated 600 V or less. Low-voltage, single-conductor and multi-conductor cables applied to navigation locks, navigation dams, and water control structures will be rated 90^{o}C, 600-V, either solid copper wire or stranded copper wire for circuit voltages of 120 V through 480 V. Low-voltage electrical wire and cable should meet the insulation and jacket requirements identified in the Unified Facilities Guide Specification (UFGS) 26 05 19.00 10, entitled INSULATED WIRE AND CABLE. Additionally, low-voltage electrical wire and cable should comply with NEMA WC 70. Solid copper conductors are easier to terminate. Stranded copper conductors are more flexible, making them easier to install. Cables must be rated for the application in which they are intended to be used. The size and ratings of an electrical cable will be printed or imprinted on the jacket of the cable by its manufacturer. Electrical cables originally installed at 1930s navigation locks were lead-covered type cables, which provided integral waterproof and mechanical protection armor and did not rely solely on the cable raceway system for physical protection. Modern electrical cables available as continuously-welded, corrugated aluminum armor Type MC (metal clad), Type AC (armor clad), and bonded-shielded cables (adhered aluminum foil with overall jacket) can provide a level of waterproof and mechanical physical protection for raceways that expose the cables or allow the cables to be submersed in water. Electrical cables intended to be submerged in water for long durations or intended always to be submerged should be rated for that specific use. Electrical cables intended to be exposed to sunlight should be rated ultraviolet (UV) light-resistant because the UV rays will degrade the cable jacket over time, resulting in failure. Electrical cables installed in a raceway system, which provides mechanical and significant environmental protection, do not require integral armor or cladding; they rather can be provided with standard cable jacket material. Electrical cable jacket and insulation installed in raceways still must be selected considering if the inside of the raceway will contain water, be wet periodically, or be a dry location. Electrical cables exposed to the sun shall be UV resistant. Jacket and insulation material also should be selected based on the environmental temperature of the facility, considering extreme cold and hot temperatures and the temperature difference between the extremes. Electrical cables expand and contract longitudinally, and the cables must be installed with adequate slack length to allow them to expand and contract inside their raceway system as ambient temperature changes. This is especially important for long cable runs and when cables cross structural expansion/contraction/deflection joints. Electrical cable jacket and insulation, which are rubber-based, synthetic rubber-based, or cross-linked polyethylene (i.e., XHHW and XHHW-2), are better suited to withstand hot and cold ambient temperature cycle changes and are the preferred choice for outdoors installation. Thermoplastic-type jacket and insulation materials (THHN and THWN) might crack when installed in an environment that experiences hot and freezing cycle temperatures. Thermoplastic-type conductors are suitable for indoor circuits and may be installed outdoors if suitable for the environment, considering the heating and cooling temperature cycles.

b. Ampacity Rating Considerations. The ampacity rating of a conductor is based on the temperature rating of the conductor and of the terminals of the equipment to which it terminates. For example, if the terminal lugs of the load equipment are rated 60^{o}C,the allowable cable ampacity must be based on NFPA 70, Table 310-16, 60^{o}C column; and, if the lugs are rated 75^{o}C, the allowable cable ampacity must be based on NFPA 70, Table 310-16, 75^{o}C column. A rule of thumb is. unless a designer knows the actual temperature ampacity ratings of the lugs of the equipment being installed, one should refer to the NFPA 70, Table 310-16, and conservatively assume that the 60^{o}C-column ampacity ratings should be used for wire sizes 14 AWG to 1 AWG, and the 75^{o}C column should be used for wire sizes 1/0 AWG to 2000 Kcmil.

11-5. Grounding and Lightning Protection.

a. Grounding System. A grounding system shall be installed in compliance with NFPA 70, National Electrical Code. It is important to bond all metal equipment and significant metal structures to the facility's equipment grounding system. Grounding systems at old facilities should be inspected periodically, in a similar manner as other electrical systems. Grounding systems intended to be rehabilitated should be replaced with new equipment. New and rehabilitated grounding systems should be provided, commensurate with the grounding needs of the other electrical systems installed at the facility, including grounding for the power distribution system, the lock-and-dam control system, the communication system, the CCTV system, the new computer system, surge protection equipment, and lightning protection system as applicable. All grounding electrodes for all systems shall be bonded together.

b. Lightning Protection. Lightning protection needs for a facility should be considered and provided as needed. The design and installation of necessary lightning protection system features shall be provided in accordance with NFPA 780, Standard for the Installation of Lightning Protection Systems, and UL 96A. It is important to bond all metal equipment and significant metal structures to the facility's equipment grounding system. The lightning protection will supplement the facility's grounding system. Grounding electrodes for all systems shall be bonded together.

11-6. Power Quality and Surge Suppression Equipment.

a. Surge Protection Equipment. Surge protection equipment is only as good as the grounding electrode system. All computer systems for lock-and-dam control should be provided with surge protection devices (SPD) to the level of protection determined to be needed. It is recommended that an SPD be installed at the service entrance, automatic transfer switch, main switchboard, or main MCC, respectively, as determined most effective for the facility. Voltage surge protection should be provided at least for each lighting panelboard and at each power distribution panel that feeds computer equipment. It is recommended that surge-protected power receptacles or power receptacle strips also be used.

b. Power-Monitoring Equipment. Metering equipment should be provided as part of the power distribution center for local indication and control. A voltmeter, an ammeter, a wattmeter, and a watt-hour meter are typical instruments installed for this purpose. A station service annunciator should be provided integral to the power distribution center for a large station service system. Multiple communication protocols and auxiliary dry contacts should be provided with the watt-hour meters for remote indication and monitoring of the facility's energy use. Switchboards and MCCs can be provided with smart meters, which are equipped for monitoring, measuring, and reporting the status of electrical power, leading again toward efficient energy usage and monitoring (i.e. current, voltage, kilowatts, phase imbalance, loss of phase, and frequency). Any changes or imbalances in measured electrical parameters might identify problems and, therefore, assist in troubleshooting.

11-7. Arc Flash. The publication NFPA 70E, *Standard for Electrical Safety in the Workplace*, defines an arc flash hazard as a dangerous condition associated with the possible release of energy caused by an electric arc. This energy might be extreme temperatures, sound, or pressure. The arc might be caused by faulty equipment or insulation, but often is a result of employees working on energized electrical systems. An arc flash event often results in severe burns. The severity of the burn is related to the amount of fault current and the clearing time of the upstream protective device.

a. NFPA 70E. While the publication covers other aspects of electrical safety, another significant article is "Work Involving Electrical Hazards." Arc flash hazards must be considered when any electrical system operates at 50 V or more. Since an arc flash hazard cannot be eliminated from electrical work, it must be mitigated to provide an acceptable level of risk. The mitigations include elimination, substitution, engineering controls, administrative controls, and personal protective equipment (PPE).

(1) An arc flash hazard analysis must be performed to determine the arc flash boundary and hazard/risk category (HRC) for each piece of equipment in the electrical distribution system. The analysis uses a one-line diagram that includes transformers with impedances, circuit breaker types and settings, conductors and their lengths, and system loads.

(a) The arc flash boundary is the distance from the arc flash at which an unprotected individual will receive a second-degree burn. This boundary must be clearly identified while performing energized work.

(b) The HRC is determined by calculating the amount of energy released during a fault. The HRC ranges from 0 to 4 and identifies the required PPE based on a table in NFPA 70E. The higher the category, the greater the level of PPE. The use of PPE will not prevent burns during an arc flash event, but limit them to second-degree.

(2) At some locations in the electrical system, the HRC might be Extreme Danger. For this category, the level of energy is so great that no PPE can protect the employee. At this level, the arc blast is more hazardous than the arc flash.

(3) Another requirement of NFPA 70E is the labeling of equipment for the arc flash hazard. The labels must include the nominal voltage, the arc flash boundary, and the hazard risk category.

(4) In lieu of the arc flash hazard analysis, NFPA 70E features tables that identify the required PPE based on the system voltage and task being performed. Use of the table might result in under- or over-protection. It is important to note that the tasks are limited to electrical systems that do not exceed specified levels of available short-circuit current and fault clearing times as described in the table footnotes.

b. Arc Flash Hazard Program (AFHP). USACE developed Engineering Regulation (ER) and Engineering Pamphlet (EP) 385-1-100 using a national product development team to address arc flash. The ER establishes the requirement for each USACE facility to develop a unique AFHP. The EP aids operations and engineering personnel in developing the program.

(1) A mandatory requirement of the AFHP is the performance of an arc flash hazard analysis for navigation locks and dams. The results of the analysis, including a one-line diagram of the electrical system, must be documented in the AFHP. The task-based tables in NFPA 70E cannot be used for work at a navigation lock and dam.

(2) ER 385-1-100 requires the arc flash hazard analysis to include recommendations to reduce all hazard/risk categories to Level 2 or lower.

(3) The AFHP also includes documentation on required training and its completion, PPE, work procedures, and inspections and program reviews.

c. Design Considerations.

(1) A well-coordinated electrical distribution system might not be the best system, in terms of arc flash hazards. A system with low hazard/risk categories might not be coordinated to serve the electrical equipment effectively or prevent nuisance tripping.

(2) While ER and EP 385-1-100 were published mainly to address hazards in existing facilities, they should be taken into consideration during the design phase of the electrical distribution system.

(3) Some might question the need to perform arc flash analysis during design, because it will have to be redone after construction to include actual make and models of the supplied equipment. However, it is more cost effective to eliminate or reduce high hazard/risk categories in the system during the design phase than after construction. HRCs should be limited to Level 2 or lower, where feasible. The arc flash hazard analysis should be performed again during the construction contract, using the rating and setting data of the actual equipment selected to be installed. This analysis can be

performed by the designer or can be made part of the construction contract requirements mandating performance of a coordination study and arc flash hazard analysis based on actual proposed equipment, but before procuring the intended equipment.

(4) Space for electrical equipment in a navigation lock is usually limited. The designer should ensure the physical environment of the electrical equipment can support the calculated arc flash boundary.

11-8. References.

a. Federal Publications.

• ER 385-1-100	Arc Flash Hazard Program
• EP 385-1-100	Implementation of Arc Flash Hazard Program
• UFC 3-520-01	Interior Electrical Systems
• UFC 3-550-01	Exterior Electrical Power Distribution
• UFGS 26 05 19.00 10	Insulated Wire and Cable

b. Industry Standards.

National Electrical Manufacturer's Association (NEMA)

• NEMA WC 70	Power Cables Rated 2000 Volts or Less for the Distribution of Electrical Energy

National Fire Protection Agency (NFPA)

• NFPA 70	National Electrical Code
• NFPA 70E	Standard for Electrical Safety in the Workplace
• NFPA 780	Standard for the Installation of Lightning Protection Systems

Underwriters Laboratory Inc. (UL)

• UL 96A	UL Standard for Safety Installation Requirements for Lightning Protection Systems - Twelfth Edition

CHAPTER 12

Equipment and Machinery Controls

12-1. Introduction. This chapter provides general guidelines and considerations for designers when developing a control system for a lock-and-dam project. It is not intended to address specifically all the locks operating in the United States. Although much of the information presented is specific to miter gate locks, the same technology can be adapted to sector gate locks and to the control of spillway gates. As referenced in this chapter, a new lock control system does not refer to only a new lock construction project. Replacement or rehabilitation of an existing electrical system constitutes, for the purposes of this chapter, a new control system.

12-2. Design Considerations.

a. General. The features and control preferences of a lock control system can vary greatly by Corps district, river system, location, weather, size, and traffic conditions. Many of the concepts provided herein will have to be modified to meet the specific needs of different lock operations personnel. It is important to remember that coordination with personnel at the lock is paramount when designing a lock control system. If the designer is considering visiting lock sites to evaluate their control systems, it is beneficial to bring along members of the lock operating staff. The lock operators might bring up areas of concern that the engineer might not have considered. For an idea of USACE projects using electronic and remote controls, it is suggested the designer review ERDC/CERL TR-00-29, Critical Evaluation of Lock Automation and Control Equipment at Corps-Operated Navigations Locks. This July 2001 report presents information, including some cost considerations, on PLC-run locks that were in operation at the time.

b. Type of Control System. There are two basic types of control systems in use at USACE navigation locks and dams: relay-based and programmable logic controller-based (PLC-based). Both have advantages and disadvantages and are covered in this chapter. The design project development team (PDT) must evaluate the needs of the facility and personnel and the complexity of the control system requirements in order to determine which type will work best. For example, a PLC system can reduce the amount of control wiring for a large navigation lock, but appropriately trained personnel might not be available to maintain and troubleshoot the PLC system. Relay-based controls might be less sophisticated, but are not as flexible as PLC-based systems. PLC-based systems can easily monitor analog signals such as water levels, gate position, and hydraulic pressure. Relay-based control systems can and are being used at many navigation locks and dams for machinery (gates, valves, and hoists) controls, area lighting controls, horn control, equipment interlocks, and emergency stop pushbutton control systems. The environment in which the controls are to be installed must be considered when selecting a control system. It is noted that this document neither prefers nor mandates one over the other. It is the designer's responsibility to select an appropriate control system based on all relevant considerations. Some considerations are:

c. Manual Operation. Manual operation of a lock is defined as equipment operation initiated by individual action from the lock operator. The system has no automated sequences.

(1) Manual Controls. The basic control system for any lock should be individual, hardwired pushbuttons, limit switches, control relays, motor starters, contactors, and pilot lights, and may include encoders, transducers, and meters for displaying positions, pressures, etc. For example, each miter gate leaf should have an OPEN, CLOSE, and STOP pushbutton or joystick for manual operation of the gate. This type of operation allows the lock personnel to control each piece of equipment individually, with the full complement of interlocks, limits, and failsafe devices. Note that the Inland Marine Transportation System (IMTS) draft interlock standard requires that interlocks between miter gates and culvert valves always be in the control system, regardless of whether it is hardwired relay controlled, PLC-run, or hardwired backup for any type of control system. This should be the minimum normal control system at any lock. Degrees of automation (using PLCs, not discrete relay logic), as discussed herein, will vary by project site. Access to the hardwired controls should be provided at every control station, as appropriate, and at other areas around the lock, as deemed necessary by the lock personnel and design team. The hardwired controls should be simple and ergonomically designed with good visual feedback such as pilot lights and/or on-screen graphics (if system is electronic). The hardwired controls should have a means of operating all the lock equipment including the miter gates, lift gates, culvert valves, tainter gates, traffic lights, bubbler systems, warning horn, emergency stop, lock lighting, small pleasure-craft controls, and other equipment unique to each lock and required for day-to-day operation. The designer must spend ample time with the operating personnel to determine all the features to include in the hardwired control system.

(2) Backup Control Box. On locks that the designer chooses to incorporate software human machine interface (HMI), such as larger ones with minimal hardwired backup, consideration should be given to providing a small, portable control box that can be conveniently plugged in near the main control console area or at other strategic operating locations. This control box would provide direct inputs to the PLC and, hence, bypass the software HMI in the event of an PC or network failure. Operation via the control box still would have all the PLC interlocks and limits in place, but the flexibility and feedback for the operator would be limited. The control box should be kept as simple as possible, consisting of pushbuttons (lighted for positive feedback) for the miter gates, filling and emptying valves, traffic lights, and other critical features unique to individual projects. While all operators should be trained thoroughly in the use of the backup control box and, while written instruction for its use should be on site at all times, it should be viewed as a seldom-used backup system to the PC network. The backup control box should be small and light enough to be portable and easy to store, yet of sufficient strength and durability. It is good practice to allow extra space on the control box panel to add additional functions as the need arises. Laminated nameplates should be screwed to the panel to allow easy replacement because the control box functions are likely to change as the operators get used to a new system. The backup

control box offers a safe, convenient way to upgrade PC software and hardware without expensive lock downtime.

(3) Minimal Emergency Backup. When implementing a PLC/PC-type control system, a minimal hardwired emergency backup system should be provided for each major piece of lock equipment. The system should be kept to a minimum number of control features and should be as simple as possible. Critical interlocks should be incorporated into this minimal emergency backup, and it is recommend that as many interlocks be included as practical. These interlock requirements might vary from project to project. Regardless of the project, the designer should utilize the IMTS interlock standards. Over-travel and other absolutely critical and failsafe devices should be hardwired into the control circuits, with no possibility of bypassing them from any of the controls. These limits should be only those designed to prevent equipment damage or personnel injury. The minimal emergency backup system should consist of only the controls needed to operate gates, valves, traffic lights, warning horns, and other features unique to individual locks that are deemed critical. In all cases, a lock control system should include an emergency stop pushbutton that is directly wired using normally closed contacts to the motor starters and variable speed drives. Activation of this button should stop all major lock operating equipment immediately, regardless of the lock's operating mode. The designer should be careful not to make the minimal emergency backup system more complicated than the normal control system. Solid state components should be avoided where possible because the purpose of the hardwired system is to provide emergency backup to the normal solid state control system. Excessive numbers of relays, wiring, displays, and solid state encoders can lead to a backup system that is more complicated and difficult to maintain and troubleshoot than the PLC system. The emergency backup system should be electrically isolated from the PLC control system.

(4) Contractor Controls. Consideration should be given to providing pushbutton controls, perhaps as part of the emergency backup system, at the MCC, variable frequency drives, and at remote motor starters, for use by the contractor during construction. These pushbuttons might save the contractor time and might take pressure off the system integrator by allowing the general contractor to bump motors and hydraulic cylinders for mechanical alignment purposes without expecting the system integrator to have large portions of the PLC/PC programming debugged before moving equipment. These controls either can be integrated as part of an emergency backup system or de-energized after construction and checkout is complete.

d. Automated Operation. Automated operation of a lock is defined as operation of major lock equipment (i.e., gates and valves) that is initiated by the PLC or the lock control system without direct intervention from the lock operators. For example, the lock operator may command a miter gate to open, but the PLC is programmed to start the hydraulic pump and operate the direction and volume solenoids. When the gate reaches the recess, the PLC will close the hydraulic pump and automatically shut off the hydraulic pump. Because of the control logic and components required to implement the algorithm, automated operation should be limited to use with PLC/PC equipment. Discrete control relay systems are not a good choice for lock automation. Also, the design of the lock control system and the extent of the automation, if any, must be coordinated with the Corps district's Operations Division (end user). When considering semi-automated control systems, the design team's concerns should include safety issues that could arise because of a lock operator not being able to respond immediately to a situation on the lock wall. This EM documents considerations when designing a lock-and-dam control system, but does not mandate a particular type of design. This EM will neither require nor restrict any particular type of control system, manual, automatic, or otherwise.

(1) Semi-Automatic Lockage. The most automated form of lock operation in the United States is called a semi-automatic lockage. In this sequence, the operator uses two pushbuttons to perform the entire lockage.

- The first pushbutton prepares the upstream end of the lock for entry or exit, and the second readies the downstream end. When a downstream vessel approaches the lock, the operator pushes the first button and the PLC checks and closes the lower miter gates, checks and closes the lower emptying valves, and opens the filling valves.

- At this point, the PLC waits until the lock chamber is at the same level as the upper pool, and initiates the opening of the upper miter gates or lowering of the lift gate and the closing of the filling valves. In a semi-automatic lockage procedure, the signaling of vessel movement (i.e., traffic lights and air horn) are done by the operator. After the operator has determined the vessel is safely in the lock chamber and secured to the mooring bitts, the second command that closes or raises the upper gates, checks and closes the upper filling valves, and opens the lower emptying valves is initiated.

- Then, the PLC waits until the chamber has lowered to the level of the tailwater, then initiates opening of the lower miter gates and closing of the lower emptying valves. When the gates are fully recessed and the vessel is clear to exit the chamber, the operator signals the vessel by sounding the air horn.

(a) This process requires one person to operate a lock, freeing that operator up to enter lockage data, arrange queues, operate tainter gates, operate adjacent locks, and perform other duties. A semi-automated system streamlines the operation of the lock, reduces delays, and increases efficiency.

(b) This type of system should be considered for all new lock control systems, particularly those with high tonnage where an operator's time is critical, those with a limited number of operators, or those with operators who perform various other duties such as maintenance. Locks that have high head, especially small locks, might require refinements to the semi-automatic lockage sequence because of extreme hydraulic conditions. For example, two modes of semi-automatic operation could be programmed easily into the PLC system to allow for differences in pleasure boat and commercial boat operation. High head causes rapid filling of small lock chambers and can create excessive turbulence if not properly controlled. By implementing a pleasure boat mode of operation, in which the valves stop at specified intervals to slow the filling rate, turbulence can be limited. Sequencing of the valves also can be controlled to pin the pleasure boats to one side of the lock, as is done when operating manually with existing control systems. Commercial boats typically can tolerate more turbulence, so the commercial boat mode of operation can be programmed to fill the chamber faster than the pleasure boat mode. Whatever sequences are programmed, filling times and methods of operation should be coordinated with qualified hydraulic engineers and lock operating staff.

(c) A semi-automated lock also should have a manual system, a hardwired emergency stop, and a minimal emergency hardwired backup system. In most cases, a semi-automated lock will have one centrally located operator. With visibility limited from a central operating point, a closed-circuit television system probably will be necessary. Various other forms of automation should be considered, as discussed herein.

(2) Fully Automatic Lockage. To date, there are no fully automated locks in the United States. The safety, security, legal, and policy issues would have to be addressed before a lock could be fully automated. There are plans to fully automate the Panama Canal, and there is one fully automated lock in Europe. Also, there are many remotely controlled locks in Europe that incorporate automated lockage sequences after the lockage is initiated by the remotely located operator. The Panama Canal, when completed, will sense the motion of the vessels using optical sensors and move the miter gates and valves accordingly. It will be an expert system that will coordinate moving of the vessels, using electric mules and winches, the filling and emptying of tandem locks, and movement of the miter gates. The system even will allow the entry of special lockage information unique to each vessel to alter the filling and/or emptying of the chambers. Such knowledge exists only in the experience of the lock operating staff, which likely will be reduced since the canal has been turned over to Panama. In the Netherlands, the Dutch operate the only fully automated lock and dam in the world. This is a trial operating mode using lasers and intelligent radar to sense the movement of vessels and to operate the gates accordingly.

(3) Filling and Emptying the Lock Chamber. Because of the control logic and components required to implement the algorithm, it is recommended that automatic operation of the filling and emptying systems be limited to use with PLC/PC equipment. One benefit of automating the filling and emptying operations is that over-emptying and over-filling can be minimized, if not eliminated, using preprogrammed algorithms based on pool and lock chamber water level differentials at the start of the filling or emptying cycle. Filling and emptying of the lock chamber can be automated in several different fashions. The first, as discussed above, is in the semi- and automatic lockages. A more simple level of automation would have both filling valves operate simultaneously with a single operator command. The command either could be from HMI software or from a hardwired pushbutton input to the PLC. When automating the culvert valves, the PLC can be programmed for any number of different sequences for different head conditions, pool levels, filling and emptying rates, delayed opening, pleasure crafts, light boats, and empty tows. Experienced operators must be consulted to determine the exact extent and requirements of automating the filling and emptying valves. This type of automation is generally simple to do, usually only requires programming once the PLC and field

devices are in place, and can be altered easily as needs change. Automating the filling and emptying of lock chambers should be considered with all new lock control systems.

(4) Water Level Sensing Equipment. Water level sensors could be used in both hardwired relay control and the PLC system. In the hardwired relay system, pool levels (upper pool, lock chamber, and lower) would be displayed on LED panel meters or similar. When making the decision to open the miter gates, the operator could compare the readings on the adjacent pool level meters. In the PLC system, the pool water level sensors would be inputs to the system. This information would be used to display pool levels on the HMI screen and to provide the operator with some indication that pool levels are equal. In addition, one of the critical procedures of an automated lockage sequence is the reliable sensing of water levels and determination that pools are equal in regards to water level. In other parts of this manual, actual hardware and installation are discussed. Discussed in this section is automation of the water level sensing system. The system should have redundant sensors for malfunction identification (i.e., at least two sensors in each measuring location). Malfunction of one sensor should lock out semi- or fully automatic operation. Under these conditions, the operator would verify visually that the water has reached a safe level for manually moving gates. Consideration should be given to providing a built-in system for determining, through a series of checks and comparisons, which sensor has failed, and allow the automated sequence to continue, if possible, with that sensor bypassed. An alarm should be generated to alert maintenance personnel of the failure, yet allow the lockage sequence to continue. Again, this type of automation does not require significant additional hardware, and the programming for such a system is fairly simple. For example, abrupt changes in signal level could be monitored through program logic to determine that a sensor has failed. If one sensor in a pair suddenly drops below or rises above the previous level indicated, it probably has failed. The water level system can include other troubleshooting features such as determination of type of sensor failure (i.e., power loss, signal loss, out-of-range, out-of-calibration), power supply failure, and PLC I/O failure. All help facilitate the troubleshooting and repair of an automated system.

(5) Remote Troubleshooting. For a PLC lock system, the designer might want to consider providing the capability of remotely troubleshooting via standard telephone lines. Designers will have to address information assurance requirements when providing outside connectivity. This provides a means for lock electricians to look at problems from an off-site location and provide guidance to lock operators when problems arise. District engineers also can provide assistance to trained, on-site lock electricians when complex programming revisions are required. The remote capabilities should allow qualified personnel to monitor and change ladder logic programming, operating screens, network parameters, and database files. Extensive changes to the control logic that are unsafe to make remotely or that radically affect control actions should be deferred until they can be tested and debugged on site. If remote troubleshooting via telephone lines is provided, designers shall incorporate appropriate security firewalls into the system to prevent intrusion by unauthorized personnel.

e. Remote Operation. Districts are exploring the use of remote operation to counteract the reduction in operation and maintenance budgets and resources. While remote operation may provide cost savings by allowing a single operator to control multiple facilities, it may add design and other operational requirements.

(1) An Engineer Circular was developed by the Tulsa District for the remote operation of spillways. The publication EC 1110-2-6071 titled “Engineering and Design: Remote Control and Operation of Water Control Systems” expired 31 July 2012, but contains very pertinent design and operational information. As of the publication of this document, it is unknown whether the EC will be updated or superseded. Pittsburgh District is remotely operating several spillways on the Monongahela River.

(2) At the time of this publication, USACE does not remotely operate any of their navigation locks, but remote lock operation is very prevalent in Europe. USACE personnel have made several site visits while participating in a PIANC working group. Germany remotely operates the 56 locks on the Danube River and canal. Germany also remotely operates several of the navigation locks in the Bremerhaven Port system. St. Louis District personnel have visited locks in Germany, France, and The Netherlands as part of a remote control study. Germany remotely operates its locks on the Upper Neckar and Rhine rivers. The Canal Saint Denis in Paris, France contains seven pairs of locks. Two control centers, located at each end of the canal are used to remotely operate the locks. Control consoles located at the Heel Lock in The Netherlands are used to remotely operate the Linne and Roermond Locks.

(3) PLCs use routable protocols, such as Ethernet, to transmit data. Because of this, information assurance activities are necessary to evaluate risk and ensure the system is secure. This includes isolated facilities that do not have outside network connectivity.

(a) Operation Order 2012-14 identifies the requirements of the Federal Information Security Act of 2002 (FISMA). FISMA (44 U.S.C. Section 3541 et seq. 2011) requires formal certification and accreditation (C&A) for all computer systems and data networks, including electronic control systems and Supervisory Control and Data Acquisition (SCADA) systems, including stand-alone (isolated) systems. The C&A process used by the Army is the DoD Information Assurance Certification and Accreditation Process (DIACAP). The end result of DIACAP is an Authority To Operate (ATO). The ATO is only good for three years, meaning DIACAP is a continuous process. While information assurance activities for a control system might not be daily, weekly work will be required.

(b) DIACAP identifies a system’s weaknesses by using an external assessment. This assessment is performed by an Agent of Certifying Authority (ACA). The ACAs are Army-approved teams trained to perform the assessments. The ACA prepares a score card outlining the system weakness for the Certifying Authority (CA). The CA generates a report for the Designated Approving Authority (DAA). The DAA is the Chief Information Officer (CIO) for USACE. The DAA evaluates the risk in the CA’s report and determines whether to issue an ATO. For systems with significant risk, an Interim Authority to Operate (IATO) can be issued and is good for six months. Updated documentation showing that weaknesses have been addressed is required before the

DAA will provide an ATO. DIACAP will be transitioning to the DoD Information Assurance Risk Management Framework (DIARMF).

(c) DIACAP emphasizes a defense-in-depth approach. Defense-in-depth uses Administrative, Technical, and Operational controls to secure information technology systems. Control systems requiring outside connectivity (remote operation or monitoring) will require a firewall (technical control) to meet the requirements of DIACAP.

(4) Remote operation of any facility will require outside network connectivity. The network connectivity may use existing USACE connections (CorpsNet) or dedicated bandwidth.

(a) DIACAP for a system with outside network connectivity is more complex and more costly.

(b) Use of the corporate network (Corpsnet) for network connectivity might require additional hardware such as a router, switch, and Intrusion Prevention System (IPS). This equipment is used to isolate the PLC or SCADA system on the Corpsnet. Router and firewall rules will be reviewed by the ACA.

(c) A dedicated network connection may avoid the above network security equipment, but its cost will subtract from the savings generated by remote operation.

f. Operator Locations. The operation of a navigation lock will vary by Corps district and location. Designers must coordinate with operations personnel during the planning stages to coordinate the control system requirements. Lock operations can be accomplished with single or multiple operators.

(1) Local control requires the lock operator to be physically near the machinery during its operation. A small control shelter is located at each end of the lock chamber. The lock operator will have direct visibility of the equipment and machinery. Local control allows the lock operator to communicate with the deck hands and to please craft from the lock wall. Dedicated operators may be stationed at each end of the lock, or a single operator may travel between locations. Navigation locks equipped with tow haulage units require the lock operator to work from the lock walls.

(2) Central control allows the operation of the entire lock chamber from a single location. The lock operator will not have direct visibility of equipment and machinery, which will require the use of a closed-circuit TV (CCTV) system. Central control will allow the lock operator to remain in the control room. Central control can be used for one lock chamber and is suited to the operation of multiple lock chambers. When central control systems are used, it is strongly recommended that the maintenance program include the requirement for lock personnel to periodically be at the equipment when it operates, looking and listening for any indications of maintenance needs.

(3) Many locks with local control feature an administration building. The lock control system, especially the PLC type, could be used to allow a lock operator to monitor the status of certain equipment on the navigation lock, but in most cases should not allow operation of the lock equipment from the administration building.

g. Site Characteristics. When designing a control system for a lock and dam, it is important to review all features of that project. Caution should be used when deciding if remote operation of a dam is the appropriate choice. There are instances when remote operation might not be preferred. For example, it might be preferred to operate the dam locally when large amounts of debris typically travel down the river through the dam. It is impossible to provide an engineer design document that will be typical for all lock-and-dam control systems. That is not the intent of this document. Rather, the intent here is to promote ideas and consideration of certain features that affect the concept and design of a control system. Obtaining and reviewing documentation from existing lock-and-dam projects that have been debugged and functioning control systems is a good idea. However, it is important to remember that each site is different, and what works well at one lock will not necessarily work well at another. It is recommended that the designer provide the contractor with the controls design (at least a performance specification) because Corps personnel are intimately familiar with the operation of a lock and dam. Designers must consider these and other features early in the design process.

(1) Ambient Environment. The environment that lock control equipment will be installed in should be considered. Equipment must have some minimum level of protection against the ambient. Installations at Corps projects have demonstrated that both hardwired relay systems and PLC systems are fairly rugged hardware systems. Discrete relay control systems have demonstrated many years of reliability with simple strip-heated enclosures for which the heaters have been sized to keep the temperature inside the enclosure to approximately 10° above the ambient temperature in order to minimize/eliminate condensation. Discrete relay systems generally do not need to be cooled, except for extreme installations. PLC equipment is designed for industrial environments, as well, and can accommodate the wide range of temperatures and humidity at most, if not all, locks and dams. For example, it is common for PLC equipment to have ratings similar to operation from 0 to 60°C (0 to 140°F), rated to operate in vibrations of 2G at 10 to 500 hz, operating shock to 30G, and relative humidity of 0 to 95% non-condensing. Most of the time, the rejected heat in the PLC enclosure can warm the equipment or, if the ambient is cold enough, heaters may be used. Generally, PLCs are environmentally rugged components. The design life of PLC systems is 15 to 20 years, compared to 40 for hardwired relay systems. Design life is a tradeoff for the desired function of the equipment.

(2) Size. The size of a lock is critical when deciding the number and location of control system equipment, such as whether or not to use discrete relays or PLC/PCs, consoles, I/O racks, cabling, fiber optics, CCTV cameras, IPCs, and emergency backup controls. While it will vary by project, larger locks generally have more of this type of equipment, and it will be located in a distributed manner to reduce cable lengths. Arranging control and communication equipment to reduce, as much as possible, the

number of lengthy cable runs is critical to ensure a reliable system. Failures from rodent damage, construction or operation procedures, weather, and lightning are less likely to occur in protected cabinets with short, protected cable runs fewer than 30 m (100 ft). While the initial cost of an extra intermediate I/O rack or communication point might not appear economical during the design phase, the cost of downtime and repair associated with failure likely will be much higher and at greater inconvenience. Larger lock-and-dam projects will have more electrical and electronic components located at greater distances from maintenance buildings and control houses. For this reason, designers should look at the possibility of installing in strategic locations network connections so lock electrical maintenance personnel can monitor all the control system features. These network connections can be used by a laptop computer to instantly obtain troubleshooting and repair information on all PLC I/O points, HMI databases, and PC operating screens. The funding required to install such communication points will be well spent, considering the time it can take to travel between locations on 366-m (1200-ft) locks or projects with 15 or more dam gates. Also, time should be spent considering the possibility of locating these connections in areas where lock personnel will be protected from adverse weather conditions. However, the farther repair personnel are removed from the trouble location the harder it will be and longer it will take to restore the system to proper working condition. When trying to show economic justification for the extra funding, be sure to look at the significant consequences of downtime in addition to the relatively low probability of incidents when determining the risk of failure.

(3) Layout. The physical arrangement of miter gates, lift gates, dams, spillways, service bridges, galleries, guide walls, maintenance buildings, control houses, and access to these features will have an effect on the arrangement of the control system components. Procedures such as flood control, winter shutdown, ice flushing, dewatering, open river operation, and general facility maintenance all can be considerations when designing a lock-and-dam control system.

(a) Single Locks. Single locks, the most common at Corps sites, have an obvious need for a reliable control system because they lack the built-in redundancy of a second lock. It is imperative to provide at these sites some means of hardwired emergency backup controls, as discussed previously. These types of locks often are equipped with tow haulage units that increase direct operator involvement in the locking process. However, this does not preclude the justification or need for a certain level of automation. A semi-automated lockage process can free the operator to assist vessel crew members with lockages while the PLC monitors water levels and operates lock gates. Fully automating a lock at which there is heavy dependence on tow haulage units is not feasible at this point, given shared responsibilities between the vessel crew and the lock operating personnel.

(b) Double Locks. Generally, double locks consist of one larger main lock and one auxiliary lock. In the past, it has required as many as five lock operating personnel to perform lockages during heavy traffic at such sites. Federal budgets are not going to support such crews in the future, so it is incumbent on the designers to consider this when laying out the control system. At some double locks, the smaller auxiliary lock is used strictly for pleasure boats. These locks might warrant consideration as user-operated facilities because an operator is on site, controlling the larger lock, and can assist if problems arise. If both the locks are used for commercial and pleasure traffic, a centralized control system in which a single lock operator can perform simultaneous lockages in both locks should be considered. Given certain types of traffic, volume of traffic, proximity to other sites, and other Corps district considerations, potential remote operation of such a facility should be reviewed during the design phase.

(c) Tandem Locks. Tandem locks consist of one long, segmented lock chamber with intermediate gates to facilitate high head lift in incremental steps. Tandem locks are used where the lift is too large to accomplish lockage in a single step. These locks, similar to those at the Panama Canal and Welland Canal, make excellent candidates for automation because the emptying of one chamber fills the other. Water levels must be closely controlled to keep from over-filling the lower chamber or over-emptying the upper chamber. A PLC-based system can repeat precisely a prescribed filling and emptying sequence to ensure a safe and efficient project. If conditions exist that alter the normal sequence, they can be programmed into the system and the PLC approaches an expert system. Depending on conditions described above, remote control of such a site is a design consideration.

(4) Traffic. The volume and type of traffic that use the lock are perhaps the biggest considerations when deciding if there should be automation and, if so, the optimal degree of automation. It also determines if remote control is feasible, the location of control points and CCTV cameras, and other electrical/electronic control design features. In the case of new locks, the size of the lock and arrangement of the lock equipment have been designed for the location and traffic concerns of the project. However, on lock rehabilitation projects, the type and volume of traffic likely have changed since the lock was built. The new control system must be designed with current and future traffic projections in mind. Automation and remote control are two features that can address these needs.

(a) High Volume. High-volume locks, defined for the purposes of this chapter as those with annual tonnage in excess of 36,300,000 tonnes (40,000,000 tons), such as those on the Mississippi and Ohio rivers, generally can support automation to streamline their lockage procedures. With this amount of traffic, the lock is busy all the time and automation of certain functions, including the dam, help free the operator's time to more efficiently operate the lock. Additional costs for redundancy, remote troubleshooting, spare parts, backup control systems, and reliability of equipment are usually easy to justify at these high-volume sites. Remote control of such a facility is a potential consideration that should require little additional cost to build into a new computerized control system.

(b) Low Volume. While lower usage locks might not support the same construction costs as higher volume locks, the control system can be designed for operation by fewer personnel. Often, a single electrician maintains more than one project, and remote troubleshooting and repair capabilities can enhance their ability to accomplish this. When several locks in a system are relatively close, remote operation from one point with automated lockage sequences at each site can enhance the efficiency of the system and cut operating costs. Control system designers should consider these possibilities, if not for immediate use, for future possibilities.

(c) Commercial. Commercial vessels comprise the bulk of the traffic at high-volume locks, while low-volume locks often have a large percentage of pleasure boat traffic. Automation is somewhat easier to implement for commercial tow lockages because vessel movement is slower, more consistent, and generally predictable, and vessel operators are usually more experienced in using the locks. For this reason, a semi-automated system will work well for high-volume locks. Remote control might be harder to justify because a single operator would have to control multiple remote locks for economic justification. The higher volume locks have enough traffic to justify an on-site operator. However, consider the potential for future remote operation, perhaps during slow traffic, because the cost is minimal after a computerized control and CCTV system are installed.

(d) Pleasure Craft. At all locks, the unpredictability and inexperience of pleasure boat operators, coupled with the lack of radio communication and the vulnerability of small-craft occupants, require lock operators to have more direct control over the lockage sequence. Automation of dam gates and lock filling and emptying can give the operator extra time to pay attention to the special needs of pleasure boat occupants.

h. Control Rooms. The operational success of a lock-and-dam PLC/PC control system can be affected greatly by the interface between the operator and the computerized control system. Usually, the point of interface is a control console located in a lock control room. Control rooms should be designed with safety and the needs of the operator in mind. From an equipment standpoint, locations of consoles, CCTV monitors and controls, marine radios, public address systems, PC monitors, printers, and vessel-logging computers can be located essentially anywhere. With today's technology, the flexibility of such equipment allows for the design of the control room to be based almost entirely on the interests of the lock operators. However, when changing from a traditional hardwired localized control system to a computerized central control room, it often is difficult for a lock operating staff to determine its exact needs for the new system. Listed below are guidelines for determining the needs of individual control rooms and meeting those needs with an effective, ergonomic design. It is important to remember that no design will satisfy the needs of every operator, and the final product might have to be modified to meet the changing needs of the lock operators as they become accustomed to a new system.

(1) Access. The designer should consider the accessibility of the control room and its equipment when determining the proper layout for a lock control system. The degree of accessibility required often will be determined by the type of control system employed, namely manual versus automated. Once this determination has been made, the control house layout design should proceed with the safety and convenience of the operators in mind. Some points to consider for accessibility of a new or refurbished control room:

(a) Manual Systems. In manual systems or systems with tow haulage units that require direct operator intervention to the locking process, the control room should be as accessible as possible from the lock wall. This might compromise the flexibility of the control room layout, but will be more efficient for an operator who must return to the lock wall often during lockages. The layout of the control system components, such as those listed herein, should give the operator access to the lock wall and as much direct visibility as possible. Equipment within the control room will have to be arranged in a different fashion to allow an operator to use it in a more mobile mode. Operators will be entering or exiting the control room often and will need quick, handy control system interfaces, rather than elaborate ones designed for the operator who is at the controls on a continuous basis.

(b) Automated Systems. Lock wall access to and from the control house is less significant with automated or remotely operated control systems. In this type of control system, the control room can be oriented to provide the most convenient accessibility to control console components without as much emphasis on access to the lock wall. Where possible, direct visibility of the lock should be provided. With access not as much of an issue, higher elevations, such as on top of a service bridge pier, provide a good location for a centralized control room. In an automated system having single operator control, CCTV monitors become the main focal point of the operator’s attention. These should be positioned to give the operator not only convenient access, but also a matching orientation of the actual lock equipment so quick reference can be made when examining the monitors. In other words, downstream views made looking at the landside of the lock should be the same when looking at the CCTV monitors. Orientation of the PC graphic operating screens also should match the physical orientation of the lock equipment.

(2) Visibility. Direct visibility of the lock and the area around the lock should be provided if economical and architecturally feasible, even in highly automated control systems. Designers should not, however, compromise the economic justification of automating a system by providing excessive means of direct visibility of the entire lock-and-dam facility. Although this sounds contradictory, and on certain designs can be a fine line for the designer to walk, the designer must determine the operator’s need for direct visibility. Often, a control room can be designed where direct visibility and CCTV monitors complement rather than compete. This balance creates a more efficient and convenient working atmosphere. An operator does not have to fumble around looking for the correct orientation if the monitors are set up to provide the same views as direct visibility. In an automated centralized control room, consider positioning and orienting control consoles and CCTV monitors so that direct visibility is available without the operator leaving the console area, or continually rotating more than 90 deg in either direction to see the lock approach areas. In manual systems with local control houses located at the lock wall level, position consoles and CCTV monitors to provide convenient views of approach areas, guide walls, culvert valve discharge areas, dam or spillway gates, and other areas that are hard to see from the lock wall level. Attention to these details will help the operator monitor the entire project while minimizing distractions. In the future, it is likely there will be less manpower at the locks, and

operators could have other maintenance and security-type duties to perform simultaneously with vessel lockages. Designers must consider and plan for this by making the system convenient and safe to operate while increasing the efficiency of the project.

(3) Layout. A good control system that is difficult to access or use is really not a good control system. Therefore, the layout of control rooms or areas is critical to the lock operator's perception of a quality system and, hence, the success of the system. Figure 12-1 shows a proposed new layout for the Melvin Price central control room. The new layout will accommodate operation of the two lock chambers at Melvin Price Locks and Dam, plus a third remote chamber operated locally at this site. Note the use of three separate and distinct work surfaces, one for each lock. Also, note the use of rack-mounted PLC and PC equipment to conserve valuable floor space.

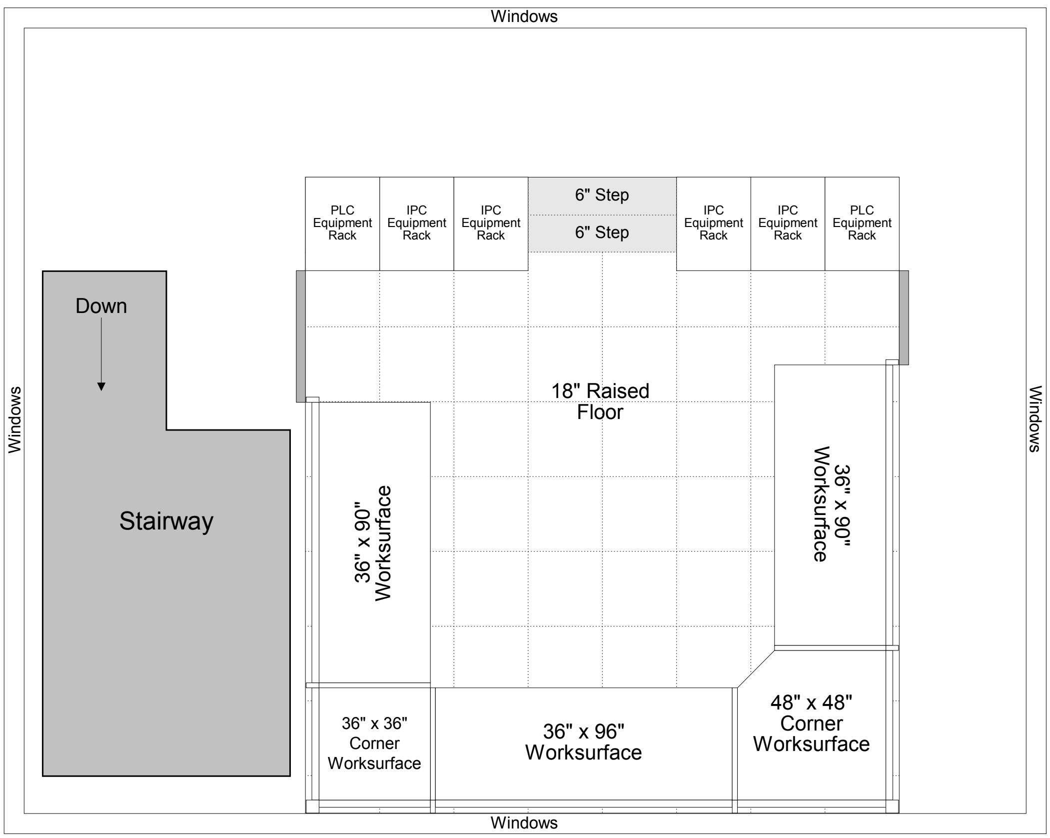


Figure 12-1 Melvin Price central control room

As stated above, the parameters guiding the control room layout will be different for a manually controlled lock and an automated control facility. The size, type, and amount of traffic also should help determine the location, size, and layout of a lock control room.

(a) Automated Control Systems. Automated systems require much less direct visibility on the part of the lock operator. However, as stated above, direct visibility will enhance the efficiency and safety of the lock. Designing a control house in an area to provide maximum visibility is a good practice, provided it is economically feasible. It is noted again that, when central control systems are utilized, it is strongly recommended that the maintenance program include the requirement for lock personnel to periodically be at the equipment when it operates, looking and listening for any indications of maintenance needs. Remember that CCTV monitors can be used to provide quick, convenient visual feedback of vessel movement, status of gates, and debris in the water, while direct visibility usually provides better feedback of large, distant areas such as approach forebays. For this reason, a centralized control room usually makes a better system if it is located at a height above the lock wall with a good view of the upstream and downstream lock approach areas. Automated systems usually justify single control rooms for centralized operation of the entire facility by a single operator. Such a control room should be kept relatively small, with all the control and monitoring equipment within convenient reach of the operator. Administrative areas, maintenance areas, and visitor accesses should be kept separate from the control room in order to minimize distractions to the operator. A reasonably accessible break room with restrooms, a stove, microwave, and refrigerator should be considered where possible. Take under review the possibility of providing a means for fresh air ventilation during pleasant weather. These all will help to increase the overall efficiency of the lock.

(b) Manual Control Systems. Because manual systems require much more direct intervention by the operator, control rooms must be kept near the lock wall area. This greatly reduces visibility of the lock approach areas and dam or spillway gate areas. A CCTV system can supplement the operator's efficiency by providing continuous views of these areas. This type of control room will require a different arrangement of equipment because an operator will be moving in and out as he performs lock wall duties, such as operation of a tow haulage unit, that are necessary in a manual control system. Control rooms of this kind will be exposed to weather, traffic, dust, and dirt much more than a centralized control house because they share duties with administrative, maintenance, and visitor access functions. These factors could shorten the life of some control system components and should be considered when laying out such a control room. Protection of the equipment and convenience for the operator might be factors that require compromise in the layout of a manual control room.

(4) Redundancy. All control systems should be provided with backup control devices as well as redundant points of control. With the exception of a remotely controlled lock and dam, for which there should always be an on-site control room with full capabilities, the need for redundant control rooms or houses is not justified at most locks. In contrast to a control point, a control room or control house is an area where lock controls, CCTV monitors and controls, marine radios, telephone lines, vessel-logging PCs, water level readouts, weather instrumentation readouts, and dam or spillway gate controls are grouped together to facilitate operation of the entire project. The enhanced reliability and piecemeal failure tendencies of modern control and CCTV systems make the probability of a catastrophic control room failure significantly less than in years past. This decreases the economic justification for a full-blown redundant control room. Consideration should be given to providing local controls near the lock

operating machinery. These controls can be in the form of a plug-in pendant, network connections for laptop PCs, permanently mounted hardwired pushbuttons with pilot lights, or a combination of these, depending on the needs of the lock operators and maintenance crews. These stations can be designed as local control points as well as redundant backup controls for the primary lock control room. Redundant control points in these forms do not add significant cost to the design or construction of a new control system and will go a long way to enhance the operator's confidence in the operational reliability of the project. Often, redundant controls can be used to perform maintenance duties without distracting the operator or affecting operation of the lock.

(5) Operating Consoles. The operating consoles are the actual point of interface between a lock operator and a computerized control system. A design engineer should spend significant time reviewing all the factors that make a control system operator interface user friendly, efficient, convenient, and, most of all, safe. What follows are discussions and guidelines for some of the equipment located on a centralized control console. Ultimately, it will be the responsibility of the designer to provide this equipment, other equipment unique to each lock, and capacity for additional equipment that will be added after lock personnel begin operating the facility. For locks that require smaller, more localized consoles, it will be up to the designer to determine the best location for the following control devices.

(a) Construction. The control console should be designed and constructed to act as a single unit. Modular, off-the-shelf component construction is a good choice, provided the components are of the same manufacturer and are intended to be connected to act as a single unit. Specifications should provide that all materials, including metal, hinges, shelves, finishes, and paint, be of top quality, with first rate workmanship and installation. This is important because repair and maintenance of the control console will affect all operations of the lock and dam. Control consoles may be many different shapes and sizes, depending on the type of control room and the functions required. All equipment contained within the control console should be easily accessible from the outside through hinged doors, slide drawers, or easily removed panels. Auxiliary equipment such as 120-V receptacles, cooling and ventilating fans, filters, uninterruptible power supplies, power strips, radio and radar power supplies, and networking hubs should be specified in the design document. If a detailed design of the consoles is not a part of the plans and specifications, a contractor or lock operating personnel will try to fit all this equipment into a console structure that does not have the capacity to accommodate it. The result will be an overcrowded and hard to maintain control console. All control console design and construction should include provisions for adding components to the existing structure and expanding the console. Equipment the lock operator does not need to access, such as PLC I/O racks and complex CCTV switching circuitry, should not be in the operating control console.

(b) CCTV System. While it is a good idea to consider a CCTV system with all lock control systems, automated or remotely controlled facilities require it. Because direct visibility will be limited, the CCTV system monitors should be considered the primary means of visual feedback to the operators. It is imperative to locate monitors where they will be convenient to view. Factors such as glare, operator comfort, viewing angle, and accessibility all should be considered when placing CCTV monitors in a control console. Generally, arranging CCTV monitors at a low level near the console work surface, at a slight incline toward the operator, will satisfy these factors. Figure 12-2 shows an example of a console with the CCTV monitors placed at the working level and with control monitors placed above them.

- When deciding the number of monitors to provide, a designer should consider that a lock operator needs to monitor several different views of the lock at all times during a vessel lockage. Certain areas of the facility must be monitored often for security reasons or for dam and spillway gate movements.

- Switching the same monitor between cameras on a continuous basis can be time consuming and inconvenient, leading to a tendency not to monitor certain areas of the lock. The cost of adding monitors to a CCTV system during the design phase is really not significant, considering the long-term flexibility, reliability, and redundancy they provide for the system.

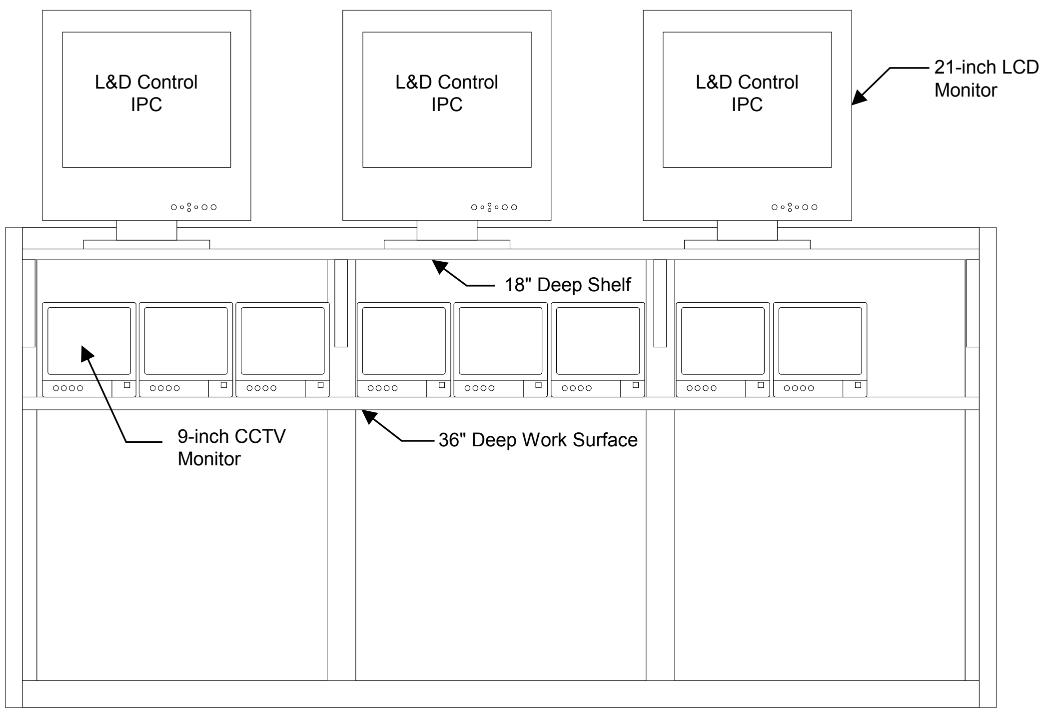


Figure 12-2. Typical control console section

(c) PC Control Network. The primary means of operating a new lock control system normally will be via stand-alone touch screens or PCs running an HMI software. The PC will connect to the PLC processor using an Ethernet connection. The PC control network should remain isolated from enterprise (corporate) networks and should be limited to HMI traffic. Internet Protocol (IP) based CCTVs should be connected on a network separate from the control system. Because the HMI operating screens are used to convey commands to the lock PLC operator, the PC monitors should be located where, aside from the CCTV monitors, they are the easiest and most convenient control system component to view. Because an operator often will be accomplishing other duties while operating the lock equipment, PC monitors located above the CCTV monitors and at approximately shoulder level will provide a good viewing angle. PC monitors should be as large as practicable, with a recommended minimum quantity of three for each lock chamber. Three monitors will allow the operator flexibility to control different features of a lock and dam without excessively switching screens. Two monitors can be used at small, low-usage locks, but this is the minimum. A single HMI screen does not provide redundancy in case of video display failure. HMI screens often use graphical representations, and the monitors should be positioned to give the operator approximately the same orientation as direct visibility. With a mouse or touch screen as the primary operator interface, convenient drawers to store keyboards are a good consideration. This will keep the console top clear and allow more space for the operator to do paperwork.

(d) Printers. Providing network printers and/or printers for the vessel-logging PC as part of the control console is a good idea. However, a printer is not used with every lockage, and its location should not compromise the location or accessibility of the CCTV monitors and controls, the PC network workstations, or whatever direct visibility might be available from the control room. Printers can be shared with maintenance computers.

12-3. Applications.

a. Lock Gates. Lock gates typically are miter gates, sector gates, or vertical lift gates. Each gate type can be motor-driven through gearing or a cable hoist or can be operated by a hydraulic cylinder. Chapter 4 discusses miter gate drives. Chapter 5 discusses sector gate drives. Hydraulic systems are discussed in Chapter 3. Hydraulic cylinders can be direct-acting or connected by a rack and sector gear. Designers must coordinate the control system requirements with the machinery designers

(1) Miter Gate.

(a) Motor-driven miter gates typically feature a motor, gearing (open or enclosed or both), and a pinion gear operating a sector gear. The sector gear connects to the miter gate through a strut arm. A two-speed motor should be used to provide a low speed for starting and mitering the gate and a high speed in between. The miter gate always should start in the low speed, even when starting from an intermediate position. A reversing, two-speed contactor is used to control the motor. The starter will be mechanically interlocked for the open/close and low/high speed functions. The starter schematic must include operation of the holding brake when opening or closing. End of travel limits can be provided by a rotary limit switch or derived from a rotary position encoder. The rotary limit switch can be used to select the gate speed. The speed also can be selected based on the gate position. Limit switches should prevent gate over-traveling when opening or closing. A single-speed, squirrel-cage motor can be used with an AC variable frequency drive in lieu of the two-speed motor. The items likely will have a shorter lead time and cost less than the two-speed motor. The AC drive also can provide a smooth acceleration and deceleration of the machinery.

(b) Hydraulically-operated miter gates require the control of the hydraulic pump and directional solenoids. A non-reversing starter is required for the hydraulic pumps. The pumps may be centrally located or dedicated to each piece of machinery. Larger pumps might require separate main and pilot pumps. The control system should provide indication of pump operation and low-oil conditions in the reservoir. Operator control and indication requirements are similar to those of the motor-driven gate, but also require controls to start and stop the hydraulic pumps and any alarms.

- The directional control valve is a solenoid-operated valve providing the open and close operations of the gate. Miter gates, including a rack and sector gear, will include physical limit switches to indicate when the rack is fully extended or retracted. Direct-acting cylinder applications must use the internal position of the hydraulic cylinder to determine when the gate is recess or mitered. The internal position may be provided by a linear differential transducer (LDT) or an integrated measurement system. The LDT provides an absolute position, while the integrated measurement system is a relative position. A relative position requires the miter gate to be moved to a known position (mitered or recessed) for re-calibration of the position.

- The gate speed is controlled by the volume of hydraulic fluid flowing into the cylinder. The volume can be set by discrete solenoids or a proportional valve. A series of discrete volume solenoids can provide multiple gate speeds. This could be low and high or multiple speeds (such as Speeds 1 through 5). The control system can select the gate speed by a limit switch and be flag mounted on the rack (only for low or high speeds) or based on the gate position. The proportional valve provides multiple gate speeds dependent upon the position of the proportional valve. The valve is controlled by an analog output signal. For example, a 4-20mA valve will open at 4mA, stop at 12mA, and close at 20mA. The farther from center (12mA), the larger the fluid volume and the faster the gate speed. The 8mA and 16mA positions will provide less volume and slower gate speeds.

- Control of the hydraulically operated miter gates can be accomplished using either a hardwired relay control system or the PLC system. When controlling the gates with a hardwired relay control system, pushbuttons or joysticks are used to control the speed (flow of oil) of the miter gates through proportional flow control valves, or multiple fixed flow valves. When PLCs are used for the control system, analog outputs serve as inputs to the valve controller, and the joysticks or multiple flow valves are used as a backup for the PLC system. A PLC system is well suited for providing an automatic closing or opening of the miter gate leaves through position feedback from an encoder or similar position sensing devices.

(c) Operator controls should include gate open, close, stop, and speed. The controls system should indicate to the operator gate closing, gate opening, gate speed. The control system also should provide indication of gate mitered (closed) and gate recessed (opened).

(2) Sector Gates. Sector gates can be operated by a wire rope and drum, a rack and pinion, or direct-acting hydraulic cylinder. Electric and hydraulic motors can be used to transmit power to the machinery. The control application will be similar to the miter gate systems. Sector gates should allow the installation of limit switches in the gate recess to provide indication of gate position when using any of the three power systems. One consideration unique to sector gates is the filling and emptying of a lock chamber. These functions usually are performed with the sector gates and might require additional limit switches and control logic.

(3) Vertical Lift Gate. There are two types of vertical lift gate hoist systems. One type of hoists consists of a continuous wire rope and sheaves to route the wire rope from the hoist drum, across the lock chamber (or gate) to the opposite side hoist machinery, then back to the driven side of the hoist. In those installations, synchronizing of the hoist is accomplished mechanically. The other type of hoist system consist of machinery on each side of the chamber or gate. This machinery is operated independently. Machinery may be a motor-driven cable hoist or a hydraulic cylinder.

(a) Motor-driven cable hoist machinery is operated by either a single-speed, squirrel-cage motor or an AC (variable frequency) drive. The AC drive permits the gate to operate at speeds less than the rated speed of the motor. The drive also provides a smooth acceleration and deceleration of the lift gate machinery. The drives may operate independently or as a master and slave when gate position and controls are joined by fiber optic cable. The AC drive should be operated closed-loop when used in any hoisting application. This will require an encoder mounted on the motor and a suitable interface card in the AC drive. The AC drive can provide control of the holding brake. The holding brake should include a limit switch to indicate to the drive when the brake has released. A rotary limit switch should be joined to the hoist drum to provide overtravel limits for the lift gate. The over-travel raised limit should not permit the gate to travel above the top of the lock wall. Control of the AC drive (raising and lowering, speed, etc.) can be performed with physical inputs to the drive from the lock control system or via commands received across Ethernet.

(b) Considerations for hydraulic cylinders will be similar to those for the miter gate system. The control application will have to provide for the operation of the hydraulic pump and valves.

(c) When controlling hoist machinery on each side of the gate that operates independently, the position of the lift gate on each side of the gate should be instrumented to ensure the gate doesn’t get skewed in the recess. A rotary position sensor on a sheave can be used to detect position of the gate. The position is scaled to an elevation in feet. The lock control system must monitor the gate movement to prevent skewing (one side of the gate higher than the other). The skew can be monitored using the rotary position encoder or with an inclinometer (for gates that are not submerged). The lock control system should stop the gate movement automatically when the skew becomes excessive, typically 1 ft. The amount of allowable skew should be coordinated with the gate and structural designer. The gate skew is corrected by raising or lowering one side of the gate independently of the other. The lock control system also can provide dynamic skew correction by slowing the machinery when it starts to lead the other side.

(d) The gate position also is used to determine when it is in the raised or lowered position, since these usually are based on pool water level. For example, the gate might be considered raised when the top of the gate is 1 ft higher than the pool water level. The gate is lowered when the top of the gate is 16 ft below the pool water level.

(e) Operator controls should include gate raise, lower, stop, and skew correction. Separate controls should be provided for raising and lowering the gate during ice flushing operations. The controls can be programmed to raise and lower the gate at a slower speed. Indication should be provided to the operator when the gate is lowering and raising. These functions should be based on outputs of the AC drive or based on motor RPM reported by the AC drive. The control system should indicate when the lift gate is raised or lowered. The system also should warn the operator when a gate has stopped at an intermediate position. For example, an audible alarm could be generated if the lift gate has stopped and the gate is not fully lowered.

b. Culvert Valves. As with miter gates, several variations of culvert valve machinery are used. Culvert valve bodies can be controlled with a cable hoist, hydraulic cylinder and bell crank, or a direct-acting cylinder.

(1) The motor-driven cable hoist can be operated by a two-speed motor or a variable frequency drive similar to the miter gate application. A rotary limit switch on the hoist drum is used to provide the opened and closed limits for the valve. The limit switch should provide indication to the control system when the valve is nearly open or nearly closed, so the valve can operate at a slower speed. A rotary encoder on the hoist drum can be used to calculate a valve position but, due to the varying diameter of the hoist drum, it will not be linear. The hoist machinery must include a limit switch to detect slack cable in the event the valve body binds or cannot close fully due to debris. When tripped, the limit switch will stop the valve immediately.

(2) A culvert valve can be hydraulically operated through a bell crank system or with a direct connection. The rotation of the bell crank can be instrumented with limit switches to indicate the valve position. Some installations have placed limit switches at the cap-end of the hydraulic cylinder to determine the valve position. A flag mounted to the cylinder can trip a proximity switch as it moves through its cycle. In a direct-acting cylinder application, the pivot point of the cylinder is at the cap-end. The rotation of the cylinder cap will be small, making it difficult to instrument the open and closed positions. A cable reel sensor or magnetostrictive sensors (Temposonics) could be used to instrument the valve movement, but the designer should consider the possibility of physical damage from drift, ice, or other debris.

(3) Operator controls should include valve open, close, and stop. The control system should indicate to the operator when the valve is opening or closing and the valve speed. The system should provide indication of valve fully closed and fully opened positions.

c. Spillway Gates. The hoist machinery for spillway gates may feature hoist systems driven by electric motors or direct-acting hydraulic cylinders.

(1) The hoist system using electric motors and gearing may use cable or chain at the hoist drum, but the gate control is independent of the material used. Hoist drums are located on the dam pier at each of the gates. The drums are connected by gears and a horizontal shaft. The drive side includes an electrical motor, holding brake, gear reducers, and starter. The motor usually is controlled by a reversing, across-the-line starter but can be controlled by an AC drive. A rotary limit switch should provide end-of-travel and over-travel limits. The limit switch is driven by the gate hoist machinery.

(2) Direct-acting cylinders can be used to operate spillway gates. Cylinders powered from independent hydraulic pumps must include a control system to raise and lower both ends of the gate equally to avoid skewing the gate in the opening.

(3) Gate position can be provided by a rotary encoder on the hoist drum or hoist machinery gearing. The relationship between the hoist machinery rotations and the vertical opening of the gate will not be linear due to the cable wraps on the drum. Gate position can be sensed by an inclinometer mounted on the upper strut of the spillway gate. Using trigonometry, the size of the opening in feet can be calculated. These forms of position indication will require a programmable logic controller to implement, but manual indication can be accomplished with a staff gage and a pointer installed on the gate or a rotary pointer and dial connected to the hoist machinery.

(4) Operator controls should include gate open, close, and stop. The controls system should indicate to the operator when the gate is closing and opening. Spillway gates are designed to operate at approximately 1 ft/min. The designer should consider a control system that provides a periodic reminder, such as an audible alarm, to the operator while a spillway gate is operating.

d. Bubbler Systems. A bubbler system uses compressed air to remove drift and other debris from miter gate recesses. The system is made of a central air compressor, distribution piping, and solenoid-operated valves (SOV). The central air compressor will start when the air pressure drops in the system. Navigation locks with significant debris may be equipped with air compressors dedicated to each gate. Dedicated compressors usually are started by the operator or control system before gate operation.

(1) SOVs open when energized, and a spring closes the valve. Depending on the environmental conditions, SOVs may remain open in cold weather conditions. Therefore, it is advisable to locate the air SOVs in a heated enclosure or environmentally controlled space.

(2) Motor-operated valves (MOVs) require electrical energy to open and close the valve. An advantage of MOVs is the ability to instrument the valve position in the control system. The MOV requires internal limit switches that stop the motor in the open and close positions.

(3) Operator controls include bubbler on and bubbler off for each miter gate (solenoid valve) location. The control system should indicate when the bubbler system is activated to prevent wear on the air compressor. For PLC-based systems, air pressure can be instrumented with a pressure transmitter.

12-4. Relay-Based (Discrete Relay) Control System.

a. General. Relay-based control systems are prevalent in USACE projects and still used frequently. Both relay-based controls, as well as PLC-based systems, should be considered for new and rehabilitated control systems and an appropriate selection made based on each facility's needs and resources. Discrete relay-based control systems typically are used in relatively simple systems, when the number of control components keeps the enclosure sizes, wiring, and troubleshooting manageable, when the control is mostly local, and when the system control logic is changed infrequently. They also might be employed where funding is limited and few personnel with the technical expertise to maintain a PLC-based system are available. All control systems have discrete relays. In a PLC-based system, these discrete relays typically are employed between output modules and field equipment. These interposing relays provide isolation to minimize transient damage to the solid-state equipment. They also are used to carry the heavier load currents that are beyond the rating of the solid-state equipment. Cost analyses (total cost of ownership) have been performed by Corps personnel when comparing PLC-based and discrete relay-based systems. Generally, the life-cycle cost comparison demonstrates that, in some instances, use of PLC-type control systems can be justified over relay-based systems. The cost comparison assumed that maintenance would be performed by lock personnel and that they are available at the project. Because there are so many factors (i.e., maintenance skill levels, proximity to equipment sources, operational preferences, legal considerations, lockage types, etc.) that vary from project to project and Corps division to division, methods of conducting cost comparisons on PLC versus discrete relay logic are not presented here. The basic control system for any lock should be a control voltage source, hardwired limit switches, pushbuttons, pilot lights, relays, contactors, E-stops, and other similar control components. A hardwired, relay-based system may also include position encoders, and transducers and other status sensing electronic devices. Relay-based control allows the lock personnel to maneuver each piece of equipment directly and individually with the full complement of interlocks and other safety devices that might be required. Typically, there is very little to no automation in projects utilizing discrete relays. Access to the discrete controls should be at every control station and at other areas around the lock, as deemed necessary by the lock personnel and design team. The discrete control components should be simple and ergonomically designed, with good visual feedback, and arranged in a logical manner. The discrete controls should have a means of operating all the lock equipment including the miter gates, lift gates, culvert valves, tainter gates, traffic lights, bubbler systems, warning horn, emergency stop, lock lighting, small pleasure-craft controls, and other equipment unique to each lock and required for day-to-day operation. The designer must spend ample time with the operating personnel to determine all the features needed for the discrete control system.

b. Advantages of discrete relay-based control. Relay-based control systems are simple to maintain and reliable. Control relays are relatively inexpensive and are produced by several manufacturers, making it relatively easy to replace them if they fail. Maintenance workers do not have to be specially trained when working on discrete control relays from project to project. Discrete relays do not require special power supplies and usually are operated at 120vac. It is common for discrete control relay system lifespan to be 35 years-plus with few instances of relay failures. There are two more common types of relays available: industrial control heavy duty relays and those called ice cube relays. Industrial control relays come with an integral mounting base, whereas ice cube relays are usually plug-in and mount-in bases. The industrial control heavy duty relays generally have a longer life and are a better choice for navigation projects that are designed for a 50-year or more life.

c. Disadvantages of discrete relay-based control. Discrete relay-based control systems can require labor-intensive wiring when building control panels and, depending on the number of relays used, can require a lot of wiring in the control enclosure. This can make the relay system harder to troubleshoot. Making changes to discrete relay-based control systems can be costly, depending on the scope of the change, especially compared to PLC systems. In larger locks or where controlled components might be quite a distance away, the designer must consider whether or not a DC control system should be used, because it is possible that inductive coupling from adjacent cables and capacitance of the control cables could keep a relay (or LED pilot light, for example) energized when it shouldn't be. Using discrete relay-based control requires much more control wiring than PLC-based systems. With the high price of copper, this could add significant cost to the project. However, once the copper is installed, it requires little to no maintenance.

d. Main Control Equipment. The lock MCC or a separate enclosure in the operations building (or similar, centrally located building) usually houses all the control relays. Because control relays are not equipment an operator will need to access on a daily basis, the enclosure housing them should be in a dry, low-humidity, low-traffic, protected area. Only qualified maintenance personnel should have access to the control relays. It would be wise to add 25% more panel mounting space to allow for changes, modifications, or expansion in the control system. Procurement, installation, wiring and startup of the relay based control system should be required by the contract documents. It is strongly suggested to provide well-written, complete specifications with well-defined control function parameters, to ensure equipment installation and operation will provide a reliable, lasting control system. It is generally not desirable to allow the contractor, who is motivated by profit, to determine the quality and performance of the control system.

e. Discrete Relays.

(1) Instantaneous relays. Relay-based control systems use discrete relays with contacts that usually are rated 10A continuous at 600vac (NEMA A600). The relays are usually used in 120vac control systems. The relays are available in the industrial control relay version (rugged, larger relays) and in a not as rugged plug-in or bolt-in relay version (sometimes called an ice cube or miniature relay). These relays usually have clear, plastic housings and usually are rated 2A at 120 vac. The plug-in version is secured in place with steel bails that attach to the relay mounting base and wrap around the relay. Relays are available in NEMA- and IEC-type ratings. A NEMA-type rated device is considered more rugged, but also is usually larger than the IEC-rated device. Both types can be successful when used within their ratings. Control relays that are used between low- and high-current portions of a control circuit are called interposing relays. Another type of relay used for this function is called a contactor or starter. A contactor is used to carry the load current of the device controlled (motor, lights, heaters, etc.). A contactor with overloads is called a starter (see below for more on overloads). Relays also are used to provide interlocks between portions of the control circuit. A common example is an interlock between the raise and lower contactors for a motor. Interlocking relays keep the contactors from being energized simultaneously, or keeps a piece of equipment from being energized under certain conditions. For example, when the upper miter gates on a lock area closed, the relay operated by the limit switches on the miter gate keeps the opposite end miter gates and culvert valves from being operated simultaneously.

(2) Timing Relays

(a) Analog type. Analog types of timers are available as motors or bellows. For critical timing functions , these timers might not be as suitable because they are less accurate than digital timers. For most USACE applications, they will be sufficiently accurate but might need adjustment during seasonal changes.

(b) Digital type. This type of timing relay is common and more accurate than the analog type. Also, they are available in a wider range of timing functions. Even though they are digital, they can be used without power supplies on standard control voltages.

(c) There are many options for timing relay contacts. Instantaneous contacts always are provided. Timed contacts include on delay, off delay, normally closed time opening (and normally open time closing) on energization, normally closed time opening (and normally open time closing) on de-energization, interval, one shot, repeat cycle (flasher) and multifunction. Input voltages vary among 12vac/vdc, 24vac/vdc, 48vac/vdc, 120vac/vdc, and 240vac/vdc. Contact ratings are usually available in B300, R300, AC15, and DC13.

(3) Undervoltage relays. Undervoltage relays can be used to detect low-voltage conditions for hoist motors (to ensure minimum torque) or to ensure available voltages do not drop below those required for proper operation.

(4) Overload relays. Overload relays are used primarily in motor circuits to ensure the driving equipment (motor, gearing, and shafts) is protected. There are two

categories of overload relays: thermal and instantaneous. Thermal overload relays have heaters or current transformers in the motor power circuit to sense overcurrents based on heat buildup in the motor. Instantaneous overload relays trip instantaneously when the current set-point is reached. This type usually is used for torque limitation on the drive machinery or load. Overload relay auxiliary contacts are located in the control circuits and serve to de-energize the control circuit upon an overload condition. Overload sensors are available in solid-state and eutectic alloy versions. Eutectic alloys melt on overcurrent. Solid-state overloads contain electronic components to sense the overload. Both types of overload relays are available in three different trip classes (10, 20, and 30), depending on the loads.

(5) Selector switches and pushbuttons. Selector switches and pushbuttons are available in different configurations such as maintained contact, momentary contact, illuminated and non-illuminated, locking, emergency stop types, and other types not as commonly used. Contact blocks are separate and available in NEMA and IEC ratings.

(6) Pilot Lights. Pilots for indicating status of equipment, alarm conditions, and other types of information are available in LED types, incandescent types (full-voltage or transformer type), or neon lamps. Push-to-test lights also are available, allowing the operator to push the lens to verify the lamp is working without having to remove the lamp for observation. The LED lamp is a preferred choice due to its long life.

f. Control wiring considerations. Hardwired relay control systems should use point-to-point copper wiring, stranded, with Type K hinge wiring to components mounted on doors of enclosures. USACE projects usually employ 120vac control circuits. For long-life single- and multi-conductor cables that are installed in conduits or cable trays, insulations and jackets should be made of cross-linked thermosetting polyethylene or ethylene propylene rubber (EPR), rated 600vac, 75 to 90°C. Copper conductors typically are specified, although aluminum conductors can be used. Wiring types that typically are used in control enclosures for control equipment are Type SIS, MTW, and XLP. Stranded wiring is preferred. Unless load calculations require #12 AWG wiring size, it is suggested that #14 AWG conductors be used for flexibility, ease of installation, wiring space, and cost savings. When specifying control equipment, consider the requirements for wiring termination. Solderless box lug or screw compression connections are required because they provide a satisfactory and easier installation, compared to terminals such as ring, spade (fork) terminations. When specifying power supplies, optional screw terminations are suggested over the standard soldered termination.

g. Motor starting. As with most other control equipment, starters are available in NEMA and IEC ratings. As stated, NEMA generally is considered a more rugged device because of the range of loads for which they are designed. However, IEC-rated devices can be used, provided the load currents and applications are within the rating of the device. Several different types of starters are available and include full-voltage, across-the-line (FVAL), WYE DELTA starters (to reduce the starting current and torque), reduced-voltage autotransformer starters (RVAT, for reduced starting currents and torques), solid-state, reduced-voltage starters, and variable frequency drives. When selecting the motor starting method, the designer must consider required torques, voltage drops as a result of the starting motor, and load requirements.

h. Equipment Enclosures. Relay-based system components should be installed in metal enclosures, complete with power sources, control transformers (if required), control relays, pushbuttons, selector switches, pilot lights, heaters, and any other equipment needed. The enclosure should be NEMA rated, as required for the area in which it will be installed, with locking door hardware. Outdoor enclosures should be rated suitably and shielded from sunlight if located in direct exposure. The control enclosures should have at least 25% spare mounting panel space for upgrades and expansions.

i. Hardwired control system interfaces. The control system interface in a hardwired control system consists of discrete pushbuttons (maintained and spring return type), selector switches (maintained and spring return type), joysticks (friction hold and spring return type), pilot lights (mostly incandescent full voltage and transformer type), and LED type. Pushbuttons and selector switches can be furnished in illuminated or non-illuminated, key-operated versions. The contacts on these control devices should be rated for the load they will be operating. The vast majority of the time, NEMA A600 contact ratings are sufficient. When used with low energy level loads (inputs to PLCs, etc), logic reed blocks or similar should be considered. These contact blocks can operate successfully with the low wiping currents of some electronic loads.

j. Grounding. All enclosures for the control components should include solid grounding provisions for operator safety. Most of the time, control circuit transformers will be grounded. However, it was common in older control systems (and some still use it today but mostly in component replacement, not new system design) not to ground the control transformer, in order to prevent loose wiring connections in any part of the control system from causing a component to inadvertently energize.

k. Operating Locations. The operating locations in a relay-based control system usually are kept to a minimum due to control wiring and wireway costs. Normally, all components are controlled from one end of the lock or the other (upstream gate or downstream gate). Additional, infrequently used control stands may be used to operate the gates or valves from the opposite side of the lock during inclement weather. When in this mode, there are two operators moving the gates, one for each gate leaf. It is common to provide culvert valve controls at the control relay location, but the miter gates always are controlled locally. It is strongly recommended that the distance from the control relays to the feature being controlled (valve, gate, motor, pilot light) be considered when laying out the control system. Inductive or capacitive coupling might keep some relays or pilot lights energized when they should not be. Manufacturers list the recommended maximum distances when using certain control relays and pilot lights, and this data should be consulted during the design of larger locks. It generally is not a problem in locks 1200 ft long and 110 ft wide or smaller.

l. Contactors and Starters. Contactors are NEMA- and IEC-rated devices that are used to carry and interrupt the larger currents required for motors and electrical loads larger than the rating of control relays. NEMA-rated devices typically are specified for Corps projects because they provide a more rugged, heavily rated component than do IEC-rated devices. IEC devices are more closely electrically sized to the load. Starters are contactors that include a thermal overload relay. Contactors and starters should be provided with additional auxiliary contacts for pilot lights, interlocking, and possible future use. Overload heater elements should be provided with auxiliary contacts for indication of trip condition to facilitate troubleshooting. Thermal overloads are available in at least two different types, eutectic alloy and solid state, and are available in three trip classes (10, 20, and 30) for various applications.

m. Variable Speed Drives. It might be desirable to use variable speed drives at some projects with relay-based controls. When using variable speed drives, it is important, for motor-cooling reasons, to coordinate the selection of the motor with the drive and the speeds at which it will be used. Not all motors are rated for use below their rated slip speed. See the PLC section below for more on variable speed drives.

n. Alarms. The lock-and-dam control system should include audible and/or visual alarms for over-travel instances, low hydraulic oil, overloads, and the like. Alarms should alert operators and maintenance personnel to failure of lock operating equipment.

o. Sensors and Field Devices for Hardwired Relay Control Systems.

(1) The input/output devices that provide information to the relay-based control system are the same for those provided for PLC systems. However, there are more electronic-type field devices used with PLC systems. Sensors and field devices are important in the overall success of a lock-and-dam control system. Failure to specify quality materials and proper installation and testing of field devices will diminish operator confidence in the entire system. Redundancy and spare parts are an important part of the auxiliary equipment design process. Typical sensors and field devices for a navigation lock include:

(2) Limit Switches. Dry contact limit switches are a common control system component. Different switches are used for different applications. These devices are particularly useful for end-of-travel or overtravel limit switches because they are absolute, passive, and require no electronic calibration. Limit switches of this type also can be provided with auxiliary contacts. Care should be taken to specify quality components because limit switches are critical to the correct, dependable, safe operation of the control system. Time should be taken to write sound specifications for procuring, installing, and testing them.

(3) Vein/Roller/Lever Operated. Simple vein or roller-operated dry contact limit switches are used for all types of moving machinery. It is a good idea to specify these with extra contacts. Specifications should include heavy-duty, oil-tight, corrosion-resistant ratings, number, rating, and type of contacts, NEMA 4X or 6P rating as appropriate, UL listing, operator lever type, and operating temperature range.

(4) Magnetic/Proximity/Photo Electric. By eliminating moving parts and providing a degree of submergibility, magnetic and proximity limit switches have replaced the vein-operated switches for use on the end of miter gate leaves for indication of proper miter, and in the miter gate recesses for indication of a fully recessed gate. To avoid damage to miter gates, HQUSACE has mandated positive indication of the miter and recess positions. These switches must be installed and used in the control system as interlocks to prevent filling of the chamber (improper miter) or changing the traffic light to green (improper recess). The use of these types of switches is a good consideration in other areas where ice can hinder the operation of vein-operated switches. When procuring magnetic or proximity switches, specifications should include number, type and rating of contacts, NEMA 4X or 6P rating as appropriate, UL listing, temperature range, copper or fiber optic leads, side or top mount, standard and extended operating ranges, and surge protection. The extra time it takes to properly specify a good switch is well spent.

(5) Encoders, inclinometers, and electronic transducers. In a relay-based control system, it might be desirable to use some electronic equipment for certain position and level measurement and display tasks. They can be used with a relay-based system, but require dedicated display devices and signal-handling equipment because PLC screens or computers are not used in the relay-based system. It is strongly recommended that absolute position encoders be furnished for control systems instead of the incremental, if at all possible and practical. This will ensure that, after loss of power to the position sensor, the position of the machinery being measured will be known. See the PLC section for more about this type of equipment.

p. Planning the design. The discrete relay main control system should be located in the electrical center of the components to be controlled, in order to minimize the length of the conductors to the field components. It is recommended that some spare mounting panel space be included or the equipment arranged to facilitate adding relays, should the system ever be revised and require more relays. The relays should be housed in a conditioned space, heated to minimize condensation. There usually isn't a significant amount of heat rejected into the discrete relay-based control systems.

12-5. Programmable Logic Controller (PLC)-Based Control System.

a. PLC Manual Controls. The basic control system should be individual PLC inputs from a Human Machine Interface (HMI) software package, hardwired pushbuttons, limit switches, encoders, transducers, and discrete PLC outputs to motor starters, contactors, and pilot lights. For example, each miter gate leaf should have an OPEN, CLOSE, and STOP pushbutton for manual operation of the gate. This type of operation allows the lock personnel to control each piece of equipment individually with the full complement of PLC interlocks, limits, and failsafe devices. This should be the minimum normal control system at any lock. Degrees of automation, as discussed herein, will vary by project site. Access to the manual controls should be at every control station and at other areas around the lock, as deemed necessary by the lock personnel and design team. The manual controls should be simple and ergonomically designed with good visual feedback such as pilot lights and/or on-screen graphics. The manual controls should have a means of operating all the lock equipment including the miter gates, lift gates, culvert valves, tainter gates, traffic lights, bubbler systems, warning horn, emergency stop, lock lighting, small pleasure-craft controls, and other equipment unique to each lock and required for day-to-day operation. The designer must spend ample time with the operating personnel to determine all the features of the PLC manual control system.

b. Programmable Logic Controllers. Programmable Logic Controller (PLC), or Programmable Automation Controller (PAC), systems are becoming increasingly common on new lock construction. For the purposes of this document, the terms PLC and PAC are interchangeable. The PLC should be industrial quality, off-the-shelf, standard equipment from a reputable manufacturer. The PLC would be the primary means of control for all lock-and-dam operating equipment. Power equipment monitoring and plant lighting control are features that can enhance the efficiency, cost, and reliability of a lock-and-dam facility. The intent of this document is to provide to designers guidelines and issues to consider when laying out a new PLC lock control system. Size, communication speeds, capacities, performance parameters, location, and the number of PLC components will vary from project to project. Ultimately, it will be the responsibility of the design PDT to determine what system is right for the project.

(1) Central Processing Unit (CPU). The central processing unit will perform all manipulations to input data, update all outputs, provide the information for HMI software to update operating screens, and accept operator commands from the HMI. The CPU should be located near the central control room area, but not in the main control console. The CPU is not something an operator needs to access on a daily basis. Whenever possible, the enclosure housing the CPU should be installed in a dry, low-humidity, low-traffic, protected area. Only qualified maintenance personnel should have access to the PLC system’s CPU. Specifying an appropriate amount of memory for the CPU is an important concern when designing a PLC system. Memory usage is different between PLC manufacturers, so it is important to specify an amount of memory that provides adequate capacity. Typically, CPU memory is specified in terms of K units, where each K unit is 1024 words (2 bytes). After becoming familiar with how memory is utilized in several PLCs, designers should determine the maximum memory requirements for the application. There are several rules of thumb, but none can be used without first knowing the approximate number of output points (i.e., real-world outputs, plus internal relay coils) in the system. Once that number is known, an estimate must be made for the amount and type of instructions associated with each output (the number of words required for each instruction is dependent on the CPU manufacturer and can be determined by consulting the manufacturer’s PLC literature). Then, the minimum amount of memory required is the estimated memory required for each output multiplied by the estimated total number of outputs. It would be wise to add an 25 to 50% more memory to allow for changes, modifications, or expansion.

(2) Procurement, installation, and programming of the CPU should be provided for in the contract documentation. When writing contract specification requirements, particular care should be taken to include a unit with the highest performance available. Such parameters include the largest amount of memory, fastest communication speed,

highest program execution rate (scan time), maximum amount and type of I/O capacity, number and flexibility of communication ports (serial RS-232, RS-422, Ethernet, etc.), special proprietary communication ports, self diagnostics, and the largest set of internal instructions. Compromise might have to be made to get the ideal processor for an individual project, but try to get the highest quality equipment with the most capacity, performance, and options to ensure adaptability to future needs. Because PLC equipment does not become obsolete as fast as PCs and software, preparing for future capacity is a good idea. Without well-written complete specifications with well-defined parameters such as those listed above, the designer is at the discretion of the contractor to provide a quality processor. Often, the processor’s quality will determine the quality of the rest of the PLC system components and, hence, a large majority of the cost. A low bid contractor is not going to provide a state-of-the-art, top-of-the-line system unless it is well specified in the contract documents.

(3) Remote Input/Output (I/O). Determining parameters for a good system of input and output PLC components is not necessarily straightforward. Manufacturers often provide several different grades of I/O components for a top-of-the-line CPU. Parameters such as isolation, density of points, response times, operating voltage levels, power requirements, fusing, and LED diagnostics should be considered when specifying I/O components.

(4) Surge Suppression. Noise suppressors should be used to protect PLC equipment from the voltage transients, spikes, and electrical noise appearing on power circuits. Such noise suppressors would be installed in each I/O rack enclosure and connected to the power supplies feeding each I/O rack. To guard both communication modules and communication cable (metallic only) from damage, I/O interface modules should be protected with overvoltage transient surge suppressors. To accomplish this, use a device that suppresses transients caused by lightning, inductive switching, and electrostatic discharge. In those applications where an inductive load, such as a motor starter or solenoid, is wired in parallel with an input module, a surge suppressor should be installed. A typical suppressor consists of a 0.5 μF, 400-V capacitor with a 220 ohm resistor in series. Procurement and installation of these should be coordinated with the PLC manufacturer.

(5) Digital Input Modules. A designer first must determine the basic requirements for I/O component types in different locations on the lock-and-dam facility. Where there are dry contact-type inputs, digital (discrete) input cards should be provided. Such inputs could include travelling nut and mercury cam limit switch assemblies, pushbuttons, selector switches, relay and motor starter auxiliary contacts, vein and/or magnetically operated limit switches, thermostats, and photocells. Digital input cards can be specified to be isolated or non-isolated. Both types serve useful purposes in lock-and-dam applications and, in some circumstances, both types should be provided in the same I/O rack location, though this increases the required number of spare I/O card types. Certain digital inputs, such as control panel pushbuttons or limit switches, on the same assemblies can be grouped together on input cards using a single reference or neutral conductor and termination point. Often, a single common conductor can be connected to these types of input groups. This requires non-isolated digital input cards.

More remote inputs operating at different voltage levels such as photocells, thermostats, and magnetic limit switches should be isolated from other inputs. Select a voltage level for each group of inputs to be the lowest possible without excessive voltage drop or capacitive coupling. In most cases, if voltage drop or capacitive coupling is a problem, an additional I/O rack assembly can be installed to reduce conductor lengths. A voltage greater than 120 vac should not be used for PLC input systems, except in special cases. With the reliability and hot-shadowing capability of most I/O power supplies, the recommended PLC control system voltage is 24 V, AC or DC. With the density of points available on today's small I/O cards, it is usually feasible to use an extra input to monitor power supplies and alert maintenance personnel to failures, as well as switching to an alternate unit without interruption in the control system process. A designer should try to maximize the number of I/O points of each type available at each location by specifying the highest available density cards. It is important, though, not to compromise other features such as isolation just to achieve more I/O points. It is unwise and ultimately will cost more in inconvenience and downtime than the money saved at first by using a minimum number of I/O cards or lower quality I/O cards with more points on each card. Therefore, first determine the type of inputs and the requirements for isolation and voltage levels. Next, specify the highest density card that meets these requirements. Specify enough cards of each type to provide a minimum of 100% spare I/O capacity beyond any known expansion plans.

(6) Digital Output Modules. There are many different types of digital outputs required in a lock-and-dam PLC system. Some of the more common types include motor starters, solenoid- and motor-operated valves, pilot lights, relays and contactors, bells, sirens, and horns. As with digital input cards, some outputs can be grouped using non-isolated output cards. These outputs, usually pilot lights or relays of the same coil voltage, utilize the same common source and the same neutral wire. Other outputs with varying voltages and/or inductive load conditions require isolated digital output cards, sometimes called relay cards. These types of cards provide a single output for each common, are electrically isolated from other outputs, and can have different voltage levels for each output. All outputs should be fused, either internally to the output cards or externally at a power supply or at the load. This protects the card as well as the field wiring. Output loads should be reviewed to ensure they do not exceed the load capacity of the output cards. In cases where the load ratings required are high or marginally high, or have high starting currents such as motors, pilot (interposing) relays should be used to provide a smaller, more consistent load on the output card as well as isolate it from more unpredictable power system faults. When specifying output cards, designers should follow a rule similar to that for input cards. First, determine the need for output cards at different locations in the system. Second, determine the operating voltage level, the need for pilot devices, the need for isolation and fusing, and the quality of card necessary for the system. Third, specify the highest density card that meets all these requirements including the spare capacity, as stated above for the digital input modules. Whenever there is doubt, remember it is easier to remove cards or exchange them rather than add them if there is insufficient space or capacity.

(7) Analog Input Modules. Analog input devices include rotating shaft encoders, resolvers, inclinometers, pressure transducers, RTDs, and hydraulic cylinder position tracking systems. Most PLC manufacturers can accommodate several different types of analog input signals, often using the same card. Input types include 4-20ma, 0-10 V, -10 - +10 V, and so forth. When specifying analog input modules, it is important to address features such as input current/voltage ranges, number of channels, impedance, resolution, accuracy as a percent of full scale, electrical isolation, shielding, fault detection, update time, I/O bus power requirements, and fusing. Analog signals present a much greater challenge to designers because inaccuracies, without failure, can occur easily due to electrical magnetic noise, improper grounding or shielding, mismatched impedances, or combinations of these. If these items are not properly addressed by the contract specifications, problems with drifting signals ultimately will occur, possibly after a contractor is long finished with the job. These types of problems can be difficult to find and correct. Specifying high-quality, isolated analog input modules will help keep these problems to a minimum. When designing a PLC system with analog inputs, a designer first should determine the location and type (current or voltage with range) of analog inputs required. Where possible, without making signal cables excessively long, these inputs should be grouped to make use of multi-channel analog input modules. Full-scale accuracy and resolution should be determined for each analog input by first determining the accuracy of measurement required to control the system. For example, if a gate raises and lowers a maximum distance of 9 m (30 ft), and the operator needs to know the position to the nearest 3 mm (one hundredth of a foot), the range of travel is 3000 counts, with each count equal to 3 mm (one one hundredth of a foot). In addition, say a rotating shaft, 4 to 20ma analog transducer mounted to the cable drum or chain sprocket produces a 5ma current signal at 0 m (0 ft) and 15ma current signal at 9 m (30 ft) of gate travel. To achieve the accuracy required by the operator, the PLC analog input card must be able to resolve 10ma of travel into a minimum of 3000 counts. Because the full-scale range of the transducer is 4 to 20ma or 16 ma, this equates to 4800 counts over the full range. This requires an accuracy of 1/4800 or roughly 0.02% of full scale, with resolution of 4800 counts or 13 bits of accuracy. Failing to perform this type of design analysis properly can result in a signal or analog input card that is not accurate enough through the entire range of travel to use for effective machinery control or position determination. When physical location, voltage/current range, accuracy, and resolution requirements for each analog input has been established, PLC analog input cards that meet all these requirements should be specified. If possible, a single type of analog input card should be used throughout the entire system. Previously, a jumper or dipswitch setting was used to configure each channel on a card for different current and voltage ranges, but now this usually can be configured in the programming software. Specifying electrical isolation levels, register update times, and fault detection features help ensure the contractor provides a quality product. The impedances of the analog input card, cable, and transmitting device should be analyzed when determining the voltage level of the power supply that drives the current loop. Shielding always should be accomplished in accordance with written recommendations from both the transmitting device and PLC manufacturers. Signal shield grounds should be isolated from power grounds.

(8) Analog Output Modules. When motion controllers, variable speed drives, hydraulic linear variable differential transformers, or other control equipment require current loops for speed or position reference, designers might want to use analog output cards to interface the PLC system with such equipment. Because a margin for error exists when using analog control signals, designers should, before deciding to use analog output cards, consider digitally integrating such equipment using serial communication standards. This will make for a simpler, more reliable control system that likely will be more flexible because of the amount of information that can be transmitted digitally. If, however, it is determined that analog output cards are necessary, the system designer should follow the same steps as outlined above for the analog input modules. Again, the objective is to perform a design analysis on each device that is driven by the analog output points to determine the exact requirements for the output cards, wiring, shielding, and power supplies. Always try to specify the highest quality component available for the system.

(9) I/O Enclosures. Remote I/O components should be installed in metal enclosures, complete with power supplies, power line conditioners, isolators, I/O component racks, ventilating equipment, desiccants, heaters, air conditioners, communication equipment, pilot devices, and uninterruptible power supplies, as needed. The enclosure should be NEMA rated, as required for the area in which it will be installed, with locking door hardware. Outdoor enclosures should be rated suitably and shielded against sunlight if directly exposed. Remote I/O racks should be as required for the type of cards specified and should have at least 50% empty slots for upgrades and expansions. Where applicable, I/O rack addressing should allow for the empty spaces to be used without readdressing an entire program. To achieve a reliable system, it is important that design engineers look at all of the components necessary in the I/O enclosures to ensure they are of proper size, rating, and capacity with proper space requirements for dissipation of heat. Calculations should be done to determine the need for and sizing of heating, air conditioning, and ventilating equipment. All this should be given in a design specifications so a contractor will provide the highest quality equipment, helping to ensure a reliable system. One of the critical parts of the I/O enclosure design work is the grounding of the electronic and communication equipment contained within the enclosure and the enclosure itself. The National Electric Code does not cover in sufficient detail the grounding of such equipment. Therefore, references to the NEC within plans and specifications do not guide a contractor very well in the area of electronic ground systems. In general, it is wise to keep electronic grounds separate from power and conduit system grounds even if, ultimately, the two are tied at some grounding point. Keeping the electronic equipment out of the path of other potential ground surges is important. I/O racks, power supplies, communication interface equipment, and other electronic equipment should be mounted to the I/O enclosure metal using rubber or plastic standoffs to isolate them from the enclosure itself. Most electronic equipment is provided with a ground terminal and instructions from the manufacturer on how to ground the equipment. It is important to ground and shield all electronic equipment properly.

c. Network Configurations. PLC systems can be networked in several different configurations. The general guideline is to locate I/O racks in areas where limit switches, motor starters, and solenoids are grouped. Networking of I/O racks on PLC communication channels should be laid out in a design document with consideration given to fail override and redundancy. Some general design guidance:

(1) I/O Rack Location. The first objective is to determine the number and location of system I/O racks. A designer first must survey and chart all the I/O points necessary to operate the lock. This includes all discrete and analog points. Sorting the I/O point list by general location will give the designer an idea of where they are concentrated. I/O racks should be in areas where a significant number of I/O points are grouped, taking care not to use so many I/O racks that the overall number impacts the availability of the system by increasing the number of failure points. It is difficult to determine the minimum number of I/O points in an area that requires installation of an I/O rack. This will vary from project to project and is relative to the overall number of I/O points. An important consideration is the number of lengthy control circuits that can be eliminated by the installation of an I/O rack. An objective a designer should have in mind is to keep hardwired, difficult-to-diagnose I/O circuits as short and accessible as possible. With fiber optic technology, lengthy communication circuits are not only possible but also easy to maintain, diagnose, and repair. Therefore, a designer should not be afraid to specify additional I/O racks in remote areas where there are relatively few I/O points. In any case, an availability analysis should be done to determine the optimum number of I/O racks for the application. In some cases, the analysis might show that fewer I/O racks with longer cable runs will provide the optimum system availability. In cases where single I/O are remotely located from I/O racks, consider using optical switches and sensors.

(2) I/O Rack Networking. After the location of I/O racks has been determined, it will be necessary to connect them in a network configuration. While different networking protocols have been used over the years, Ethernet IP is becoming the industry standard for PLC networks, as many devices are built with Ethernet capability. Using Ethernet also makes it easier to connect PCs and HMIs to the network for monitoring, logging, and control, as needed. Disclaimer: Using Ethernet protocol in no way implies or advocates connecting a navigation lock network to the Internet, or any external network for that matter. Avoid a linear network configuration, where loss of one rack could impair or stop communications with devices past that on the network. While a star configuration can offer the most reliability, it also requires the most cabling (whether copper or fiber optic). A ring network also should be considered. If possible, adjacent locks should be on separate PLC processors and networks. Communication between I/O racks generally should be via high-quality fiber optic cables, as recommended by the manufacturer of the PLC system. Specifications should be so written to provide the fastest I/O network communication speed available. This will help ensure the system the contractor provides is of high quality. Make sure Ethernet switches used on the network are of industrial quality and capable of supporting the chosen network. Enable IGMP (Internet Group Management Protocol) snooping to reduce bandwidth consumption by reducing multicast streams. Converters, power supplies, and other communication

equipment all should be supplied by the PLC manufacturer. A third party may provide such equipment, but only as recommended by the PLC manufacturer. This also will help ensure a quality PLC system.

d. Human Machine Interfaces (HMIs). Input devices can consist of personal computers (PCs) with a mouse and keyboard running HMI software, touch screens, physical controls (pushbuttons, switches, etc.) wired to PLC I/O, or a combination of the above.

(1) Keyboards. Because of its widespread use, standard IBM AT keyboards should be used with PCs. Small footprint versions may be used, depending on console design. Custom-built consoles or racks with pullout drawers may be used to house keyboards. Protection against dust and liquid can be achieved by using keyboard overlays.

(2) Mouse. Either a standard two-button Microsoft-compatible mouse or a heavy-duty unit may be used. Considering the heavy 24-hour-per-day, 7-day-per-week use, a mouse with no moving parts, such as an optical or laser mouse, is preferred.

(3) Touch Screen. Touch screens come in several varieties, including infrared, surface acoustic wave (SAW), and resistive types. Of the touch screens implemented for lock-and-dam control, the SAW type is less prone to false inputs. The infrared type touch screens are best used in applications that require a second acknowledgment for confirmation of each input. SAW-type touch screens are much more forgiving due to their design. They require firm pressure applied to the screen before registering as an input. Touch-screen HMIs can be used as a stand-alone terminal, or monitors with touch-screen capabilities can be used with a PC, instead of a mouse and keyboard.

(4) Video Display Monitors. There are two common types of video display monitors: cathode ray tube (CRT) and liquid crystal display (LCD). The disadvantages of CRT monitors, among which are excessive power consumption, heat generation, and large footprints, have led to their declining use. Advantages of LCD monitors are low heat generation, small footprint, and lightweight build. LCD flat screens are available in large sizes, with high resolutions, and wide viewing angles. Power consumption is typically less than half that of CRT monitors, and heat generation is correspondingly less. Their flat design, small footprint, lightweight build, wide viewing angle, and brightness are ideal for lock-and-dam control. Note, however, there are many flat-screen display manufacturers and not all models are equivalent. The primary characteristics to look for are high resolution and wide viewing angles. Touch-screen LCD monitors also can be used, eliminating the need for a keyboard and mouse in daily use.

(5) Industrial Personal Computers. In applications where a personal computer is being used for monitoring and/or control, industrial personal computers (IPCs) can be considered. IPCs are built to withstand a wide variety of environments. They are specially shielded against electromagnetic and radio frequency interference and certified to meet the Federal Communications Commission’s EMI and RFI regulations.

They also are built for a wide range of temperature extremes and can withstand extreme thermal stress. IPCs are built with shock-and-vibration resistance for rugged and heavy-duty use. IPCs can be considerably more expensive than name-brand PCs with similar specs, so ease of replacement and downtime impact should be taken into consideration. IPCs also are called blind nodes and non-display computers, depending on the manufacturer. In the following discussions, the terms PC and IPC can be used interchangeably.

(6) Power Systems

(a) Uninterruptible Power Supplies. Clean, reliable power is perhaps the most important factor in assuring long life of a computer system. The simplest way to achieve this is through the use of uninterruptible power supplies (UPS). Careful consideration should be given to designing the power supply for the computer system. Large harmonic currents caused by nonlinear loads should be reduced with larger, grounded (neutral) conductors and with K-rated transformers suitable for nonlinear electronic loads. Note that a UPS is not a harmonic eliminator, and that multiple harmonic generating devices connected to the UPS output could affect each other. The power system design should take into account the harmonic distortion caused by the UPS on its line side. UPS should be circuited separately. For centrally located computers, a large, floor-mount UPS might be suitable for providing backup power and surge protection. Several smaller floor-mount UPS also can be used. For remote, stand-alone computers, a small UPS should be provided. Units should be sized carefully to handle the power loads for all the computer components including the computer chassis and all internal devices, as well as the video display monitor. Peripheral components such as printers, modems, and network hubs (if serving non-critical computers) might or might not need continuous power. Consider connecting these devices to the UPS, however, to provide them with lightning and surge protection. Rack-mounted UPS can be used for rack-mounted computer equipment. Depending on the application, several rack-mounted units might be required because of their limited power capacity. It might be beneficial to use UPS that include communications capabilities. The UPS can be monitored and controlled with software through a communication link to the computer.

(b) Surge Protection. All computer systems for lock-and-dam control should have surge protection. As a minimum, voltage surge protection should be provided for each lighting panelboard and at each power distribution panel that feeds computer equipment. It is recommended that surge-protected power receptacles or plug-in strips also be used.

(c) Grounding. Surge protection equipment is only as good as the grounding electrode system. A good grounding system should be installed in compliance with the National Electrical Code. Ground systems at old facilities that are retrofitted or rehabilitated for computer control should be upgraded to meet the needs of modern computer equipment.

(7) Rack Mount versus Stand-alone Systems. For installations where several PCs are grouped in a single control room, rack-mount systems might offer a better

solution than stand-alone systems. All the computer equipment can be mounted in a single, 19-in. equipment cabinet outside of the control room. Monitors, keyboards, and mouse units would be located in the control room and extended with special devices. This setup provides the ideal arrangement for upgrading, troubleshooting, and maintaining the computer equipment because all the major components are in the same enclosure. Also, work on the computers can be done without disturbing operators. Rack-mounted equipment should not limit replacement or retrofit options because industrial computer manufacturers always provide components suitable for rack mounting in standard racks.

(8) Printing. Printers are convenient during the development and testing stages of a new control system, but also can be used for data logging and vessel report generation during normal use. Consider using networked printers so any PC on the network can print to all of the printers. This provides the most flexible printing arrangement. With Windows XP or later, printers can be connected directly to one PC, yet shared among all PCs. Alternatively, and perhaps the more flexibly, printers can be purchased or provided with their own network interface card so they act as an addressed network device. This allows PCs to be shut down without disrupting printing capability.

e. Network Fundamentals. It is important to review the fundamentals of networks before discussing their applications to lock-and-dam control systems. A network is a group of two or more linked computer systems. There are many types of computer networks, including local-area networks (LANs) and wide-area networks (WANs). With LANs, the computers are geographically close (i.e., in the same building or group of buildings). With WANs, the computers are farther apart and connected by telephone lines or radio waves. In addition to these types, the following characteristics are used to categorize different types of networks.

(1) Topology. The geometric arrangement of a computer system. See Figure 12-3 for the three principle topologies used in LANs.

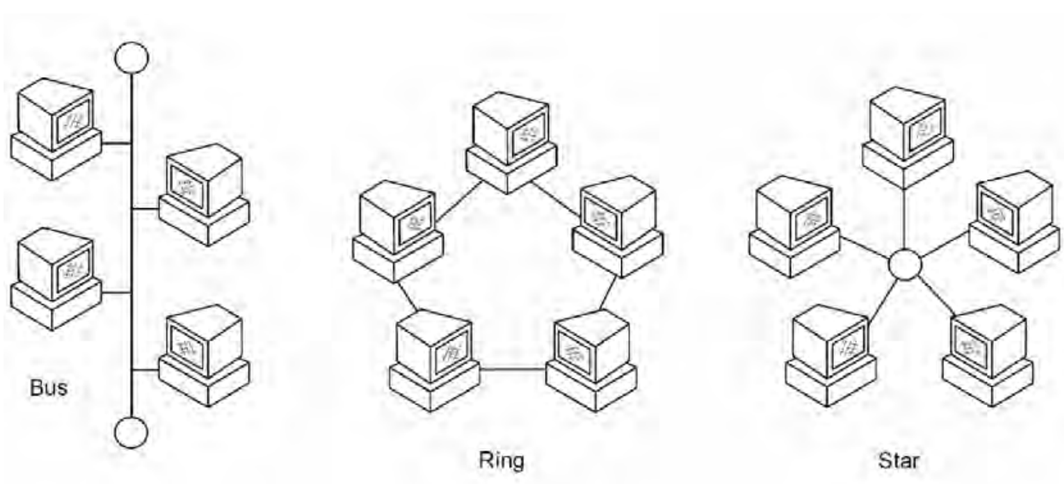


Figure 12-3. LAN topologies

(a) Bus Topology. All devices are connected to a central cable, called a bus or backbone. Bus networks are relatively inexpensive and easy to install. Ethernet systems typically use a bus topology. The downside is that a failure anywhere along the bus will isolate all computers past the failure.

(b) Ring Topology. All devices are connected in the shape of a closed loop, so that each device is connected directly to two other devices, one on either side. Ring topologies are more expensive and can be more difficult to install, but they offer high bandwidth and can span large distances. Another advantage is a single failure anywhere along the ring will not isolate any devices, keeping the network active to all but the failed device.

(c) Star Topology. All devices are connected to a central hub. Star networks are relatively easy to install and manage, but bottlenecks can occur because all data must pass through the hub. Star topology is usually the most expensive, as long cable runs might be needed to connect all devices to the central hub. The advantage of star topology is that failure anywhere on the network, other than the central hub, will not affect any other devices on the network.

(d) Variations on these topologies exist. The bus and star topologies, for example, can be combined to form a hybrid LAN. This arrangement is useful where several remote PCs need to be networked to a central LAN, such as in a control room. The remote PCs connect in a star configuration to a network hub that is connected to a local bus. See Figure 12-4 for an example of a hybrid LAN. This topology has worked well at Locks No. 27 on the Mississippi River.

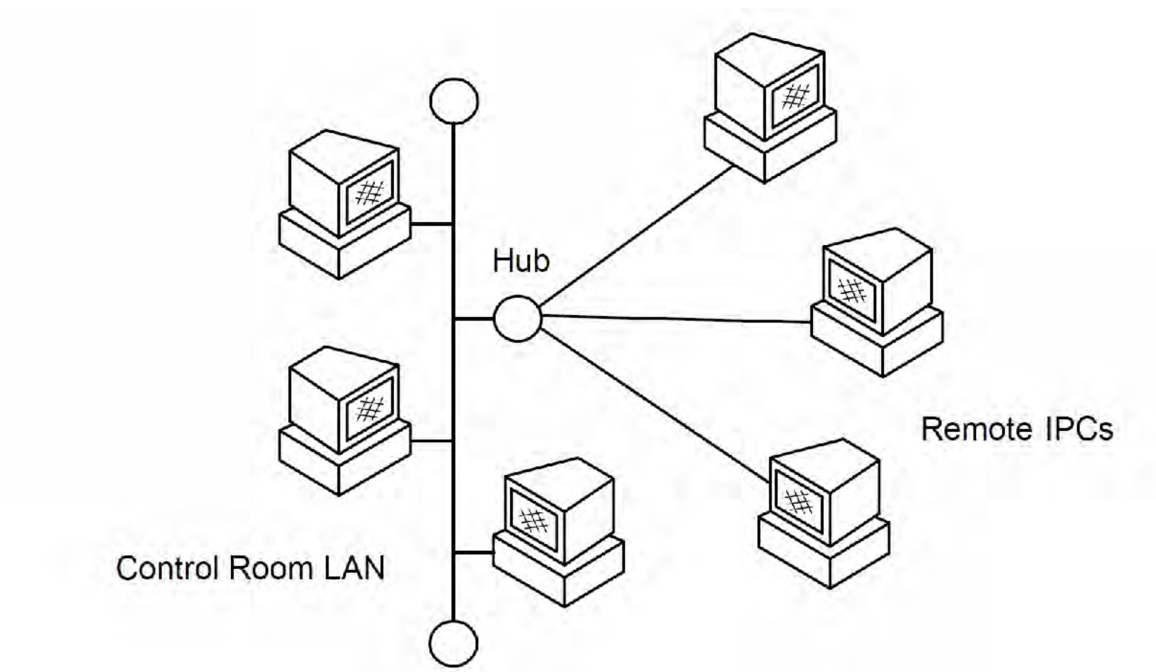


Figure 12-4. Hybrid bus and star LAN topology

(2) Protocol. The protocol defines a common set of rules and signals that computers on the network use to communicate. One of the most popular and widely used protocols for LANs is Ethernet. Ethernet was developed by Xerox Corporation in cooperation with DEC and Intel in 1976. Ethernet supports data transfer rates up to 1 Gbps.

(3) Network Considerations for Lock-and-Dam Control. The same considerations apply to lock-and-dam networks that apply to any sizeable network. They include reliability, availability of management and troubleshooting tools, scalability, and cost.

(a) Reliability. Highly reliable networks are critical to the success of a network at a lock and dam, so ease of installation and support are primary considerations in the choice of network technology. Ethernet networks are by far the most widely used. Because of this popularity, equipment and wiring systems have become increasingly reliable. They are also relatively simple to understand and administer.

(b) Availability of Management and Troubleshooting Tools. Management tools for Ethernet, made possible by widespread adoption of management standards, including Simple Network Management Protocol (SNMP) and its successors, allow an administrator to view the status of all desktops and network elements, including redundant elements, from a central station. Ethernet troubleshooting tools span a range of capabilities, from simple link indicator lights to sophisticated network analyzers. As a result of Ethernet's popularity, many people have been trained on its installation, maintenance, and troubleshooting.

(4) Network Design for Lock-and-Dam Control. To design a reliable network, it is important to understand some of the limitations of the different technologies available and to decide which features are required for the particular application. An important design consideration for a lock-and-dam network is the size of the facility. If the lock and dam is controlled from a single point, with no plans to add control points away from the main control room, distance is not a problem. However, for distributed control, in which control points might be hundreds or thousands of feet apart, distance becomes a critical factor in the design of the network.

(a) Table 12-1 shows that Fast Ethernet and Gigabit Ethernet may be implemented only at large facilities with widely separate control points using fiber optic cable. In most instances, Fast Ethernet also will require fiber optic cable between distant control points. For 183- and 366-m (600- and 1200-ft) locks, control points located at each end of the lock exceed the maximum network distance for copper wiring.

Table 12-1. General rules for maximum network distance

	Fast Ethernet 100 BASE-T	**Gigabit Ethernet 1000 BASE-T (LX)**
Data Rate	100 Mbps	1 Gbps
Cat 6 Unshielded Twisted Pair	100 m	100 m
Shielded Twisted Pair/Coax	100 m	100 m
Multi-mode Fiber	400 m (half duplex) 2 km (full duplex)	550 m
Single-mode Fiber	10 km	2 km

(b) Noise. Another important consideration in network design is noise from radio and electromagnetic interference. Locks and dams can be extremely noisy environments, especially with large motors, variable speed drives, and so forth. Lightning and surge protection are other important considerations. Fiber is naturally suited to protect against noise, lightning, and surges because it is non-conductive, using glass as the medium of transmission instead of copper.

(c) Operating Plan. During design of the network, an operating plan must be developed, and a network topology based on the operating plan, a suitable network protocol, and the network architecture must be selected. The operating plan includes selection of operating locations, number of control points at each location, and primary versus secondary control points.

(d) Operating Locations. The operating locations may include a central control room with backup control points situated at strategic locations at opposite ends of the lock chambers. At each of these control points, the total number of operating work stations must be determined. Typically, primary control points, or those that are used for normal operation, might have two workstations in case one fails. Secondary, or backup, operating points might have only one work station, depending on how critical and how frequently that location is used.

(e) Operating Points. Using the maximum network distances as a guideline, the operating points should be grouped by distance of separation. Those that are within 100 m of each other should be considered local, while those beyond 100 m should be considered remote. One operating point should be selected as the primary control point if there is no central operating location. The most logical network topology should become clear from this analysis. In most cases, the topology will be the bus topology, star topology, ring topology, or a combination.

(f) Protocol Selection. Next, the network protocol must be selected. Ethernet is the protocol of choice, but consideration should be given to other protocols depending on the application. Ethernet has several shortcomings of which network designers should be aware. Foremost, it is a non-deterministic protocol, meaning there is no guaranteed time in which communication between two or more nodes has been completed. This probably is not a problem for lock-and-dam control, but some industrial applications require a deterministic network in order to ensure control of critical processes.

(5) Design Details. Once the network is designed, the details can be completed. This includes selection of hardware and software (operating system).

(a) Network Routers. Routers are used for controlling communications between two locations. Routers typically are not used in lock-and-dam networks. They are used, however, to provide control communication from the lock and dam back to the district office for the lock performance monitoring system (also called the OMNI system). Routers might have application in remote control of locks and dams. The network should be designed keeping the option of remote control in mind. Refer to Engineering Circular EC 6071, Remote Operation of Water Control Systems for more information.

(b) Network Hubs. A hub can be the center of a star topology network system. Hubs can be used to convert between different physical network media, such as fiber to twisted pair and vice versa.

(c) Network Repeaters. Repeaters are used to extend the length, topology, or interconnectivity of the physical network medium beyond the limits imposed by a single segment. They perform the basic actions of restoring signal amplitude, waveform, and timing applied to normal data and collision signals. The designer should be aware that there are limitations in extending networks using repeaters and plan the network accordingly. For example, repeaters add delay when re-transmitting communication signals. The overall delay for all repeaters in a chain must not exceed the acceptable limit.

(d) Network Interface Cards. Network interface cards (NICs) provide the connection between the computer and network. NICs are available in many varieties for connection to ThinNet, ThickNet, twisted pair, and fiber networks. Consideration should be given to using the type of NIC best suited for direct connection to the network. For example, fiber NICs should be used for direct connection to fiber media, without converting to a different media using a transceiver. The drawback to this approach is that more than one type of NIC is required for the computers attached to the network. The advantage of this approach is that fewer components are required and the system reliability consequently should be higher.

(e) Network Transceivers. Network transceivers provide a single physical connection between standard Ethernet and Ethernet communication equipment. Transceivers typically are used to convert between media types. For example, a four-port hub with ThinNet network connection and AUI (attachment unit interface) ports might be available, while the same unit with fiber ports is not. Fiber can be connected to the AUI ports using fiber-to-AUI transceivers. This is an important consideration in network design because communication equipment is not always available in the desired configuration and port types.

(f) Dial-Up Networking. Dial-up networking is becoming increasingly popular. A dialup network router typically is attached to the LAN and is provided with analog telephone ports. The router allows remote users to dial into the network and access it off site for software maintenance and troubleshooting. The router should support TCP/IP, IPX routing, PPP, and Multilink PPP (MP) protocols. It also should provide user authentication for security management and firewall technology. Proper configuration of dial-up networking or other means of remote access must be reviewed and confirmed under DIACAP (see paragraph 12-2.c.(3).(c))

(g) Wireless Networking. For rehabilitated locks and dams, as well as new facilities, wireless LANs might provide an excellent alternative or supplement to wired versions. For control points that are used infrequently, for example, it might make more sense economically to operate from that location with a notebook computer equipped with a wireless Ethernet card. Such devices can interface with existing Ethernet LANs and operate at high data rates up to 3 Mbps using frequency-hopping, spread spectrum technology at 2.4 GHz. The range of wireless LAN devices varies from 1 km to several kilometers (3000 ft to several miles), depending on the antenna used. Wireless LANs also allow mobility and roaming; gates can be operated while positioned at the gate. This ability to operate equipment from anywhere can be invaluable for troubleshooting and maintenance. The use of wireless networking, even in an isolated or stand-alone control system, may generate additional requirements under DIACAP.

(6) Cabling. Good cabling is essential to a reliable network. There are three types of cables used for transmission in a network: coaxial, twisted-pair, and fiber optic.

(a) Coaxial Cable. A type of wire that consists of a center wire surrounded by insulation, then a grounded shield of braided wire. The shield minimizes electrical and radio frequency interference. 10BASE-2, also called ThinNet, is one adaptation of the IEEE 802.3 Ethernet standard and uses 50-ohm coaxial cable (RG-58 A/U) with maximum lengths of 185 m. This cable is thinner and more flexible than that used for the original 10BASE-5, also called ThickNet, Ethernet cabling standard and is less expensive and easier to install. The maximum cable length in 10BASE-5 is 500 m. Cables in the 10BASE-2 system connect with BNC connectors. Network devices connect to the network with a T-connector so that cables can connect to other devices. Any unused connection must have a 50-ohm terminator.

(b) AUI. AUI, short for Attachment Unit Interface, is part of the Ethernet standard that specifies how a cable is to be connected to an Ethernet card. AUI specifies that a coaxial cable connects to a transceiver that plugs into a 15-pin socket on the network interface card.

(c) Twisted-Pair Cable. This type of cable consists of two independently insulated wires twisted around one another. One wire carries the signal, while the other wire is grounded and absorbs signal interference. Twisted-pair cable is the least expensive type of LAN cable. 10BASE-T is one adaptation of the IEEE 802.3 Ethernet standard and uses a twisted-pair cable with a maximum length of 100 m. 100BASE-T, also called Fast Ethernet, is another adaptation of the Ethernet standard and is 10 times faster than 10BASE-T. Cables in both systems connect with RJ-45 connectors. Star topologies are common in 10BASE-T and 100BASE-T systems.

(d) Fiber Optic Cable. This type of cable uses glass or plastic fibers to transmit data. A fiber optic cable consists of a bundle of glass threads, each capable of transmitting messages at close to the speed of light. Fiber optic cables have several advantages over traditional copper communication lines: much greater bandwidth, meaning they can carry more data; less susceptible to electromagnetic interference; and much thinner and lighter. The main disadvantage of fiber optics is that the cables are more expensive to install. There are two basic types of fiber: multimode and single mode. Multimode fibers have a large core (25 to 300 μm) and permit non-axial light rays or modes to propagate through the core. Single mode fibers have a small core (5 to 10 μm) and allow only a single light ray or mode to be transmitted through the core. This virtually eliminates any distortion due to the light pulses overlapping, as in multimode fiber. Multimode fiber is used more commonly in small LANs, while single mode fiber, because of its higher capacity and capability, is used for long-distance transmission. Telephone companies typically use single mode fiber because of its ability to transmit long distances without the need for repeaters. There are a variety of connector types for fiber optic cable. Two common ones are ST and SMA (sub-miniature assembly). ST connectors are metal with a straight tip and are the preferred type. They join by pushing and twisting. SMA connectors are used with multimode fiber and are the screw-on type. Both types of connectors are available with metal, plastic, ceramic, and glass tips.

(e) Motor Control Centers. The demarcation point between the PLC/PC computerized control system and the traditional electrical distribution and control system is the MCC or, in some cases, a 480-V switchboard-type motor controller enclosure. See Chapter 11 for more information.

(7) Motor Starters. Starters should have sufficient auxiliary contacts to provide both hardwired and PLC feedback. It is always a good idea to have extra contacts for future use. Overload heaters should have auxiliary contacts for input to the PLC system. This will provide enhanced remote troubleshooting capabilities. Solid-state motor starters with PLC network connections are also good considerations for critical motors.

While the primary means of energizing motor starters will be the PLC I/O system, it is a good idea to provide a means of energizing the starters at the MCC. This can be via simple deadman-type pushbuttons that can be de-energized when the PLC and minimal hardwired control systems are completely functional. When using a hydraulic system, these MCC pushbuttons should be accompanied by similar pushbuttons for energizing hydraulic solenoids. By doing this, the designer provides a way for the general contractor, during construction, to bump motors and cylinders for shaft alignment, cylinder attachment, clevis pin attachment, shim and key installation, and so forth. This will relieve the system integrator of the burden of providing untested solid-state controls before actual gate operation. This also will provide the system integrator with opportunities to check transducer signals and feedback devices without the responsibility of operating the equipment.

(8) Control Relays. In general, motor starters should be provided with pilot control relays to interface them with the PLC I/O cards. While some PLC isolated output cards are rated for enough current to energize smaller starters, larger starters will require pilot relays. Pilot relays provide a way to isolate the PLC system from starters and potential damage from 480-V system faults.

f. Variable Speed Drives. Mechanical loads that require soft starting, multiple speeds, or ramp up/ramp down features warrant consideration for a variable speed drive. This could be a DC drive or a variable frequency AC drive. Traditionally, DC drives have been used because they provide much greater control of motor speed and torque. Varying the voltage changes the speed and inversely changes the torque of a DC motor in a linear manner. In contrast, AC drives provide very fine control of speed with varying or constant torque. In addition, AC drives can be supplied with the inherent feature of an across-the-line bypass contactor in the event of solid-state inverter failure. Therefore, when specifying variable speed drives, first consider an AC adjustable frequency drive (AFD), also called a variable frequency drive (VFD), with either constant or varying torque. It is a good idea to require that the AFD be of the same manufacturer as the MCC. Large drives that do not fit in MCC-type construction will require free-standing enclosures to house them. Space is an important requirement, and a designer should consider that an engineered drive occupies significantly more space than the inverter itself. An engineered drive consists of:

(1) Isolated Bypass. Except in cases where emergency, across-the-line starting will damage mechanical or structural equipment, AFDs should be provided with an isolated bypass, across-the-line contactor. The AFD should include full controls, accessible from the PLC system, for switching to the inverter-bypass starting mode. Determine the control features and/or operational procedures that will require special consideration in order to start the load in this fashion. Such considerations should be programmed in the PLC and/or AFD controller. These details should be well covered within the specifications, to insure proper coordination by the AFD manufacturer and the system integrator.

(2) Network Communications. The AFD should be provided with a means to communicate digitally with the PLC processor, via serial interface. This networking

capability should be an inherent feature of the drive and might require the AFD to be of the same manufacturer as the PLC system. This might limit the number of PLC manufacturers that can supply the system, but it is a necessary requirement when providing a reliable, coordinated PLC/AFD system. The network communication should provide all status and diagnostics of the drive to the PLC system for remote troubleshooting capability. Specifying the network communication speed is also a good way of specifying a higher quality product.

(3) Hardwired Stop Override. The AFD should be provided with a means of stopping the drive independent of the PLC system. Activation of this override should come from the lock hardwired emergency stop system. Indication of the status of the stop override should be available on the PLC network.

(4) Dynamic Braking. All AFD systems require dynamic braking of the load. Gates and bridges that are lowered by gravity require excessive dynamic braking to control the speed of the falling load with the electrical-magnetic braking torque of the motor. Manufacturers should be consulted to calculate the exact amount of dynamic braking required for the system. A conservative approach to such calculations will prolong the life of the resistor banks and possibly the inverter. Failure of the dynamic braking system, such as overheating, should be an input function of the PLC system to provide help in troubleshooting.

(5) Isolation Transformer. All AFDs reflect harmonics back to the power distribution system. Such harmonics can damage transformer neutrals as well as affect other digital switching loads. For this reason, an appropriately sized K-factor-rated isolation transformer should be provided with each AFD. The designer should consult the AFD's manufacturer to determine the exact size and ratings of the isolation transformer.

g. Software. Because most general contractors do not have the expertise to install and configure PC and PLC networks, a system integrator should be used for this purpose. A system integrator is a company that designs, installs, programs, and provides start-up and maintenance services for commercial/industrial control and computer systems. The successful implementation of a computerized control system depends largely on the capabilities of the system integrator. It also depends on the capabilities and support of the engineers responsible for the control system after it is installed. Few control systems are perfect immediately after installation. Most require tweaking to incorporate missing features or to adjust parameters that were not defined during testing and startup. On lock-and-dam applications, often after the project is operating, the lock personnel will request some changes as they become accustomed to the system. There are three major software components in a computerized lock-and-dam control system: operating system, PLC, and HMI.

h. Operating System Software.

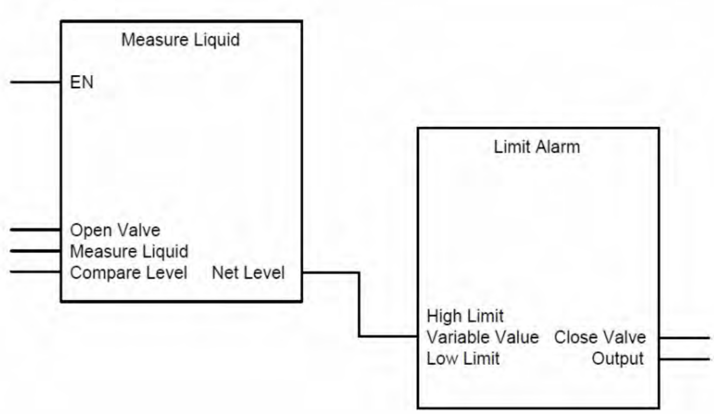
(1) The operating system is software that every computer must have to run other applications. It performs basic tasks, such as recognizing input from the keyboard or

mouse, sending output to the display screen, keeping track of files and directories on the disc, and controlling peripheral devices such as modems and printers. Operating systems provide a software platform on which other programs, called applications, can run. The application programs are written to run on a particular operating system. The choice of operating system determines which applications can be run. There are many different operating systems, including DOS, Microsoft Windows, OSX, and Unix. In the Corps, Microsoft Windows 7 is the preferred operating system for industrial control.

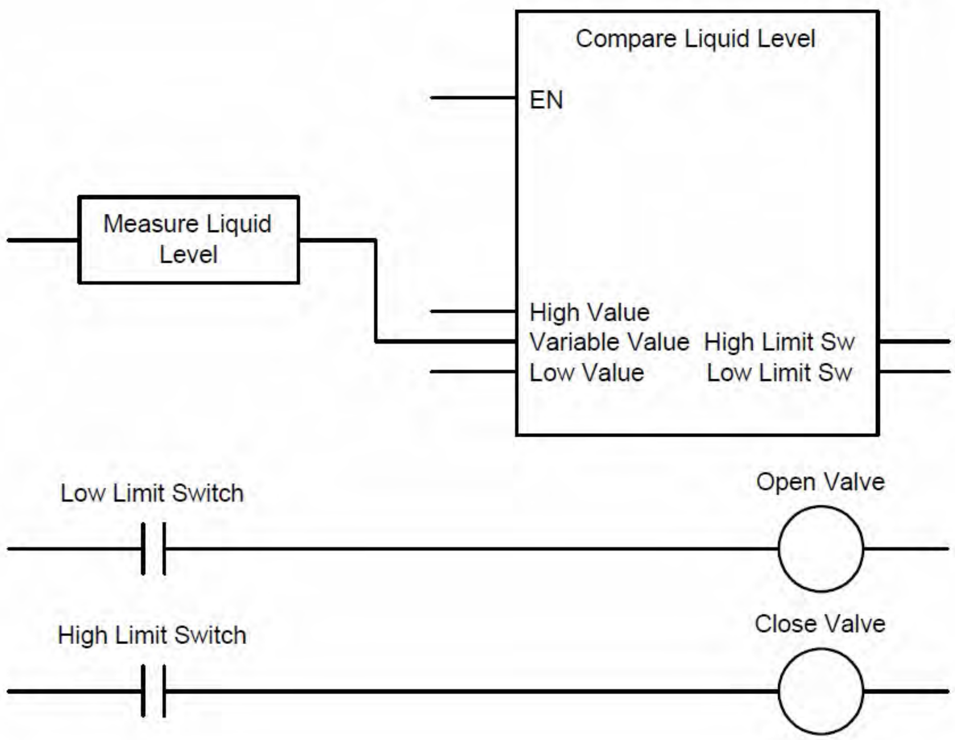
(2) The designer should work with the system integrator during initial configuration of the operating system and network to set up user accounts and passwords. Since operating computers typically are used 24/7, a single operator account should be created so all shifts use the same account and password. Administrator accounts should be set up for system administrators or engineers who administer the computers and network.

i. PLC Programming Software. The PLC processor programming software should be as provided by the PLC manufacturer. It might be a third-party product the PLC manufacturer recommends in its written literature. The programming software should have provisions for configuring I/O rack addresses, simulating program execution for debugging, and downloading to the PLC processor. The programming software should conform to Part 3 of IEC 61131, the standard for PLC programming languages, should operate on a Microsoft Windows 7 platform, and should include editors such as:

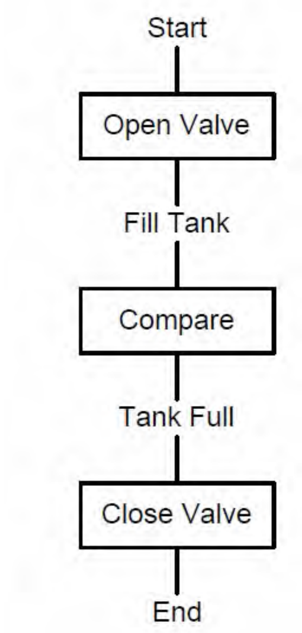
(1) Function Block Diagram. This editor depicts process data flow suited for discrete and continuous control application functions, and should include predefined elementary function blocks as well as user-defined function blocks. Language written in other editors, as listed below, can be nested within the Function Block Diagram (FBD). In the FBD, control sequences are programmed as blocks that are wired together in a manner resembling a control circuit.



(2) Ladder Diagram. This language allows programming in the familiar left-to-right contact and coil arrangement in an order familiar to most electricians and maintenance personnel. The ladder logic editor should allow the use of other editors, such as Function Block and Structured Text, to be incorporated into the ladder programming. This will simplify the programming because electricians and maintenance personnel will not have to be familiar with the complex logic of the other editors. The Function Block or Structured Text editors simply perform predetermined logic within the body of the ladder diagram. These can be grouped in subroutines to simplify the appearance of traditional ladder logic.



(3) Sequential Function Chart. This editor provides a graphical method of organizing a control program using programming from other editors nested within. The SFC editor should include three main components: steps, transitions, and actions. Steps are individual control tasks in which programmed logic operators are used to perform a particular control function. Actions are the individual operators of that task. Transitions are merely mechanisms to move from one task to another. With SFC, the processor continues to perform the actions in a step until the transition conditions is true (i.e., repeat the step containing the action of filling a tank until the transition condition of comparing level against full is true, then move to step of closing the valve).



(4) Structured Text. The Structured Text editor is a high-level language resembling Pascal or Basic and is used to perform control logic programming. Structured Text often is the easiest way for the novice to write and understand control logic because of its inherent resemblance to sentences.

```
LET LowLimit = Low_Limit

LET HighLimit = High_Limit

LET Value = Liquid_Level

WHILE Value < HighLimit

IF Value < LowLimit THEN

DO Open_Valve

END IF

END WHILE

DO Close_Valve
```

(5) Instruction List. Instruction List editor is a text-based Boolean language. The basic Boolean operators can be used to create more complex control applications. Similar to Assembly Language, the Instruction List editor is a low-level language useful for simple control processes; its logic is repeated often. Instruction List allows the logic

for these processes to be programmed once, then recalled in latter instances in the program.

```
Start:  LD Liquid_Level     :Move value of liquid level into argument

GT Low_Limit        :Compare with Low Level Limit

ST Open_Valve       :Move (1 or 0, based on above) into output

GT High_Limit       :Compare with High Level Limit

ST Close_Valve      :Move (1 or 0, based on above) into output

End:
```

It is important to specify the PLC programming software in sufficient detail using the IEC 61131-3 standard because this will insure the contractor provides a quality software package that complies with worldwide industry standards.

j. Human-Machine Interface (HMI) Software. HMI software provides the graphical user interface for operating the lock and dam. This software can run on any of the PCs on the network, or on a dedicated touch-screen panel running its own operating system (typically Windows CE), and can communicate to the PLC through the network. The designer should plan the system to determine which PCs or panels need to communicate directly to the PLC. Most HMI software can operate either as a client node, in which it piggybacks off another computer for access to the PLC, or as a server node, in which it communicates directly with the PLC processor. In general, a small number of HMI servers on the system will enhance the communication speed with the PLC processor and reduce the risk of one server inhibiting operation from another one.

k. User Interfaces. One of the key ingredients to a successful computerized control system is the general perception of the system by the operators. Most operators are not going to care what operating software is used or how much memory it has or about the scan time of the PLC processor. Therefore, an argument can be made that the most important part of the control system is the interface that allows the operator to use the system. For most new lock-and-dam control systems, this will be the operating screens, whether stand-alone, touch-screen panels or monitors with PCs. These must be designed and programmed with considerable care to ensure a completely user-friendly interface that is convenient and, most of all, safe to operate. The HMIs in the main control console should have all the operating screens necessary to control and monitor the entire project.

l. Semi-Automatic Operating Screen. On locks with semi-automatic operating systems, a screen, such as in Figure 12-5, should be included to facilitate complete normal lockage from one operating screen. The screen should include control of both ends of the lock, the traffic lights, warning horn, emergency stop, and other critical features unique to each project. The designer should try to keep the semi-automatic or automatic operating screens (Figures 12-5 and 12-6) complete, yet compact and concise, because too much information or control of auxiliary equipment can make the screen confusing to operate. Animated graphics, based on real-time data from transducers and sensors, makes the screen more friendly to the operators. The screen should be designed to allow a busy operator to look and very quickly ascertain the status of the major lock operating equipment.

(1) Manual Operating Screens. The system should include screens for manually operating each piece of equipment (Figures 12-7 and 12-8). Special, seldom-used, operating procedures such as interlocks bypassed should be included on these screens. The system probably should have a special operating screen for each major piece of lock equipment (i.e., gates and valves). This control will be accomplished through the PLC system with the full complement of safety interlocks and permissives, but it will allow independent, non-automatic operation of the individual pieces of equipment.

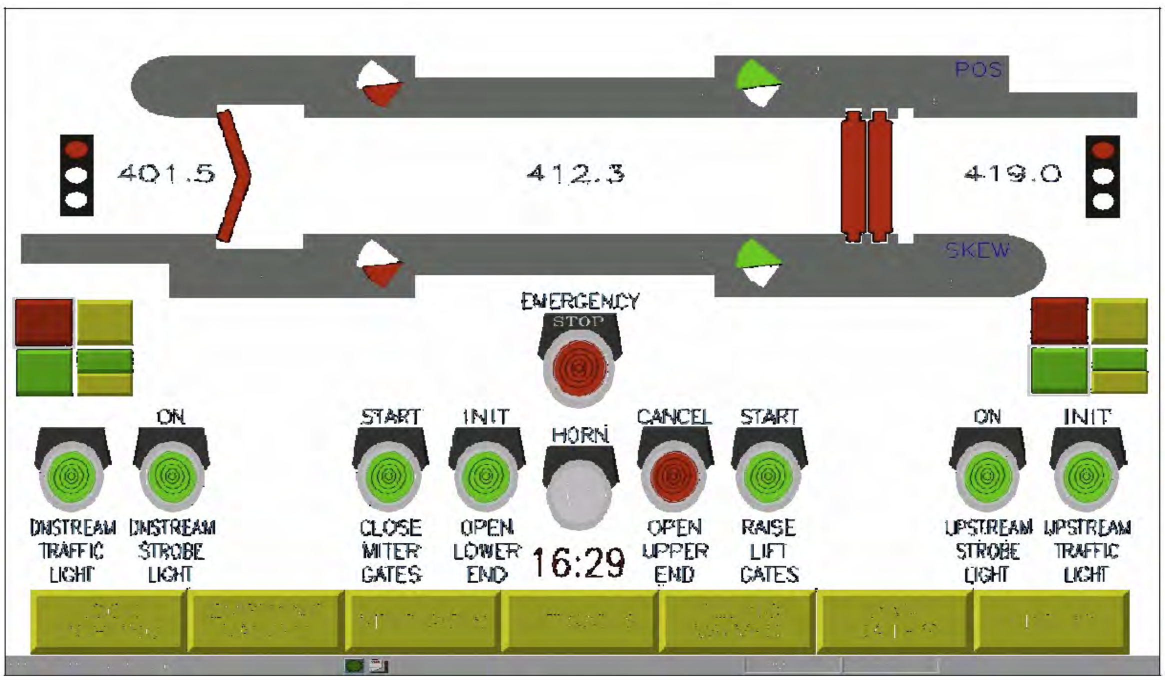


Figure 12-5. Semi-automatic operating screen used at Melvin Price Locks and Dam

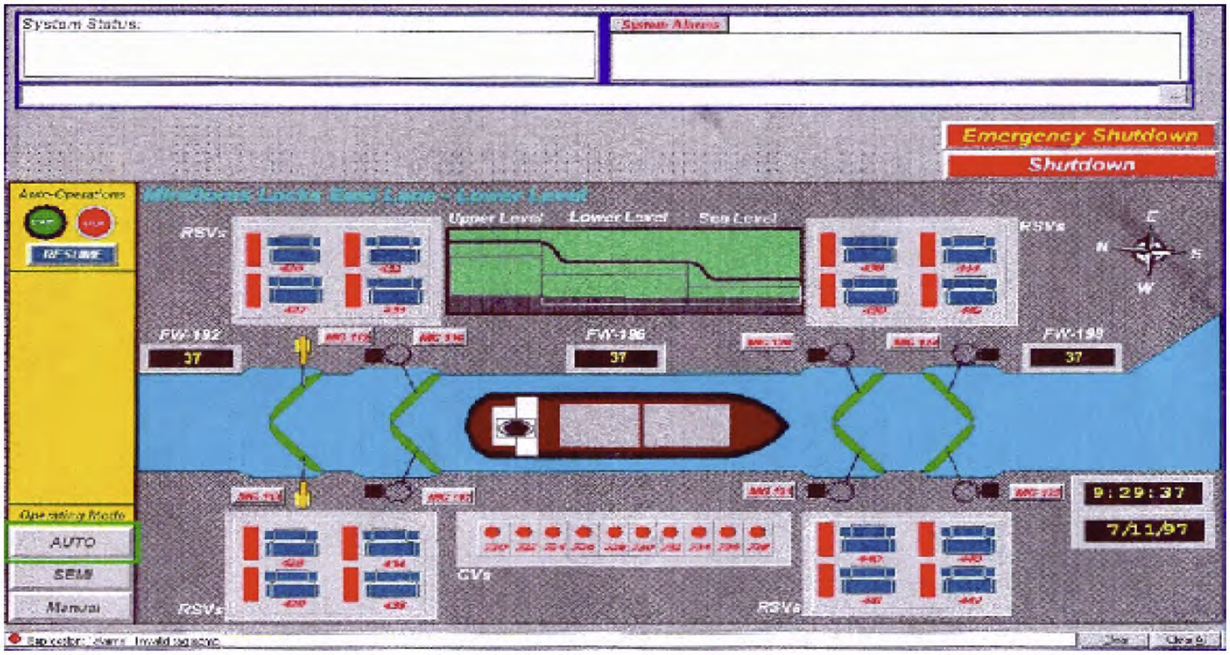


Figure 12-6. Automatic Operating Screen proposed for use on the Miriflores Locks, Panama Canal

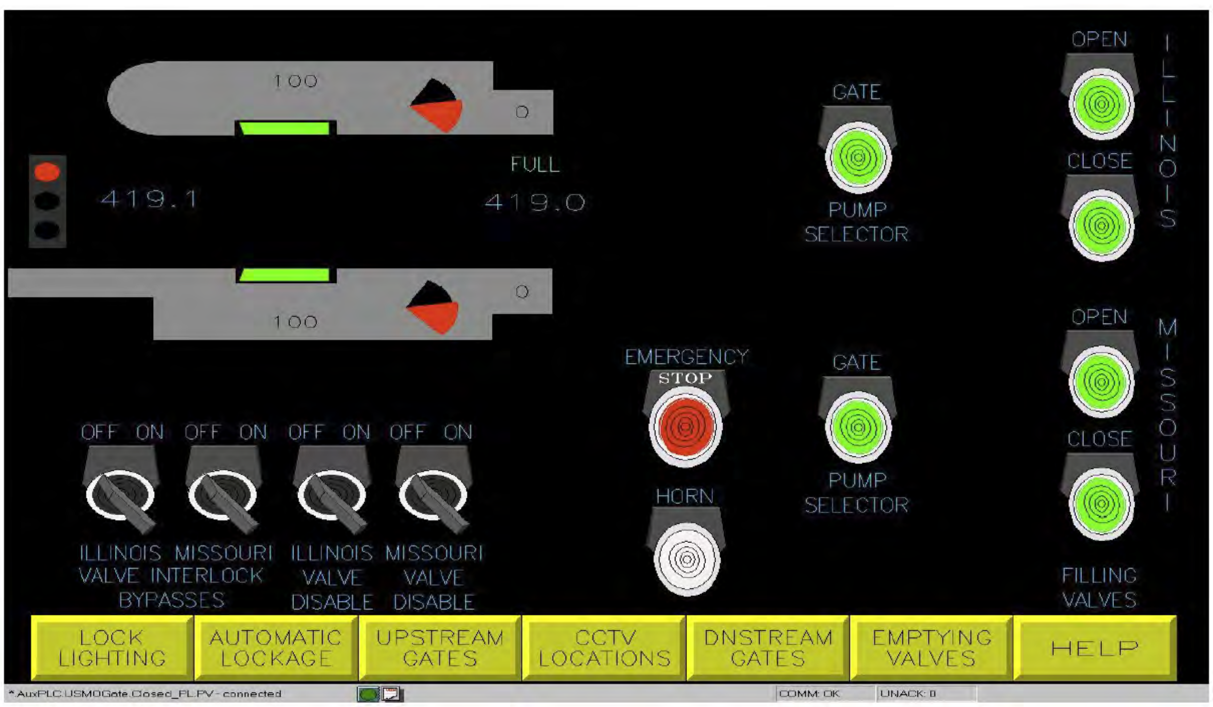


Figure 12-7. Miter gate manual operating screen used at Melvin Price Locks and Dam

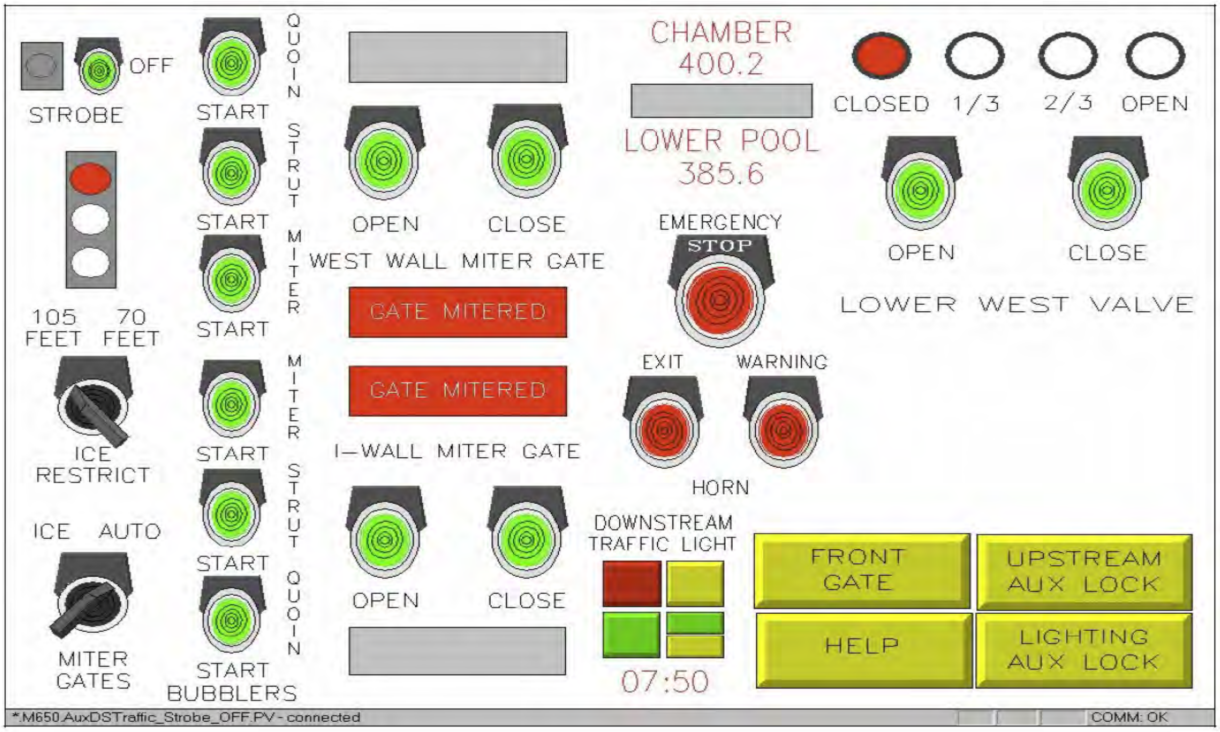


Figure 12-8. Manual operating screen used at Locks No. 27

(2) Troubleshooting Screens. A good operating system interface should include simple troubleshooting screens the shift chief can use to identify problems. These screens alert the shift chief of situations that can stop or affect the operation of the lock but are not necessarily equipment failures. These instances could be emergency stop pushbuttons that have been left depressed; equipment access doors with lockout switches; interlocks that have been bypassed on an auxiliary control panel; or emergency controls that are being used somewhere on the locks. The troubleshooting screen is intended to alert the shift chief of these situations so he can begin investigating and correcting. This should be the first screen the shift chief pulls up when equipment will not operate properly.

(3) Alarm Screens. Alarm screens should be provided to alert operators and maintenance personnel to failure of lock operating equipment. Included on such a screen should be failure of any PLC component that can be diagnosed by reading registers in the processor (most PLC equipment failures can be found this way). These typically would include PLC communication failures, I/O failures, AFD failures, water level sensor failures, and position sensor failures.

12-6. <u>Sensors and Field Devices for PLC Systems</u>.

a. Selection and installation of a quality PLC/PC system is only part of the work necessary to complete a successful lock control system. The I/O devices that provide information to the computerized control system are just as critical in the overall success of a lock-and-dam control system as the PLC itself. Failure to specify quality materials and proper installation and testing of field devices will diminish operator confidence in the whole system, regardless of how good the PLC system might be. Also, consider that these devices often serve a dual purpose when they are used in the emergency backup system. Redundancy and spare parts are an important part of the auxiliary equipment design process. Discussed next will be typical sensors and field devices for a navigation lock, but the list is not complete or all inclusive.

b. Limit Switches. The most common PLC input device is a dry-contact limit switch. Different switches are used for different applications. These devices are particularly useful for end-of-travel or over-travel limit switches because they are absolute, passive, and require no electronic calibration. Limit switches of this type also can be provided with auxiliary contacts for a hardwired backup system. In this day of not using brand-name and “or equal” specifications, carefully specifying all the important features is essential to ensure the contractor provides a quality product. Limit switches are as critical, if not more, than any other feature of the control system, and time should be taken to write sound specifications for procuring, installing, and testing them.

(1) Vein/Roller/Lever-Operated. Simple vein- or roller-operated dry-contact limit switches are used for all types of moving machinery. It is a good idea to specify these with extra contacts for hardwired applications. Addressable limit switches that can be connected directly to the PLC network, eliminating the need for I/O racks, are becoming increasingly popular and are a good consideration for non-critical applications in which there are few limit switches and great distances between them. Their usage on locks and dams has been limited generally because lock operating machinery is grouped together and has enough limit switches in one location to justify an I/O rack. Specifications should include heavy-duty, oil-tight, corrosion ratings, number, rating, and type of contacts, NEMA 4X or 6P rating, UL listing, operator lever type, and operating temperature range.

(2) Magnetic/Proximity/Photo Electric. By eliminating moving parts and providing a degree of submergibility, magnetic and proximity limit switches have replaced the vein-operated switches for use on the end of miter gate leaves for indication of proper miter, and in the miter gate recesses for indication of a fully recessed gate. HQUSACE, to avoid damage to miter gates, has mandated positive indication of the miter and recess positions. These switches must be installed and used in the control system as interlocks to prevent filling of the chamber (improper miter) or changing the traffic light to green (improper recess). The use of these type of switches is a good consideration in other areas where ice can hinder the operation of vein-operated switches. When procuring magnetic or proximity switches, specifications should include number, type and rating of contacts, NEMA 4X or 6P rating, UL listing, temperature range, copper or fiber optic leads, side or top mount, standard and extended operating ranges, and surge protection. The extra time it takes to specify a good switch properly is well spent.

c. Electronic Sensors. Position and level measurement require the installation of an electronic encoder such as those mentioned below. Encoders should be specified without the need for external or third-party converters, decoders, linearizers, or signal conditioners. These devices have much higher failure rates than PLC I/O cards and cannot always be diagnosed for problems through the PLC system. Encoder signals should be fed directly into a PLC analog input card. It is usually not a good idea to use electronic encoders in a hardwired backup system. The need for scaling and offsetting factors, easily accomplished in the PLC software, coupled with the distances that the signal must travel often will cause reliability, noise, and calibration problems when used without a PLC system. These problems could, depending on how the system is wired, interfere with the normal PLC system. Electronic encoders require special installation, wiring, and shielding to provide long-term reliability and accuracy. Large electrical contractors often will not provide this type of detailed quality installation unless specifications are written tight enough to force them.

(1) Pressure Transducers. Pressure transducers are used to measure the level of water in and around the lock and dam. A pressure transducer produces an analog signal directly proportional to the amount of water in which it is submersed. Typically, a designer should put at least two transducers in the upstream pool, two in each lock chamber, and two in the downstream tailwater. Such transducers also can be used to measure the amount of leakage in manholes and crossover tunnels. Placing two transducers in areas of critical applications allows for reliability and accuracy checks of the PLC software. In an automated system, the pressure transducers become one of the most critical control components because each automated sequence depends on determination of equal water levels without visual check from an operator. In this case, it is essential to have good reliability checks and failover programming built into the PLC program. When specifying submersible pressure transmitters, design engineers should take the time to consider transmitter construction (titanium provides excellent corrosion control), unique pressure rating, excitation power supply level, output signal and wiring, accuracy, repeatability, electrical connection (should include molded integral cable of sufficient length for each application), resolution, and installation instructions. It is important to note that merely stating that the transducer shall be installed in accordance with manufacturer’s recommendations might not be sufficient. There are many different applications for such transducers in industry, and many are not in as harsh an environment as a river near a lock and dam. The installation should provide protection from ice, debris, and zebra mussels; should facilitate easy maintenance and replacement; and should provide unobstructed atmospheric reference pressure to the breather tube.

(2) Rotating Encoders. Angular position of rotating machinery, such as cable spools, gears, chain sprockets, and miter gate machinery, is accomplished through the use of rotating angle encoders. Rotating encoders could include absolute encoders or single and/or multi-turn resolvers. It is strongly recommended that absolute position encoders be furnished for control systems instead of the incremental, if at all possible and practical. This will ensure that, after loss of power to the position sensor, the position of the machinery being measured will be known. Typically, these are mounted adjacent to the machinery and attached via shaft couplings. Shaft couplings should be one-piece, flexible stainless steel and sized to match exactly the encoder shaft and adapt to the machinery shaft. Installation details should include provisions for making the machinery shaft extension absolutely true and for aligning the shafts as true as possible through the use of properly installed stainless steel shims. Details to include in

the specifications are NEMA 4 or 6P housing, output signal and wiring, power supply and excitation requirements, number of turns needed to resolve entire travel of machinery, repeatability, accuracy, resolution, lightning and surge protection for primary and secondary windings, shaft size, operating temperature range, and installation instructions. Also, it is important to ensure that the rotation of the shaft extension to which the encoder is coupled is linear with the movement of the machinery. If not, programming must be added to the software to correct this. Keep mind that wire cable wound on a drum is not a linear application because the length of cable unreeled during each rotation varies with the amount left on the drum. Each successive wrap of cable on the drum is shorter than the previous one by the cable diameter multiplied by 2π. Gear tooth counters also would fall in this category.

(3) Inclinometers. Inclinometers are used to track angular tilt of the machinery or structural member on which they are mounted. A good application for such a device at a lock-and-dam project is the position of the tainter gates. Angular rotating encoders do not work as well in this application because the rotation of the cable drum or chain sprocket is not linear with the change in opening between the gate and the sill. Also, because a tainter gate can become frozen during times of heavy ice and because drift can be lodged under the gate, the rotation of the machinery is not necessarily indicative of gate movement. A submersible inclinometer mounted on the tainter gate strut will give an accurate indication of gate movement. By programming simple interlocks in the PLC software, slacking of the hoist cables or chains can be avoided in the event the gate does hang up. As stated, the signal from the inclinometer should be fed directly into the PLC control system as a raw electronic signal. All scaling, trigonometry, and linear and angular offsetting should be done in the PLC software. It is important that this type of device be thoroughly engineered in a design document. Some parameters to determine in a specification are housing construction, NEMA 6P (might require separate purged enclosure), angular operating range, resolution, accuracy, repeatability, vibration sensitivity, axis of measure (to ensure proper mounting), excitation power supply requirements, output signal and wiring, temperature operating range, and detailed installation requirements.

(4) Hydraulic Cylinder Position Transducer. With the increasing popularity of hydraulic cylinders in navigation lock equipment design, several manufacturers offer position-sensing transducers integral to the cylinder construction. The type of transducer and the output signal vary by manufacturer, therefore, a design engineer should consider this and specify the type of position-sensing system that is best for the equipment. The output signal should be directly compatible with the PLC system I/O cards available. As stated above, it is not a good idea to use third-party converters and signal conditioners to massage the signal before it is read by the PLC.

12-7. Control Cabling.

a. General. Wire and cable to be used in locks and dams are specified in various Uniformed Facilities Guide Specifications (UFGS). Control cabling can consist of exclusively copper wiring or a combination of copper and fiber optic cabling. The copper-only control cabling system is usually at control voltages from 24V to 120V, sometimes 240V, and used in discrete relay control systems. In the case of PLC/computer/electronic control systems, control cabling will be a combination of copper wiring (from 5vdc to 120 to 240vac) and fiber optic cabling.

b. When specifying wire and cable for a project, the designer should consider the environment in which the cables will be permanently installed and the design life of the project. For the usual project design life (40 to 50 years), it is desirable to specify cable insulations and jackets that are of the cross-linked, thermosetting polyethylene type. See NEMA WC70 and UFGS 26 05 19.00 10 for details on specifying cables with thermosetting insulations and jackets. This cable has a rugged outer jacket, insulation, and typically has desirable characteristics with regard to resistance to oils and to water absorption in long-term, submergence conditions. When selecting the cables, pay attention to the duration and depth of submersion of the cable. It is noted that 120-vac control cables typically can be procured with thermosetting jackets and insulations. However, control cables such as fiber optic and wire line data transmission systems (i.e., small cables such as Ethernet, PLC communication cables) might not be readily available, if at all, with thermosetting jackets. The reliability and costs should be considered when specifying these wire line data transmission and fiber optic cables to ensure that additional costs that might be required for thermosetting, durable jackets (if available) are justified. Most cables are available with PVC, thermoplastic jackets and are best suited for use above grade in buildings, but might be used on the locks and dams to avoid cost premiums, if acceptable to the designer and end user.

c. The 120-vac control cables can be rated at 75°C, but are usually available to 90°C if desired. Wire line data cables usually are rated to 60°C and have been used in USACE locks and dams. The designer might choose to require a special insulation and jacket, if available from a cable manufacturer, if it is determined that reliability and cost justify the use of the cable.

d. Splices in copper, 120-vac control cables are acceptable but not preferable. Splices in fiber optic or wire data transmission system cables, such as Ethernet, are not preferable and should be avoided if possible. If splices are used in fiber optic cables, losses at splices or any connecting points should be considered when performing optical power budget calculations. If required, use higher power transmitters or repeaters to maintain sufficient optical power for successful communications.

12-8. Interlocks.

a. General. The purpose of interlocks on navigation locks is to prevent damage to machinery or to prevent water flow through the lock chamber. The requirement or authority for interlocks on navigation lock equipment can be found in the Inland Marine Transportation System’s National Interlock and Maintenance Standard. As of 2012, a draft copy of an interlock standard by the IMTS exists and is in the appendix of this document. Once the draft standard is reviewed and approved, it will become a part of the maintenance standard referenced above. Design of a new or rehabilitated lock shall incorporate the requirements of this interlock standard.

b. Inland Marine Transportation System. The IMTS was created to implement improvement ideas from the marine transportation work force. The IMTS provides a mechanism to take a process or best practice from one Corps district or division and implement it across the nation. The IMTS is not a new organization. It is made of virtual USACE personnel from across the nation. IMTS is supported by a Board of Directors and a Working Group. The Board is led by the Deputy Commanding General for Civil and Emergency Operations. Members of the Board are Division Commanders and the HQUSACE Chief of Operations. The Working Group features a cross section of skills from the IMTS. The IMTS also is supported by Action Teams that are created as needed to address specific concerns or maintenance issues. Output from the Action Teams is reviewed by the work force and industry prior to review and approval by the Board.

c. National Maintenance Standard. The goal of the National Maintenance Standard is to provide a world-class maintenance program for the aging navigation infrastructure while accommodating shrinking budgets and resources. The standard includes a national baseline and annexes. The baseline is titled *IMTS Standard, Maintenance of Navigation Locks and Dams*. Each Major Subordinate Command (MSC) will include an annex documenting any supplements or exceptions for the Corps Division's infrastructure. Additional annexes also will supplement the baseline standard. For example, the interlock requirement is an annex titled *Minimum Interlock System Performance Requirements for IMTS Lock Projects*.

d. Minimum Requirements. USACE has experienced several miter gate failures. Most are attributed to the closing of the miter gate while water flow is present in the lock chamber. The interlock requirements in the IMTS standard include gate-to-valve, valve-to-gate, and valve-to-valve. The standard includes logic diagrams to implement the required interlocks.

(1) The gate-to-valve interlock prevents the operation of a lock gate while any valve at the opposite end of the lock chamber is open. For example, the upstream miter gate should not close while an emptying valve is open.

(2) The valve-to-gate interlock prevents the opening of a valve unless the lock gate at the opposite end is in the closed position. A filling valve should not be opened unless the downstream miter gates are closed (mitered). Likewise, an emptying valve should not be opened unless the lift gate at the upstream end of the lock is in the raised position.

(3) The valve-to-valve interlock prevents the filling and emptying valves from simultaneously opening, allowing water flow in the lock chamber. For example, all filling valves must be closed before opening an emptying valve.

e. Additional Interlocks. The lock control system designer should consider the requirement for additional interlocks beyond the minimum in the interlock standard.

(1) Water Levels. A miter gate should not be operated while a head exists on the gate. The lock control system can be designed to compare the water level on each side of the gate and permit opening of the gate when they are equal. Similarly, a lift gate should not be lowered unless the pool and chamber water levels are equal. These interlocks would be difficult to implement in a relay-based control system, but can be done easily in the PLC-based system.

(2) Ice Flushing. A key advantage of a lift gate installed on the upstream end of the lift gate is the ability to flush ice downstream. This is accomplished by lowering the lift gate below the pool water level and allowing the ice to flow over the top of the gate. Obviously, this condition will create water flow in the lock chamber. The downstream lock gates must be in a suitable position before the ice flushing operation. Ideally, the downstream gates will be mitered, but ice flushing operations could be performed when the gates are recessed fully. In periods of severe ice conditions, the buildup of ice behind miter gates might prevent them from recessing completely. USACE has experienced a miter gate failure during ice flushing operations when the gates were not recessed fully. Ice flushing requirements and their interlocks should be coordinated between the designers and operations personnel.

f. Bypassing Interlocks. Operation of a navigation lock likely will require the operator to bypass the interlocks for certain conditions. For example, heavy rains might send drift downriver to collect upstream of the miter gates. Bubbler systems can move drift to permit opening of the lock gates, but the drift must be locked downstream. Bypassing the valve-to-valve and valve-to-gate interlocks will permit the opening of the emptying valves, create water flow, and move the drift into the chamber. Once the lock chamber has emptied and the lock gates are recessed, the filling valves are opened to create flow and move the drift out of the lock. Designers must plan accordingly when overriding the interlocks. The system should be designed to minimize the possibility of the lock operator leaving the interlock bypass enabled. The bypass system should include key switches, indicating lights, etc. When using a key switch, keys should be removable only when the interlocks are not bypassed. PLC-based systems can employ logic to automatically turn off interlock bypasses after a time delay. The lock must develop a Standard Operating Procedure (SOP) for the interlock bypass system, so designers should coordinate its requirements with operations personnel.

g. Field Devices. Input devices such as limit switches are necessary to implement the interlocks. Limit switches are used to indicate valve closed, gate mitered, gate recessed, etc. The inputs must be reliable to ensure proper interlocks. The reliability of the input can be increased by installing instruments on the gate or valve, not its operating machinery. The designer should consider the environmental conditions in which the switch is installed and provide redundancy.

(1) USACE has experienced a miter gate failure when the machinery (rack and sector gear) indicated the gate was closed, but was open due to a failed strut arm. Limit switches should be located at the miter end of the gate. A magnetically-operated switch will ensure a gate has mitered correctly. Limit switches should be mounted in the gate recess. Rocker switches will be susceptible to damage from debris and high water.

Proximity switches can trip from a flag mounted on the gate, and there is little chance of experiencing a false-positive condition.

(2) It is difficult to instrument the valve body on culvert valves. Doing so would require the limit switch to be submerged where it would be susceptible to damage from ice and other debris. Replacement of the limit switch would be difficult and costly. The culvert valve can be instrumented at the bell crank. The pivot point of the bell crank might not always be above the water level, but is fairly accessible. Hoist-cable operated valves can be instrumented with a rotary limit switch on the hoist drum. A slack-cable limit switch can prevent the machinery from reaching the valve-closed position, in the event the valve body cannot close fully due to a piece of debris.

(3) Permanently submerged lift gates cannot be instrumented directly at the gate. While the lowered position might be constant, the raised position is based on the pool water level. The gate-raised and gate-lowered positions are determined by a rotary encoder on a sheave or drive gear and hardwired, rotating cam or travelling nut limit switches.

(4) Limit switches used for interlocks should be wired normally-open to provide a fail-safe condition. This will require the application of electrical voltage to the limit switch and, to complete the interlock, it be in the tripped (closed) position. When limit switches are used for end-of-travel indication, they usually are wired normally-closed. If power is lost to the limit switch or a wire is broken, the machinery will not move because the end-of-travel condition is made. A failed condition should not permit completion of the interlock.

12-9. Emergency Stop.

a. General. It is likely that many of the current lock control systems are equipped with an emergency stop pushbutton. This pushbutton may be an input to a PLC system or part of a hardwired system designed to stop the operation of any machinery on the navigation lock. The term “emergency stop” has specific requirements in terms of NFPA 79, *Electrical Standard for Industrial Machinery*.

b. Supply Circuit Disconnecting Means. Chapter 5 of NFPA 79 requires the electrical supply for the machine to terminate at a disconnecting means. This may be a molded-case circuit breaker or switch, motor circuit switch, or even a plug and receptacle for a cord connection.

c. Machinery Stop Functions.

(1) Chapter 9 of NFPA 79 defines three categories for a machinery stop function:

(a) Category 0 is an uncontrolled stop by immediately removing power to the machine actuators.

(b) Category 1 is a controlled stop with power to the machine actuators available to achieve the stop, then remove power when the stop is achieved.

(c) Category 2 is a controlled stop with power left available to the machine actuators.

(d) Each machine must be equipped with a Category 0 stop. Most navigation lock machinery operates with the Category 0 stop by de-energizing a reversing-contactor to a motor and setting a holding brake.

(e) Emergency Stop Operation. Emergency stop must meet the following requirements from Chapter 9 for NFPA 79:

- It shall override all other functions and operations in all modes.
- Power to the machine actuators, which causes a hazardous condition, shall be removed as quickly as possible without creating other hazards.
- The reset of the emergency stop command shall not restart the machinery, but only permit the restarting.
- The emergency stop shall function as a Category 0 or Category 1 stop. A risk assessment for the machinery will determine which category is required. NFPA requires, “Final removal of power to the machine actuators shall be ensured and shall be by means of electromechanical components.” While this a reference to a relay, the standard does permit a solid-state device (drive) to be the final switching device when it meets the “relevant safety standard.”

d. Machinery Risk Assessment. The requirement to perform a risk assessment of the machinery is referenced in many sections of NFPA 79. The risk assessment must determine the circuit reliability required for the emergency stop function, but the standard does not define the reliability classes. The risk assessment also may dictate the monitoring of safety circuits and the use of safety relays and circuits.

e. Emergency Stop System Design. The scope of NFPA 79 does not include the design requirements for the emergency stop system and its components. Annex A of NFPA 79 refers to ISO 13850, *Safety of Machinery - Emergency Stop - Principles for Design*. When related to solid-state devices used in safety functions, the annex refers to IEC 61508, *Functional safety of electrical/electronic/programmable electronic safety-related systems*, and IEC 61800-5-2, *Adjustable speed electrical power drive systems – Part 5-2: Safety requirements.*

f. Emergency Stop Devices. While the standard does not address the design of the emergency stop function system, it does detail the requirements for emergency stop devices. The devices must be readily accessible and continuously operable. A device should be located at each operator control system and other locations that require emergency stop. Devices might be required on the lock wall near the machinery for use by operations and maintenance personnel. The device may be, but not is limited to, a pushbutton-operated switch, pull cord-operated switch, foot-operated switch (no guard), push bar-operated switch, or a rod-operated switch.

(1) Pushbutton-type devices must be red with a yellow background. The device must be self-latching with direct operation. The actuator of the pushbutton shall be of the palm or mushroom-head type and shall affect the emergency stop when depressed.

(2) Devices cannot be flat switches or “graphic representations based on software applications.” This means an object on an HMI cannot serve as an emergency stop device.

g. Other Considerations. Consideration should be given to illuminating the pushbuttons to indicate when the emergency stop is active. Illuminated pushbuttons also can be programmed to flash to remind the operator that equipment is operating. This might be important if one is operating from a centralized control room that is remote from the equipment. It also serves as a reminder, should the HMIs fail.

12-10. Procurement Considerations.

a. Computer Hardware. One of the most difficult tasks of putting together a computerized lock control system, whether automated or not, is the acquisition of quality, state-of-the-art computer hardware. Often, large construction contracts have too long a construction period to write specifications around state-of-the-art computer equipment. Even PLC components might be superseded in design during major construction of a new lock and dam. Government-furnished equipment (GFE) is not always a good idea because of the issue of responsibility during installation. In theory, GFE is a good idea; but, in practice, many agencies have had trouble administering contracts with a large amount of GFE. Optional bid items and change orders are usually difficult to negotiate and costly when issued near the end of a large contract. Some guidelines for procurement of such equipment:

(1) Personal Computers. Computer hardware will usually only stay state-of-the-art for about a year and somewhat current for up to three years. Therefore, it is not recommended to specify PC-type hardware in a contract of duration more than nine months. Even at that, with reproduction, advertisement, contract award, notice to proceed, and shop-drawing phases, the equipment model design likely will be well over a year old when the government assumes ownership of the control system. Contracts of duration longer than nine months should be looked at with the possibility of doing a small follow-up contract to install, configure, and program the computer network. When upgrading the PC hardware and software, use purchase orders with hired labor for installation or write small contracts with a system integrator. Trying to incorporate computer upgrades into another contract usually is not a good idea because of the different trades involved and the contract duration time. An important part of the successful procurement of a quality system is complete engineered plans and specifications that do not allow a contractor much room to substitute cheaper components or installation methods.

(2) Programmable Logic Controllers. Having better stability than PC components, PLC equipment generally will stay current on the market for five to ten years. Once installed, a PLC system should be expected to require a complete upgrade

every ten to fifteen years. However, it is critical at the time of project start-up to have a state-of-the-art PLC system. Therefore, if the construction contract exceeds one year in length, it might be worthwhile to leave the PLC system out of the contract and write a follow-up one to install and program the PLC system. In the big construction contract, items such as raceways, cable trays, field devices, and even some field wiring can be put in place to minimize the effort of the PLC system installer. The first contract must be managed well to ensure everything is in place. Too often, we rely on operational tests to tell us the contractor has completed all his work and, in this case, the first contractor might be long gone when the operational tests are performed. It is important to develop some in-house expertise on PLC systems to assist in administering both contracts and providing startup, assistance, and long-term maintenance to the lock. This will prove to be the key, whether the control system is successful or not. Down the road, the problems with administering the contract mostly will be forgotten, but the reliability of the PLC system always will be an issue.

b. Computer Software. Not unlike computer hardware, software does not stay current for more than three years in most cases, and often is superseded by new versions within a year. It may be written in the specifications that the contractor must provide the latest release of a particular software; but, at some point, the contractor must purchase the software and that needs to be written in the specifications so that there is no dispute as to what revision is the latest at the time of purchase.

(1) Operating System. The operating system software should be purchased at the same time as and by the same contractor as the computer system hardware. This will alleviate compatibility problems between the operating system software and other software or hardware.

(2) Human-Machine Interface Software. The latest version of HMI software and the PLC system should be purchased at the same time. With most HMI packages, upgrades should take place probably on a three-year basis. Problems arise when operating systems, such as Windows XP or Windows 7, are revised to the degree that a plant's current version of the HMI software will not run on the new operating system software. These are times when a design engineer or software maintenance personnel must be careful when upgrading software. Databases usually can be transferred when the HMI software is upgraded, but not always. Before writing plans and specifications or a purchase order, it is recommended that the designer gain a thorough understanding of the marriage of the operating system software and the HMI, as well as a thorough knowledge of what is on the market. All projects should factor in monies to upgrade the HMI software every three years.

c. Sensors. Non-electronic sensors such as dry-contact, vein-operated limit switches, travelling nut limit switches, and magnetic limit switches may be purchased and installed in a large construction contract. However, if the construction contract exceeds one year, consider purchasing encoders, pressure transducers, inclinometers, hydraulic cylinder position-tracking devices, and other solid-state sensors in a separate, follow-up contract perhaps with the PLC/PC system hardware. Again, raceway and wiring can be put in place for these items; but, to ensure that the latest versions are acquired, it might be best to purchase the sensors later.

d. Training.

(1) Training should be provided for in all contracts and purchase orders. Generally, a contractor will not raise his bid price too much to cover training; so, consider putting a generous amount of training in the specifications package. Furthermore, it is a good idea to provide for enough training, stressing quality and qualifications of instructor, to force the contractor to put some extra money in his bid. This will ensure better training when the time comes. If the system is easy to operate and maintain, delete some training or reserve it for cross training personnel from other locks or projects. Ideally, training should be provided well in advance of equipment installation. It is a good idea to videotape all training sessions, to be used later as a reference and training tool. Training requirements are covered below. The design team should select the applicable portions of the training below for the control system furnished: hardwired relay or PLC type.

(2) PLC/PC System Training. The contractor’s system integrator should provide the PLC/PC system training. Schedule enough sessions for enough personnel to more than ensure that everyone gets sufficient instruction. Three things to include in the contract are number and length of training sessions, qualifications of instructors, and material to be covered and training aids to furnish.

(a) Hardware. The PLC hardware training should cover everything from simple I/O card installation and removal to termination of fiber optic and copper communication cables. Discussion should cover hardware diagnostics, interpretation of system LED indicators, automatic failover of communication channels, power supply connections, fuses, line conditioners, lightning protection, and all other hardware in the PLC I/O racks. The instructor should provide hardware similar to that used on the project, complete with power supplies, I/O cards, communication cards, and all equipment necessary to allow trainees to assemble a small PLC system ready to program, as stated below. The PC system hardware training should cover all connections to the chassis, including network, printer, mouse, keyboard, and power. The discussions should cover the installation and removal of CPU, video, and network cards. The contractor should discuss all connections to equipment such as routers, modems, printers, hubs, and backup tape drives.

(b) Software. Software training should be included with the hardware training and accomplished on the same hardware during the same training sessions. This will help bridge the gap between the hardware and software.

(c) Operating System Software. The software training should start with a thorough review of how to navigate the system operating software. Particular attention should be paid to how the system operating software interacts with the PLC programming software and the HMI software. System log-in and boot-up procedures should be shown. Passwords and restricted access should be discussed and explained.

Location of directories and file storage folders for the PLC programming software and the HMI should be addressed in the training. Things such as operating-screen file transfers and file backup procedures should be discussed. Any special custom icons used to short-cut loading of the PLC or HMI software should be reviewed. The contractor should show how to reload and configure the system operating software in the event of failure. This is a good time for government personnel to point out some features they would like to see changed, such as passwords, initial loading screens, profiles, etc. At this point, a system integrator usually will make these changes at no additional cost because it does not impact the schedule. Sometimes these changes can be made during the training session, enhancing the quality of the training.

(d) PLC Programming Software. The contractor should demonstrate how to install and configure the PLC programming software package. The training should cover complete I/O rack and slot addressing, as well as communication software installation and configuration. PLC ladder logic should be developed for the mock PLC system developed, as stated above. All common programming features such as coils, contacts, timers, counters, shift registers, LET, IF, and compare statements, GOTOs, and other common ladder logic notations should be covered. Documentation of the program also should be covered. An application for the training PLC system should be developed, loaded to the processor, and shown to be working.

(e) HMI Software. The same procedure should be followed for the HMI software training. After demonstrating how to navigate the package, an application for the training PC/PLC system should be developed and shown to function properly.

1. Sensors Training. The contractor should provide training on all field devices including limit switches, pressure transducers, encoders, inclinometers and other I/O equipment. The training should cover the installation and replacement of these devices, as well as their respective interfaces to the PLC system if a PLC control system is used.

2. CCTV System Training. It is important to mandate in a construction contract sufficient CCTV system training. As with the PC/PLC training, the specifications should include three things: the qualifications of the instructor(s), the amount of training, and the topics to cover. The instructor should be an integrator who has commissioned CCTV systems at other projects with the same components. The instructor need not be a representative of the hardware manufacturer and, in fact, the quality of training probably will improve if the instructor is an integrator who actually makes the equipment function, rather than a manufacturer’s representative. Specify enough training in the contract to force the CCTV supplier to build sufficient monies into his bid. This is because large manufacturers often will tell electrical contractors during the bid process that they will provide the training at no cost if they use their equipment. In this case, a salesman will come to the site and read from an operating manual. This is not quality training. If a supplier has to build some cost into his bid, he will be more flexible in providing the training and, after award of the contract, quality rather than quantity can be stressed. Topics should cover everything from installation and configuration of all hardware and software to long-term access and maintenance of the equipment. Constructing a mock

system as part of the training is always a good way for maintenance personnel to learn the new equipment.

3. CCTV System. CCTV equipment does not evolve as fast as computer hardware and software. However, if your contract duration is more than two years, it is best to furnish and install the equipment with a follow-up contract to ensure procurement of the latest technology. In all cases, it is probably best to have a CCTV system supplier and integrator do the work. They should be a firm regularly engaged in providing, installing, and performing startup and long-term maintenance of CCTV systems. As with the PC/PLC equipment, thoroughly engineered plans and specifications are the key to the procurement of a successful CCTV system.

e. Warranties and Service Contracts. Troubleshooting, repair, and long-term maintenance of computerized control equipment is a critical part of a successful lock control system implementation. It is recommended that a plan for such action be in place before procurement of such equipment. Some things to consider:

(1) Warranties. Most construction contracts include a one-year warranty on labor and materials. It is a good idea to have the contractor make a written transfer to the government of any manufacturer warranty on equipment covered by this manual. Some of these warranties might exceed one year. Procure additional years of extended warranty with caution because there is a point of diminishing returns. Namely, some equipment will become outdated before it fails. Also, it has been shown that a majority of equipment failures occur within the first year or so of operation. Once the wrinkles are worked out of a system, many of the components can be expected to last a reasonable amount of time relative to the time it takes for their obsolescence to drive replacement.

(2) Service Contracts. The requirements for service contracts should be commensurate with the type of control system furnished. The more electronics furnished on a project, the more consideration should be given to service contracts. For hardwired relay control systems with a small-to-moderate number of electronics, a service contract usually is not needed. For systems with PLC and PC control, it might be desirable for the construction contract to require a service contract or service agreement with the contractor’s system integrator. This is usually more preferable than searching for a low-bid third party to service the system. It is recommended that this agreement be limited to one year with the government’s option to renew for a second. By doing this, the Corps district can evaluate the contractor’s performance and become more familiar with the control system. Specify in the service contract that it shall be the responsibility of the contractor to furnish spare parts, as he feels necessary, above and beyond that specified in the original contract. Also, specify response and repair times as well as a list of government points of contact (POCs) to avoid confusion when problems arise. Require that the contractor make a full report of changes and procedures that were made to correct the problem, both temporary and permanent. This report should specify all parts that were replaced.

(3) In-house Maintenance. The best way to maintain systems such as these is through the use of well-trained, in-house electricians and electronic technicians

supplemented with help from Corps district engineers when required. When personnel and budget restrictions allow, a district should have at its disposal a crew of such personnel that can perform routine maintenance as well as emergency repairs at all district locks. Busy locks might be able to justify a highly trained electrician/electronics mechanic dedicated to that lock. This is the best way to maintain the system. Keep in mind that, in addition to the high skill level necessary to service computerized control systems, this person must have a thorough knowledge of fundamental electrical principles. This is important because the computerized control system is only a convenient front end to make operations and maintenance safer, more reliable, and more flexible. Beneath all this is the basic concept of electrical power transmission and conversion into mechanical work via a motor or solenoid. For this reason, it is better to put a sound electrician through the proper training and upgrade him to an electronics technician than to take a computer programmer and try to teach him fundamental electrical principles.

f. Testing and Startup. All construction projects should have well-defined testing and startup procedures in the contract requirements. All testing should be witnessed by government representatives from engineering, construction, and operations. This will ensure that the test procedures are correct, relevant, and that all hardware and software (in the case of a PLC control system) pass the test. Each system should be tested individually, then as a total lock control system. The design team should select and modify, as appropriate, the applicable portions of the testing and startup requirements below for the control system furnished, whether it be hardwired relay or PLC type.

(1) PLC System. Require the contractor, before installation, to assemble a small system using the proposed PLC components to demonstrate how the system will work. This also will be a good training tool for district personnel. This should be done off site, at the system integrator's shop, to enhance the training aspects of the testing. Mixing training with testing at this point can be a good thing, as long as there are clear requirements for each, as stated herein and above, provided for in the contract. Once the system is in place at the lock, the contractor should be required to test each I/O point on the system, making a chart to verify each has been tested. This test should be done independent of field devices using dry-contact switches and pilot lights for digital I/O, and function generators and meters for analog I/O. This will test the entire PLC communication network. The test should be performed using each of the redundant PLC communication channels. The contractor should be responsible for correcting any and all deficiencies shown by the test. Field devices such as limit switches, encoders, and transducers now can be tested using the PLC system, with confidence that the communication and I/O rack wiring are properly in place.

(2) PC System. The system integrator should be required, at the same time as the PLC factory test, to set up a mini version of the PC network. The network should include the PLC processor and all appropriate software to allow programming, troubleshooting, and execution of a ladder program and an HMI application. During this test, in the presence of government personnel, at least one PC should be brought online, from the ground up, including the following steps:

(a) Connect all peripherals such as monitor, mouse, modem, etc.

(b) Install the network cards.

(c) Install and configure Windows operating software.

(d) Verify communications with all other network devices.

(e) Install and configure all PLC programming software.

(f) Install and configure all HMI programming software.

(g) After this has been done, the system integrator should demonstrate that numerous large files and directories can, with speed and accuracy, be copied over the network. At the same time as the PLC factory test, a mock ladder program and HMI application should be developed to ensure compatibility of all hardware and software. Again, this test procedure serves as a good training exercise for government personnel.

(h) After the system is in place at the locks, all network communications should be verified again to check all field wiring. Knowing from the factory test that all network software parameters are configured correctly will aid in determining startup problems at the site. While testing the PLC I/O points, a test HMI application can be used. This will make the I/O testing easier, as well as demonstrate further that the PC network is properly in place.

(3) Field Devices. Immediately after installation, the contractor should check all field devices using meters to determine that they have freedom for necessary motion, put out the correct signal when input power is applied, operate correctly with the movement of machinery, and are protected from operations, debris, and weather. After all I/O points have been checked, field devices should be wired to the I/O racks and tested using the PLC/HMI network to verify that all field device outputs are compatible with the PLC system.

(4) Total System. After the above tests have been completed successfully and the lock machinery is ready to be moved, the entire system should be checked for proper operation. All systems should be checked via:

- Water Level Sensing System. Because it is used as a safety interlock in PLC control systems, the water level sensing system should be tested before operation of any lock machinery. Each water level sensor first should be checked for a proper and accurate level, as displayed on the HMI operating screen. Because the transducer has been tested, this test should concentrate more on the decoding in the ladder logic. Require the contractor to verify that the level displayed corresponds to water level changes. In the case of a submersible pressure transducer, raise each unit exactly 1 ft and verify that the level displayed responds accordingly. Repeat this step for several feet and back down again. While not the most accurate test, this will give a good indication that the unit and programming are responding correctly. Fine tuning the calibration can be done when the lock chamber is filled and emptied and large changes in water level can be tested.

- Lock Gates. Motor rotation always should be checked before movement of any equipment. When all limit switches and safety interlocks have been tested and verified for proper operation, the contractor should begin moving each lock gate one at a time in slow speed. Immediately check to ensure that the position displayed on the HMI screen is responding correctly. The contractor should stop and restart the gate from several different positions to ensure the machinery starts in slow speed and changes speeds correctly. All limit switches, including overtravels, should be checked for proper operation and indication. Each gate should be run through a minimum of 10 cycles for testing purposes. Miter gates should be run together after it is shown that each leaf operates correctly. The position indication should be monitored continuously for glitches or spikes. End-of-travel limits should be checked for proper mitering and recessing of the gates. Bubbler system compressors and solenoids should be checked for proper operation and indication.

- Culvert Valves. The same basic procedure for the miter gates should be followed for startup and testing of the culvert valves.

- Dam Gates. After rotation has been checked, the dam gates should be operated individually through as much of their full range of travel as pool conditions will allow. If possible, placing stop logs will give the contractor the chance to test the full range of travel of the gates. All limit switches and overtravels should be checked for proper operation. Again, the position readout of the gate on the HMI should be monitored for proper response to the movement of the gate. The contractor should stop the gate periodically and verify the position against a known benchmark such as a staff gauge.

- Lock Lighting. All PLC-run lighting systems should be checked for proper operation and indication. Feedback indication on the HMI screens should be from auxiliary contacts on the lighting contactors. Integrators often will use PLC output status as an indication of light operation, but this is not reliable feedback. Traffic lights also should be tested for proper operation.

- Alarms. All alarms such as transducer failure, fire and smoke detectors, motor overload, machinery overtravel, communications fault, power failure, etc., should be simulated and checked for proper indication on the HMI screens.

- Remote monitoring and troubleshooting capabilities of the system should be tested by the contractor for proper operation and security.

- Miscellaneous Control Features. Miscellaneous control features unique to each lock should be checked for proper operations. When all these systems have been verified to function properly, the contractor should run the lock through at least five complete lockage cycles, using all the equipment mentioned above. Pay particular attention to the position indication of the gates and valves and to the level readouts for the pool, tailwater, and chamber water level sensors. Make sure that the pools equal interlock is functioning correctly at each end of the lock. It is usually better to specify requirements for extra testing rather than not enough. A contractor usually will not add too much to his bid to cover testing, but will try to reduce the amount of testing when the project comes to an end. This test procedure might be the last chance for the contractor correct some deficiency, so it is important to be clear about the extent of testing the government requires.

g. Documentation. The contractor should be required to provide complete system documentation for all hardware and software used on the system.

(1) PC Network. The documentation should show all network and communication parameters and give detailed drawings showing the complete Ethernet network including all PCs, modems, routers, fiber optic equipment, communication cables, hubs, transceivers, network cards, video and sound cards, data storage devices, uninterruptible power supplies, etc. Manufacturer names and model numbers should be listed for all devices.

(2) PLC Network. The contractor's system integrator should provide drawings showing the complete PLC network with all I/O racks and cards, fiber optic converters and power supplies, lighting panels, AFDs, network communication equipment, power supplies, uninterruptible power supplies, and other equipment. The drawings should show all manufacturer names and model numbers, how all devices are interconnected, and all PLC network addresses including addresses for each individual I/O rack slot. A list should be provided showing the location, address, type, designation, tag, and purpose of every I/O point in the system.

(3) Ladder Logic and HMI Applications. The system documentation should include complete, up-to-date listings of the entire ladder logic program with all I/O points, cross referencing, and labels listed. The HMI software documentation should include a cross reference of every I/O point monitored, a list of all tags, showing type and designation, and a printed copy of every operating screen. All software configurations necessary to establish proper communications with the PLC processor should be in the documentation.

(4) Field Devices. The contractor should include a complete listing of every field device, including transducers, encoders, limit switches, photocells, and motion detectors. The documentation should include manufacturer names and model numbers, voltages, input and/or output parameters (if selectable), dip switch settings, wiring, power supplies, and all information relative to the job.

(5) Input/Output Rack Wiring. The contractor should include as-built documentation of all equipment and wiring in each I/O rack enclosure. This information

should be detailed enough to show point-to-point wiring, with terminal board designations for all connections.

CHAPTER 13

Electrical Support Systems

13-1. Support Systems. This chapter addresses the electrical ancillary features of a navigation lock and dam. These include, but are not limited to, raceway systems, lighting, traffic signals, closed circuit television (CCTV) systems, security systems, communication systems, and life safety systems. Detailed consideration of these systems in the early stages of the design can add a great deal to the successful operation of the facility. It is recommended that the design engineer consult operations personnel regarding this ancillary equipment, especially locations of CCTV cameras and monitors, communications systems, and the life safety systems. It also is recommended that the design engineer work with the District Security Officer when planning the security system.

13-2. Raceway. Types of Raceway Systems. Design of appropriate and sufficient raceway systems for lock and dam gate operating and control systems can facilitate control system installation and maintenance. Raceway materials include galvanized steel, PVC, liquid tight metal and non-metal conduits, cable tray, and other special raceway configurations that might be required by the equipment. Generally, it is better to provide a large number of relatively smaller conduits than a few large ones. Fewer, smaller conduits will facilitate replacement of cables/wiring, should it be needed during the life of the project. Raceway size primarily is based on the volume of the cable(s) installed. However, in the case of conduit, it might be desirable to size the conduit to allow the installer to pull through connectors that can be attached to the equipment, in lieu of requiring tedious, final connections in the field. The raceway system should be designed with utilization voltage levels in mind. For instance, all cables in a raceway must be insulated to the highest voltage level in the raceway (unless barriers, such as in cable trays, are used). In addition, it is desirable to separate low-voltage control wiring from instrumentation type cables (4-20mA dc, RS232, RS485, RS422, etc.) in an effort to minimize noise on the communication lines. Raceway minimum size requirements can be found in NFPA 70, National Electrical Code.

13-3. Lock and Dam Lighting.

a. General. Many types of lighting are provided for a navigation lock: building interior lighting, security lighting for parking lots and esplanades, roadway entrance lighting, and lock-and-dam lighting. Because the Illumination Engineering Society (IES) addresses building lighting, only lock-and-dam lighting is addressed in this manual.

b. Lock Lighting. Lock lighting can be accomplished with either high-mast poles (50 to 100 ft) or standard poles (usually 30 ft). Normally, the length and width of the lock chamber(s) will determine the best choice of fixture height. However, other factors such as boat height, maintainability, spill lighting, and architectural/aesthetic preferences should be considered when selecting fixture heights. It is recommended that the customer be consulted at the beginning of the design phase. Additional lighting requirements on a lock include mooring bitt recesses, and navigation guard and traffic control lights.

(1) High-mast lighting usually is employed where a large area of lighting is required, while using as few standards (poles) as practical. High-mast lighting also is used to provide a relatively uniform lighting intensity. High-mast lighting usually is provided with one of two types of HID lighting fixtures: high-pressure sodium or metal halide. High-pressure sodium fixtures can provide more lumens per watt but, because of the poorer color rendition (compared to metal halide), seem to provide no more, or even a lower, lighting intensity. Computer programs that make the lighting design fast and economical are available. Often, the light fixture manufacturer is willing to provide a lighting layout based on input from the customer. High-mast lighting has been used to illuminate many lock areas: walls, upper pool, lower pool, near upstream and near downstream pool. In addition, high-mast lighting has been used for parking areas and other large places within the lock project. High-mast lighting poles are usually sectional, from 50 to 100 ft tall, galvanized, and provided with a luminaire ring that can be lowered to within 5 ft of the base of the pole, for easy maintenance of the luminaires. The luminaire ring is lowered with a portable, heavy-duty drill that connects to an input shaft of a gearbox that is located inside the pole. The design of the pole, anchorage, and foundation must account for the equipment mounted on the pole and the wind conditions that will be experienced in geographical area of the project. If high-mast poles are to be used in the project lighting, it is highly recommended that the engineer coordinate with the lock wall designers to ensure the walls allow the poles to be mounted as close as possible to the design location.

(2) Lock lighting also can be accomplished using standard poles. The standard pole is 20 to 35 ft tall. Depending on the size of the lock, this approach might require more poles to be used, compared to lighting using high-mast poles. In addition, compared to using high-mast poles, uniformity usually is not as high with standard poles,. Standard poles mostly are used to illuminate the approach walls of a navigation lock, to fill in areas not illuminated by the high-mast lighting, and for projects where the use of high-mast lighting is not desirable (not cost effective, spill lighting in residential areas, etc.). Poles are mostly galvanized, round, steel but might be aluminum, square, and sometimes concrete. If possible, hinge these poles for ease of maintenance of fixtures or other equipment installed on the pole. They may be either hinged at the base or at some point in the middle third of the pole. If access to the pole is limited, which is usually the case on an approach wall, a base hinge point is not desirable. When using a hinged pole, the designer should specify the orientation of the pole on the drawings to ensure the installed pole will hinge in the direction desired. It is suggested that the poles be installed in a straight line, especially on approach walls, as this helps tow operators line up with the approach wall and is aesthetically more pleasing.

(3) There is no Corps standard for lighting intensity levels at navigation locks. The St. Lawrence Seaway design guidance for lighting has been provided as a reference in Appendix A. The lighting intensity requirements vary to some degree from district to district, even project to project. Some projects are illuminated to the same intensity all the time. Some projects illuminate to a security level (lower intensity level) when not

locking, and switch on additional lighting (higher intensity level) during the lockage. The designer should consider lighting intensities at other locks in the district when deciding what level will be used, to ensure uniformity throughout the district. This is important because it gives a tow operator uniformity from project to project, minimizing the chance of an accident caused by sufficiently different lighting. The following suggestions are for areas bounded by the lock miter gates and the outside walls of the lock chambers; all intensities are average maintained footcandle values. It is suggested that top-of-wall lighting levels be designed for approximately 2.0 footcandles (FC), average maintained; that a lock chamber at lower pool be designed for approximately 1.25 FC, average maintained, with uniformity of at least 70 to 80%. When a lockage isn't being made, the low-level (or security) lighting intensity may be chosen. It is suggested that low-level mode lighting intensities for areas outside the lock chamber be designed for 0.5 to 1.0 FC. This can be accomplished with fewer energized fixtures on the high-mast poles, or fewer standard pole fixtures being energized. Lower values of uniformity will result when using the standard poles, but might not be important during low-level or security mode. Lighting fixture types used for lock lighting are high-pressure sodium (yellowish, more footcandles per watt than MH) and metal halide (better color rendition, whiter light, fewer footcandles per watt that HPS). Luminaires on high-mast poles are usually similar to Holophane refractor-type HMST type. Luminaires on standard poles are usually the cobra-head type with wide distribution. With lock lifts greater than 30 ft, the designer might want to consider using floodlight-type fixtures with a more narrow distribution than the HMST-type fixtures. In that case, additional poles/fixtures might be needed for lighting on top of the lock wall.

c. Traffic Signals. Traffic signals are required at navigation projects to communicate to the tow operator that the lock is ready to be entered. The traffic signal lights are provided on the land wall of the landward lock and middle wall of the riverward lock, just upstream of the upper miter gates. Mounting height of the signal lights is 10 to 15 ft and might vary, depending on the tows entering the lock. The navigation lights typically are provided in a street traffic light configuration, with red conveying that tow should not enter the chamber and an amber-green combination allowing the tow to enter the chamber with caution. The lights are to flash in a duty cycle specified in the U.S. Coast Guard regulations. Mechanically driven flashers have been used to generate the flash duty cycle, but these are slowly giving way to PLC logic to provide the timing. Discrete, solid-state timers do not provide the flash duty cycle required for the traffic signal lights.

d. Navigation and Spillway Signals. The lock and dam should be marked in accordance with Coast Guard requirements or with those of the entity responsible for defining the marking requirements of the project. The designer should refer to 33 CFR Part 207, Navigation Regulations. These are also USACE regulations. At the time this document was published, the link to the CFR was www.gpo.gov/fdsys/pkg/CFR-2012-title33-vol3/pdf/CFR-2012-title33-vol3.pdf. The lock usually is marked with a single red light on each end of the land wall, 3 green lights on the upstream end of the river wall and 2 green lights on the downstream end of the river wall. The specific configuration of the lock-and-dam structure might necessitate a modified marking scheme, as determined by the Coast Guard or other appropriate entity. The dam should be illuminated with walk and security lighting across the bridge. Each pier should be provided with floodlight fixtures to illuminate at least the upstream portion of the gate bay.

13-4. Closed Circuit Television (CCTV) Systems.

a. General. CCTV systems provide multiple functions in a lock-and-dam control system. A CCTV system provides greater visibility of lock activities and can allow the operation of multiple locks from a single control room. Employing digital video recorders (DVRs) or network video recorders (NVRs), the CCTV system also can store video information that can, among other things, be used as forensic evidence in accident investigations. With the appropriate cameras in place, the system can inform the lock operator of fisherman or other boats near discharges, dam spillways, or other hazardous areas. CCTV cameras also can be placed to provide additional security assessment for entrance gates, storage areas, and visitor access areas.

b. System Purpose. Every lock-and-dam facility will place different requirements on the CCTV system. The system designer must identify the requirements and determine all the roles of the CCTV system to provide a suitable design. To begin the design, ask:

- Will the system be used to provide better visibility for operators in control rooms at both ends of the lock?
- Is the system supposed to provide a means of operating a single lock or multiple locks from only one central control room?
- Do the lock operators need to watch the dam or spillway gates?
- Does the system provide security assessment for the facility?
- Will remote monitoring or control be a possibility?

Answers to these questions can help identify where video monitors and cameras are needed. Refer to United Facilities Criteria, UFC 4-020-04A, for detailed application guidelines.

c. Lock Control Room. For surveillance of the project during operation, control rooms equipped with monitoring stations will be required. On facilities requiring a complete CCTV system, lock operations will be conducted from a single control room. However, even on these facilities, there might be other CCTV control locations, as described herein.

(1) Single Lock, Multiple Operators. In this method of lock operation, a control room usually is positioned at each end of the lock. Each control room should be equipped with a video client or server. A video server is the NVR or DVR itself. A video client is a PC configured to access the video files on the NVR or DVR. The video client's monitor can be split into four or more views to provide a look at each side of the lock gates, an upstream (or downstream) view of the lock approach, and a general view of

the lock. Views can be added as required by the facility for viewing the security cameras, coverage of the dam, discharges, etc.

(2) Single Lock, Centrally Operated. A central control room should be equipped with at least six views: each side of the gates, upstream and downstream, the upstream and downstream approaches. Views can be added as required by the facility for viewing the security cameras, coverage of the dam, discharges, etc.

(3) Multiple Locks, Centrally Operated. A central control room should be equipped with at least one video client, with a monitor split into multiple views, for the operation of each lock. Most central control rooms have separate control consoles for each lock. Placing a CCTV monitor at each console reduces operator movement between the consoles. The six views per lock provide a view of each side of the gates, upstream and downstream, and the upstream and downstream approaches. Views, and monitors if necessary, should be added as required by the facility for viewing the security cameras, coverage of the dam, discharges, etc.

- Lockmaster Office. The lockmaster might require a video client in his office. This provides a means to monitor the lock activities without disturbing operations in the main control room.

- Lock Electrician Office. It also might be convenient to provide a separate video client for the lock electrician. He can troubleshoot system problems or test the cameras without disturbing operations in the main control room.

- Visitor Center. A visitor center could be equipped with a video client. The public could view the same cameras as the lock operators. The center would not have control over the camera views or video playback.

d. Camera Requirements. The designer should consider the appropriate use of pan/tilt/zoom (PTZ) versus fixed mountings for cameras. Certain types of coverage, such as fixed security views or dam gates, do not necessarily warrant the extra expense of a PTZ camera. Typically, however, complete coverage of a lock requires multiple views from each camera, which means cameras should include PTZ.

(1) Lock. Camera coverage of the lock chamber is provided for all lock gates. Lift gates require one camera on each side of the lock. Miter gates require four cameras: one upstream and one downstream of the gate leaf for both sides of the lock. Because the lock chamber alternates between two water levels, cameras must be placed so it still can see the water level when the chamber is empty. Cameras also must provide a good view of the gate recesses. Cameras at the ends of the upper and lower guide walls provide coverage of the approaches. These cameras all should be equipped with PTZ capability.

(2) Dam. Cameras can be placed on the upstream and downstream sides of the service bridge to provide coverage of the pool and tailwater. Some facilities might

require a camera at each tainter gate to watch its movement during pool changes. Consider using PTZ cameras.

(3) Security. Cameras may monitor entrances to the lock and dam, storage areas, and visitor access areas. An integrated dome camera can provide indoor coverage of visitor centers or public access areas. PTZ might not be required for these cameras.

e. Documentation of Accidents. Recording the CCTV cameras can provide valuable documentation for accidents at a lock and dam. The designer must determine the level of documentation required for the facility. Previous generation CCTV systems relied on VCRs and multiplexers, in which all recordings where done by taking a 3-hr tape at 25 images per second (ips); recording video was indexed and could be played back using a multiplexer. But the maximum frame rates were only 3 ips when recording 16 cameras.

(1) As computer processors became faster, modern CCTV systems could store video and its associated metadata (time, date, location, etc.) digitally on hard drive disks (HDD), with the DVR being the standard for the past decade. This standard is being replaced quickly by NVRs.

(2) It is important to differentiate between DVRs and NVRs, as both often are called digital. A DVR digitally compresses analog video feeds and stores them on a hard drive, the term “digital” referring to the compression and storage technology, not the transmitted video images. The DVR, therefore, must be located near the analog feeds. In contrast, an NVR stores digital images directly from the IP network.

(3) Digital Video Recorder. DVRs utilize proprietary, PC-based devices containing analog-to-digital capture cards. Analog video is compressed by the DVR’s processor and video cards, and is stored digitally on the DVR’s HDDs. Remote video access is accessible only through the actual DVR itself. DVRs are used primarily for recording images from analog video cameras. Disadvantages of DVRs include:

- CPU/Video cards are doing all of the compression.
- Hard drives have no redundancy and are prone to failure due to their 24/7/365 usage.
- Loss of image quality due to the DVR converting the image from analog to digital, then compressing it so it can fit the information on a hard drive.
- Maximum resolution is VGA quality (approximately 640x480).
- Analog cameras all must be run back to the head end where the DVR was located.
- Hard to upgrade and expand because the video cards, operating system, and software are proprietary.

(4) Network Video Recorders. The big differences between DVRs and NVRs are, whereas a DVR digitally compresses analog video feeds and stores them on a hard-

drive, an NVR stores digital images pre-compressed from an IP camera. This subtle difference makes the NVR an efficient recorder.

(a) All the compression is being done in the field, by the cameras, allowing the CPU to lay down pre-compressed files in an easily searchable format. The NVR can use a conventional computer, meeting certain minimum specifications, and is easily upgradable to provide additional storage or new functionality as features become available. NVRs can accommodate resolutions of 480p, 720p, 1080p, and greater. NVR software can accommodate up to 3500 ips on a single server. NVRs also can take advantage of hot-swappable disks, and mirroring can be used to duplicate the recording of video streams on additional hard drives to minimize the chance of a single point of failure; if one drive goes down, the other is there as a backup.

(b) NVR minimum requirements. NVR hardware and software are advancing at an appreciable rate. Today's minimum standard will be obsolete in a couple of years. Therefore, it is recommended that the designer review current available technology with proven track records to determine what CCTV recording system is best for the facility for which they're designing.

f. Lighting Conditions. For a CCTV system to work properly, adequate lighting must be provided. The sensitivity of a camera refers to its ability to produce a usable image given a minimum lighting level. The CCTV market contains many cameras with various sensitivities, but the designer must remember that, just because a manufacturer's specification sheet states a certain sensitivity, it does not mean the picture will be usable at that level. For exceptionally low light areas, consider using cameras specially designed for these levels.

g. Lightning Protection. While fiber optic communication and composite video signals are immune to problems caused by lightning, cameras, coaxial cable, pan/tilt units, and receivers are not. Therefore, it is critical to the success of the CCTV system to provide proper protection from transient surges caused by lightning. All metal portions of camera and receiver mounting material should be well grounded, and the CCTV system manufacturer should be consulted for proper lightning protection components for the cameras and receivers.

h. Video Monitor.

(1) Cathode Ray Tube (CRT). The CRT used to be the most common video monitor until liquid crystal display (LCD) monitors became more affordable. Due to the CRT's large footprint and considerable power consumption, compared to LCDs, it is not recommended that new installations use CRT monitors and it is recommended that existing installations replace CRTs with LCDs as they fail.

(2) Liquid Crystal Display (LCD). LCD displays, also called flat screens due to their very narrow profile, have dropped considerably in price. This, coupled with their small footprint and lowered power consumption, make LCD monitors ideal for CCTV systems. Flat-screen monitors are available in a multitude of sizes, 27 in. and larger.

(3) Mounting. Mounting of the monitors should minimize the strain on the lock operators. Commercial enclosures with many different configurations are available. Usually, it is more cost effective to use commercially available consoles and avoid specially constructed consoles. A modular approach also reduces impacts from upgrades and equipment repair. The designer must determine the size and mounting of the video monitors, in addition to the number required.

i. Camera Control. NVRs not only receive video and feedback from IP cameras over Ethernet, but also send control signals over Ethernet to them. PTZ cameras typically are controlled with simulated joysticks and/or point-and-click interfaces. Multiple, scalable, camera views can be set up easily on a single monitor, and multiple monitors can be used if the NVR server is equipped with multiple outputs, or more outputs can be added with additional video cards. Analog cameras also can be wired into an NVR with the addition of analog input cards.

j. Interconnection Methods. There are different connection methods to the cameras, depending on the type of camera. For IP cameras, everything is transmitted over Ethernet. For analog cameras, the video signal from the camera is transmitted over a cable separate from the receiver’s serial connections.

(1) Analog Cameras. The receiver serial connections are made using a dual twisted-pair cable, and the video connection to the camera is made using coaxial cable. Both connections are distance limited. While distance might not be a problem with the lock and dam, the designer must consider other problems. Lightning and noise can cause major problems with the CCTV system, including equipment loss. Camera receivers, especially the serial connection, are susceptible to damage.

(2) Upload via Coax. CCTV system manufacturers have developed methods of transmitting the camera commands (PTZ) over the coaxial connection to the receiver. While this method does not offer complete immunity to noise and surges, it does simplify installation and eliminate the need to route dual twisted-pair cable all over the facility.

(3) Internet Protocol (IP) Cameras. Ethernet over copper (CAT5 or CAT6 cables) is limited to runs of 100m (~300 ft) or fewer. Any distance greater than this should be transmitted over fiber optic lines.

(4) Fiber Optic Receivers/Transmitters. While more expensive to implement, use of fiber optic transmission offers immunity to noise and the greatest isolation between the receivers. These factors will reduce and limit equipment damage. Use of fiber optics also can increase the distance between equipment. The maximum distance is related to the losses associated with the fiber. These might be splices, connectors, etc. Multi-mode fiber can be used for distances between 3.2 to 4 km (2 to 2.5 miles). Single-mode fiber (more expensive than multi-mode) can provide distances from 10 to13 km (6 to 8 miles). The standard optical connection is the ST type, and most units are configured for 62.5/125 micron fiber optic cable.

k. Camera Mounting. Manufacturers providing equipment to wall-mount, corner-mount, parapet-mount, and pole-mount camera equipment should be consulted. The designer must ensure that the hole pattern in the mounting equipment is compatible with the camera housing chosen. In a lock-and-dam application, pole mounting is used most often. Pole mounting provides the heights needed to obtain a good view in the lock chamber. Parapet mounting may be used on a dam structure to provide camera coverage of either the lock or dam.

l. Integrated Dome Cameras.

(1) Interior Dome Cameras. Typically housed in a dome unit with an integral pan/tilt and receiver, these cameras should be high-resolution and color equipped, with a suitable zoom lens.

(2) Exterior Dome Cameras. These are similar to interior dome cameras, but are equipped with an integral heater sized to maintain the lower dome above the dew point.

m. Video Analytics. Video analytics is software that provides motion detection triggered by pixel changes in the surveillance camera field of view. Current software can be configured to filter rain, snow, cloud shadows, small animals, and so forth. Also, areas within the camera's field of view can be masked so that movement in that area doesn't trigger an alarm. It also can be configured to display intruder paths for later viewing, should the operator not notice the intruder immediately. Cameras dedicated to security monitoring around a lock and dam can benefit from video analytics . One drawback, though, is it really only works well with a fixed camera.

n. Motion Detection. Hardware motion detection such as radar, infrared, or ultrasonic detectors has become sophisticated enough that it should be considered when designing a CCTV system. Detection also can be connected to audible alarms for immediate notification of a security problem. Motion detection also can have application at low-volume locks to detect small vessels in the lock approaches.

o. Upgrade Frequency. Generally, the upgrade frequency of a CCTV system depends on the type of device or component to be upgraded. Cameras and monitors usually will need to be upgraded most frequently in a CCTV system. Ultimately, equipment serviceability and availability of spare parts will determine the upgrade frequency of CCTV system components.

13-5. Security Systems.

a. Physical Security. Physically limiting access to the navigation lock control spaces, electrical and mechanical rooms, and other critical areas is the most basic and effective security measure. Keep doors locked, areas well lit, and report anyone who looks suspicious.

(1) Access Control System (ACS). The function of an ACS is to ensure only authorized personnel are permitted ingress and egress from a controlled area. The ACS

should be able to log and archive all transactions and alert authorities of unauthorized entry attempts. ACS can be interfaced with the CCTV system to assist security personnel in the assessment of unauthorized entry attempts.

(a) An ACS can have many elements, including electric locks, card readers, biometric readers (when required, but not always part of every system), alarms, and computer systems, to monitor and control the ACS. An ACS generally includes some form of enrollment station used to assign and activate an access control device.

(b) In general, an ACS compares an individual's credential against a verified database. If authenticated, the ACS sends output signals that allow authorized personnel to pass through controlled portals such as gates or doors. The system has the capability of logging and archiving entry attempts (authorized and unauthorized). Typically, the ACS interfaces with an Intrusion Detection System (IDS) for input of digital alarm signals at access portals controlled by the ACS. An example of this would be door forced alarms at a card reader controlled door. Similarly, the ACS interfaces with the CCTV system, in that cameras could be placed at remote locations to verify identity of entrants before operators at a central control station can manually actuate the remote gate.

(2) Navigation lock control rooms, entrance into mechanical and electrical spaces, and/or gates accessing the navigation lock area are all suitable candidates for inclusion in an ACS. The designer should evaluate each location and determine the best way to incorporate it into existing systems, or provide for future ACS. See UFC 4-021-02NF for further electronic security design criteria.

b. Computer System. With so much of the control system integrated with the PC/PLC system, computer system security is a major concern. In general, access to the lock operating computer system should be limited to qualified lock operating personnel. Access to the computer hardware, as well as the ability to alter programming, should be limited to the lock electrician and to district electrical engineers assigned to assist the lock personnel in maintaining and providing training for new computerized control systems.

c. Hardware. The biggest threats to the security of computer hardware are unauthorized use and the introduction of foreign software, exposing the system to computer viruses and hardware compatibility problems. While features such as dial-up routers, modems, and Internet connections offer greater flexibility in maintaining and troubleshooting the lock control network, they also are means for external break-ins to the system. Properly updated anti-virus software, firewalls, and switches to disconnect external access are ways to combat this problem.

d. Software. The lock operating systems should be equipped with a password security system that allows only qualified personnel to access the lock operating screens. Administrative tools and development software privileges should be limited to the system administrator account, to which only qualified personnel should have access. Backup directories should exist on each computer so that databases on any of the machines can be rebuilt quickly and correctly in the event of failure. Backup directories should be updated automatically when there are changes to the database. External access to the network also creates a way for software hackers to infiltrate the system and corrupt its software. Secure passwords and network firewalls can help, but they are not guarantees against system intrusion.

e. Communications. Most of the communications between computer systems and the machinery PLC I/O racks is via fiber optic cable. Therefore, noise and outside interference with the communication signal should not be an issue. The fiber optic communication cable will be run, in most cases, in secure areas around the project. Remote communication and access via the dial-up router, modem, or district intranet as stated above is more likely the method for compromise or security breach of the lock control communications system. Isolation of the lock control network will keep the system secure from would-be intruders.

13-6. Communication Systems.

a. General. While the power distribution and lock control systems are the key electrical systems for a navigation lock, designers must provide other communication systems for lock operators, and maintenance and administration personnel. Designers should plan for raceways and other necessary infrastructure to support the following communication systems. Some of these systems will be used heavily by the lock operators and should be accounted for in the control house or control console designs.

b. Telephone. Telephone systems have advanced from the local private branch exchange (PBX) to the latest Voice over IP (VOIP). Many Corps districts are converting their telephone systems to VOIP technology. USACE and ACE-IT have standardized hardware and software for VOIP communications. New installations must be coordinated with ACE-IT. Designers must address requirements for phone extensions, intercoms, and paging. Utility room space will be required for the telephone backboard and VOIP equipment. Lock operators will require phone access in the performance of their duties, but phone extensions at key machinery locations might help with troubleshooting. Consider consolidating the telephone system and the lock public address system into one integrated telephone communication system with dial-out and paging capabilities. Having such a system installed by the local telephone company will result in a better, more flexible system requiring less space on a control console. Multiple lines, zone ringing, paging, voicemail, and other features will provide even more convenience and efficiency for the lock operators and will cost less than separate telephone and public address systems.

c. Enterprise Network. Administration and maintenance personnel will require access to the enterprise or corporate network. The Ethernet connections to the network may be provided by twisted-pair or fiber optic cable, but it is important to isolate the enterprise network from the lock control network. The designer should provide adequate data drops in the necessary locations and account for utility room space to support the equipment racks for the various routers, switches, etc. Network requirements must be coordinated with ACE-IT.

d. Vessel-Logging PC. Major locks usually have a PC dedicated to logging vessel cargo and lockage information. Busy locks, such as key locks that are the first on a river to enter vessel data, require frequent monitoring and data entering into such a system. An operator will spend more time at this computer than at the lock operating work stations. Therefore, this PC should be located where the operator can conveniently sit down and log information (i.e., at the standard 760-mm (30-in.) desktop elevation with convenient keyboard and mouse, extra space for vessel lists and other paper work, and a comfortable standard office type chair). The Lock Performance Monitoring System (LPMS) will require access to the USACE corporate network. Lock operators may have restricted rights or capabilities on the logging PC and will require a separate PC for access to e-mail and the Internet.

e. Marine Band Radio. An operator will spend significant time on the marine radio, arranging queues and acquiring vessel cargo information. Lock operators likely will move the radio to several different locations on the console while they are getting used to the new system. For this reason, it is probably not a good idea to provide a permanent location for mounting the radio. Rather, provide means to move the radio to any location on the console allowing the operators, after they are accustomed to the new control system, to station the radio where it is most convenient for them. Designers must plan for power supplies and antenna connections.

13-7. <u>Life Safety Systems</u>.

a. Personnel. Safety for lock personnel, commercial vessel crew members, pleasure craft occupants, and public visitors is the single most important consideration when designing a quality operating system for a lock-and-dam facility. This should be considered in every aspect of the electrical system design, including the features discussed below. It is strongly recommended that designers of such systems spend sufficient time observing the day-to-day operations at a lock-and-dam project. While this will not qualify a designer to operate a lock and dam, it might give him some safety concerns to consider when designing a replacement electrical system or one for a new lock and dam. There are numerous procedures (e.g., locking tows and ice maneuvers) at each lock that, while not obvious to the casual observer, can place the operators in serious danger if the equipment is not located strategically, fails to function properly, or interlocks and safety features do not operate in a timely and correct fashion. All must be considered at each step of the design.

b. Accessibility. All electrical and electronic equipment must be installed in a way that it is safely accessible by lock maintenance personnel. All equipment that has energized circuits should be marked properly. At times, it might be necessary to perform maintenance on this equipment in the energized state. At all times, proper arc flash protection and procedures must be followed per the operating district’s policies. Equipment for which entry will cause a shutdown of the lock control system also should be marked with such a warning. All machinery and electrical gear that is controlled from a remote location should have warning labels and a means for disabling the remote control. Also, it is a good idea to provide a means for authorized personnel to operate the machinery from local controls in the event of emergency, or during routine maintenance.

c. Operating Locations. When locating lock operating stations, designers must provide visibility to all aspects of the project as necessary to safely control the project. This can be done with direct visibility or with CCTV cameras. Only experienced lock operators will be able to determine exactly what features of the project require constant, periodic, or occasional surveillance during a lockage. Other areas might require surveillance for security reasons.

d. Machinery Safety Interlocks. All operating machinery should offer safe access to the portions that require maintenance. In addition, interlocks should be in place to ensure that machinery cannot be started remotely while being serviced. Interlocks can consist of such things as machinery room door switches used as inputs to the PLC system. Auxiliary contacts on these switches can be used as hardwired interlocks. It is a good idea to have PLC inputs from these switches because of the flexibility and remote indication that can be provided.

e. Emergency Stop/Hardwired Backup. All lock control systems should have emergency stop pushbuttons at various points around the lock. These areas can include, but are not limited to, all lock control consoles, each gate machinery area, along the lock walls in areas frequented by lock personnel, at MCCs and switchgear locations, and in all galleries where electrical and mechanical machinery are located. The emergency stop pushbuttons should be large, red, mushroom-head type, clearly marked, and hardwired directly to motor starters. Using illuminated emergency stop buttons, wired such that they light when activated, should be considered because they provide visual indication to the operator that an emergency stop has been pressed. Consider an auxiliary MCC starter bucket with relays for use in each gate and valve starter circuit. Auxiliary contacts on the emergency stop buttons should be wired to PLC system inputs for indication only.

f. Motion Detectors. Some locks have submersible walkway bridges, walkways across miter gates, or other traffic areas that can be compromised by operation of lock equipment. In these locations, particularly if the lock is automated or remotely operated, motion detectors can provide important safety interlocks to prevent movement of machinery when lock personnel are passing through these areas. It is also a good idea to have some type of visual and/or audio indication prior to actual movement of lock equipment to allow personnel to stay clear of these areas.

CHAPTER 14

Installation, Operation and Maintenance, and Inspection

14-1. General. The design engineer's responsibilities do not end with the preparation of quality plans and specifications. In an ideal world, the design engineer would remain involved in all phases of the life cycle of the project, from construction support during installation, to support of operation and maintenance, and to regular inspection of the project. Guidance in these areas is given below.

14-2. Installation. This section discusses some of the factors that affect the quality of construction during the manufacturing and installation of gate operating equipment for new construction, major rehabilitation, and major maintenance work. It discusses the gate operating equipment and does not include the gate itself, except when the operation of the gate (e.g., misalignment of the rollers or maladjustment of the gate seals) affects the operation of the gate operating equipment.

a. Designer Involvement. It is important that the designer be involved with construction activities from shop drawing review through shop inspection to final field inspection. ER 1110-2-112 establishes policy that requires field construction participation by design personnel. The designer must maintain a good working relationship with the Corps construction office and must include engineering's involvement in the Engineering Considerations and Instructions to Field Personnel, as provided in ER 1110-2-1150.

b. Contract Requirements. The contract specifications should require the contractor to perform all testing associated with the manufacturing and installation of equipment. The Contractor Quality Control (CQC) representative and a representative from Design should witness any testing. The contract documents should identify the test data that needs to be recorded, and the contractor should record all test data required by the contract. The contract documents should hold the contractor responsible for any re-testing required due to the contractor's inability to fulfill the contract requirements.

(1) Shop Assembly and Test. Each hoist or machinery unit should be fully assembled in the shop. Alignment of component parts, correctness of fabrication, and tolerances, as shown on the shop drawings, should be checked. Each hoist should be given a no-load operational test in the shop. In addition to the proper operation of the hoist assembly, items such as limit switches, the motor brake, torque switches, and the control panel should be checked for proper settings and operation, in accordance with the manufacturer's recommendations.

(2) Field Installation and Tests. The contractor should be required to submit a detailed installation procedure. The installation procedure should show items such as storage and handling requirements, installation sequence, alignment techniques and criteria, bolt torque requirements, anchorage requirements, fits and tolerances, lubrication requirements, fluid levels, inspection and testing requirements, and operation and maintenance information. The hoist units should be shipped assembled, ready for

field installation, in accordance with the contractor's installation procedure. All installed equipment should undergo field testing. This assessment should include operational tests of the installed equipment with the gate in the dry and, if possible, at or near design head. The gate should be operated through a number of complete cycles in the dry to check the alignment of gate operating equipment and gate rollers and seals, and operation of the controls, limit switches, and brake. The gate operating system also should show that it can hold the gate in any position on demand. A load test then should be conducted at or near design head, if possible, with the measurements of the motor current, voltage, temperature, and vibration taken. For hydraulic operating systems, the pressure and leakage tests should be witnessed and the operating pressures should be checked against the design pressures. Relief valve and torque switch settings should be checked.

c. CQC/QA Responsibilities. For the CQC program to be effective, it must be enforced. The designer, as a member of the command's QA team, can assist the construction office in program enforcement.

(1) The designer should have the opportunity to review the CQC plan to ensure it includes the manufacturing and installation of the gate operating equipment. A preparatory inspection should be conducted for this phase of the contract work, and the designer should have the opportunity to attend the preparatory and initial inspections. The designer should also have the opportunity to review follow-up inspection reports.

(2) Also as part of the QA responsibilities, the designer should attend the shop inspection and test. To prepare, the designer should review the appropriate technical and non-technical parts of the contract documents. The designer also should review the referenced industry standards and shop drawings and check with the CQC representative to make sure the contractor is working from the same documents. Both CQC and QA should spot check to confirm the equipment is being manufactured in accordance with the shop and manufacturing drawings.

(3) When multiple hoists or machinery units are being built, as is common to many Corps facilities, the designer should witness the shop and field assembly and testing of at least the first unit. The assembly and testing procedures should be reviewed with the contractor's CQC representative and the calibration of the testing equipment should be checked by the CQC.

(4) During installation and prior to operation, the CQC representative should perform the following and the QA representative should spot check for compliance:

- Verify that an approved welding procedure has been submitted and qualified welders are on site when field welds are required.
- Witness the tensioning of the wire rope for multiple rope hoists.
- Witness dynamometer testing.

- Visually check alignment of shafts, couplings, gears, etc.

- Check operation of electrical components (i.e., motors, controls, limit switches, brakes, etc.).

- Verify that all fluid levels and lubrication of components are in accordance with the manufacturer's recommendations.

- Check that the effects of corrosion have been minimized by a properly applied paint or coating system and that there is adequate drainage designed into the hoists to prevent water retention.

- Verify that all reporting requirements have been met.

d. Acceptance Criteria. The designer should assure that the installation acceptance criteria are provided in the contract documents. When a performance specification is developed for contractor-designed equipment, the specification also should require the contractor to develop the acceptance criteria. The acceptance criteria should be based on Corps standards or applicable industry standards when such standards exist.

e. As-Built Drawings. The development of as-built drawings is a continuous process and should be a contract requirement. While the contractor is manufacturing and installing the equipment, the as-built drawings should be revised to reflect actual conditions. The CQC and QA representatives should monitor this process. All proposed changes should be coordinated with the designer. The as-built drawings, shop drawings, assembly drawings, and installation procedures should be revised as changes occur. The contractor should be required to furnish CADD drawing files on disc or CD, compatible with the customer's existing CADD system.

f. Operation and Maintenance (O&M) Manual. Similar to the as-built drawings, the development of the O&M manual is a continuous process and its development should be a contract responsibility. The manual is developed based on the equipment manufacturer's recommendations. It gives basic operating and maintenance procedures, guidelines for troubleshooting and repair procedures, and assembly/disassembly details. It also should include the procedure and frequency of the testing and inspection of components or systems. The contract specifications should list the items to be included in the O&M manual, and the designer should be involved in review and coordination with the contractor to ensure the inclusion of necessary information. Sufficient contract funds should be retained to ensure completion of O&M manuals. Because O&M manuals contain maintenance procedures that in some cases must be started immediately, consideration should be given to not starting to the warranty period until approved O&M manuals are received. The O&M manual should be considered a living document. This means that, as the project ages and equipment is changed, the manual should be updated to reflect those changes. The O&M manual produced by the contractor will become part of the project O&M manual developed by the designers.

g. Submittal Review. The designer should review the shop drawings, assembly drawings, installation drawings, installation procedure, O&M manual, and as-built drawings. The assembly drawings, installation drawings, and installation procedure can be submitted for information purposes only. Shop drawings that detail components specifically fabricated for the project should be submitted for approval. Shop drawings for purchased components should be submitted for approval. They should include catalog cuts and sufficient information to determine compliance with the specifications.

14-3. Operation and Maintenance. The operation of a Corps of Engineers lock and dam is the direct responsibility of the on-site lockmaster. However, everyone connected with the project has a responsibility to improve safety and increase efficiency. The design engineer must consider life cycle maintenance costs in all aspects of planning and design. This should include consideration of all costs to replace or rehabilitate gates or machinery.

a. O&M Manual. All projects should be operated within the guidelines provided in the project O&M manual. The manual should be updated to reflect all changes in operating procedures at the project. The manual should contain provisions to record equipment failures and to post maintenance records to enable operators to identify developing trends and avoid an unexpected failure.

b. Benefits of Automation for Improving Operation and Maintenance Procedures. As discussed in Chapter 12, the level of automation of a lock-and-dam project can range from none to the simple manual start-auto stop of a single gate or the complete lockage and recording of a vessel without operator intervention. The appropriate level of automation for a project is a judgment made by all involved with the project. To improve the effectiveness and safety of the project, the process of locking a boat and operating a dam can be examined for inherent inefficiencies or hazards. The PLC can be a tool for monitoring such parameters. The PLC never forgets to monitor and log movement and position of equipment, perform operations in a prescribed sequence, or record gate operating times and other parameters accurately. Inefficiencies will become evident when such data is monitored closely. The time it takes for a vessel to approach the lock might seem to be based entirely upon the vessel operator, but the truth is that the operation of the lock and dam has a lot to do with the length of time required for this process. Circumstances such as traffic-light signaling, traffic queuing, outdraft, operation of the dam, inefficiencies in direction change, operation of adjacent locks, pleasure craft, and visibility all might cause a traffic delay. Only careful monitoring of the process, along with these and other circumstances surrounding it, will present useful data to operations management. This data can be used to improve the operation of the project while maintaining safety. The amount of improvement will vary by project, but the information gathered will be useful and much of it will come for a one-time set-up cost of programming the PLC to monitor particular portions of the project.

14-4. Inspection and Testing. Project-specific inspection requirements (i.e., items to be inspected, inspection procedure, and inspection frequency) should be included in the O&M manual. These inspections also are part of the Periodic Inspection and Periodic Assessment program, as defined in ER 1110-2-1156. Typical items include motors, brakes, gears, shafts, couplings, bearings, controls, limit and torque switches, hydraulic systems, wire rope, chain, structural base frames, emergency generators, and any other integral parts that transmit the power to operate a gate. Machinery should be inspected not only for its current condition, but also for its condition relative to the last inspection. The operation and maintenance procedures should be reviewed for adequacy. Operational tests should be performed on a regular basis.

a. System Condition. The general condition and operation of the gate operating equipment should be observed. Operation should be smooth, and any abnormal performance should be noted. Noise and vibration also should be noted and the source determined. The inspector should report unsafe or detrimental procedures followed by the operator that could cause injury to personnel or damage to the equipment. The condition of the paint system also should be recorded. Maintenance procedures should be in accordance with the O&M manual. Maintenance records should be reviewed with maintenance personnel. Maintenance procedures should include the periodic operation of equipment that sits idle for long periods.

b. Inspection Guidelines. Some condensed guidelines for the major components of gate operating equipment:

(1) Open Gearing. Open gearing should be inspected for alignment, including under/over-engagement, and wear patterns away from the gear pitch line. Alignment problem indicators can predict bent shafting, misaligned bearings, loose mounting bolts, improperly fitted keys, or eccentric loading. Excessive or abnormal wear of the tooth-mating surfaces should be noted, including pitting, scoring, spalling, and plastic flow. Most tooth wear problems are related to improper fabrication or lubrication, or misalignment. Inspect the teeth, spokes, and hub for cracks, which might be the result of fabrication, heat treatment, or mishandling during installation. Cracks often are obscured by a coating of paint. Examine lubrication quality and quantity. Meshing gear surfaces that are scarred in the areas from slightly below the pitch line to the tooth tips is an indication of lubrication failure. Check gear teeth for excessive backlash, pitch line mesh, dirt, and corrosion. Inspect all keys, keyways, retainer caps, and bolting materials for proper fit, alignment, and tension.

(2) Speed Reducers. Speed reducer housings and mounting base should be inspected for cracks. All seals and gaskets should be inspected for lubricant leaks. All fasteners should be inspected for corrosion and proper tension. After removing the inspection cover, the interior should be examined for signs of condensation, corrosion, general condition of the gears (see open gearing), and excessive shaft movement and backlash. The lubricant level should be checked daily, while oil samples should be laboratory tested quarterly. Oil test results should be examined for the presence of wear particles, other contaminants and water, viscosity breakdown, and the presence of sufficient additives.

(3) Shafts and Couplings. Shafting should be inspected for cracks, twist, bending, strain, and misalignment. Suspicion of cracking or excessive strain should be verified by dye penetrant testing. Bending can be estimated by using dial indicators.

Coupling components should be examined for adequate lubrication, proper fastener tension, damaged keys, and improper alignment.

(4) Bearings. Bearing housings, pedestals, and supports should be inspected for cracks and misalignment. Fasteners should be checked for tightness and corrosion. All bearings should be checked for condition and quantity of lubricant. Plain bearings (bushings) should be examined, using feeler gauges, for excessive wear, as well as the condition of any seals, as applicable.

(5) Brakes. All braking devices should be inspected for proper braking torque setting and complete release at actuation. On shoe brakes, check brake wheels and shoes for wear, corrosion, misalignment, and proper clearance at release. Linkages should be free but not loose. Ensure there is no leakage at connections or seals on enclosed hydraulic disc brakes. All limit switches should be tested for proper setting and actuation.

(6) Hoist Motor. Motors should be inspected to ensure nothing is interfering with the motor ventilation. Any unusual noise or odor, such as from scorched insulation varnish, will require a more detailed inspection. Bearings should be examined for adequate lubrication, and for indications of wear (free movement), vibration, and seal leakage. The motor should be started several times to ensure that it comes up to proper operating speed. Operation of winding heaters should be verified. Fasteners should be tight and in good condition.

(7) Hydraulic System. All hydraulic components should be inspected for signs of leakage. All flexible hoses should be examined for deterioration, flaking, cracking, kinks, and wear. Hydraulic pumps should be checked for noise and vibration. Hydraulic fluid should be tested for viscosity, moisture content, and other contamination. Hydraulic fluid should be sampled and analyzed at least quarterly. Filters, tank trappers, breathers, and other devices should be examined for contaminants or replacement indication. Personnel should review, at least quarterly, maintenance records for filter and fluid changes. Hydraulic cylinders should be inspected regularly for misalignment, seal leakage, piston rod coating deterioration, and proper function of associated valves. All valves should be inspected for loose locknuts, damaged handles, stems, or wiring connections. All limit switches, speed-change switches, and pressure switches should be tested for proper function. All pressure relief valve settings should be verified annually by an independent, properly calibrated pressure gauge. All flow rate settings should be verified. Seasonal changes to pump, valve, or other equipment adjustable settings should be recorded.

(8) Machinery Supports. Machinery support frames should be inspected for cracking in the steel, grout pad, and concrete. All welds should be examined for cracking. Any corrosion should be noted and scheduled for repair. Deformation of any steel members, or anchor bolts, should be cause for immediate analysis of the safety of continued operation. All drain holes should be clear so there is no standing water.

(9) Wire Rope, Drums, and Sheaves. Wire rope, drums, and sheaves should be inspected in accordance with EM 1110-2-3200.

(10) Interlocks and Limit Switches. All safety interlocks should be tested regularly, along with periodic inspections. These include the miter gate to culvert valve, culvert valve to miter gate, and culvert valve to culvert valve interlocks. See Chapter 12 for descriptions of these interlocks. Also, the proper function of position limit switches should be tested regularly along with periodic inspections when made possible by allowable gate opening.

(11) Filling/Emptying Valve Control. The proper function of overfill and overempty and valve synchronization culvert valve controls should be tested.

c. Test Operation of Equipment. ER 1110-2-1156 requires "frequent observations of the dam and appurtenant structures." This includes all systems necessary for the project to serve its intended purpose in a safe and efficient manner. These components should be test operated using emergency power. The emergency power generator should be full-load tested at more frequent intervals, such as every other month, to maintain its integrity. To satisfy these requirements, the standard operating procedure at most lock and dams is to annually operate one third of the spillway gates through a full open and close cycle. Bulkheads or stop logs might need to be installed at the times when pool levels prohibit the full opening of the gates. This establishes a maximum interval between full travel tests of three years for each gate. Partial opening of the gates does provide some test benefits, but it is essential that full travel tests be performed to verify non-binding operation at all gate positions and the proper function of travel limit switches. A partial opening test of each gate should be conducted at least yearly, except for the year that a full travel test is scheduled. The gate should be raised as high as pool conditions allow without the use of bulkheads or stop logs.

14-5. Access for Inspection and Maintenance Activities. Designers and those responsible to review completed designs should ensure that access has been provided to inspect and maintain equipment and components. Planning this access should happen early in the design phase of the project and must be coordinated with the structural engineer and others responsible for the design and layout of the structure. The designer also needs to develop an inspection and maintenance plan and include that plan in the O&M manual. The details of the plan must be coordinated with those responsible to operate and maintain the project. Details should include locations of removable inspection covers, permanent or portable access platforms, etc. Provide proper lighting and a safe working environment to perform the inspection and maintenance procedures.

APPENDIX A

Part 1 - Design Guidance and References

Note: The following documents and files are provided as part of this engineering manual in electronic portable document format. The numbers listed below correspond to the electronic document title.

A-1 Design of Locks Part 1**,** Ministry of Transport, Public Works and Water Management, Netherlands

A-2 Directive 2005/32/EC of the European Parliament and of the Council**,** Commission Regulation No. 640/2009, Ecodesign Requirements for Electric Motors

A-3 The Engineer School 1940, The Engineer School. 1940. Design of locks: Engineering construction, canalization, Vol. II. Fort Belvoir, VA.

A-4 NAMUR 2006, German Standard NAMUR Recommendation NE 107. 2006. Self-monitoring and diagnosis of field devices (bilingual version), VDI/VDE-GMA Technical Committee 6.14. In System Monitoring and Diagnosis in Process Control. NAMUR-Geschäftsstelle, Leverkusen

A-5 Kingsbury 2010**,** Kingsbury, Inc. 2010. A general guide to the principles, operations and troubleshooting of hydrodynamic bearings

A-6 ASCE et al. 2012**,** ASCE, FEMA, and USACE 2012. Guidelines for evaluation of water control gates draft

A-7 NAVFAC P-307**,** Weight Handling Equipment, Naval Facilities Engineering Command, Navy Crane Center, 2009

A-8 UFC 3-320-07N**,** Weight Handling Equipment, Naval Facilities Engineering Command, 2004 and 2007

A-9 Netherlands 2011a**,** Water Management in the Netherlands, Rijkswaterstaat Ministry of Infrastructure and the Environment

A-10 Netherlands 2011b**,** Waterway Guidelines, Rijkswaterstaat Directorate General for Public Works and Water Management

A-11 PIANC Working Group 106 Report, Innovations in Navigation Lock Design, 2009

A-12 PIANC Working Group 138 Report, Navigation Locks and Dams, Mechanical and Electrical Lessons Learned, 2012

A-13 PIANC Final Report, International Commission for Study of Locks, 1986

A-14 PIANC Workshop, Use of Synthetic Materials, R.A. Daniel and J. Augustijn, 2011

A-15 Kumar et al., Kumar, A., A. A. Odeh, and J. R. Myers. 1989. Mechanical properties and corrosion behavior of stainless steel for locks, dams, and hydroelectric plant applications. REMR-EM-6 USACERL. Champaign, IL: U.S. Army Corps of Engineers

A-16 Clifton and Beitelman 1989, Clifton, W. B., and A. Beitelman. Lubricants for hydraulic structures. REMR-EM-5 USACERL. Champaign, IL: U.S. Army Corps of Engineers

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APPENDIX A

Part 2 - Industry References and Industry Standards, Literature References, and other Corps Documents

Note: Source files (electronic files or hard copies) are not provided for these references.

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EM 1110-2-2602 Planning and Design of Navigation Locks

EM 1110-2-2607 Planning and Design of Navigation Dams

EM 1110-2-2608 Navigation Locks: Fire Protection Provisions

EM 1110-2-2701 Vertical Lift Gates

EM 1110-2-2702 Design of Spillway Tainter Gates

EM 1110-2-2704 Cathodic Protection Systems for Civil Works Structures

EM 1110-2-3105 (Change 2) Mechanical and Electrical Design of Pumping Stations

EM 1110-2-3200 Wire Rope Selection Criteria for Gate Operating Devices

EM 1110-2-3400 Painting: New Construction and Maintenance

EM 1110-2-3401 Thermal Spraying: New Construction and Maintenance

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EM 1110-2-4205 HydroElectric Power Plants: Mechanical Design

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TM 5-811-1 Electric Power Supply and Distribution

TM 5-853-4 Security Engineering Electronic Security Systems

USACE Engineering Regulations:

ER 1110-2-100 Periodic Inspection and Continuing Evaluation of Completed Civil Works Structures

ER 1110-2-112 Required Visits to the Construction Sites by Design Personnel

ER 1110-2-1150 Engineering and Design for Civil Works Projects

ER 1110-2-1156 Engineering and Design Safety of Dams – Policies and Procedures

ER 385-1-100 Arc Flash Hazard Program

USACE Engineering Pamphlets:

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INDUSTRY STANDARDS

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American National Standards Institute (ANSI):

ANSI B29.10M Heavy Duty Offset Sidebar Power Transmission Roller Chains and Sprocket Teeth

ANSI B29.15M Steel Roller Type Conveyor Chains

ANSI B93.30M Hydraulic Fluid Power-Contamination Analysis Data-Reporting Method

American Society for Testing and Materials (ASTM):

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ASTM A36 Standard Specification for Carbon Structural Steel

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ASTM A148 Standard Specification for Steel Castings, High Strength, for Structural Purposes

ASTM A290 Standard Specification for Carbon and Alloy Steel Forgings for Rings for Reduction Gears

ASTM A291 Standard Specification for Steel Forgings, Carbon and Alloy, for Pinions, Gears and Shafts for Reduction Gears

ASTM A519 Standard Specification for Seamless Carbon and Alloy Steel Mechanical Tubing

ASTM A564 Standard Specification for Hot-Rolled and Cold-Finished Age-Hardening Stainless Steel Bars and Shapes

ASTM A668 Standard Specification for Steel Forgings, Carbon and Alloy, for General Industrial Use

ASTM A705 Standard Specification for Age-Hardening Stainless Steel Forgings

ASTM B148 Standard Specification for Aluminum Bronze Sand Castings

ASTM B271 Standard Specification for Copper-Base Alloy Centrifugal Castings

ASTM B505 Standard Specification for Copper-Base Alloy Continuous Castings

ASTM B584 Standard Specification for Copper Alloy Sand Castings for General Applications

American Petroleum Institute (API):

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American Society of Mechanical Engineers (ASME):

ASME B17.1 Keys and Keyseats

ASME B30.2 Overhead and Gantry Cranes (Top Running Bridge, Single or Multiple Girder, Top Running Trolley Hoist)

ASME B30.4 Portal, Tower and Pillar Cranes

ASME B30.5 Mobile and Locomotive Cranes

ASME B30.6 Derricks

ASME B30.8 Floating Cranes and Floating Derricks

ASME B30.9 Slings

ASME B30.22 Articulating Boom Cranes

ASME B46.1 Surface Texture (Surface Roughness, Waviness and Lay)

Crane Manufacturers Association of America (CMAA):
http://www.mhia.org/industrygroups/cmaa

CMAA 70 Multiple Girder Cranes

CMAA 78 Single Girder Cranes

Deutsches (German) Institut Fur Normung (DIN):

DIN 22252 Steel Chains (Round Link) for Use with Conveyors and Coal-Getting Equipment in Mining: Calibrated, Tested

DIN 19704 Hydraulic Steel Structures – Part 1: Criteria for Design and Calculation

DIN 19704 Hydraulic Steel Structures – Part 2: Design and Manufacturing

DIN 19704 Hydraulic Steel Structures – Part 3: Electrical Equipment

NAMUR NE 91 Requirements for Online Plant Asset Management Systems

International Organization for Standardization (ISO):

ISO 281 Rolling Bearings – Dynamic Load Ratings and Rating Life

Japanese Standards Association (JIS):

JIS B1519 Dynamic Load Ratings for Rolling Bearings

Dutch Standardization Institute (NEN):

NEN 6786, Article 5.2.7 Life Span of Machinery, Rules for Design of Movable Bridges Netherlands (in Dutch)

National Association of Corrosion Engineers (NACE):

NACE SP0169 Control of External Corrosion on Underground or Submerged Metallic Piping Systems (Current Edition)

NACE SP0188 Standard Recommended Practice Discontinuity (Holiday) Testing of Protective Coatings (Current Edition)

National Fire Protection Association (NFPA):

NFPA 70 National Electrical Code (Current Edition)

NFPA 70E Standard for Electrical Safety in the Workplace (Current Edition)

NFPA 780 Standard for the Installation of Lightning Protection Systems (Current Edition)

NFPA 20 Standard for the Installation of Stationary Pumps for Protection (Current Edition)

National Electrical Manufacturers Association (NEMA):

NEMA WC 70 Power Cables Rated 2000 Volts or Less for the Distribution of Electrical Energy

Occupational Safety and Health Administration (OSHA):

OSHA 1910.179 Overhead and Gantry Cranes

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UFC 3-520-01 Interior Electrical Systems

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Internet References:

Knovel (online engineering database and search engine): http://why.knovel.com/

Tribology-abc (online machinery and lubrication and bearing database and unit conversion calculator): http://www.tribology-abc.com/

Whole Building Design Guide (online data base for industry specifications, guide specifications, and building materials): http://www.wbdg.org/

APPENDIX B

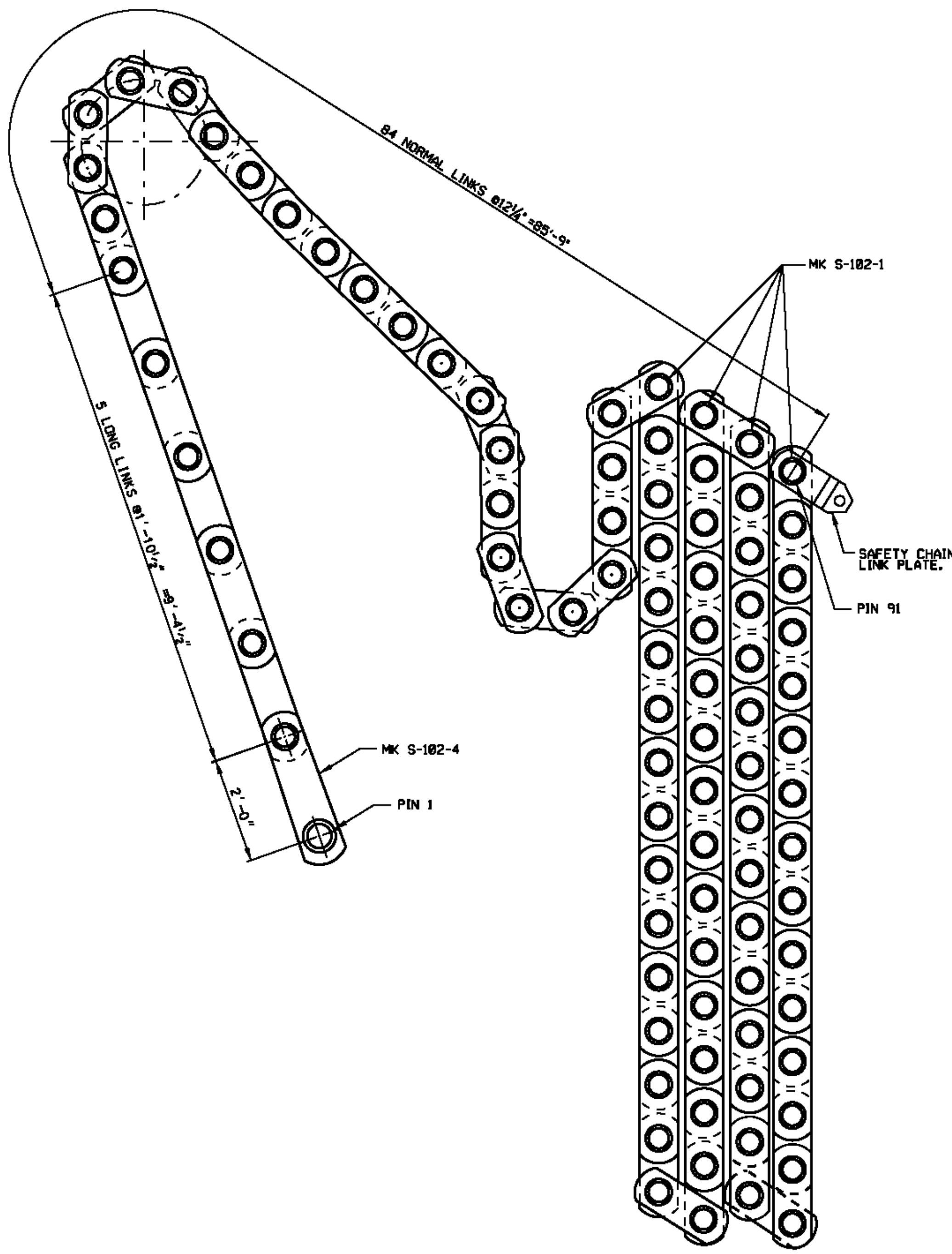
Illustrative Plates - Index

Chapter 6 – Tainter Valves

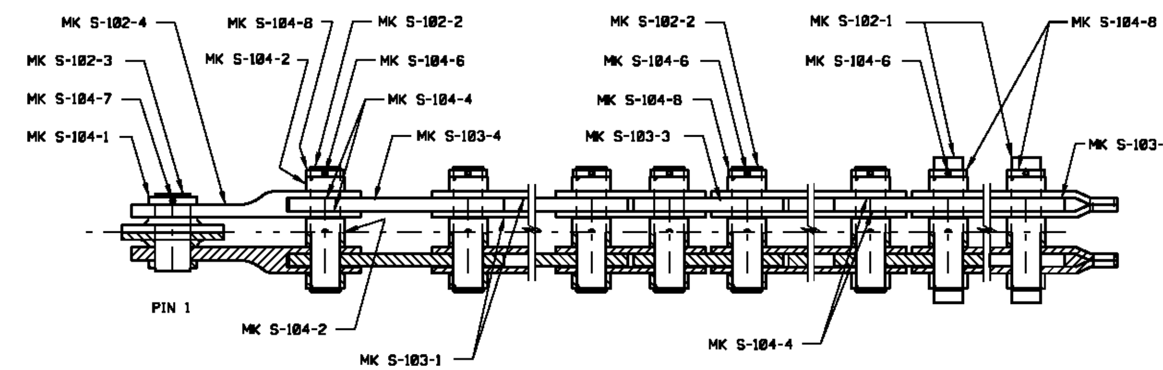
Chapter 7 – Vertical Lift Gates

Chapter 9 – Tainter Gates

Chapter 9 Wicket Gates



CHAIN ELEVATION



CHAIN ASSEMBLY

NOTES:

1. CHAIN SIDE BARS, ROLLERS, AND PINS SHALL BE DRILLED AND ASSEMBLED IN MATCHED SETS FOR BOTH THE NEW CHAIN AND REWORKED CHAIN (2 HEAVY BARS AND 4 NORMAL BARS PER SET). TOLERANCES APPLY TO MATCHED SETS.

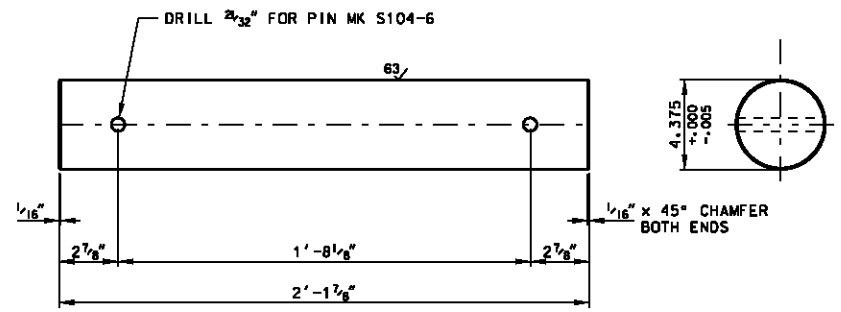
2. ANTI-SEIZE LUBRICANT SHALL BE APPLIED TO ALL MATING SURFACES OF THE CHAIN PRIOR TO ASSEMBLY OF THE CHAIN COMPONENTS. PRIOR TO INDUCEMENT OF LOAD ON THE CHAIN AND IMMERSION OF THE CHAIN, ALL CHAIN SURFACES SHALL BE LUBRICATED WITH AN ENVIRONMENTALLY FRIENDLY, BIODEGRADABLE INDUSTRIAL GREASE IN ACCORDANCE WITH MANUFACTURER'S RECOMMENDATIONS.

3. NO WORK IS TO BE PERFORMED ON THE SAFETY CHAIN LINK PLATE.

4. THE COLLARS FOR THE CHAIN SHALL BE ATTACHED TO THE PINS WITH COLLAR PINS AND THE COLLAR PINS PEENED ON EACH END TO PREVENT COLLAR PIN FROM FALLING OUT.

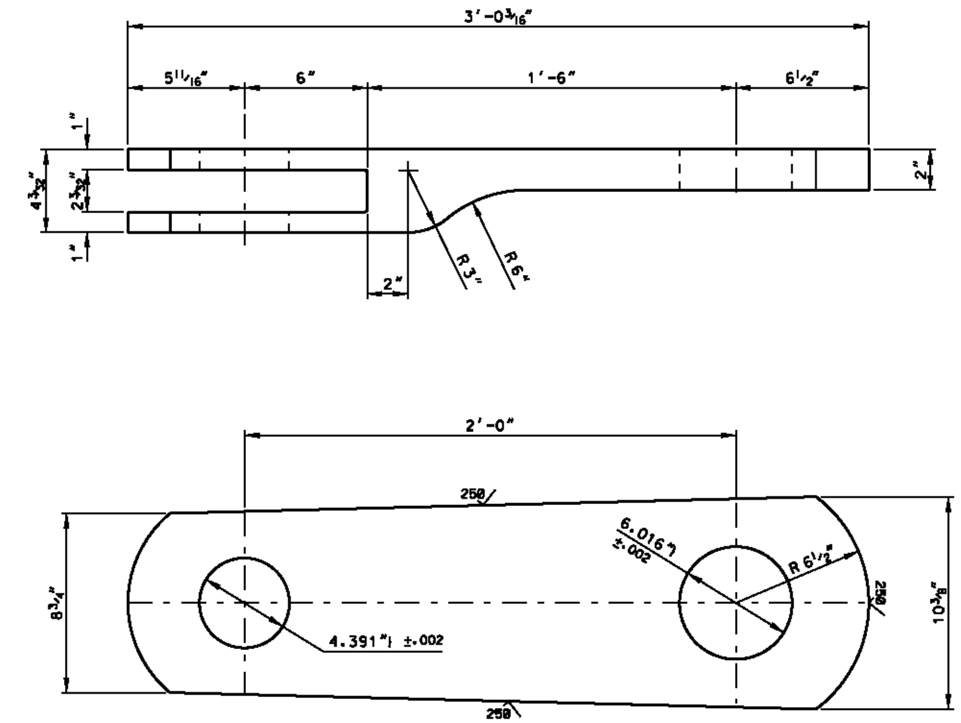
5. NEW CHAIN ASSEMBLIES SHALL BE ENTIRELY SHOP ASSEMBLED WITH THE EXCEPTION OF PIN 1, WHICH MUST BE INSTALLED IN THE FIELD.

ROLLER CHAIN
ASSEMBLY



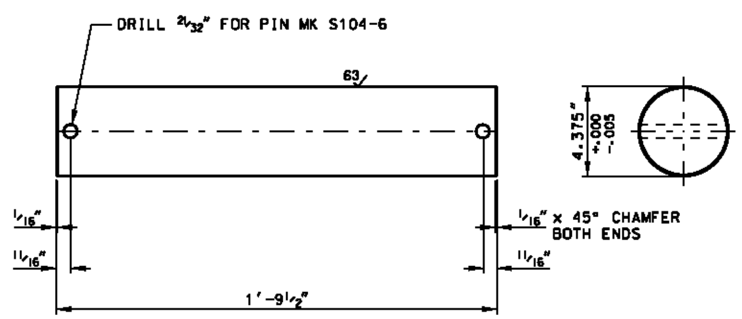
LONG PIN

MARK S-102-1 WT. 109 LB.
HEAT TREAT 350 BHN MIN.
MAKE 4 (1 SPARE)*



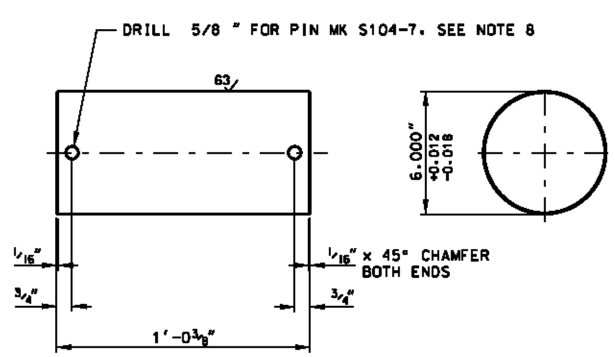
END SIDE BAR

MARK S-102-4 WT. 190 LB.
HEAT TREAT 260-300 BHN
MAKE 2 (1 SPARE)*



NORMAL PIN

MARK S-102-2 WT. 86 LB.
HEAT TREAT 350 BHN MIN.
MAKE 86 (1 SPARE)*



LOWER PIN END

MARK S-102-3 WT. 98 LB.
HEAT TREAT 350 BHN MIN.
MAKE 1 (1 SPARE)*

NOTES:

1. MATERIALS:

ITEM	MARK (MK)	MATERIAL DESIGNATION
LONG PIN	S-102-1	SEE NOTE 2, HEAT TREAT AS SHOWN
NORMAL PIN	S-102-2	SEE NOTE 2, HEAT TREAT AS SHOWN
LOWER END PIN	S-102-3	SEE NOTE 2, HEAT TREAT AS SHOWN
END SIDE BAR	S-102-4	SEE NOTE 2, HEAT TREAT AS SHOWN

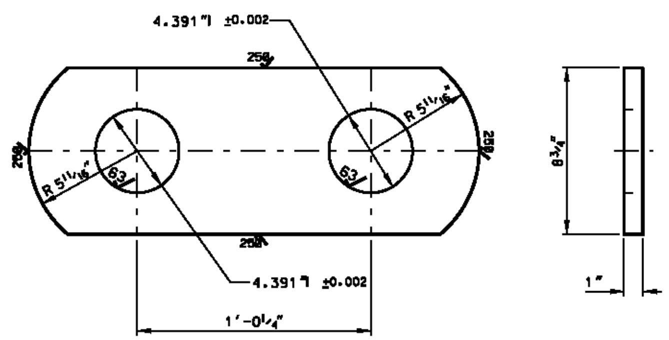
2. MATERIAL DESIGNATION FOR THE NEW ROLLER GATE CHAIN COMPONENTS SHALL BE ASTM A564, TYPE 630 OR TYPE XM-25, HEAT TREAT AS SHOWN. SEE SPECIFICATIONS FOR MATERIAL PROPERTY REQUIREMENTS.
3. PRIOR TO THE DRILLING OF THE HOLES ON THE EXISTING CHAIN, THE CONTRACTING OFFICER'S REPRESENTATIVE MAY SUPPLY THE CONTRACTOR WITH A NEW CHAIN CONFIGURATION (SOME CHAIN COMPONENTS MAY BE REARRANGED FROM THE ORIGINAL CHAIN CONFIGURATION).
4. FINISH ALL SURFACES OF ALL ITEMS 250 EXCEPT AS NOTED.
5. ROUND ALL SHARP CORNERS TO 0.02-INCH RADIUS.
6. QUANTITIES SHOWN ARE PER CHAIN ASSEMBLY FOR EACH ROLLER GATE. THERE ARE THREE ROLLER GATES TOTAL.
7. CHAIN COMPONENTS SHALL BE SAW CUT. FLAME OR ABRASIVE WHEEL CUTTING IS NOT PERMITTED FOR FABRICATION OF ANY COMPONENT.
8. CONTRACTOR SHALL FIELD MEASURE EXISTING LOWER END PIN DIMENSIONS PRIOR TO DRILLING HOLES.

SPARE PARTS:

1. ALL SPARE ROLLER GATE CHAIN COMPONENTS SHALL BE DELIVERED TO THE STORAGE YARD AT LOCK AND DAM 11 (LOCATION TO BE DETERMINED BY C.O.R.) PACKED IN GREASE AND SEALED AGAINST THE ENVIRONMENT. SEE SPECIFICATIONS FOR REQUIREMENTS. ALL SPARE PARTS SHALL BE STAINLESS STEEL OF THE TYPE DESIGNATED IN NOTE 2 ON THE COMPONENTS SHEET.
2. ALL SPARE COMPONENTS NEED TO HAVE ITEM NAMES AND MARK NUMBERS CLEARLY MARKED ON PACKING MATERIAL.
3. QUANTITIES SHOWN IN PARENTHESES ()* ARE THE TOTAL NUMBER OF SPARE COMPONENTS REQUIRED FOR THE ENTIRE CONTRACT, NOT PER CHAIN. THE QUANTITIES IN PARENTHESES ARE IN ADDITION TO THE PER CHAIN QUANTITIES LISTED BELOW THE CORRESPONDING MARK NUMBER.

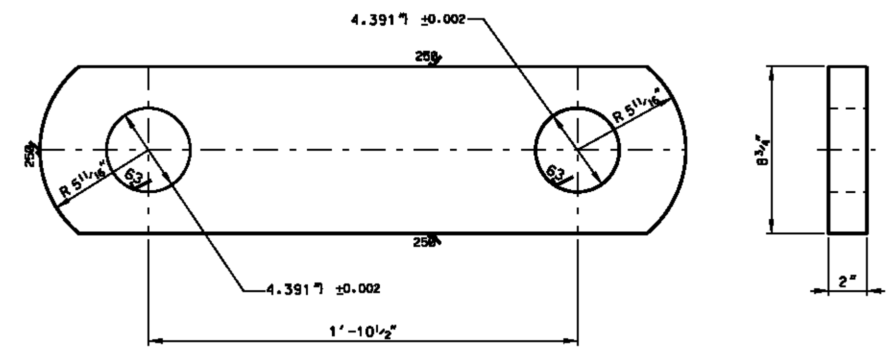
ROLLER CHAIN ASSEMBLY PARTS

PLATE B-2



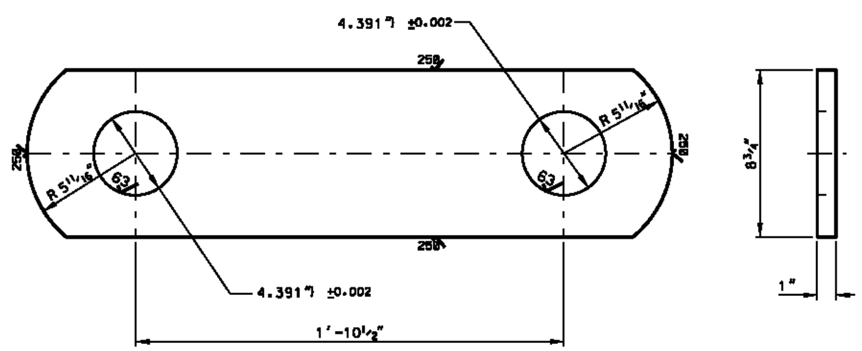
NORMAL SIDE BAR

MARK S-103-1 WT. 47.9 LB.
HEAT TREAT 260-300 BHN
MAKE 168 (4 SPARE)*



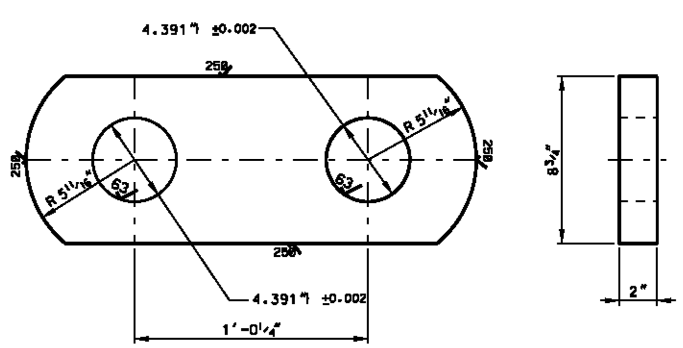
LONG HEAVY SIDE BAR

MARK S-103-4 WT. 145 LB.
HEAT TREAT 260-300 BHN
MAKE 6 (2 SPARE)*



LONG SIDE BAR

MARK S-103-2 WT. 73 LB.
HEAT TREAT 260-300 BHN
MAKE 8 (4 SPARE)*



NORMAL HEAVY SIDE BAR

MARK S-103-3 WT. 94 LB.
HEAT TREAT 260-300 BHN
MAKE 84 (2 SPARE)*

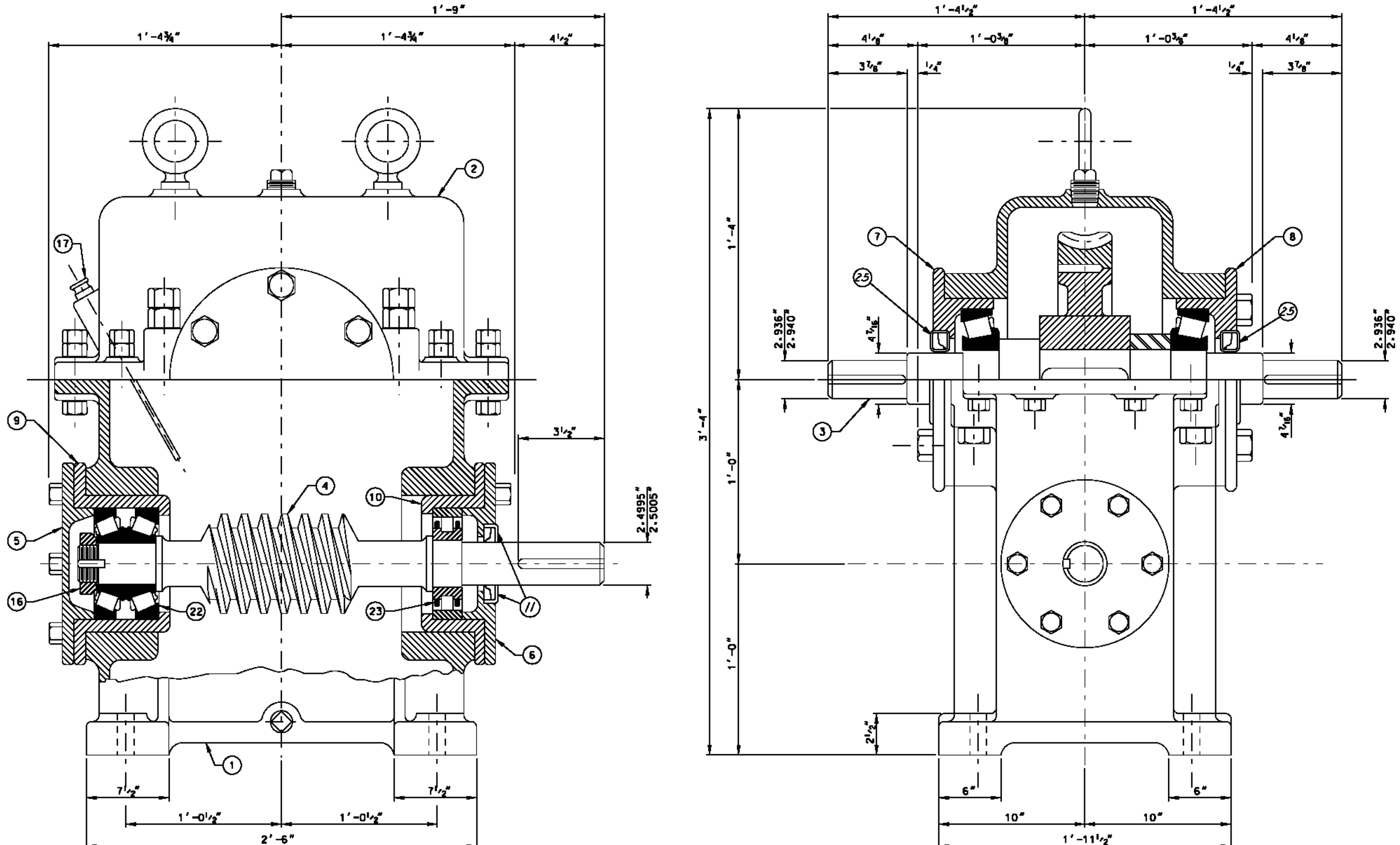
NOTES:

1. MATERIALS:

ITEM	MARK (MK)	MATERIAL DESIGNATION
NORMAL SIDE BAR	S-103-1	SEE NOTE 2, HEAT TREAT AS SHOWN
LONG SIDE BAR	S-103-2	SEE NOTE 2, HEAT TREAT AS SHOWN
NORMAL HEAVY SIDE BAR	S-103-3	SEE NOTE 2, HEAT TREAT AS SHOWN
LONG HEAVY SIDE BAR	S-103-4	SEE NOTE 2, HEAT TREAT AS SHOWN

2. MATERIAL DESIGNATION FOR THE NEW ROLLER GATE CHAIN COMPONENTS SHALL BE ASTM A564, TYPE 630 OR TYPE XM-25, HEAT TREAT AS SHOWN.
3. PRIOR TO THE DRILLING OF THE HOLES ON THE EXISTING CHAIN, THE CONTRACTING OFFICER'S REPRESENTATIVE MAY SUPPLY THE CONTRACTOR WITH A NEW CHAIN CONFIGURATION (SOME CHAIN COMPONENTS MAY BE REARRANGED FROM THE ORIGINAL CHAIN CONFIGURATION).
4. FINISH ALL SURFACES OF ALL ITEMS 250 EXCEPT AS NOTED.
5. ROUND ALL SHARP CORNERS TO 0.02-INCH RADIUS.
6. CHAIN COMPONENTS SHALL BE SAW CUT. FLAME OR ABRASIVE WHEEL CUTTING IS NOT PERMITTED FOR FABRICATION OF ANY COMPONENT.
7. QUANTITIES SHOWN ARE PER CHAIN ASSEMBLY FOR EACH ROLLER GATE. THERE ARE THREE ROLLER GATES TOTAL.

ROLLER CHAIN ASSEMBLY PART DETAILS



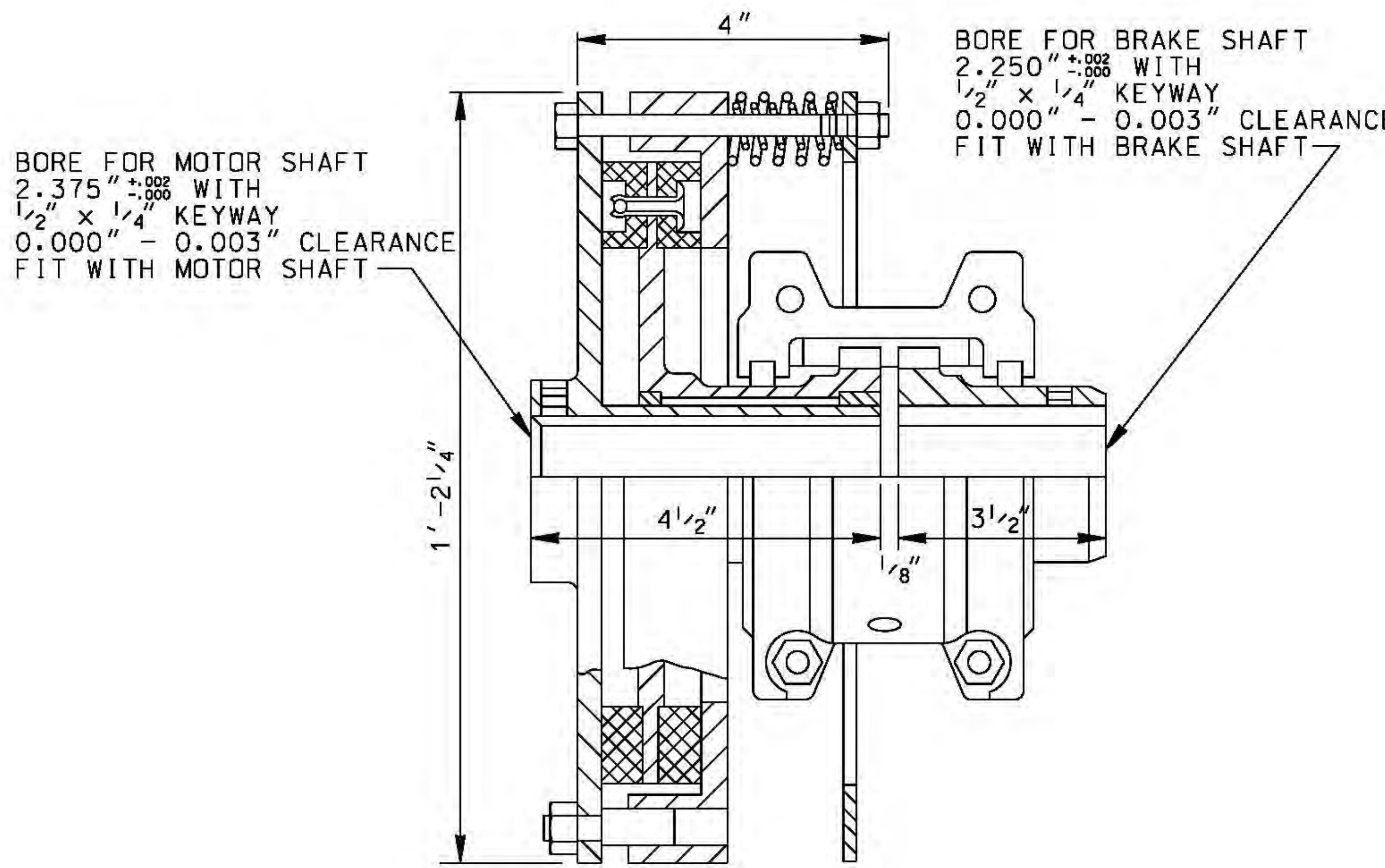
PART NO.	NAME	PART NO.	NAME
1	WORM AND GEAR HOUSING	15	
2	WORM AND GEAR COVER	16	SKF LOCKNUT & LOCK WASHER
3	SLOW SPEED GEAR SHAFT	17	BAYONNET OIL GAUGE
4	WORM SOLID ON SHAFT	18	
5	HIGH SPEED CLOSED END PIECE	19	
6	HIGH SPEED OPEN END PIECE	20	
7	SLOW SPEED OPEN END PIECE	21	
8	SLOW SPEED OPEN END PIECE	22	H.S. TIMKEN BEARING
9	TIMKEN BEARING CASE	23	NORMA HOFFMAN BEARING
10	HOFFMAN BEARING CASE	24	
* 11	**OIL SEAL (HIGH SPEED)**	25	**OIL SEAL (SLOW SPEED)** *
12			
13			
14			

NOTE: REPLACE ITEMS SHOWN IN BOLD

SIZE 1200 WORM GEAR SPEED REDUCER

TAINTER GATE
SPEED REDUCER
DETAILS

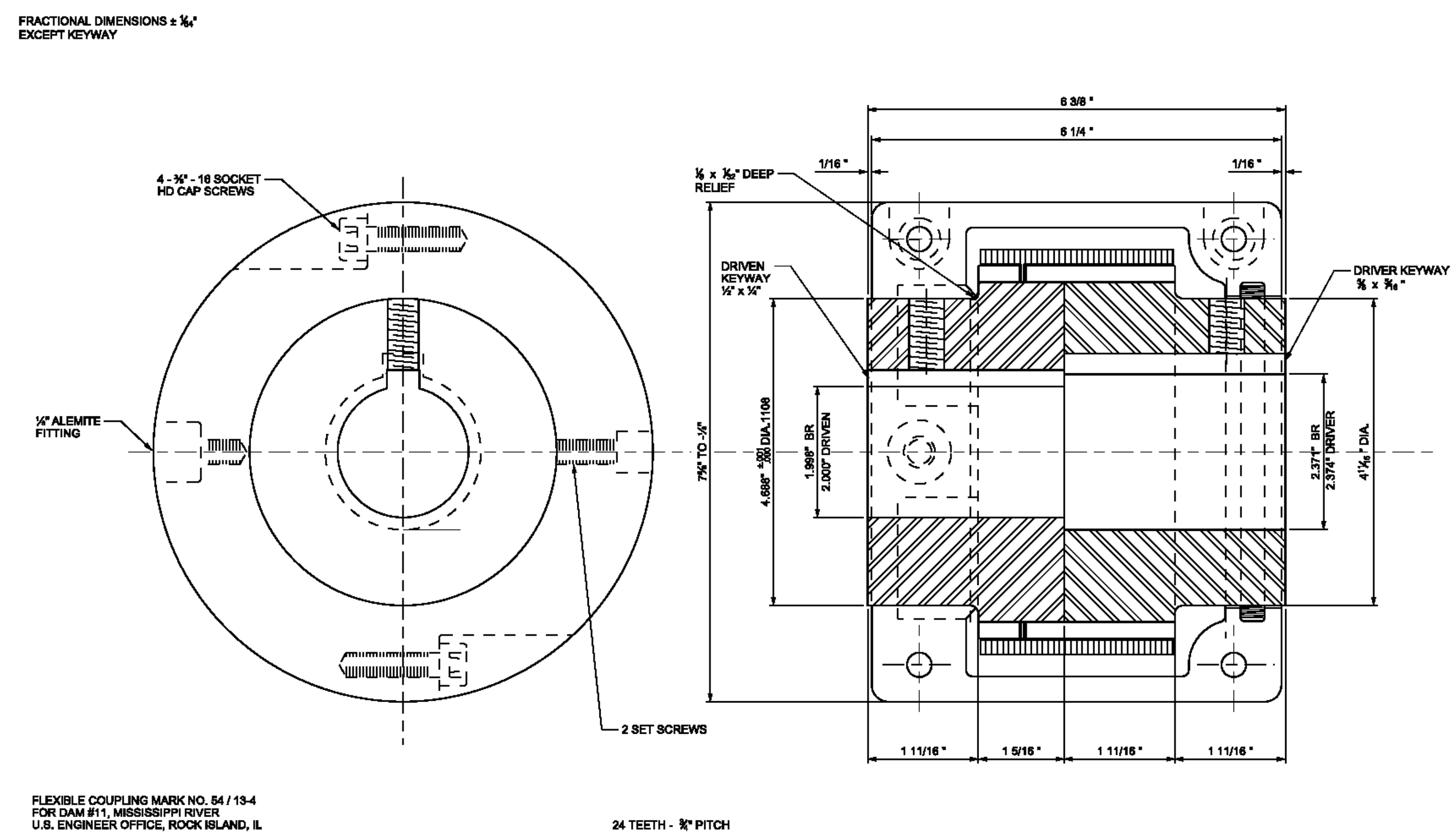
PLATE B-4



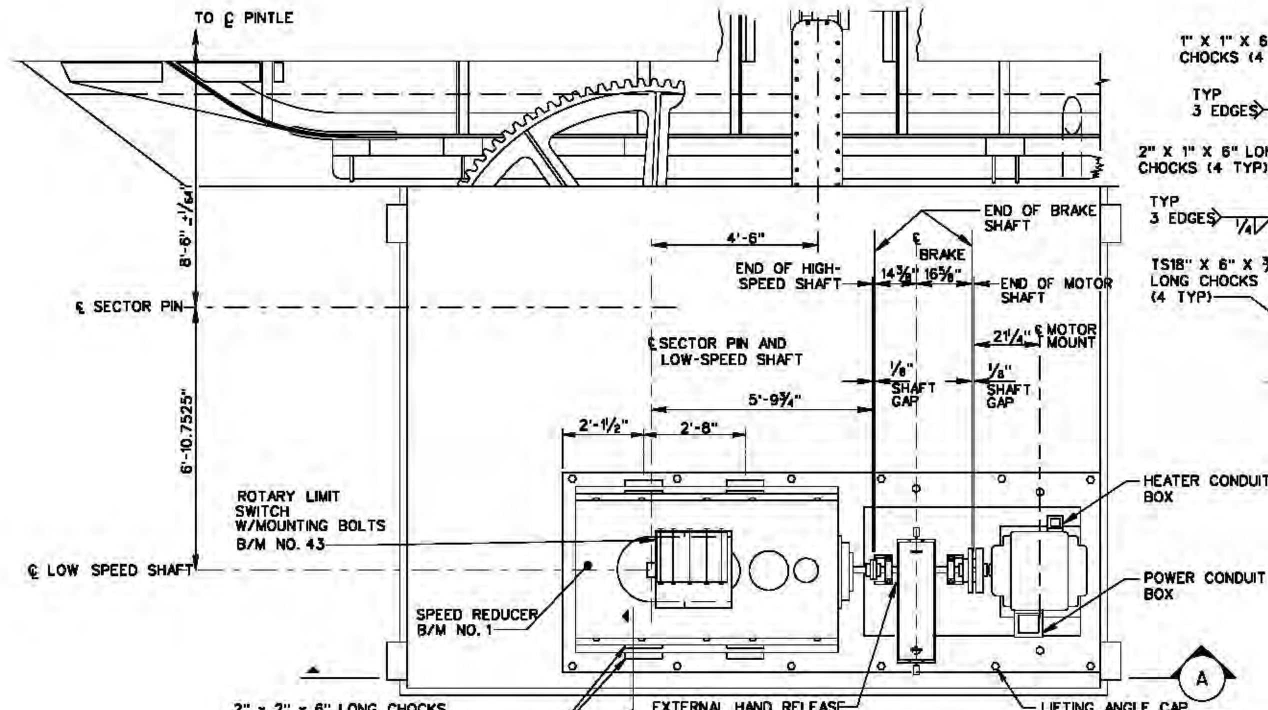
TORQUE LIMITING COUPLING DETAIL

B/M NO. 4

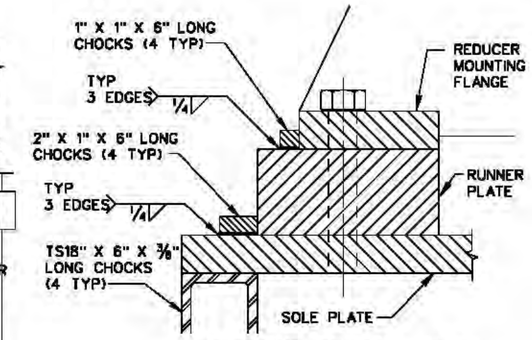
TORQUE LIMITING
COUPLING DETAIL



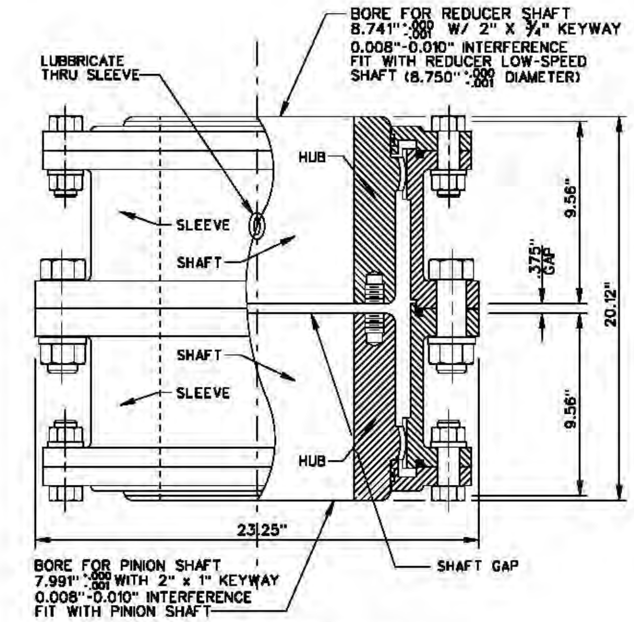
CHAIN COUPLING



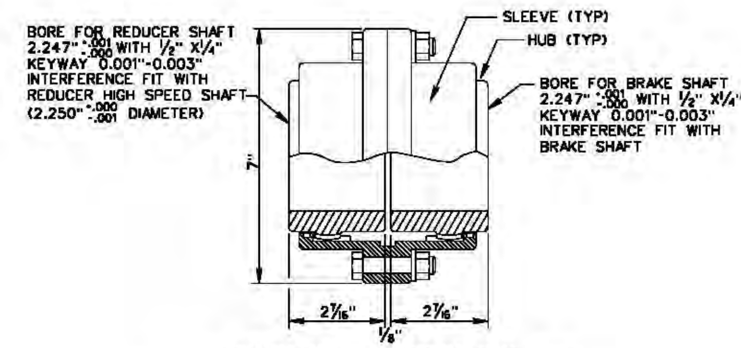
MACHINERY PLAN 1



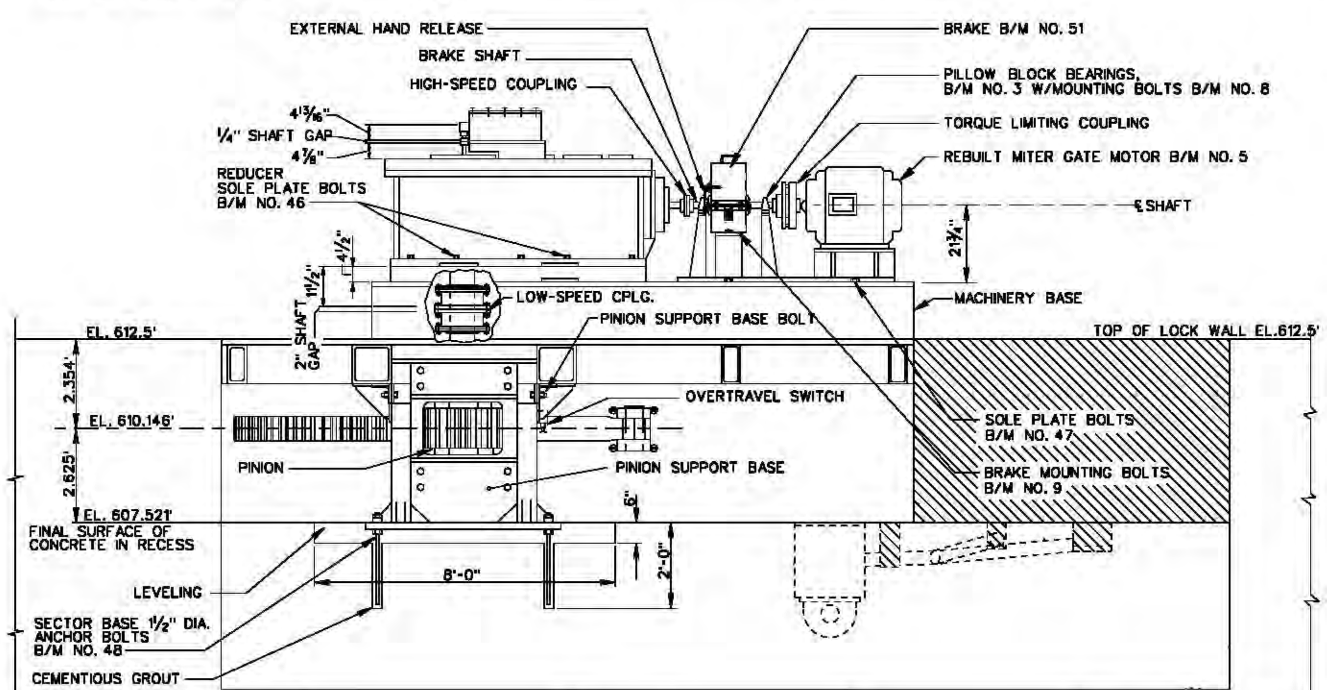
CHOCKS DETAIL 5



LOW SPEED COUPLING 3



HIGH SPEED COUPLING 4

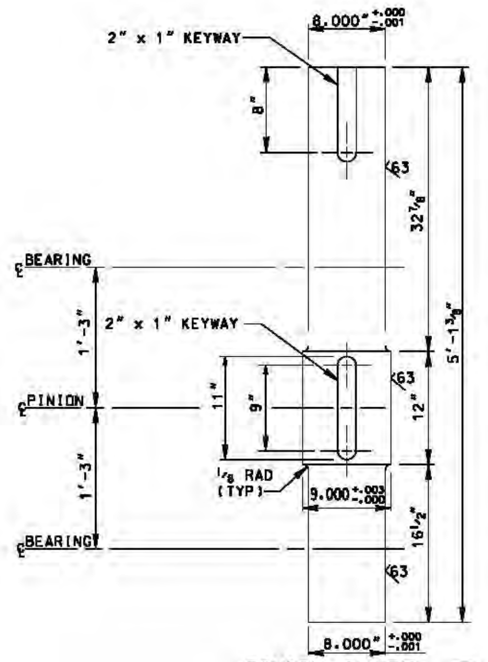


SECTION A

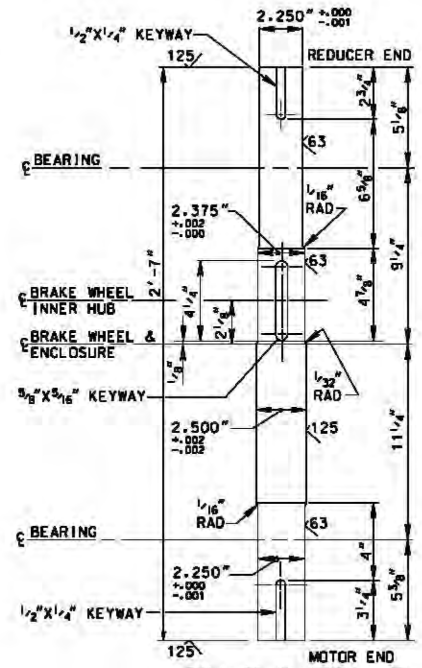
NOTES:

1. A 11½" LOW-SPEED REDUCER SHAFT WITH A 0.375" LOW-SPEED COUPLING SHAFT GAP DIMENSION IS USED AS A DESIGN BASIS. THE CONTRACTOR SHALL COORDINATE WITH THE MACHINERY MANUFACTURERS TO MEET THEIR DIMENSIONAL REQUIREMENTS.
2. MACHINERY LAYOUT IS TYPICAL FOR ALL FOUR MITER GATES, HOWEVER, THIS LAYOUT IS INDICATED FOR MACHINERY SETS LOCATED ON THE RIVER WALL. MACHINERY SETS LOCATED ON THE LAND WALL ARE OPPOSITE HAND, EXCEPT MOTOR POWER CONDUIT BOXES WHICH SHALL NOT BE MIRRORED.

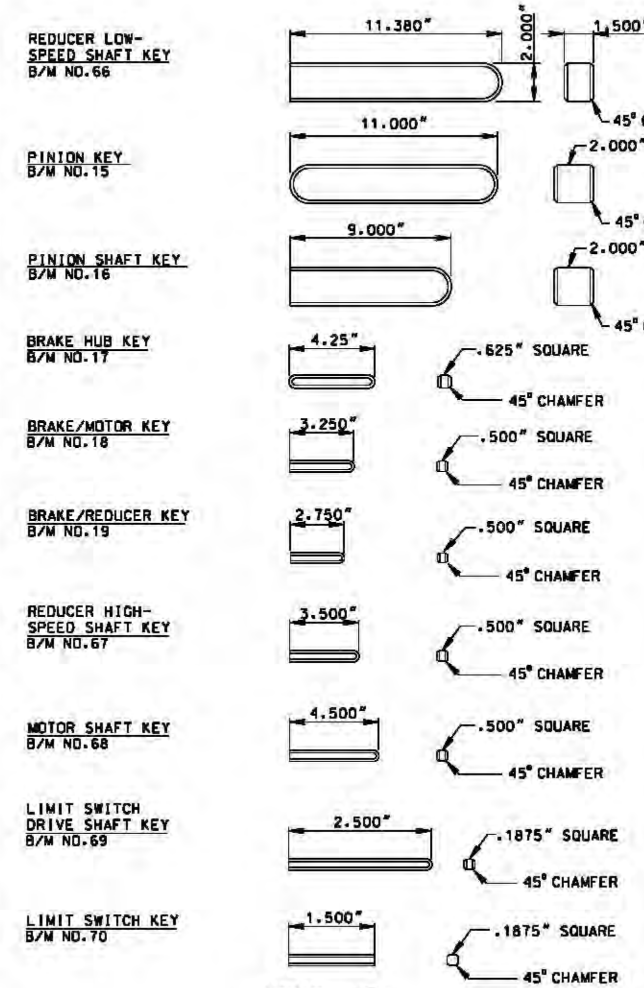
MITER GATES
MACHINERY
PLANS & SECTIONS
MECHANICAL DRIVE



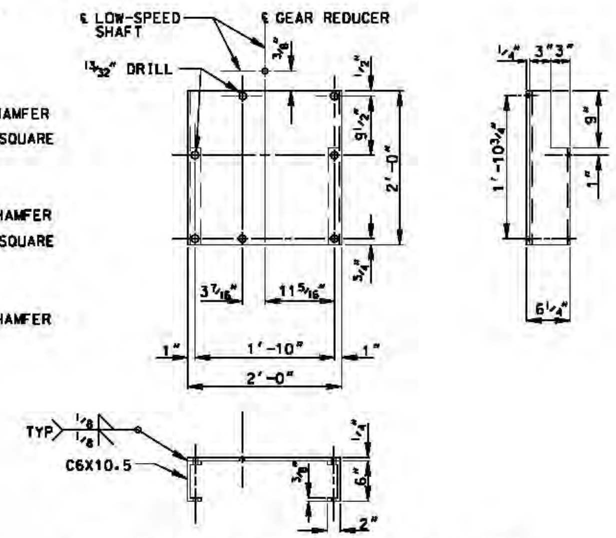
PINION SHAFT 1



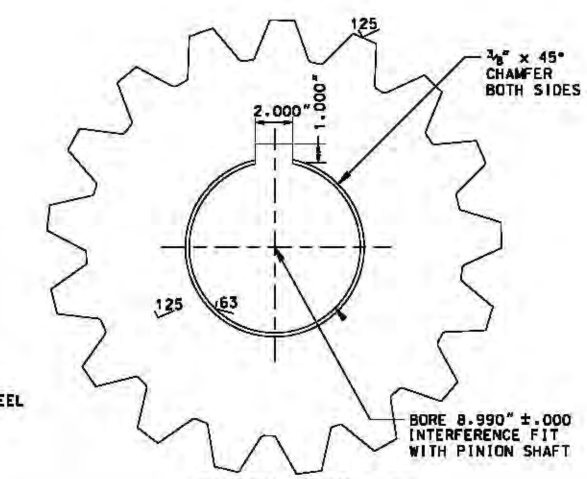
BRAKE SHAFT 2



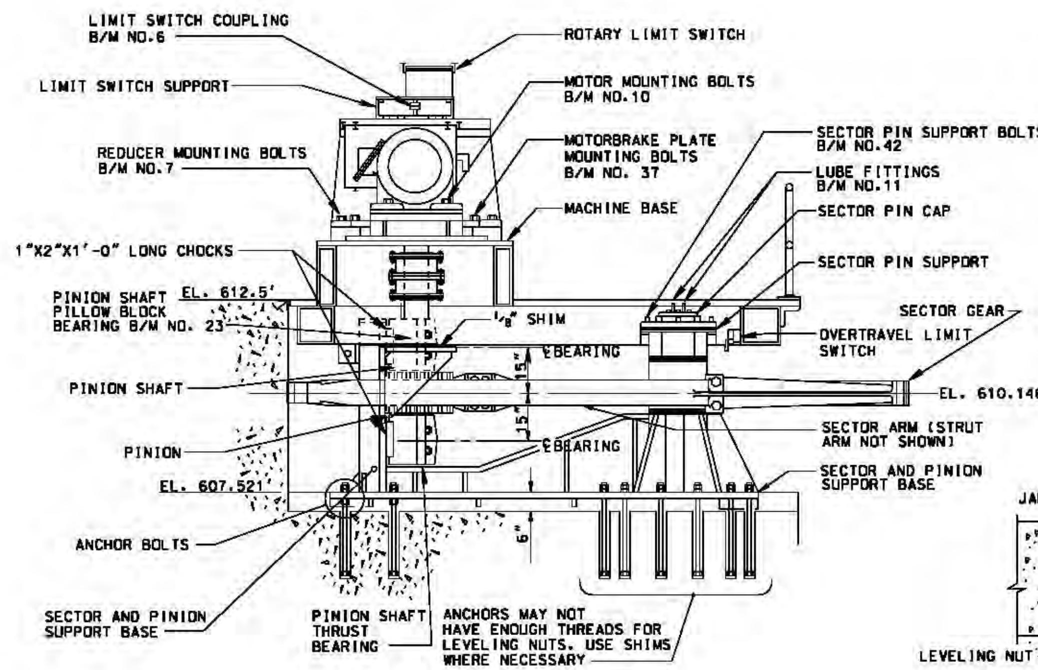
KEYS 3



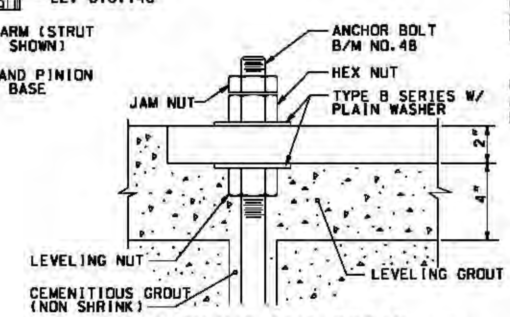
ROTARY LIMIT SWITCH SUPPORT 4



PINION GEAR 5



SECTION 6



LEVELING NUT DETAIL 7

PINION DATA:

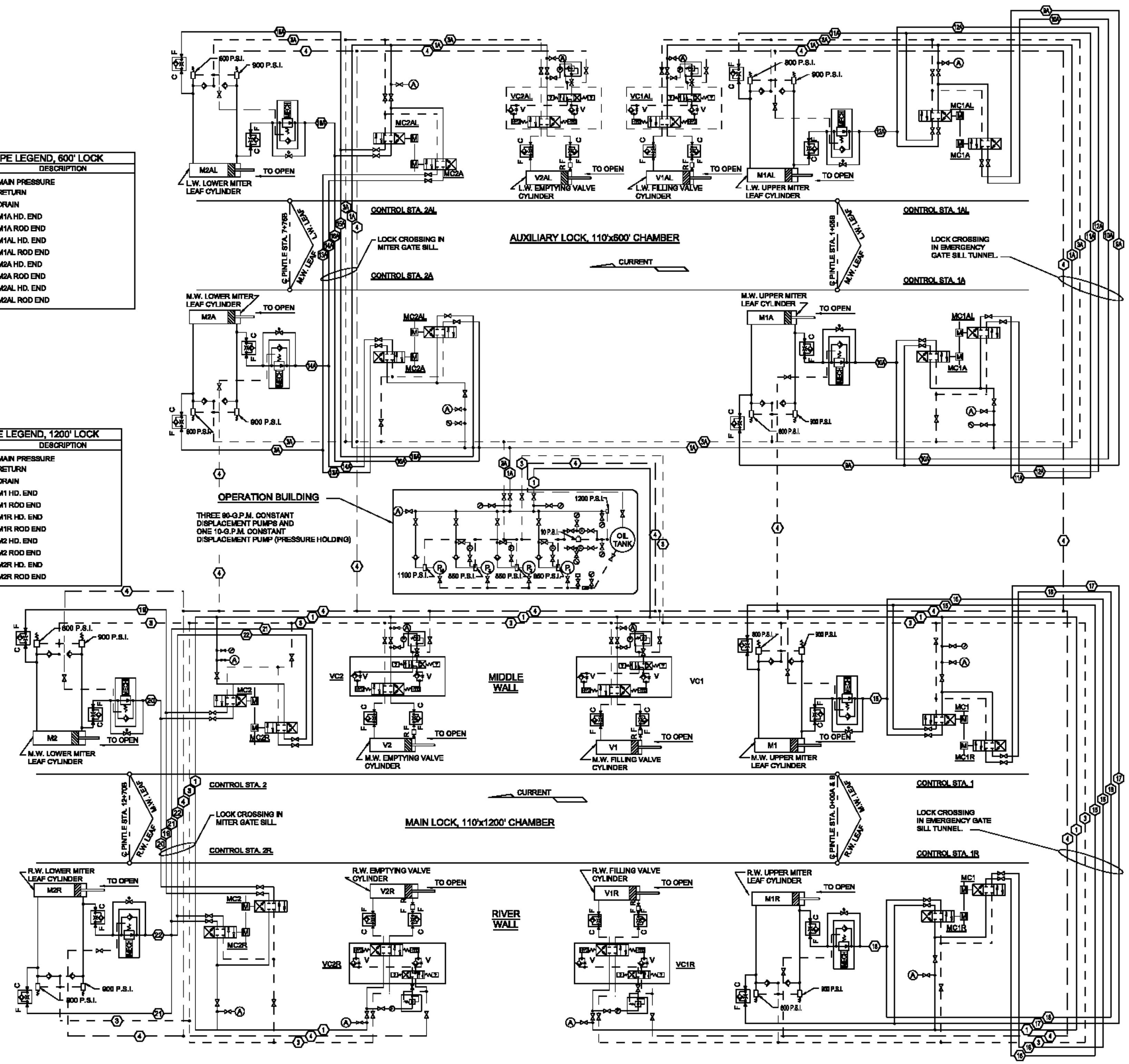
MATERIAL---------AISI 4340 FORGED STEEL
FACE WIDTH-------12.000" ±.010"
HEAT TREATMENT---300 BHN (MIN)
TOOTH FORM-------20° NUTTAL STUB
NUMBER OF TEETH--17
CIRCULAR PITCH---4.0"
DIAMETRAL PITCH--.7854
PITCH DIAMETER---21.645" (THEORETICAL): 21.629"(OPERATING) SCRIBE ON BOTH RIM FACES
OUTSIDE DIAMETER-23.771"
AGMA QUALITY-----8
BACKLASH---------.020"-.040" (ALL CUT IN PINION)

NOTES:

1. ON PINION AND BRAKE SHAFTS UNLESS NOTED OTHERWISE: CHAMFER ENDS 1/16" x 45°; BREAK SHARP EDGES 0.02" MAX.; CORNER RADII 0.02" - 0.04".

MITER GATE
MACHINERY SECTIONS
& DETAILS
MECHANICAL DRIVE

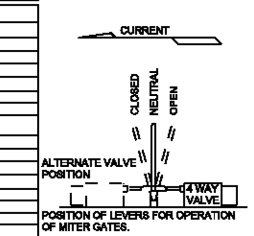
PLATE B-8



PIPE LEGEND, 600' LOCK	
SYMBOL	DESCRIPTION
1A	MAIN PRESSURE
3A	RETURN
4	DRAIN
9A	M1A HD. END
10A	M1A ROD END
11A	M1AL HD. END
12A	M1AL ROD END
13A	M2A HD. END
14A	M2A ROD END
15A	M2AL HD. END
16A	M2AL ROD END

PIPE LEGEND, 1200' LOCK	
SYMBOL	DESCRIPTION
1	MAIN PRESSURE
3	RETURN
4	DRAIN
5	M1 HD. END
16	M1 ROD END
17	M1R HD. END
18	M1R ROD END
19	M2 HD. END
20	M2 ROD END
21	M2R HD. END
22	M2R ROD END

SYMBOL	DESCRIPTION	
MAIN LOCK 1200'		
STATIONS		
1	CONTROLS FOR M.W. & R.W. FILLING & EMPTY'NG VALVES & UPPER MITER LEAVES	
1R	CONTROLS FOR M.W. & R.W. FILLING VALVES & UPPER MITER LEAVES	
2	CONTROLS FOR M.W. & R.W. FILLING & EMPTYING VALVES & LOWER MITER LEAVES	
2R	CONTROLS FOR M.W. & R.W. EMPTYING VALVES & LOWER MITER LEAVES	
S1	SUPPLEMENTAL CONTROLS FOR FILLING VALVES (ELECTRICAL CONTROLS ONLY)	
S2	SUPPLEMENTAL CONTROLS FOR EMPTYING VALVES (ELECTRICAL CONTROLS ONLY)	
CYLINDERS		
M1	OPERATES M.W. UPPER MITER LEAF	
M1R	OPERATES R.W. UPPER MITER LEAF	
M2	OPERATES M.W. LOWER MITER LEAF	
M2R	OPERATES R.W. LOWER MITER LEAF	
V1	OPERATES M.W. FILLING VALVE	
V1R	OPERATES R.W. FILLING VALVE	
V2	OPERATES M.W. EMPTYING VALVE	
V2R	OPERATES R.W. EMPTYING VALVE	
CONTROL VALVES		
MC1	OPERATES M.W. UPPER MITER LEAF	
MC1R	OPERATES R.W. UPPER MITER LEAF	
MC2	OPERATES M.W. LOWER MITER LEAF	
MC2R	OPERATES R.W. LOWER MITER LEAF	
VC1	OPERATES M.W. FILLING VALVE	(SOLENOID OPERATED)
VC1R	OPERATES R.W. FILLING VALVE	(SOLENOID OPERATED)
VC2	OPERATES M.W. EMPTYING VALVE	(SOLENOID OPERATED)
VC2R	OPERATES R.W. EMPTYING VALVE	(SOLENOID OPERATED)
AUXILIARY LOCK 600'		
STATIONS		
1A	CONTROLS FOR L.W. FILLING & EMPTYING VALVES & UPPER MITER LEAVES	
1AL	CONTROLS FOR L.W. FILLING VALVE & UPPER MITER LEAVES	
2A	CONTROLS FOR L.W. FILLING & EMPTYING VALVES & LOWER MITER LEAVES	
2AL	CONTROLS FOR L.W. EMPTYING VALVE & LOWER MITER LEAVES	
CYLINDERS		
M1A	OPERATES M.W. UPPER MITER LEAF	
M1AL	OPERATES L.W. UPPER MITER LEAF	
M2A	OPERATES M.W. LOWER MITER LEAF	
M2AL	OPERATES L.W. LOWER MITER LEAF	
V1AL	OPERATES L.W. FILLING VALVE	
V2AL	OPERATES L.W. EMPTYING VALVE	
CONTROL VALVES		
MC1A	OPERATES M.W. UPPER MITER LEAF	
MC1AL	OPERATES L.W. UPPER MITER LEAF	
MC2A	OPERATES M.W. LOWER MITER LEAF	
MC2AL	OPERATES L.W. LOWER MITER LEAF	
VC1AL	OPERATES L.W. FILLING VALVE	(SOLENOID OPERATED)
VC2AL	OPERATES L.W. EMPTYING VALVE	(SOLENOID OPERATED)



LEGEND

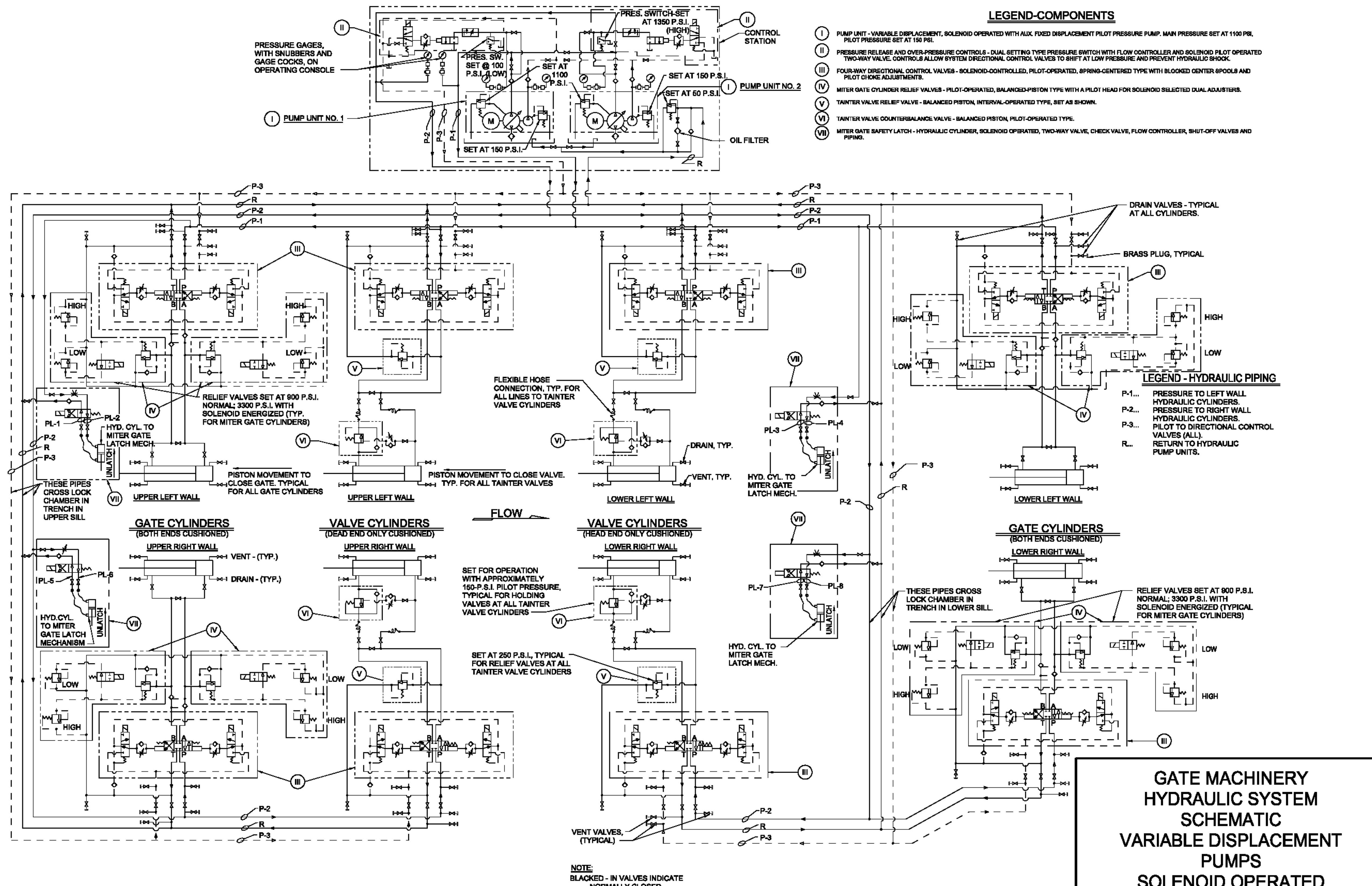
- R.W. RIVER WALL
- M.W. MIDDLE WALL
- L.W. LAND WALL
- MAIN PRESSURE LINE
- MAIN RETURN LINE
- CONTROL LINE
- DRAIN LINE
- 4-WAY CONTROL VALVE, LEVER OPERATED, NO SPRING, CLOSED CENTER, NO DETENTS
- 4-WAY VALVE, SOLENOID PILOT OPERATED, CLOSED CENTER, SPRING CENTERED, PILOT CHOKE ADJUSTMENTS, EXTERNAL PILOT PRESSURE, SUBMERSIBLE
- FLOW CONTROL VALVE, VARIABLE ORIFICE TYPE WITH INTEGRAL CHECK C DIRECTION OF CONTROLLED FLOW, ____ DIRECTION OF FREE FLOW
- PRESSURE-REDUCING VALVE.
- DECELERATION VALVE, CAM AND ROLLER OPERATED, NORMALLY OPEN, WITH INTEGRAL CHECK. VALVE SHALL BE SUITABLE FOR 15-P.S.I. DRAIN LINE PRESSURE WITHOUT LEAKAGE
- SHUT-OFF VALVE
- CHECK VALVE
- RELIEF VALVE
- * COUNTERBALANCE VALVE, F FREE FLOW, R RESTRICTED FLOW
- ACCUMULATOR
- FILTER
- RELIEF VALVE, BALANCED PISTON TYPE
- UNLOADING VALVE, REMOTE CONTROLLED, INTERNALLY DRAINED
- SERVICE PUMPS
- HOLDING PUMP
- STRAINER
- PRESSURE GAGE WITH GLOBE OR NEEDLE VALVE
- COMPOUND GAGE WITH GLOBE OR NEEDLE VALVE
- 1/8" NEEDLE VALVE

NOTE:
PRESSURES SHOWN ARE FOR CLARIFICATION AND MAY VARY WHERE NECESSARY.

RELEIF VALVE ARRANGEMENT SHOWN MUST BE MODIFIED FOR HYDRAULIC CONTROL OF TEMPORAL LOADS; SEE PLATE B-62 FOR MODIFICATION.

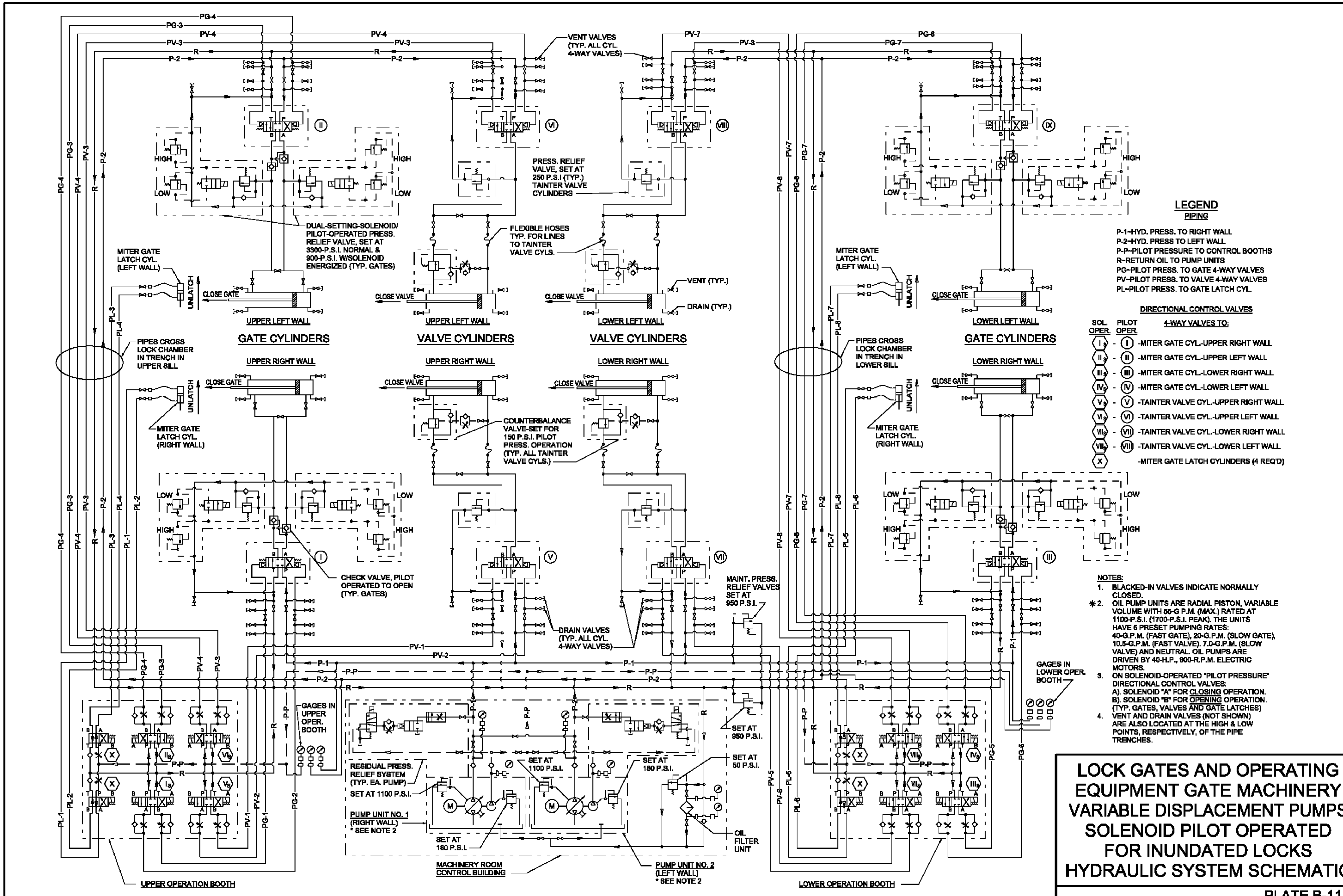
GATE MACHINERY HYDRAULIC SYSTEM SCHEMATIC CONSTANT DISPLACEMENT PUMPS MANUALLY OPERATED

PLATE B-9



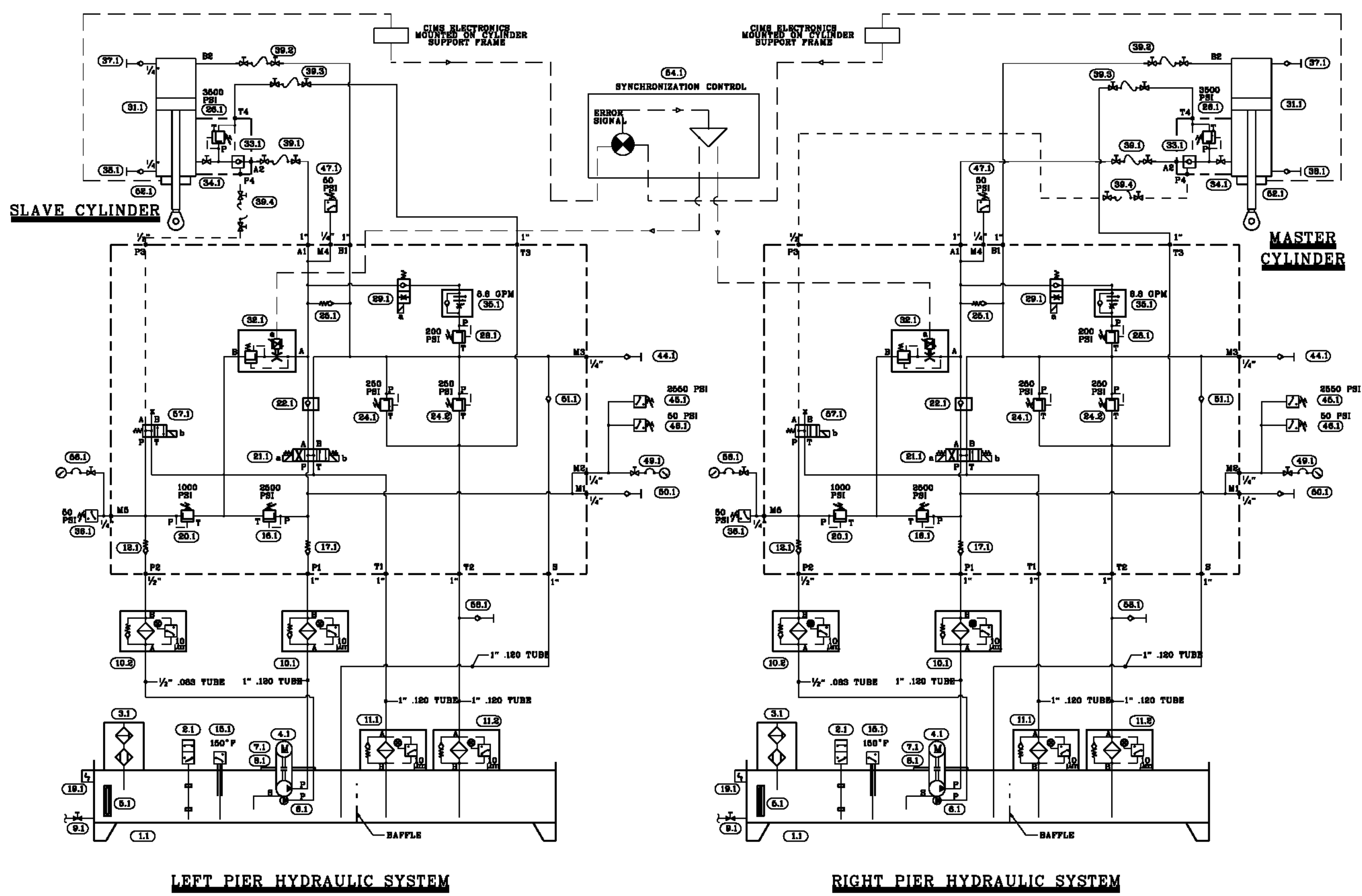
GATE MACHINERY
HYDRAULIC SYSTEM
SCHEMATIC
VARIABLE DISPLACEMENT
PUMPS
SOLENOID OPERATED

PLATE B-10



LOCK GATES AND OPERATING EQUIPMENT GATE MACHINERY VARIABLE DISPLACEMENT PUMPS SOLENOID PILOT OPERATED FOR INUNDATED LOCKS HYDRAULIC SYSTEM SCHEMATIC

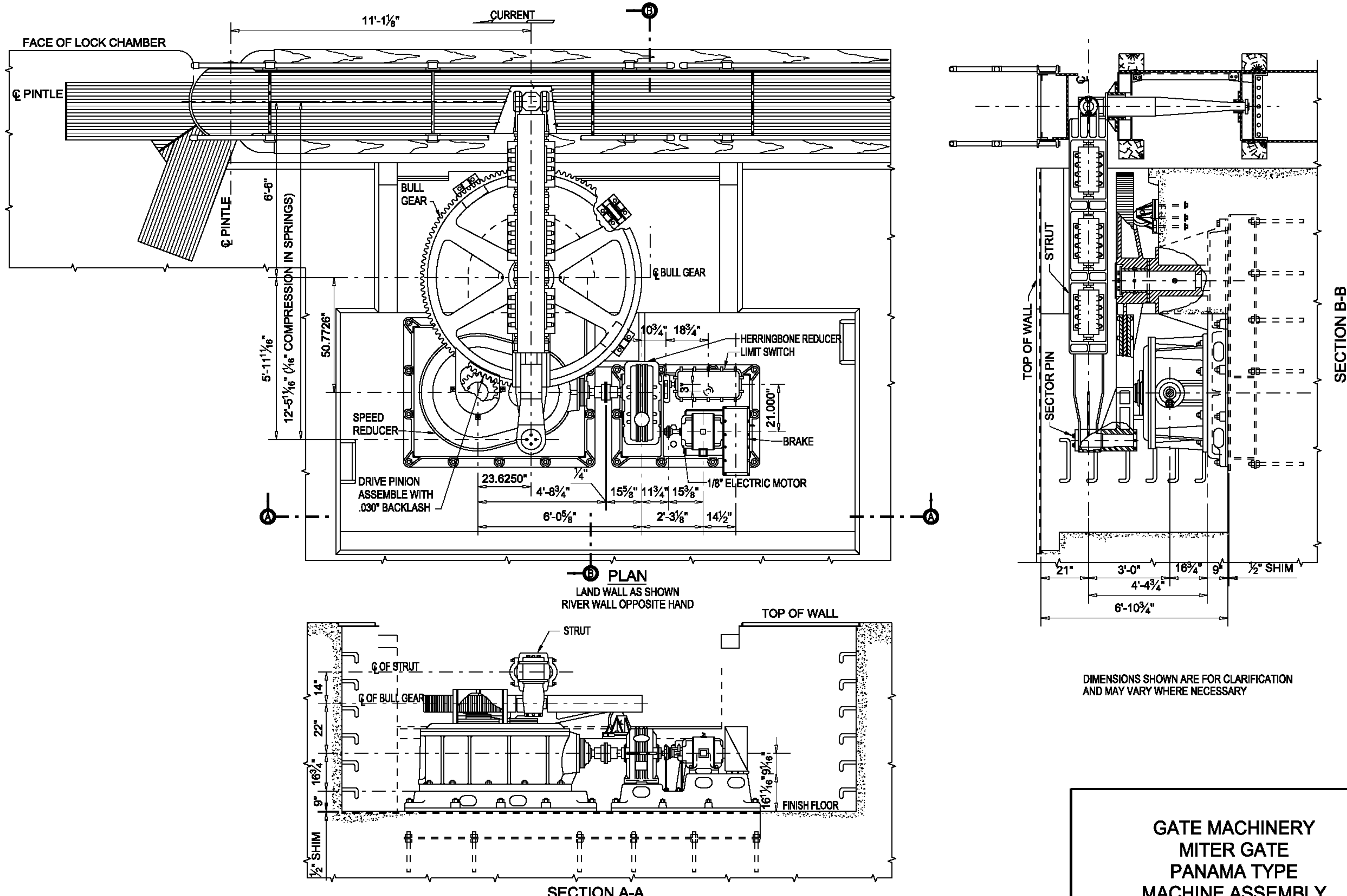
PLATE B-11



ITEM(S)	NAME/SIZE/FUNCTION	MODEL NO.	QTY.
1.1	HYDRAULIC RESERVOIR WITH SIGHT LEVEL INDICATOR	300 GAL. WORKING VOLUME	2
2.1	LEVEL SWITCH TO SWITCH OFF IN CASE OF LOW AND HIGH FLUID LEVELS	500 PU-A31-66	2
3.1	DES CASE HYGROSCOPIC BREATHER AND DRYER.	DES CASE DC-8-2-12	2
4.1	ELECTRIC MOTOR, 15HP, 1800 RPM, 4.1 FOR DUAL PUMP	OPEN VENTILATED DRIP PROOF TYPE	2
5.1	COMBINATION SIGHT LEVEL GAUGE AND THERMOMETER WITH "HIGH OIL" AND "ADD OIL" MARKS	MANUFACTURER'S STANDARD	2
6.1	DUAL HYDRAULIC GEAR PUMP WHICH CONSIST OF A SYSTEM PUMP AND A SMALLER PILOT PUMP. SYSTEM PUMP, 8.7 GPM @ 3500 PSI. PILOT PUMP 3.3 GPM @ 1000 PSI TO OPEN CHECK VALVE 33.1 DURING LOWERING.	1016-10-4 IOH1-R	2
7.1	PUMP-MOTOR COUPLING	MAGNALOY 100	2
8.1	PUMP & MOTOR SUPPORT	BSF 1102-380	2
9.1	SHUT-OFF VALVE FOR DRAINING OF THE HYDRAULIC RESERVOIR.	APPOLO 150- ¼"	2
10.1	PRESSURE FILTER WITH VISUAL CLOGGING INDICATOR, BYPASS CHECK AND PRESSURE SWITCH TO INDICATE ON THE CONTROL PANEL PRESSURE FILTER'S CLOGGED CONDITION.	NSFBN/HC160 G10040/L24-12-BYP	2
10.2	PRESSURE FILTERS WITH VISUAL CLOGGING INDICATOR, BYPASS CHECK AND PRESSURE SWITCH TO INDICATE ON THE CONTROL PANEL PRESSURE FILTER'S CLOGGED CONDITION.	LFBN/HC G10010/L24-12-BYP	2
11.1 & 11.2	RETURN LINE FILTERS WITH BYPASS CHECK, VISUAL CLOGGING INDICATOR AND PRESSURE SWITCH TO INDICATE ON THE CONTROL PANEL RETURN LINE FILTER'S CLOGGED CONDITION.	RFBN/HC240 G10002.1/L24-12	4
12.1	CHECK VALVE TO PREVENT BACKFLOW INTO THE PILOT PUMP.	M-SR8KE30-1X/5	2
15.1	TWO SET POINT THERMOMETER	PU-A31-166 604150	2
16.1	SYSTEM PUMP RELIEF VALVE	DBDS10K1X/200/12	2
17.1	CHECK VALVE TO PREVENT BACKFLOW INTO THE SYSTEM PUMP.	M-SR10KE30-1X/5	2
19.1	GROUNDING CONNECTION		2
20.1	PRESSURE RELIEF VALVE FOR PILOT PUMP	DBDS6K1X/100/12	2
21.1	FOUR-WAY, THREE-POSITION DIRECTIONAL VALVE TO CONTROL THE DIRECTION OF CYLINDER MOVEMENT.	4WE10H3X/AW120N9Z45	2
22.1	CHECK VALVE TO PREVENT REGENERATIVE FLOW FROM GOING TO SUMP DURING LOWERING OF GATE.	Z1S10C1-3X/V	2
24.1 & 24.2	PRESSURE RELIEF VALVES TO LIMIT CYLINDER TOP CHAMBER PRESSURE TO 250 PSI IN CASE CLOSURE END STOP FAILS OR IF DIRECTIONAL VALVE 29.1 FAILS TO SHIFT DURING LOWERING OF GATE	DBDS6K1X/50/12	4
25.1	CHECK VALVE CONNECTING CYLINDER TOP AND BOTTOM CHAMBERS SO THAT THE PRESSURE IN THE BOTTOM CHAMBER IS NEVER LESS THAN THE PRESSURE IN THE UPPER CHAMBER, TO LIMIT THE BUCKLING LOAD ON THE PISTON RODS TO 250PSI MULTIPLIED BY πD²/4, WHERE D=ROD DIAMETER.	M-SR8KE02-1X/5	2
26.1	PRESSURE RELIEF VALVE CONNECTED TO CYLINDER BOTTOM CHAMBER TO GUARD AGAINST EXCESSIVE CYLINDER BOTTOM CHAMBER PRESSURE RISE CAUSED BY TEMPERATURE INCREASE, WHILE THE GATE IS OPEN.	DBDS8K1X/315/12	2
28.1	BACK-PRESSURE VALVE TO ASSIST CONTROL LOWERING OF GATE.	DBDS10K1X/50/12	2
29.1	TWO-WAY SOLENOID-OPERATED DIRECTIONAL CONTROL VALVE TO CONNECT CYLINDER TOP AND BOTTOM CHAMBERS DURING LOWERING OF THE GATE.	MODULAR CONTROLS SV1-16-C-0-120AG	2
31.1	HYDRAULIC CYLINDERS.	SEE SPECS.	2
32.1	2-WAY PROPORTIONAL FLOW CONTROL VALVE TO CONTROL CYLINDER SYNCHRONIZATION BY BLEEDING OFF HYDRAULIC FLUID DURING RAISING AND LOWERING.	2FRE10-3X/60LBKV	2
33.1	PILOT-OPERATED CHECK VALVE, MANIFOLD MOUNTED TO THE CYLINDER BOTTOM CHAMBER TO HOLD THE CYLINDER IN RAISED POSITION IN CASE OF PIPE RUPTURE AND TO CONTROL CYLINDER DRIFT.	SL20PA1-3X/12	2
34.1	CYLINDER-MOUNTED MANUAL SHUT-OFF VALVE TO FACILITATE HOLDING OF THE GATE AFTER PIPE RUPTURE.	[illegible]	2
35.1	ADJUSTABLE FLOW CONTROL VALVE TO CONTROL LOWERING SPEED OF THE GATE.	2FRM10-3X/50LB	2
36.1	PRESSURE SWITCH TO DETECT PILOT PUMP FAILURE.	HED10A4X/50/12	2
37.1	TEST AND BLEED PORT FOR CYLINDER TOP CHAMBER.	BETA MICRO 20.10.50.V	2
38.1	TEST AND BLEED PORT FOR CYLINDER BOTTOM CHAMBER.	BETA MICRO 20.10.50.V	2
39.1, 39.2, 39.3 & 39.4	FLEXIBLE HOSES EQUIPPED WITH SHUT-OFF VALVES.	SAE 100R2 HOSE SEE SPECS	8
44.1	TEST PORT	BETA MICRO 20.10.50.V	2
45.1	PRESSURE SWITCH TO INDICATE HIGH SYSTEM PUMP OUTPUT PRESSURE.	HED10A4X/350/12	2
46.1	PRESSURE SWITCH TO DETECT SYSTEM PUMPS FAILURE BY MONITORING THE DEVELOPMENT OF PUMP PRESSURE WITHIN A SPECIFIED TIME AFTER "START MOTOR" PUSH BUTTON IS PRESSED.	HED10A4X/50/12	2
47.1	PRESSURE SWITCH TO AUTOMATICALLY SWITCH-OFF THE WORKING PUMPS IN CASE OF HOSE/PIPE RUPTURE.	HED10A4X/50/12	2
49.1	PRESSURE GAUGE WITH ISOLATOR (SHUT-OFF) VALVE TO MONITOR SYSTEM PUMP PRESSURE	GSA 4/3000 832350	2
50.1	TEST PORT	BETA MICRO 20.10.50.V	2
51.1	CHECK VALVE TO PREVENT BACKFLOW TO THE SUCTION LINE. SUCTION LINE FEEDS CYLINDER UPPER CHAMBER TO PREVENT CYLINDER CAVITATION.	M-SR20KE00-1X/5	2
52.1	POSITION TRANSDUCER TO MONITOR CYLINDER STROKE (GATE POSITION).	HYDRAUDYNE CMS SYSTEM	2
54.1	SYNCHRONIZATION CONTROL INCLUDES ALL ELECTRONIC CONTROL EQUIPMENT REQUIRED TO SYNCHRONIZE THE TWO CYLINDERS. SEE SPECIFICATIONS FOR DETAILED DESCRIPTION OF CONTROL SYSTEM.	SEE SPECS	1
56.1	PRESSURE GAUGE WITH ISOLATOR (SHUT-OFF) VALVE TO MONITOR PILOT PUMP PRESSURE	GSA 4/1500 832349	2
57.1	4-WAY, 2-POSITION SOLENOID-ACTIVATED DIRECTIONAL VALVE TO DIRECT PILOT PUMP FLOW TO THE PILOT-OPERATED CHECK VALVE 33.1 DURING LOWERING OF THE GATE.	4WE6D6X/AW120-60NZ45/5	2
58.1	ONE-HALF OF QUICK CONNECT, FEMALE THREADED NIPPLE FOR FILLING OF SYSTEM.	PARKER HS-63	2

CONNECTION TYPE	
A1 & A2	1" SAE 6000 PSI 4-BOLT FLANGE TO CYLINDER LOWER CHAMBER.
B1 & B2	1" SAE 6000 PSI 4-BOLT FLANGE TO CYLINDER UPPER CHAMBER.
T3 & T4	1" SAE 6000 PSI 4-BOLT FLANGE TO CYLINDER MANIFOLD ON LOWER CHAMBER.
P3 & P4	½" SAE 6000 PSI 4-BOLT FLANGE TO CYLINDER MANIFOLD ON LOWER CHAMBER.
P1, T1 & T2, S	SAE-16 STRAIGHT THREAD PORT
P2	SAE-8 STRAIGHT THREAD PORT
M1-M5	SAE-4 STRAIGHT THREAD PORT

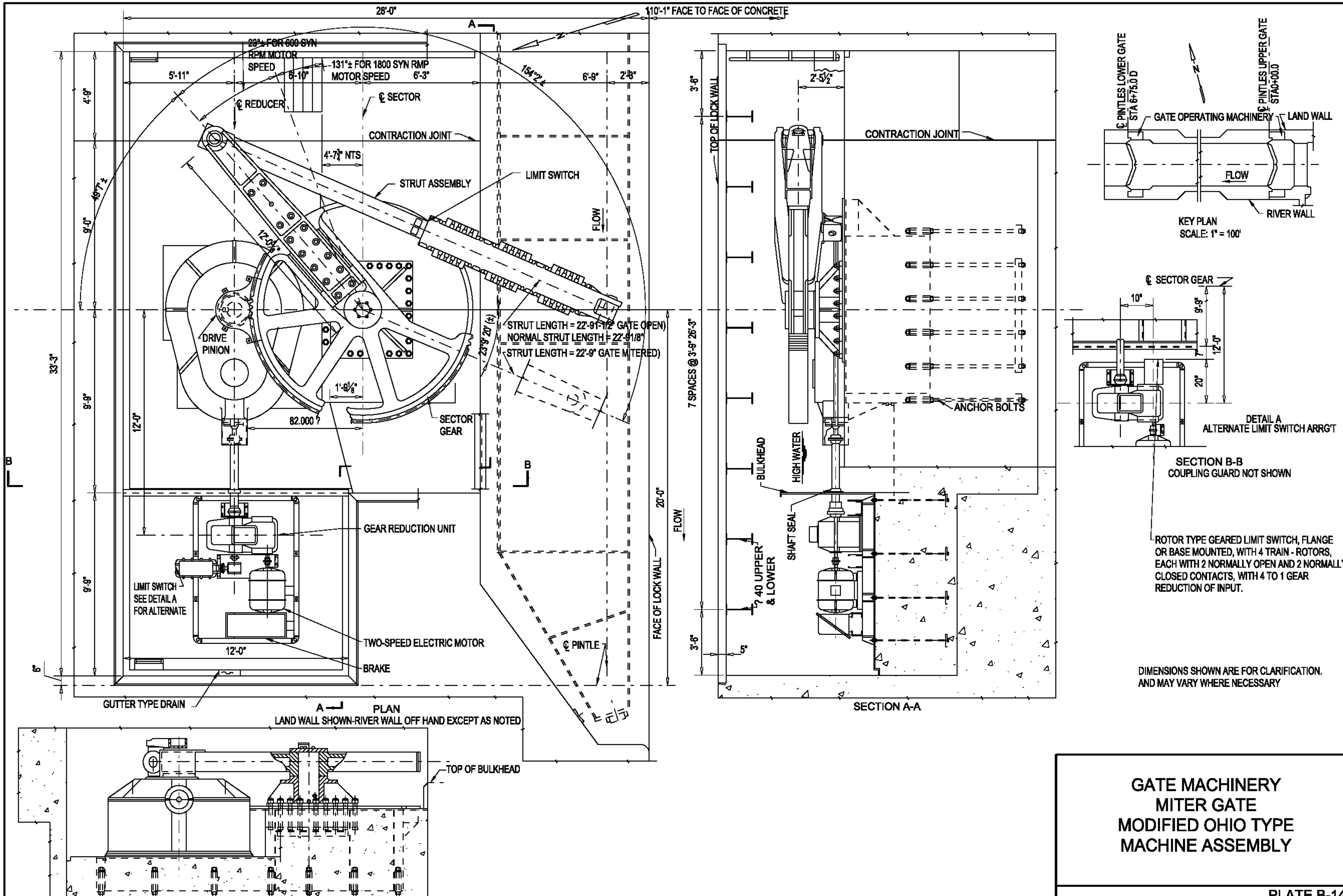
TAINTER GATE MACHINERY
HYDRAULIC CYLINDER
HYDRAULIC SCHEMATIC



DIMENSIONS SHOWN ARE FOR CLARIFICATION AND MAY VARY WHERE NECESSARY

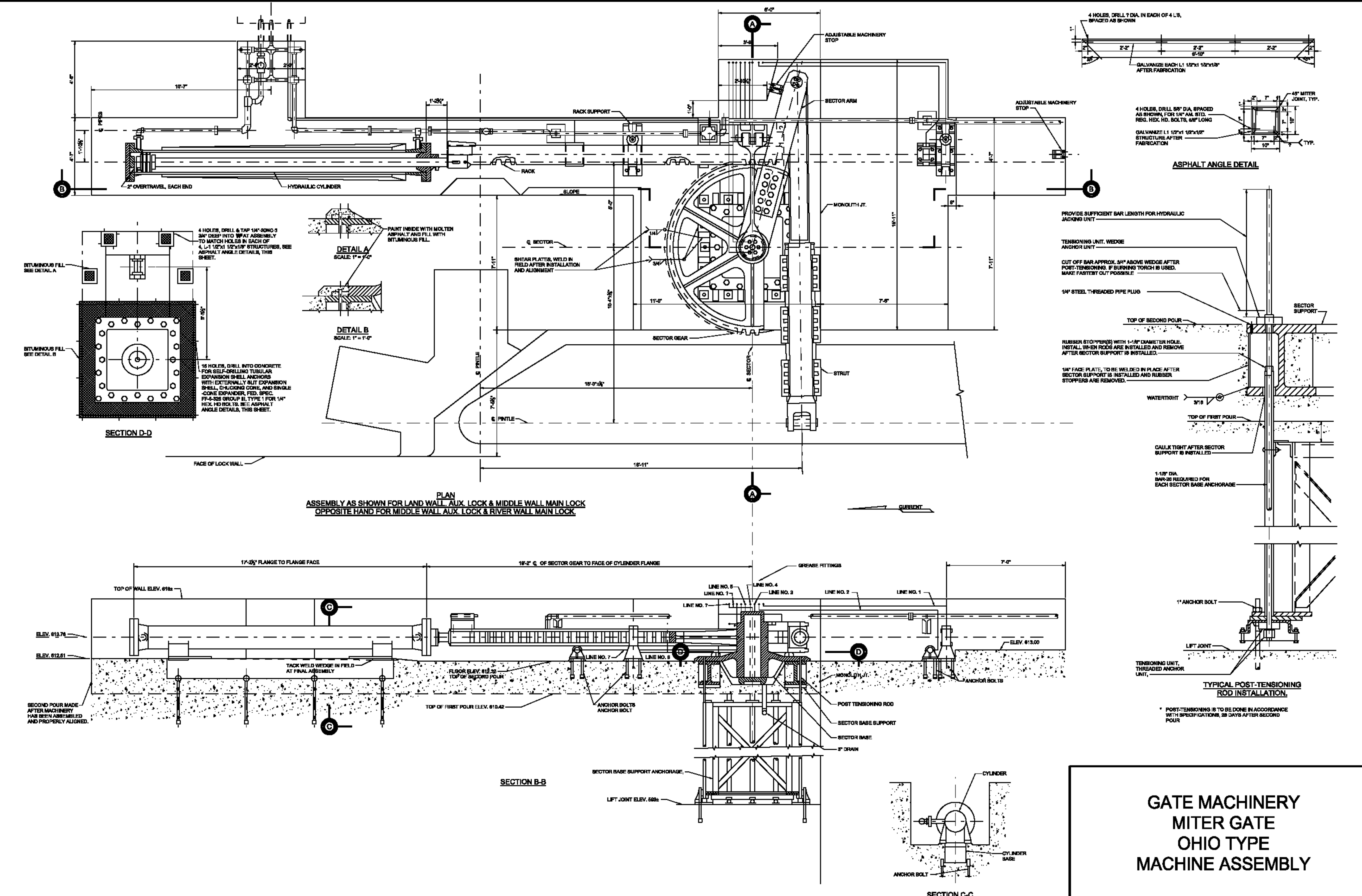
GATE MACHINERY
MITER GATE
PANAMA TYPE
MACHINE ASSEMBLY

PLATE B-13



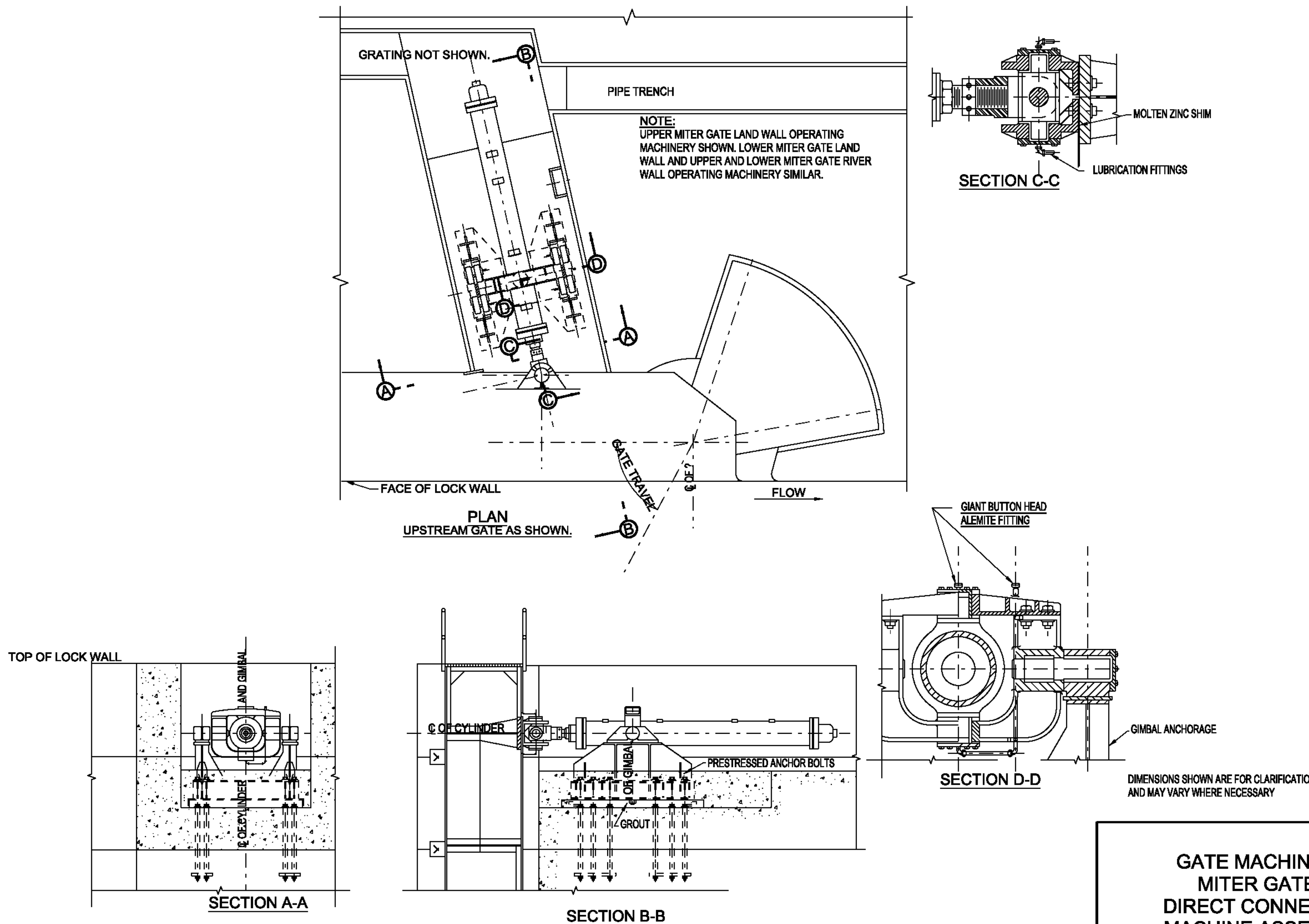
GATE MACHINERY
MITER GATE
MODIFIED OHIO TYPE
MACHINE ASSEMBLY

PLATE B-14

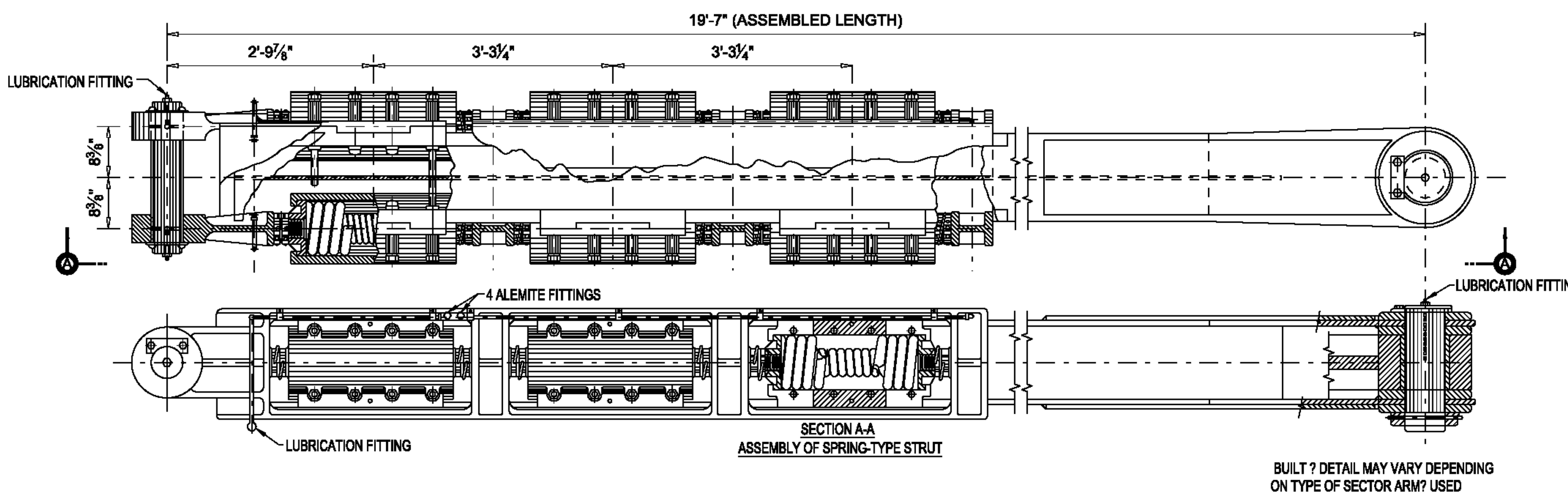


GATE MACHINERY
MITER GATE
OHIO TYPE
MACHINE ASSEMBLY

PLATE B-15

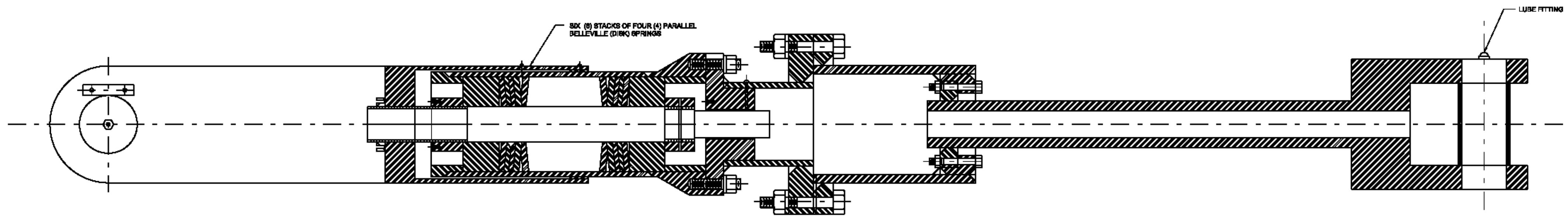


GATE MACHINERY
MITER GATES
DIRECT CONNECTED
MACHINE ASSEMBLY

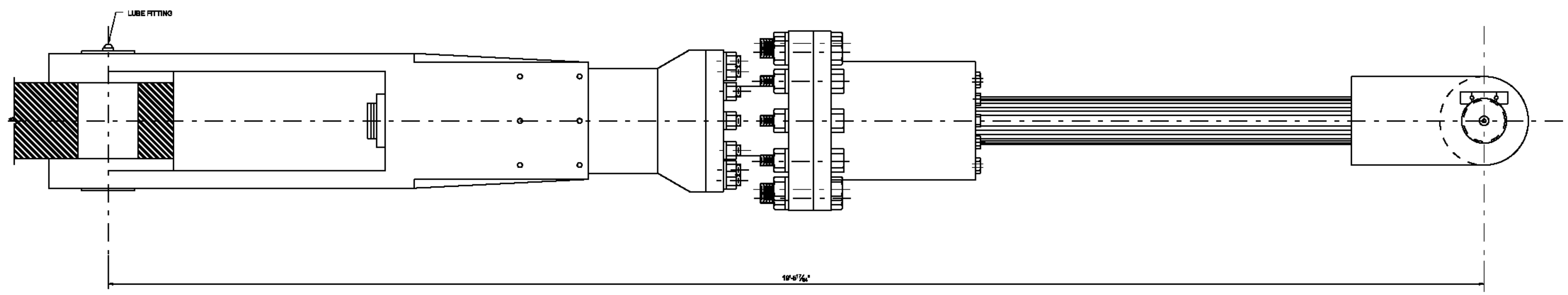


DIMENSIONS SHOWN ARE FOR CLASSIFICATION
AND MAY VARY WHERE NECESSARY

GATE MACHINERY
MITER GATE
SPRING-TYPE
GATE-STRUT



SECTIONAL PLAN



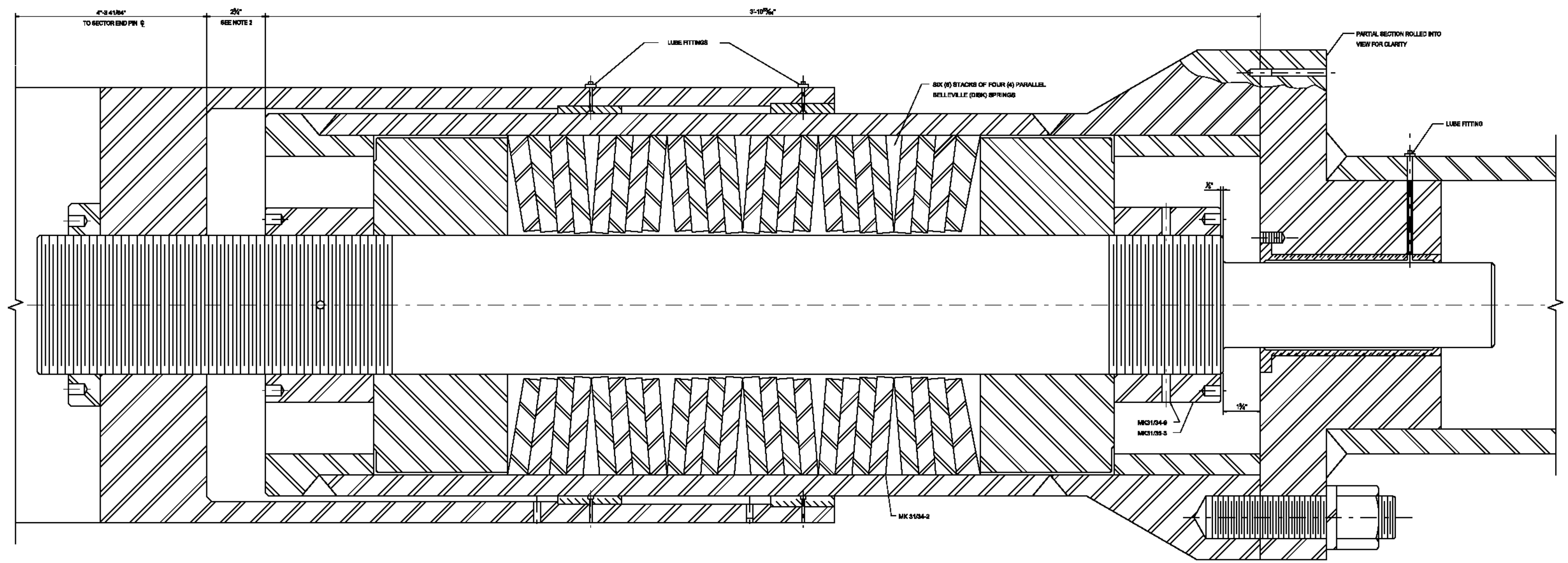
ELEVATION

MITER GATE STRUT ASSEMBLY

SCALE: 1 1/2" = 1'-0"

GATE MACHINERY
MITER GATE
BELLEVILLE (DISK)
SPRING-TYPE GATE STRUT
STRUT ASSEMBLY

PLATE B-18



NOTE: THIS ASSEMBLY IS SHOWN WITH 21/32" SPRING PRESET APPLIED

SECTIONAL ELEVATION

STRUT SPRING ASSEMBLY

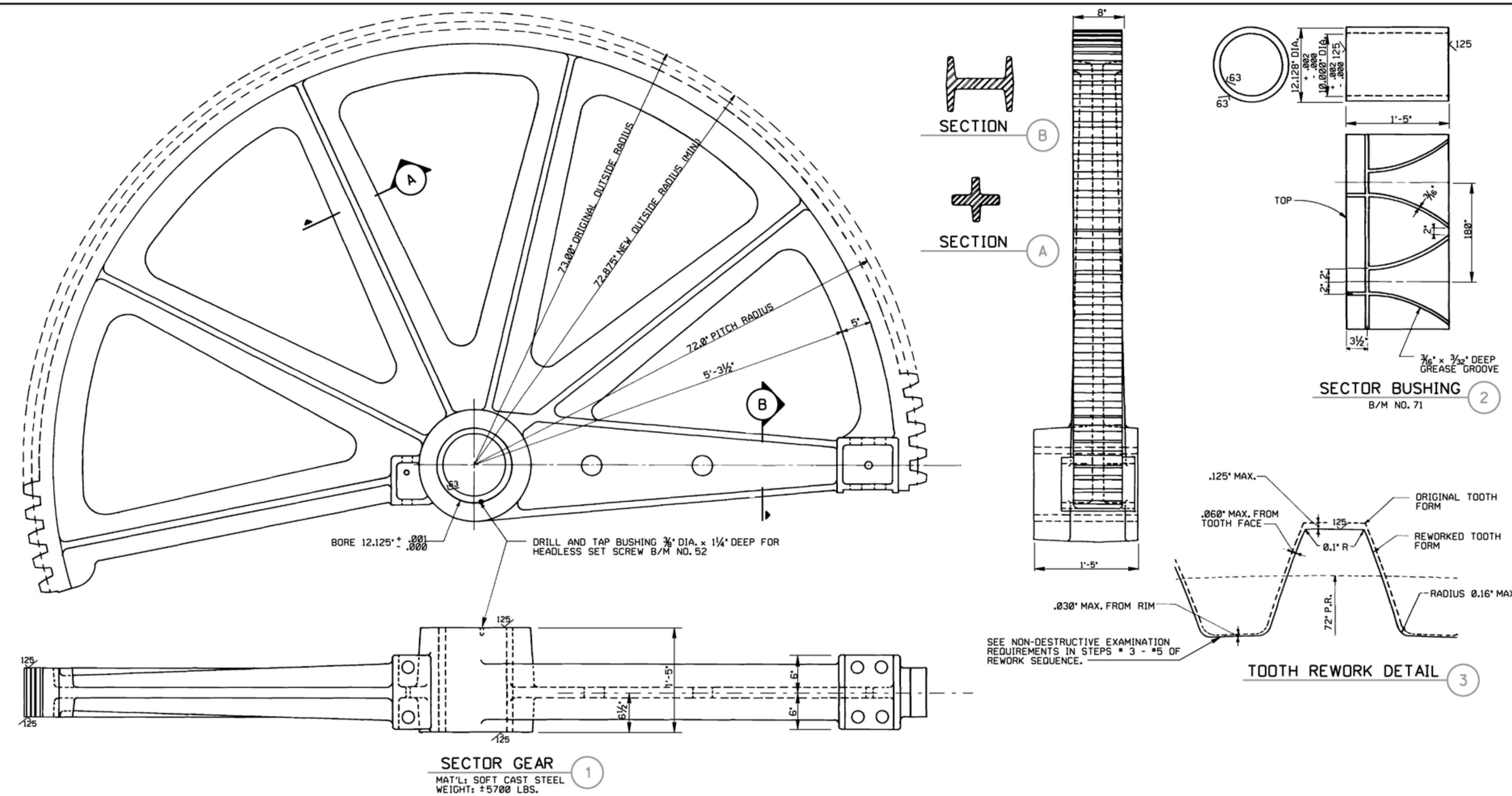
SCALE: 6" = 1'-0"

0 1" 2" 3" 6"

NOTES:

1. SIX STACKS OF 4 PARALLEL DISK SPRINGS REQUIRED PER STRUT ASSEMBLY.
2. DIMENSION SHOWN IS DESIGN VALUE. ADJUST AT INSTALLATION AS REQUIRED BETWEEN A MINIMUM OF 1 3/4" AND 3 3/4" MAXIMUM.
3. TRADE NAMES AND CATALOG NUMBERS SHOWN INDICATE EXAMPLES OF STANDARD ITEMS. OTHER MANUFACTURERS-PRODUCTS MAY BE SUBSTITUTED IF APPROVED EQUAL.

GATE MACHINERY
MITER GATE
BELLEVILLE (DISK) SPRING
TYPE GATE STRUT
SPRING ASSEMBLY



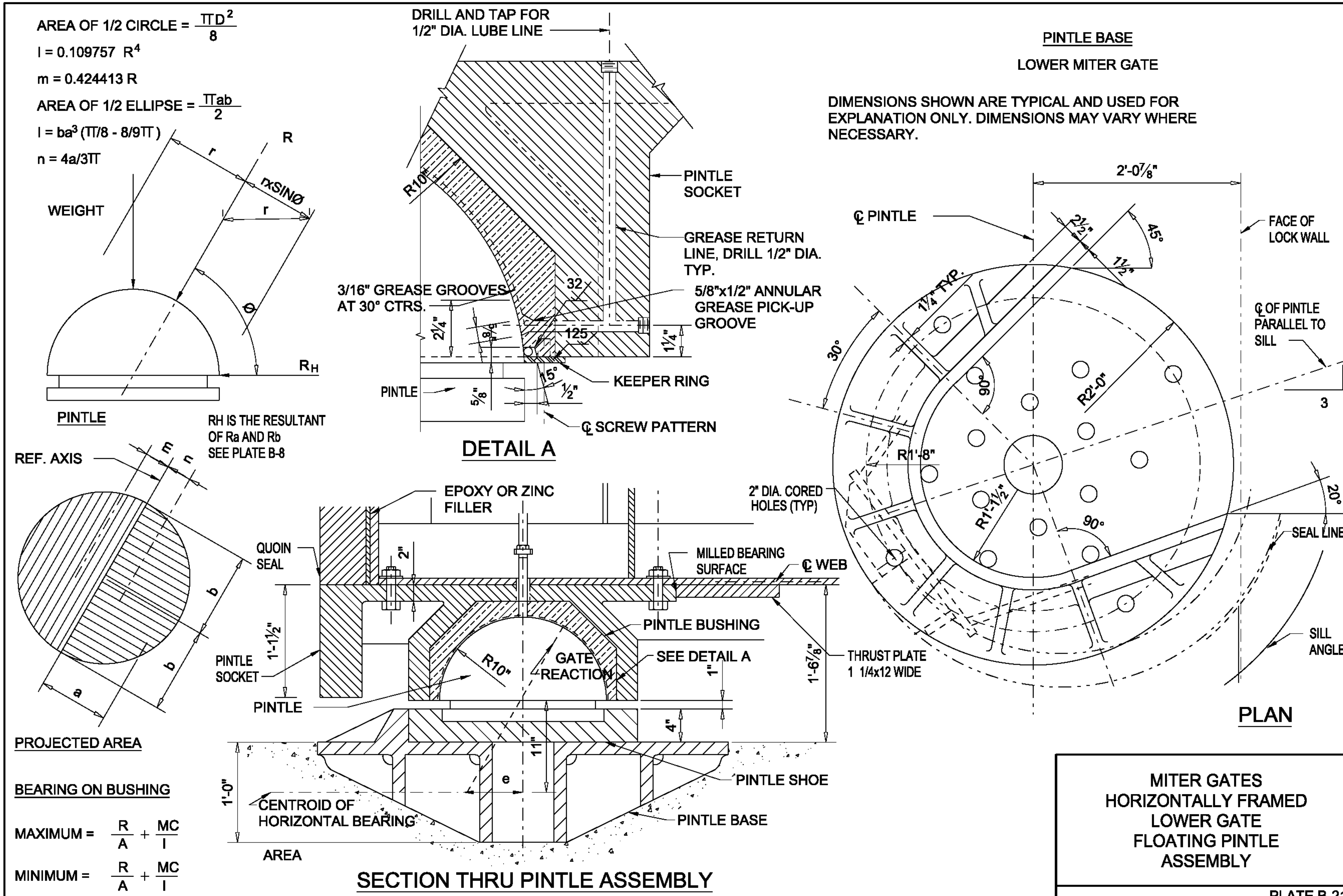
SECTOR GEAR REWORK SEQUENCE:

1. REMOVE EXISTING BRONZE BUSHING.
2. BLAST CLEAN ENTIRE GEAR INCLUDING BORE.
3. VERIFY ORIGINAL GEAR DIMENSIONS AND TOOTH PROFILE GIVEN ON REFERENCE DRAWING R55. CHECK FOR VARIATIONS IN RUNOUT, PITCH DIAMETER, AND CIRCULAR PITCH IN ACCORDANCE WITH AGMA 2000-A88. SUBMIT DETAILED REPORT SHOWING ACTUAL DIMENSIONS. SUBMIT DETAILED REWORK CRITERIA IF IT VARIES FROM THE PROPOSED CRITERIA IN ITEM 6.
4. INSPECT GEAR TEETH FOR CRACKS BY MAGNETIC PARTICLE INSPECTION METHOD OR DYE PENETRANT METHOD. INSPECT ENTIRE REMAINDER OF GEAR BY DYE PENETRANT METHOD. SUBMIT DETAILED RESULTS WITH PHOTOGRAPHS (SEE SECTION 15110). SEE SPECIFICATIONS FOR NDE ACCEPTANCE REQUIREMENTS.
5. GOVERNMENT PERMISSION IS REQUIRED PRIOR TO RECUTTING ANY GEAR. PERMISSION WILL BE GRANTED BASED ON THE DIMENSIONAL AND NON-DESTRUCTIVE EXAMINATION RESULTS.
6. REMACHINE ALL GEARS IDENTICALLY AS FOLLOWS:
   - A. ESTABLISH A DATUM SIDE. THE SAME FOR ALL GEARS.
   - B. TRUE HUB FACES BY REMOVING APPROXIMATELY .010" OR A MINIMUM OF 90% CLEANUP.
   - C. TURN RIM FACES TRUE BY REMOVING APPROXIMATELY .010" OR A MINIMUM OF 90% CLEANUP.
   - D. SCRIBE OPERATING PITCH DIAMETER ON RIM FACE.
   - E. TURN O. D. TO 145.750" + 000" - .010".
   - F. REBORE TO 12.125" + .001" - .000" OR 100% CLEANUP. MANUFACTURE NEW BUSHING AS SHOWN.
   - G. FINISH BORE AND INSTALL NEW BUSHING.
   - H. RECUT TEETH AS SHOWN IN DETAIL. DO NOT UNDERCUT RIM MORE THAN .030".
   - I. WELD REPAIR POROSITY ENCOUNTERED IN GEAR TEETH (BASED ON INSPECTION RESULTS FROM STEP 4). SUBMIT REPAIR PROCEDURE FOR APPROVAL PRIOR TO PERFORMING ANY REPAIR.

DATA ON RECUT SECTOR GEARS:

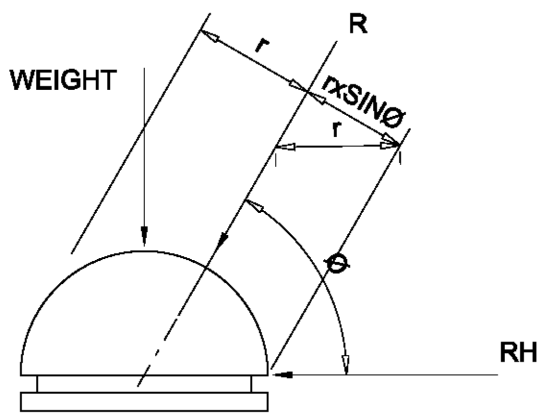
PITCH RADIUS - 72.0"
FACE WIDTH - 8.0"
TOOTH FORM - 20° NUTTALL STUB
NUMBER OF TEETH - 63 (113.097 FULL GEAR, REF)
DIAMETRAL PITCH - 0.7854
CIRCULAR PITCH - 4.00"
NEW OUTSIDE RADIUS - 72.875"
OPERATING PITCH DIA. - 143.876"

SECTOR GEAR REWORK DETAILS



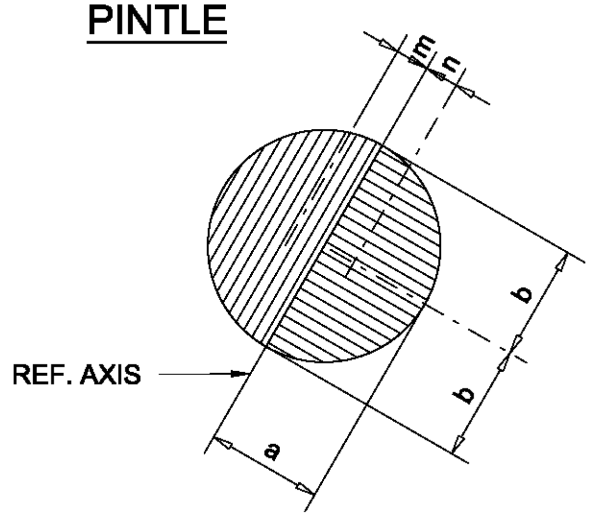
MITER GATES
HORIZONTALLY FRAMED
LOWER GATE
FLOATING PINTLE
ASSEMBLY

PLATE B-21



PINTLE

RH IS THE RESULTANT OF Ra AND Rb (SEE PLATE B-8)



PROJECTED AREA

AREA OF 1/2 CIRCLE = $\frac{\pi D^2}{8}$

$I = 0.109757\ R^4$

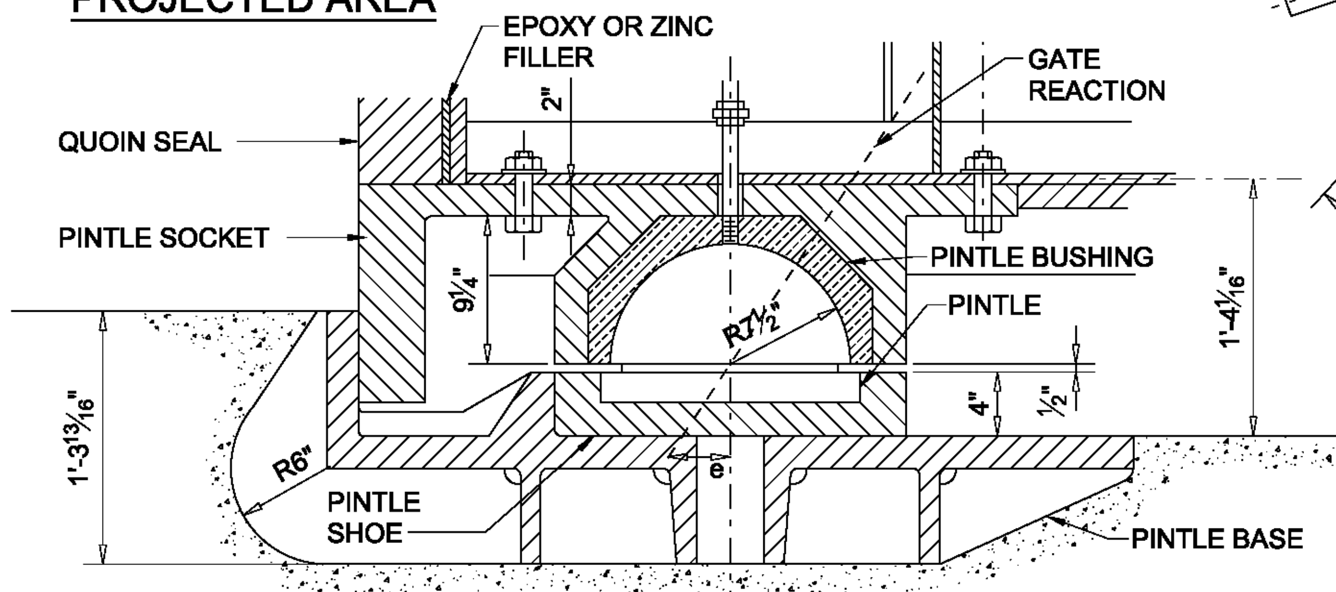
$m = 0.424413\ R$

AREA OF 1/2 ELLIPSE = $\frac{\pi ab}{2}$

$I = ba^3\ (\pi/8 - 8/9\pi)$

$n = 4a/3\pi$

BEARING ON BUSHING = $\frac{R}{A} \pm \frac{MC}{I}$

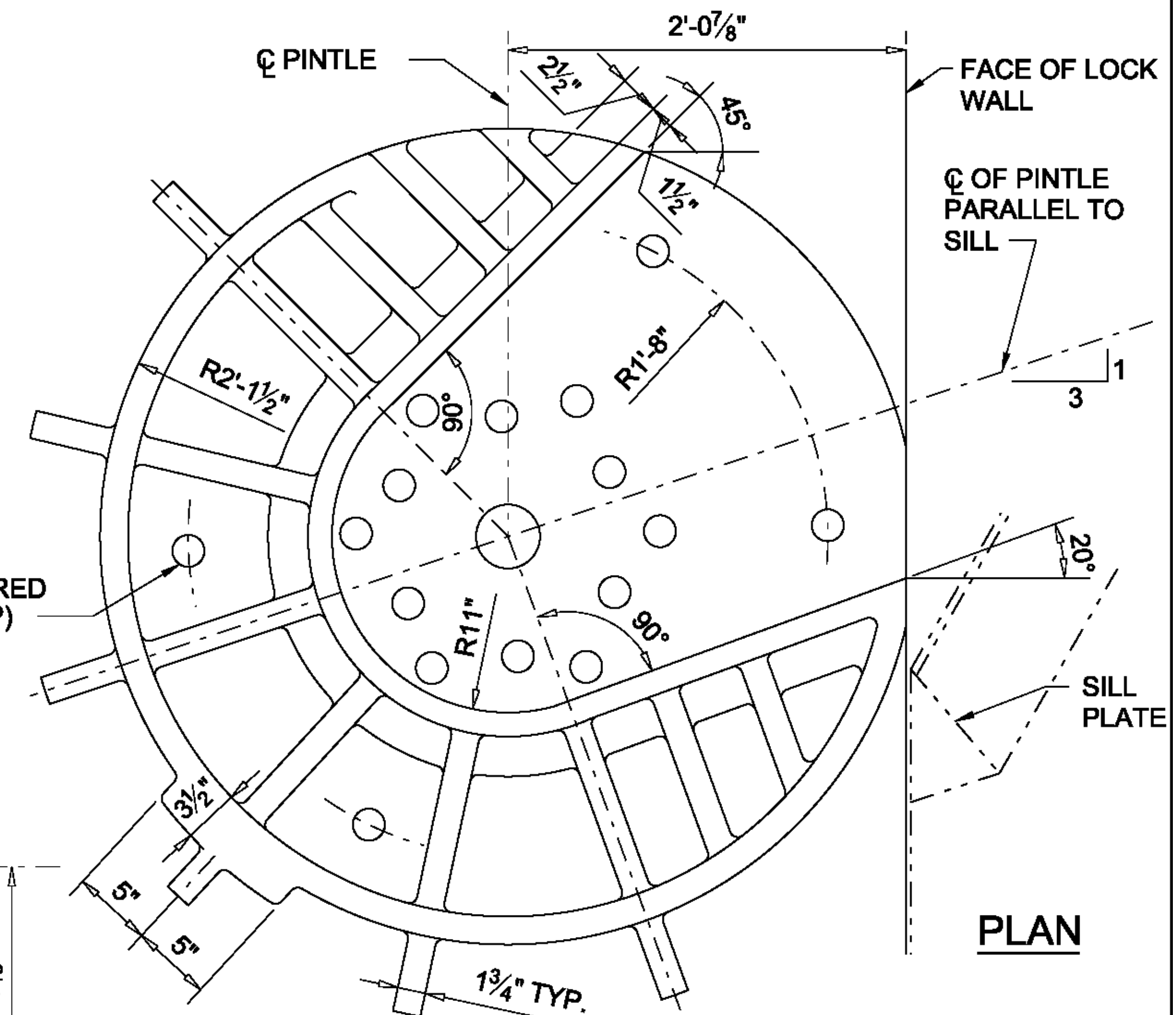


SECTION THRU PINTLE ASSEMBLY

PINTLE BASE

(UPPER MITER GATE)

DIMENSIONS SHOWN ARE TYPICAL AND USED FOR EXPLANATION ONLY. DIMENSIONS MAY VARY WHERE NECESSARY.

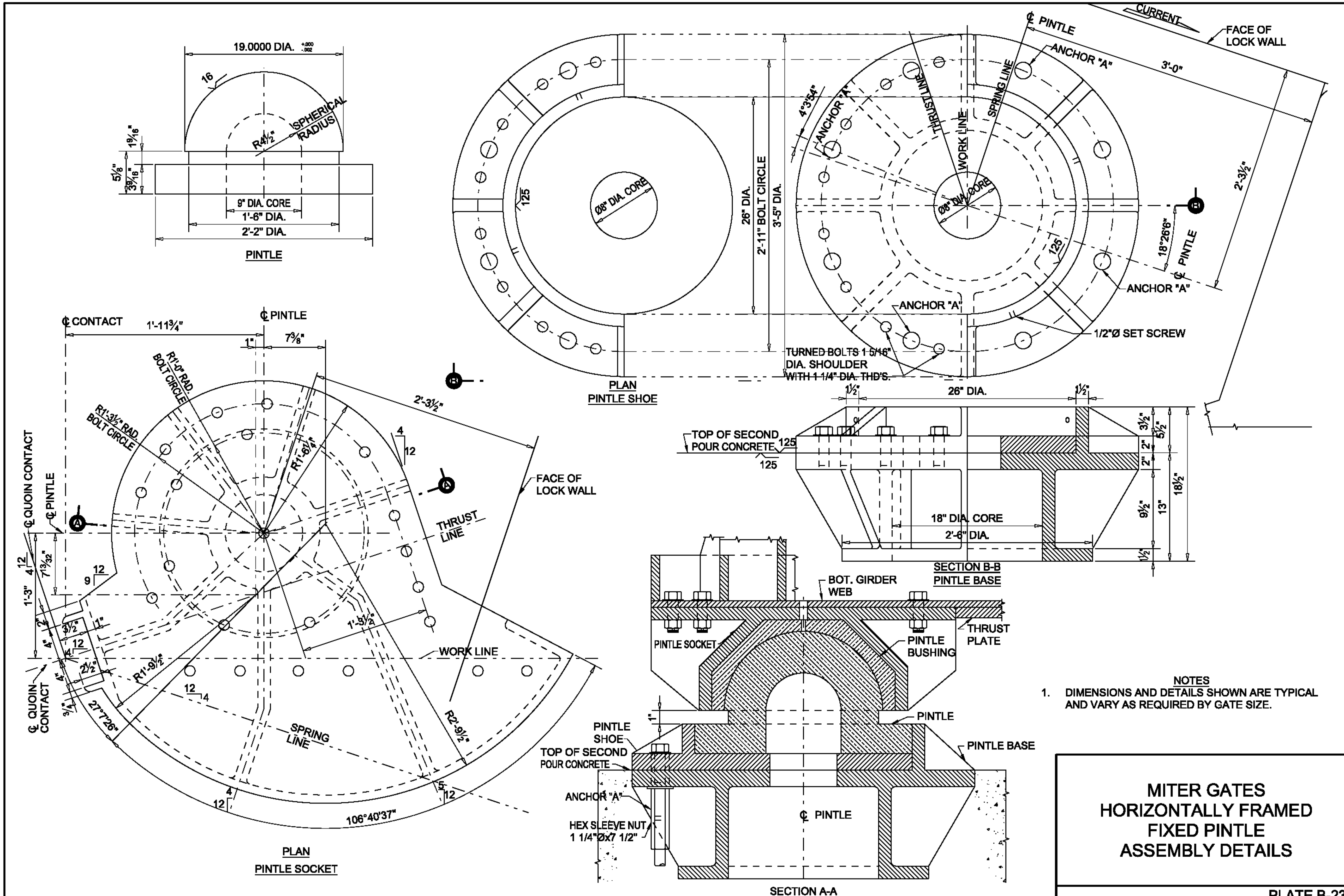


PLAN

NOTE:

PINTLE BASE FOR VERTICALLY FRAMED GATES SIMILAR TO THE PINTLE BASE SHOWN. SEE PLATE B-32 FOR PLAN OF PINTLE BASE FOR VERTICALLY FRAMED GATE.

MITER GATES
HORIZONTALLY FRAMED
UPPER GATE
FLOATING PINTLE
ASSEMBLY

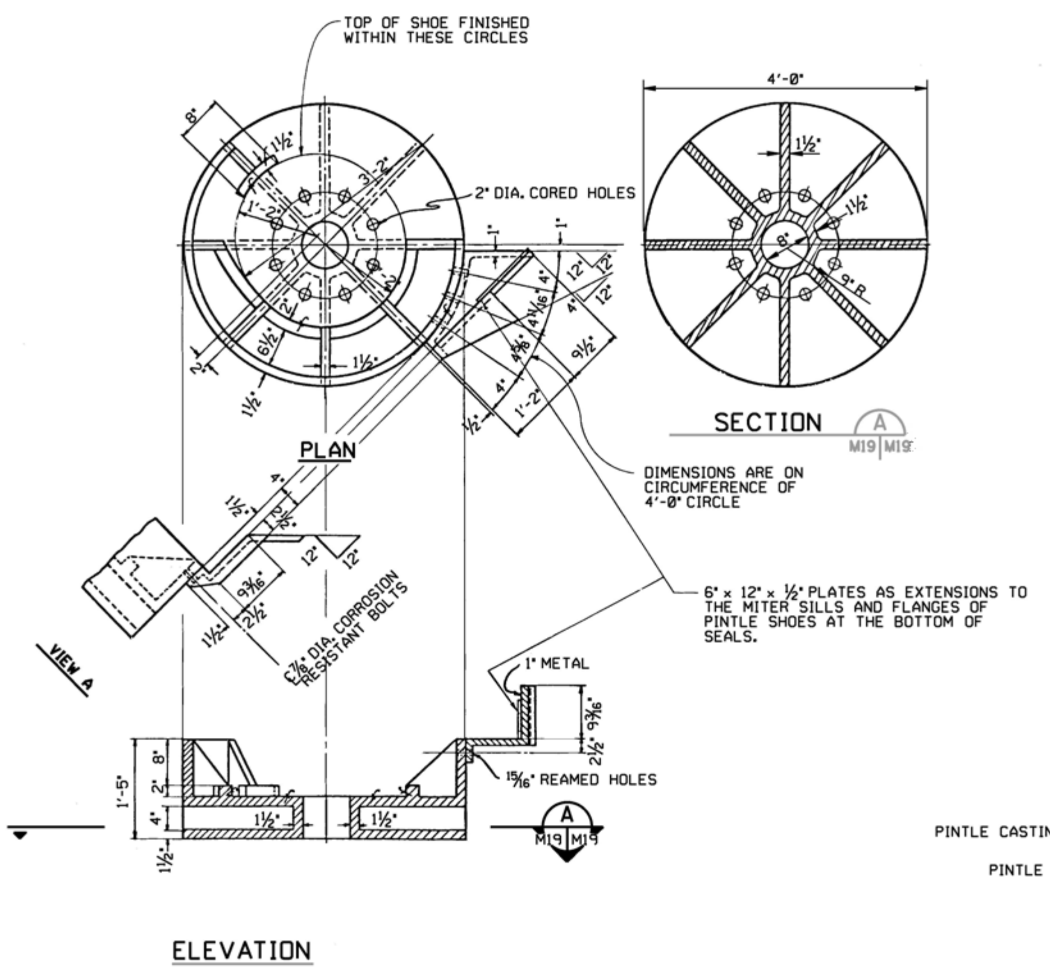


NOTES

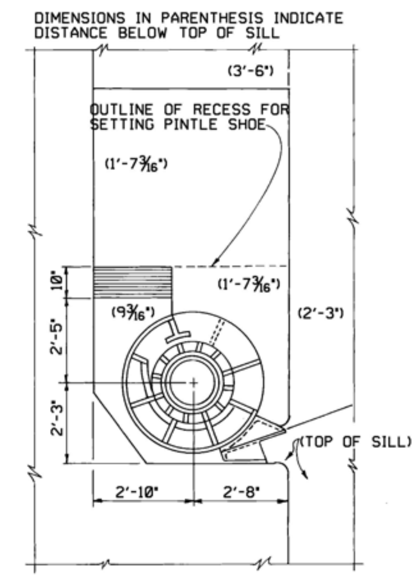
1. DIMENSIONS AND DETAILS SHOWN ARE TYPICAL AND VARY AS REQUIRED BY GATE SIZE.

MITER GATES
HORIZONTALLY FRAMED
FIXED PINTLE
ASSEMBLY DETAILS

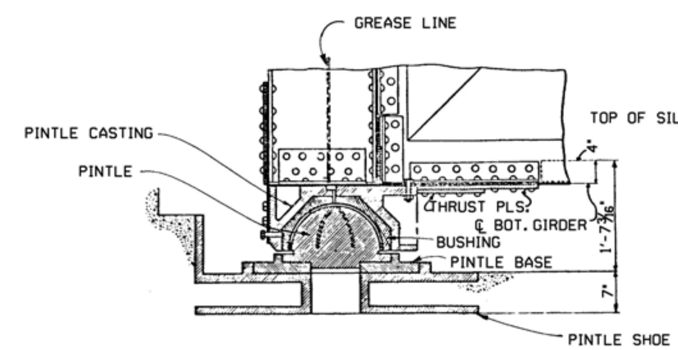
PLATE B-23



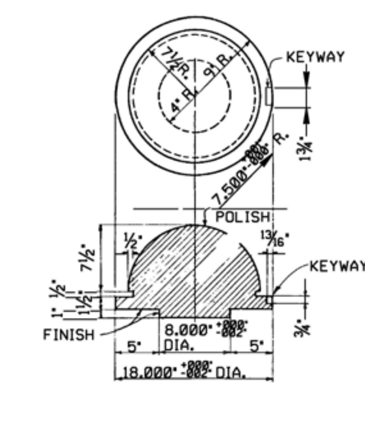
PINTLE SHOE
WT. OF LARGE CASTING 2324#
WT. OF SMALL CASTING 140#
CAST STEEL



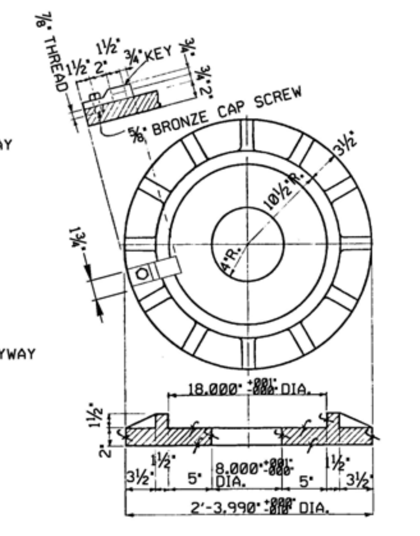
PINTLE AREA PLAN



SECTION THROUGH PINTLE
NO SCALE



PINTLES
CAST NICKEL STEEL - WT. 394#



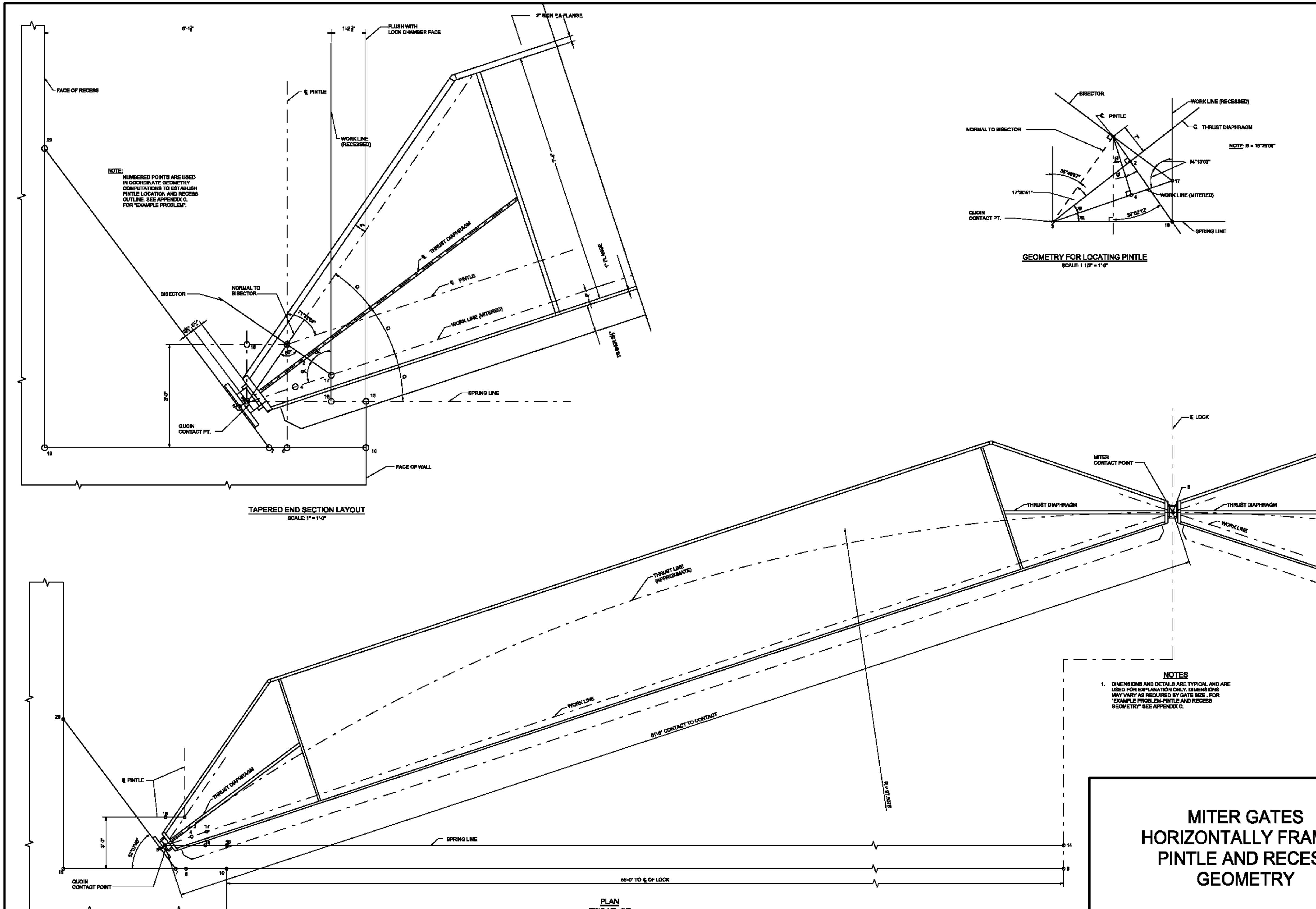
PINTLE BASES
CAST STEEL WT. 375#

NOTES:

1. AFTER MITER GATE REMOVAL AND DEWATERING OF THE LOCK, CAREFULLY MATCH MARK AND REMOVE ALL (4) PINTLES AND PINTLE BASES FROM THEIR RESPECTIVE PINTLE SHOES, TAKING NECESSARY PRECAUTIONS TO PREVENT DAMAGE TO THESE ITEMS.
2. SEPARATE EACH PINTLE FROM ITS BASE AND CLEAN ALL EXPOSED SURFACES OF EACH PINTLE SHOE, PINTLE BASE AND PINTLE. CLEANING SHALL REMOVE ALL GREASE AND RUST SUCH THAT THE GOVERNMENT CAN ACCURATELY DETERMINE ANY LOSS OF SECTION, CRACKS OR OTHER DEFECTS IN THESE ITEMS.
3. POSITION COMPONENTS ON SUITABLE SUPPORTS IN ORDER TO PERMIT THE NOTED INSPECTION. AFTER COMPLETION OF THIS INSPECTION, COAT ALL SURFACES WITH NEW GREASE (PROVIDED BY THE GOVERNMENT) AND PROTECT THE ITEMS FROM DAMAGE DURING THE REMAINING LOCK-DEWATERED PERIOD.
4. REINSTALL PINTLES AND BASES IN THEIR ORIGINAL LOCATIONS PRIOR TO FLOODING OF CHAMBER.
5. ITEMS 1-3 SHALL BE DONE IMMEDIATELY AFTER DEWATERING IS ACCOMPLISHED.

MITER GATES
PINTLE DETAILS

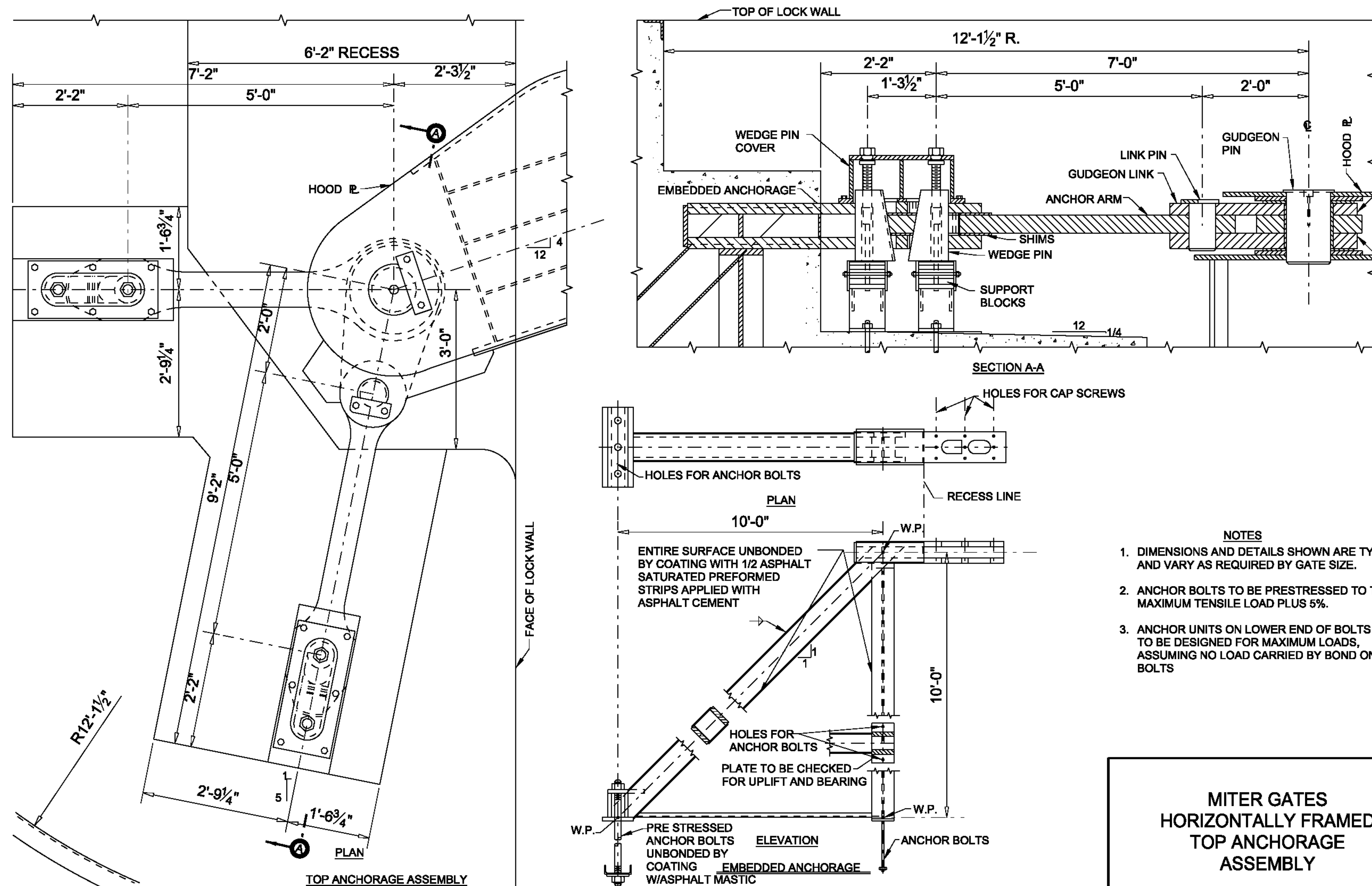
PLATE B-24



NOTES

1. DIMENSIONS AND DETAILS ARE TYPICAL AND ARE USED FOR EXPLANATION ONLY. DIMENSIONS MAY VARY AS REQUIRED BY GATE SIZE . FOR "EXAMPLE PROBLEM-PINTLE AND RECESS GEOMETRY" SEE APPENDIX C.

MITER GATES
HORIZONTALLY FRAMED
PINTLE AND RECESS
GEOMETRY

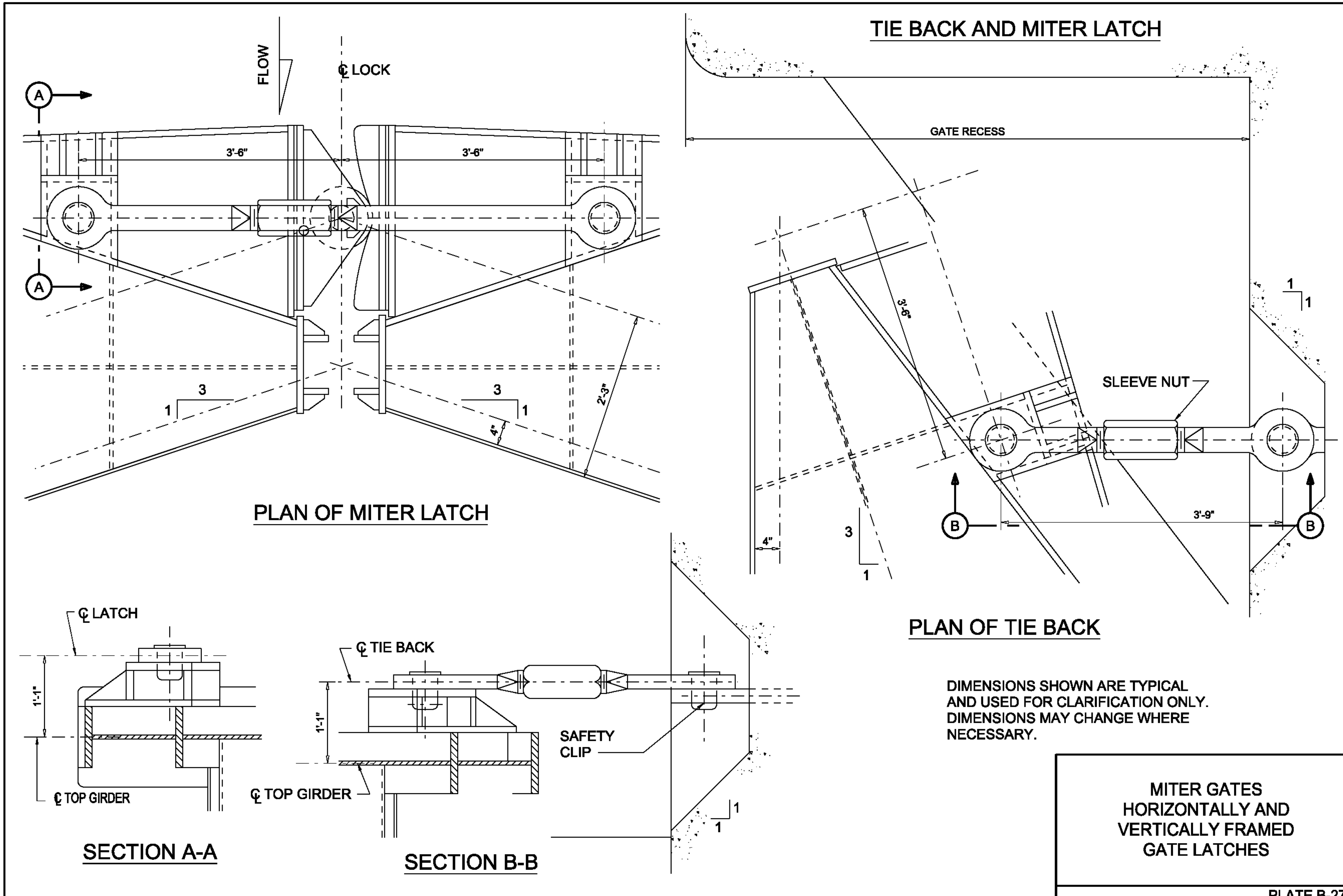


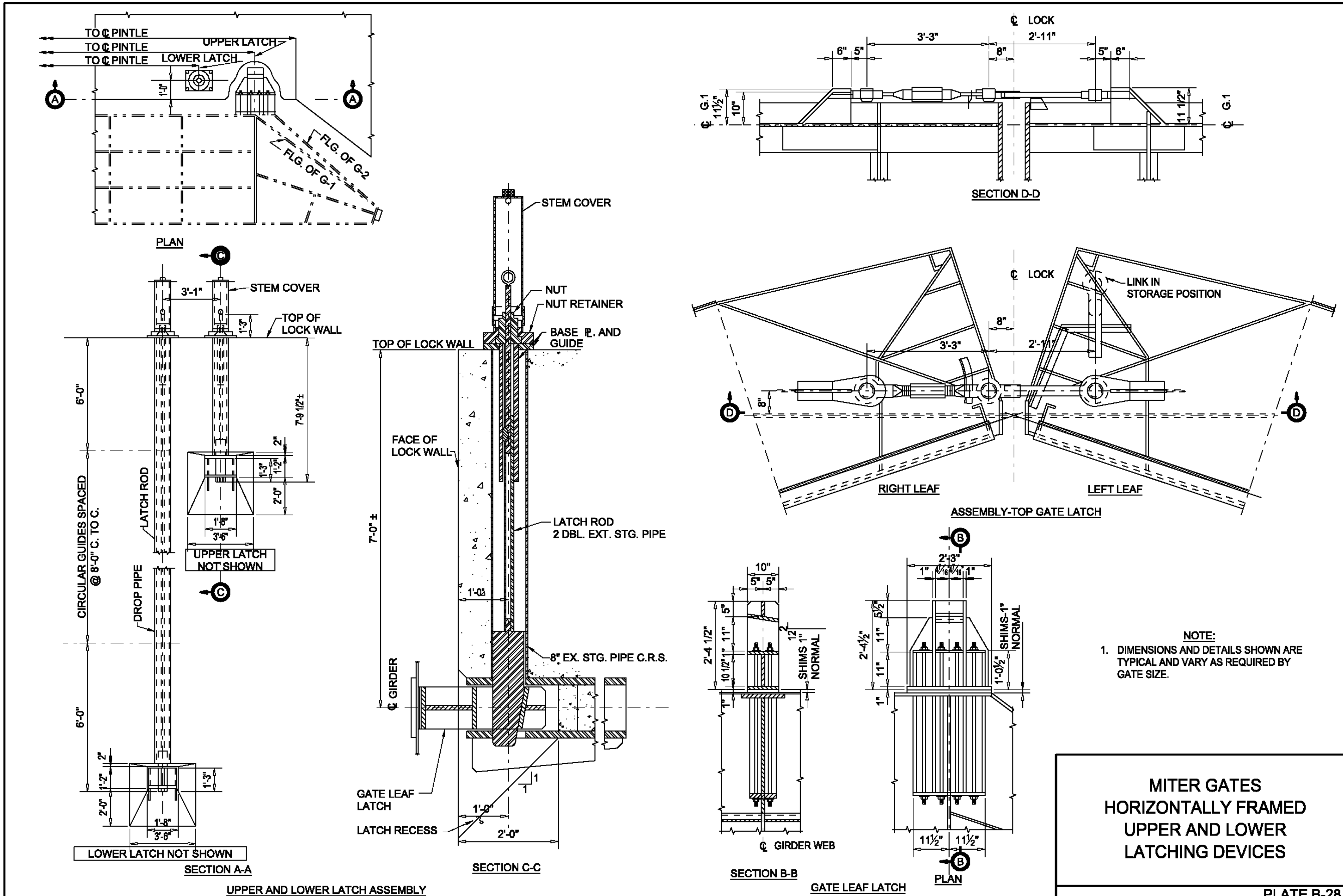
NOTES

1. DIMENSIONS AND DETAILS SHOWN ARE TYPICAL AND VARY AS REQUIRED BY GATE SIZE.
2. ANCHOR BOLTS TO BE PRESTRESSED TO THE MAXIMUM TENSILE LOAD PLUS 5%.
3. ANCHOR UNITS ON LOWER END OF BOLTS TO BE DESIGNED FOR MAXIMUM LOADS, ASSUMING NO LOAD CARRIED BY BOND ON BOLTS

MITER GATES
HORIZONTALLY FRAMED
TOP ANCHORAGE
ASSEMBLY

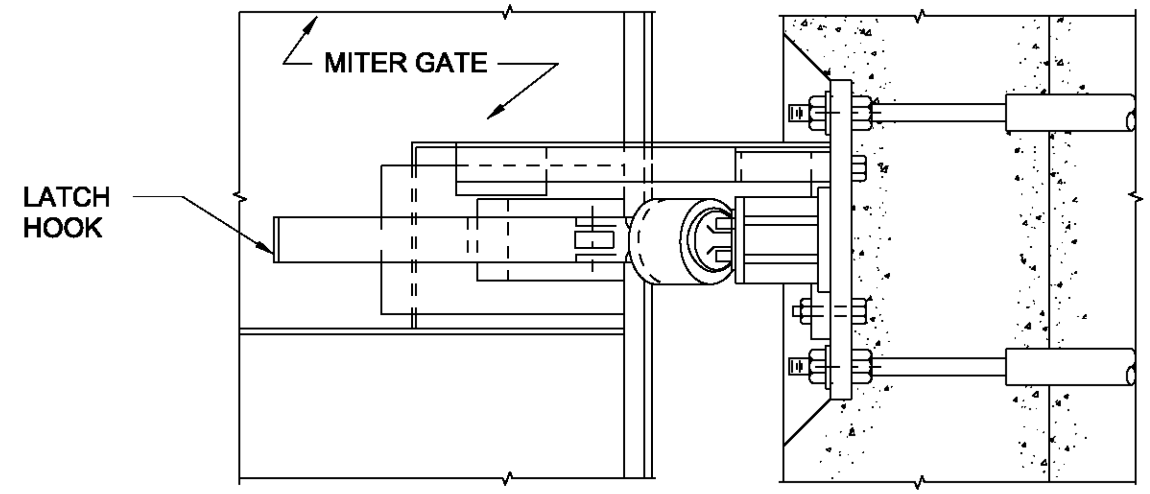
PLATE B-26


TIE BACK AND MITER LATCH
FLOW
℄ LOCK
GATE RECESS
3'-6"
3'-6"
3'-6"
2'-3"
4"
3'-9"
SLEEVE NUT
PLAN OF MITER LATCH
PLAN OF TIE BACK
℄ LATCH
1'-1"
℄ TOP GIRDER
℄ TIE BACK
SAFETY CLIP
SECTION A-A
SECTION B-B
DIMENSIONS SHOWN ARE TYPICAL AND USED FOR CLARIFICATION ONLY. DIMENSIONS MAY CHANGE WHERE NECESSARY.
MITER GATES
HORIZONTALLY AND
VERTICALLY FRAMED
GATE LATCHES

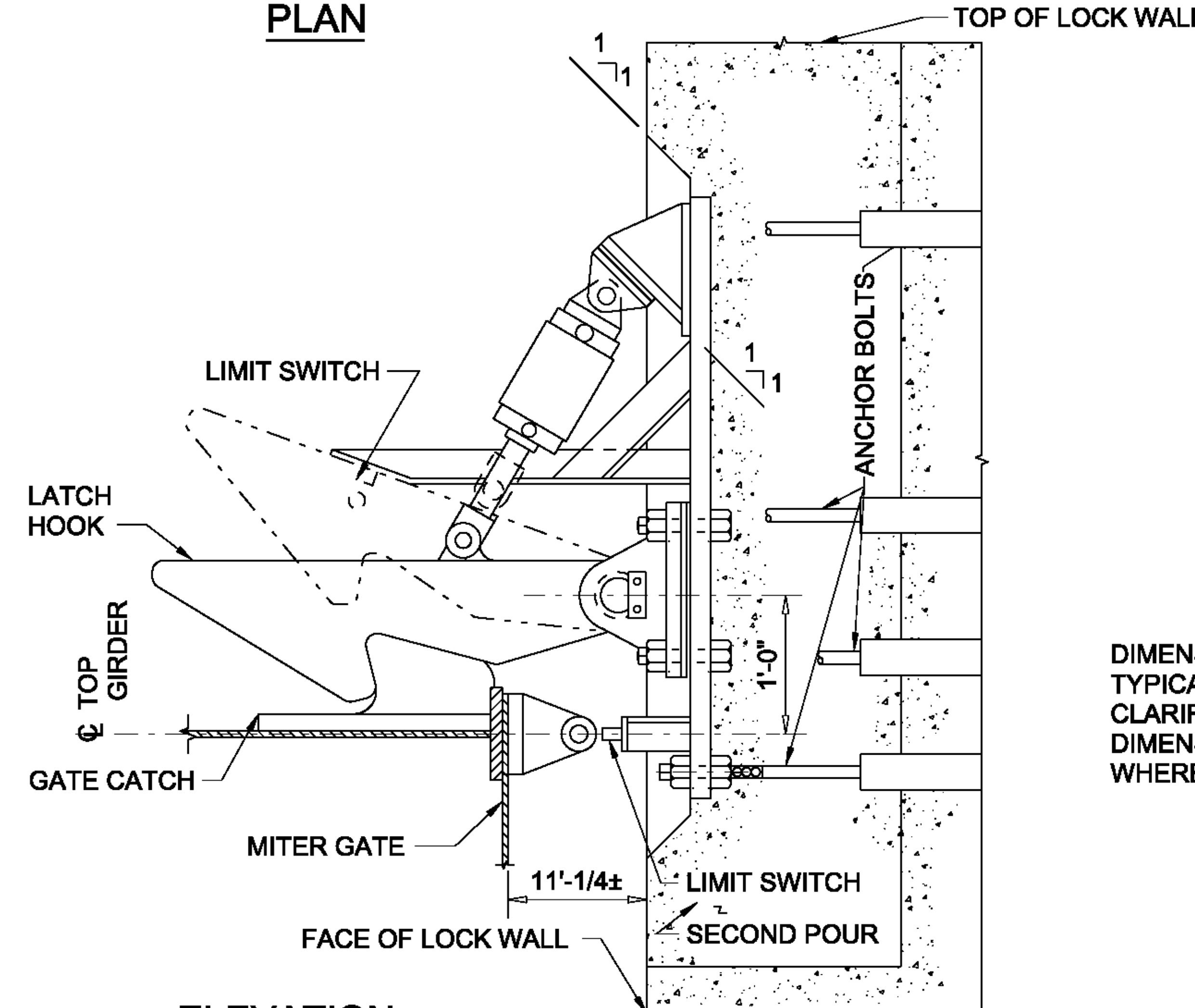


MITER GATES
HORIZONTALLY FRAMED
UPPER AND LOWER
LATCHING DEVICES

PLATE B-28



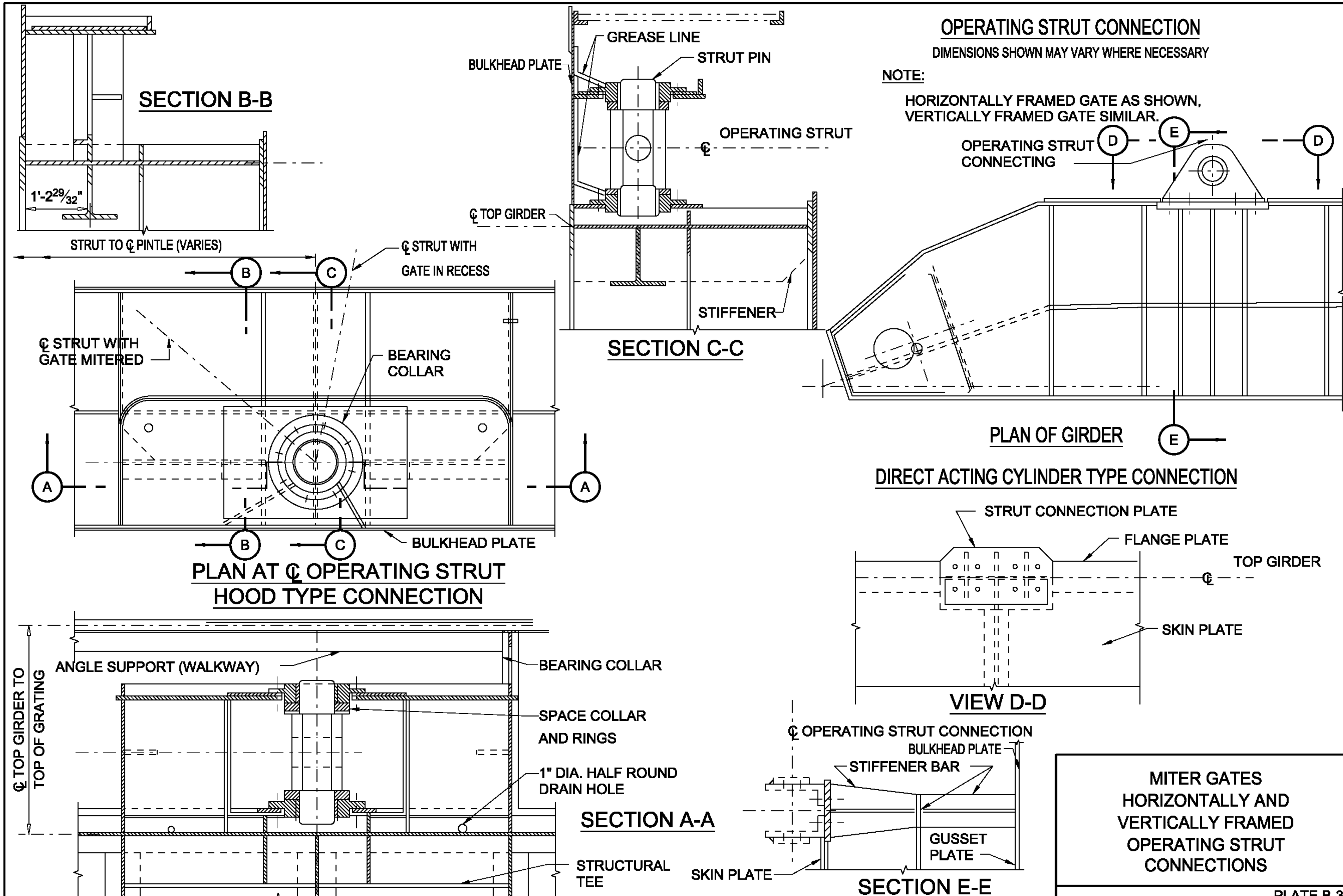
PLAN



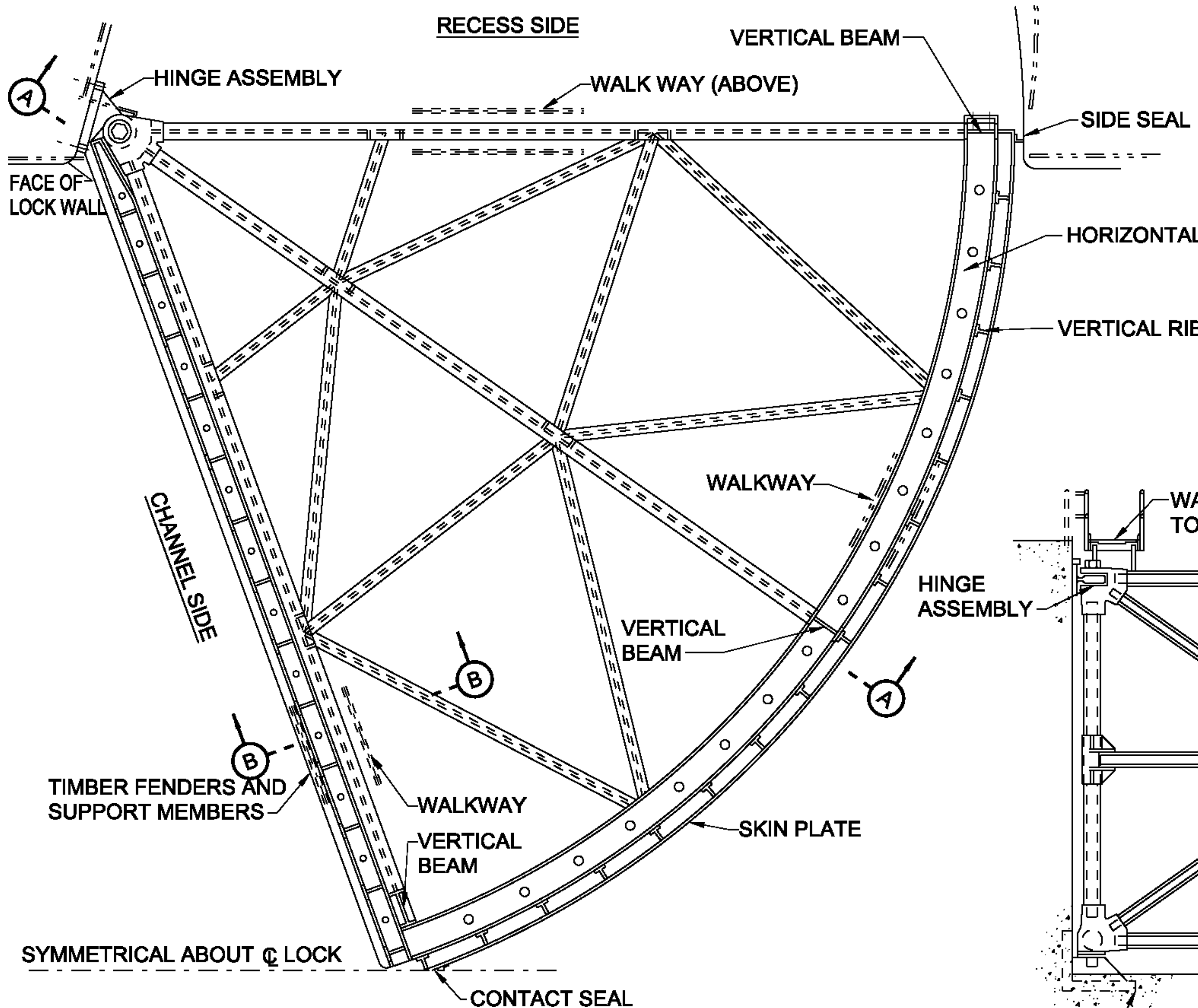
ELEVATION
LATCHING DEVICE

DIMENSIONS SHOWN ARE TYPICAL AND USED FOR CLARIFICATION ONLY. DIMENSIONS MAY CHANGE WHERE NECESSARY.

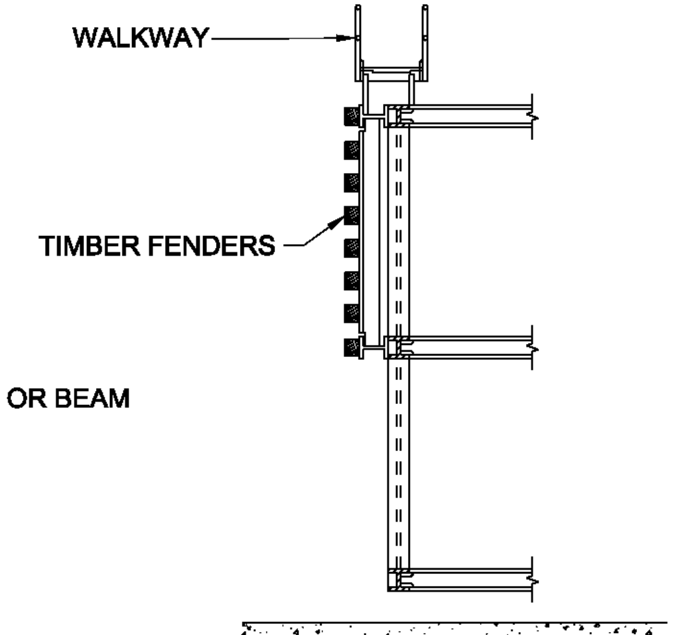
MITER GATES
HORIZONTALLY FRAMED
AUTOMATIC
LATCHING DEVICE



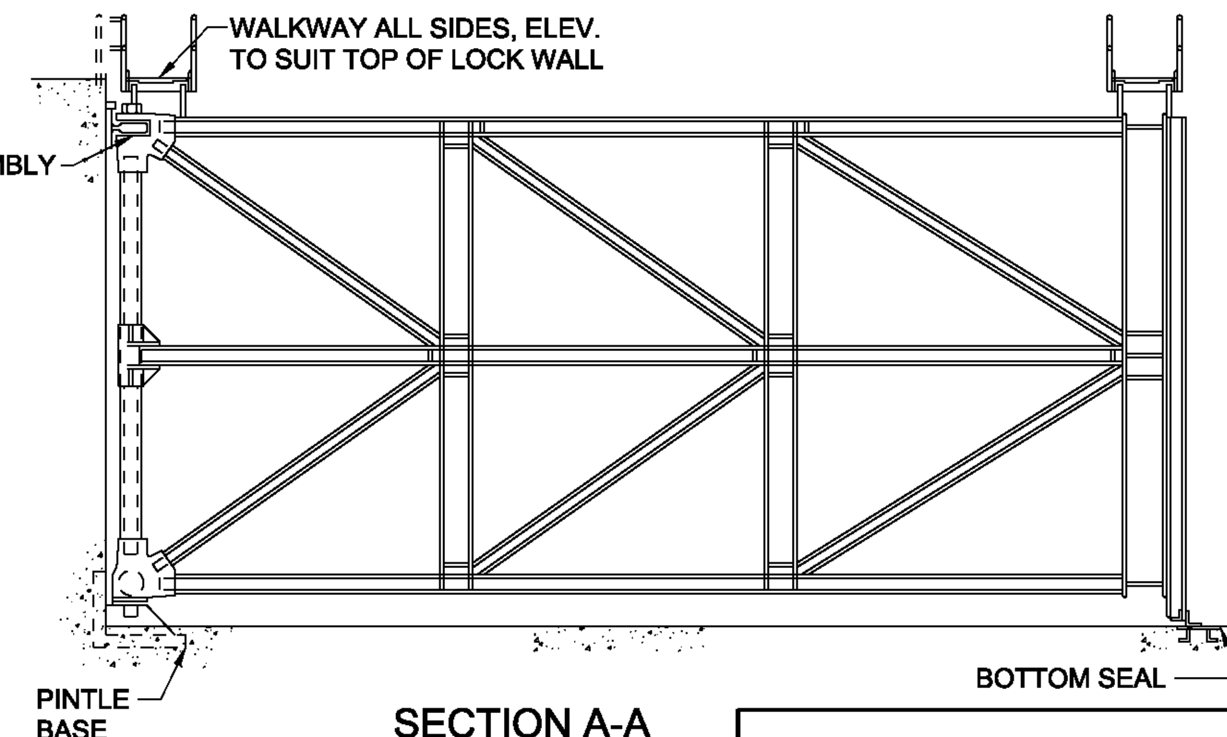
MITER GATES
HORIZONTALLY AND
VERTICALLY FRAMED
OPERATING STRUT
CONNECTIONS



PLAN
(SHOWING HORIZONTAL TRUSS)

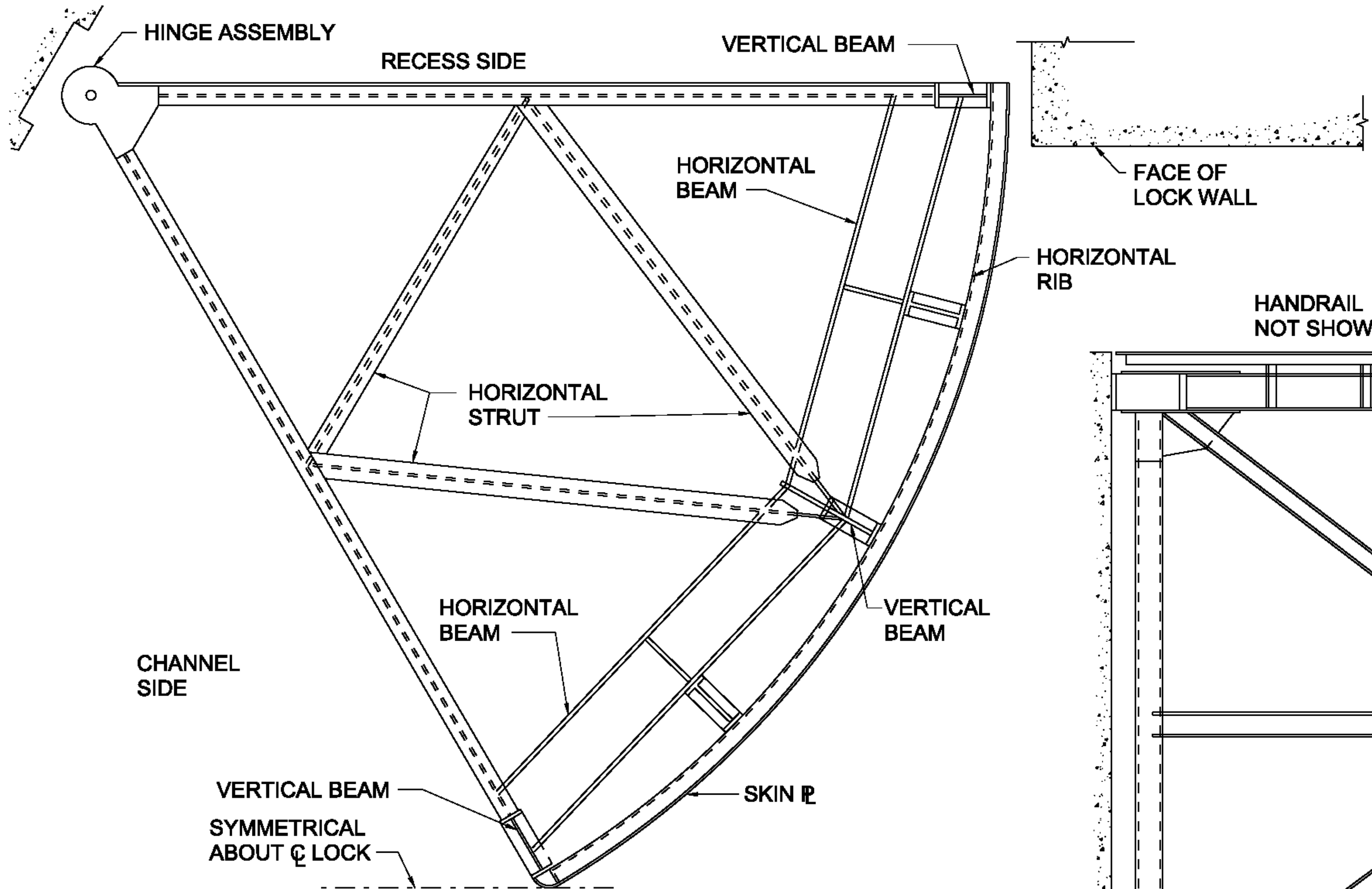


SECTION B-B

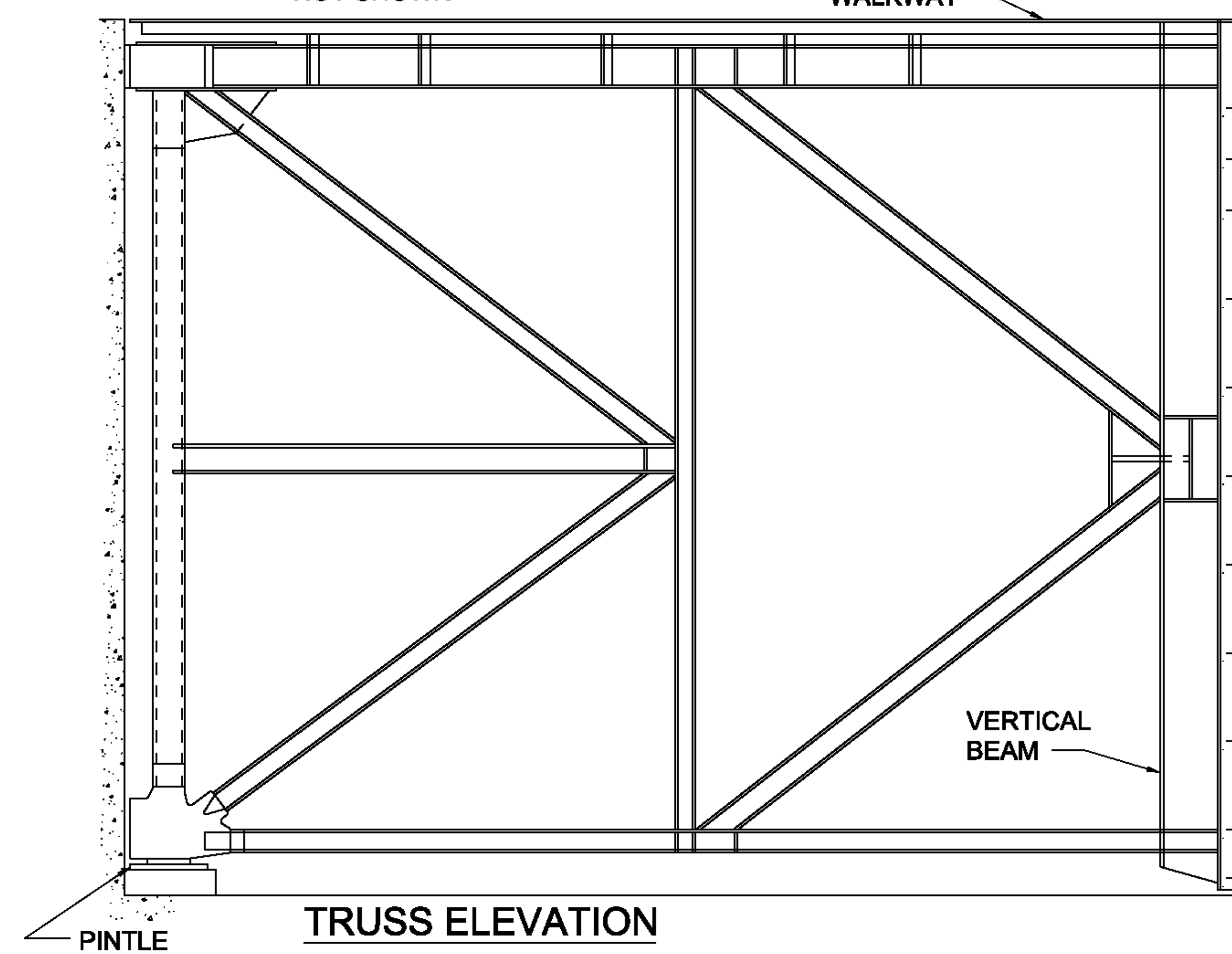


SECTION A-A
(SHOWING HORIZONTAL TRUSS)

SECTOR GATES
GENERAL PLAN
AND SECTIONS



PLAN

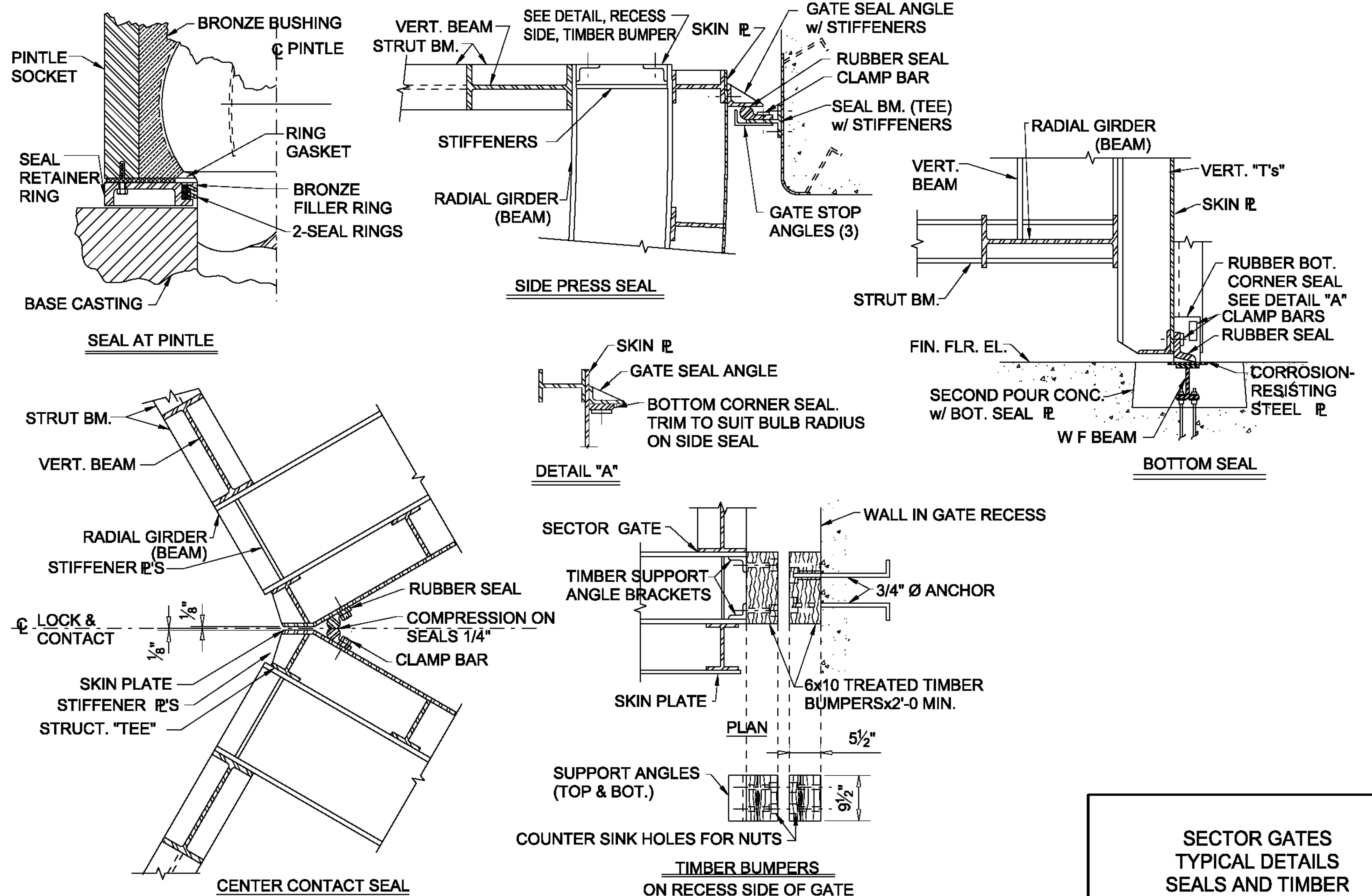


TRUSS ELEVATION

(CHANNEL SIDE)

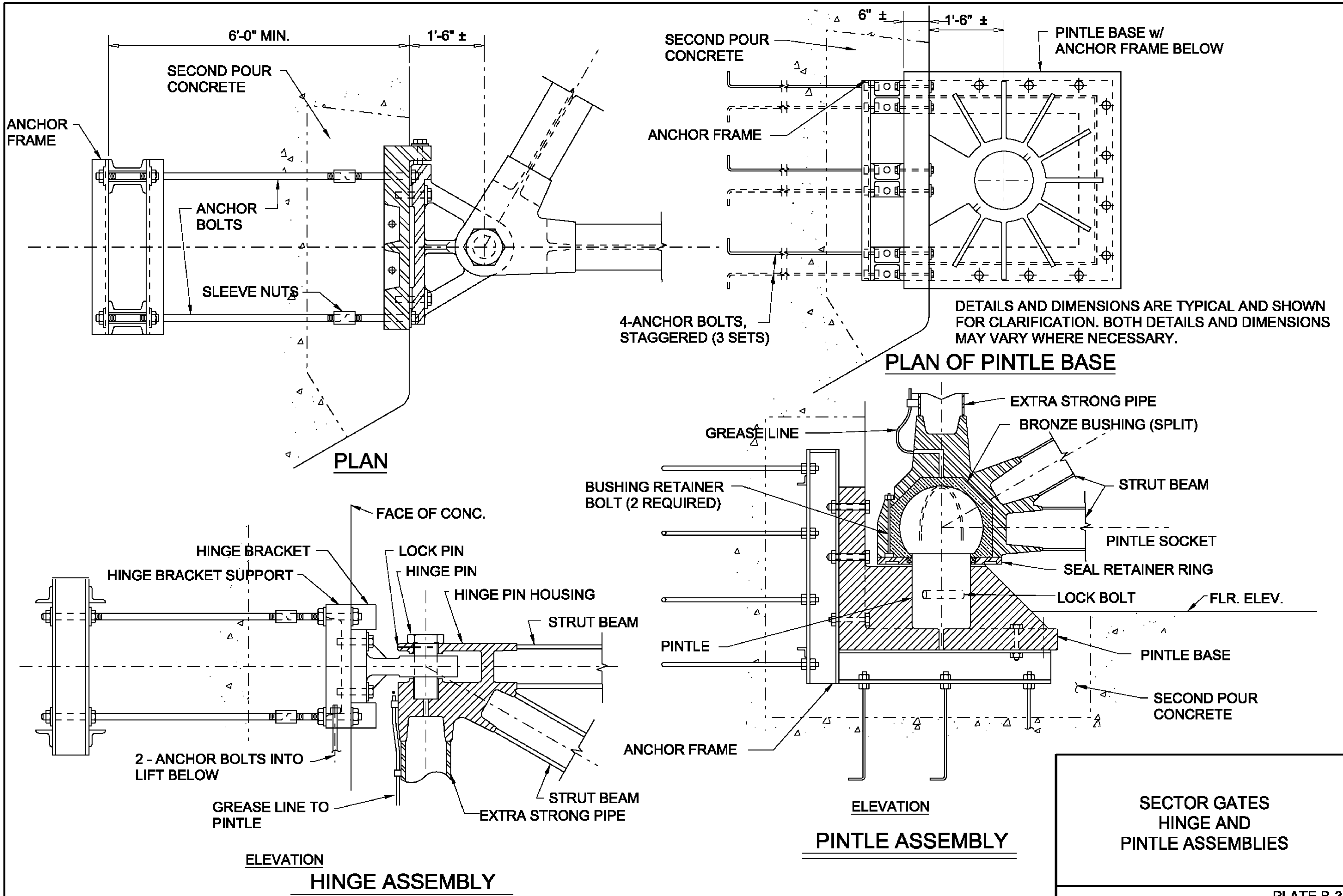
NOTE: NUMEROUS EXISTING SECTOR GATES ARE FRAMED AS SHOWN. THIS TYPE OF FRAMING IS NOT RECOMMENDED FOR NEW CONSTRUCTION.

SECTOR GATES
GENERAL PLAN
AND ELEVATION



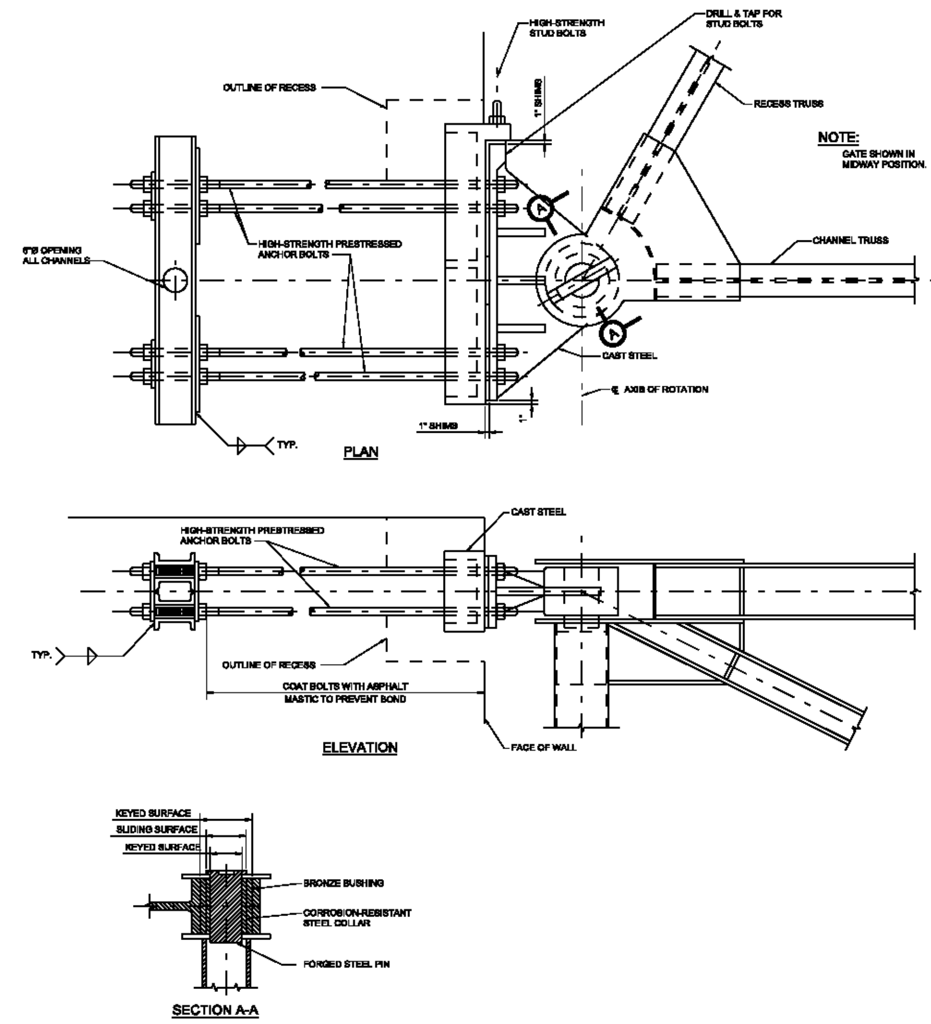
SECTOR GATES
TYPICAL DETAILS
SEALS AND TIMBER
BUMPERS

PLATE B-33



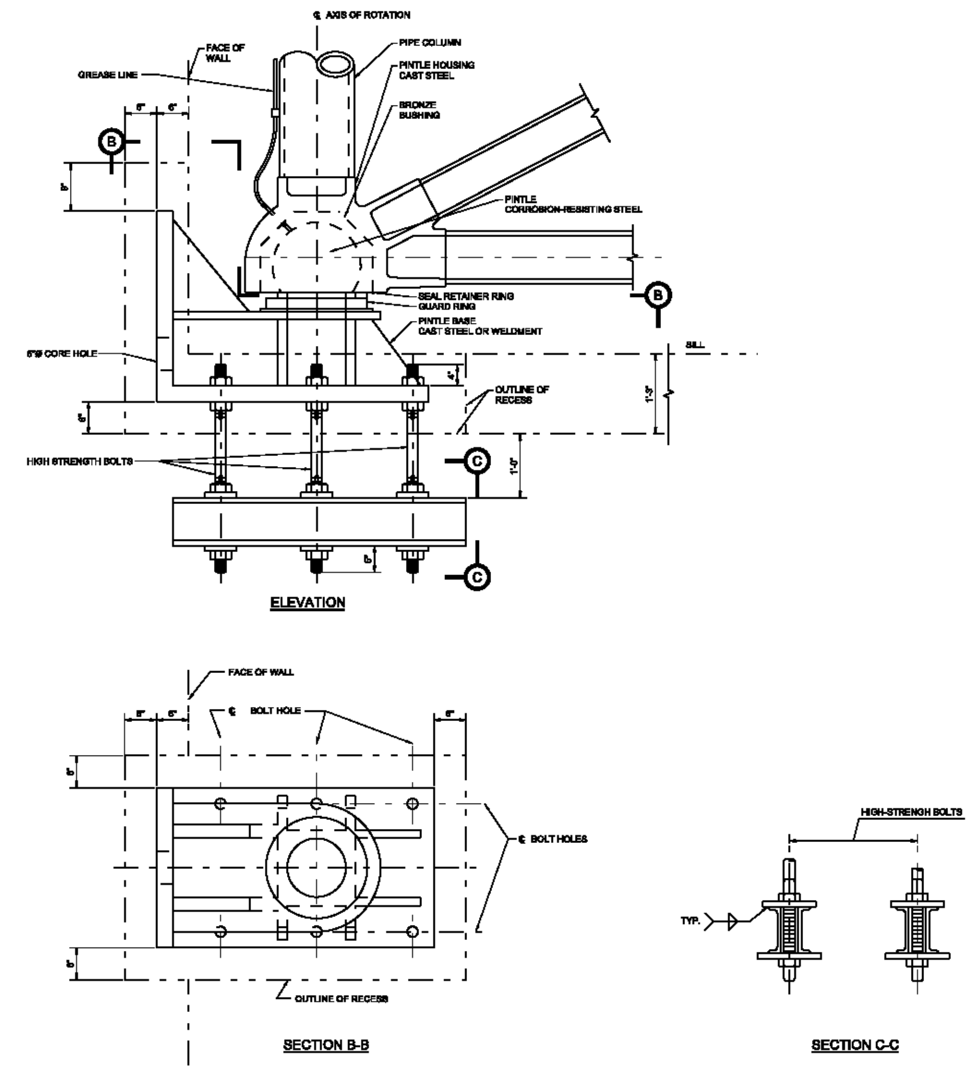
SECTOR GATES
HINGE AND
PINTLE ASSEMBLIES

PLATE B-34



HINGE ASSEMBLY

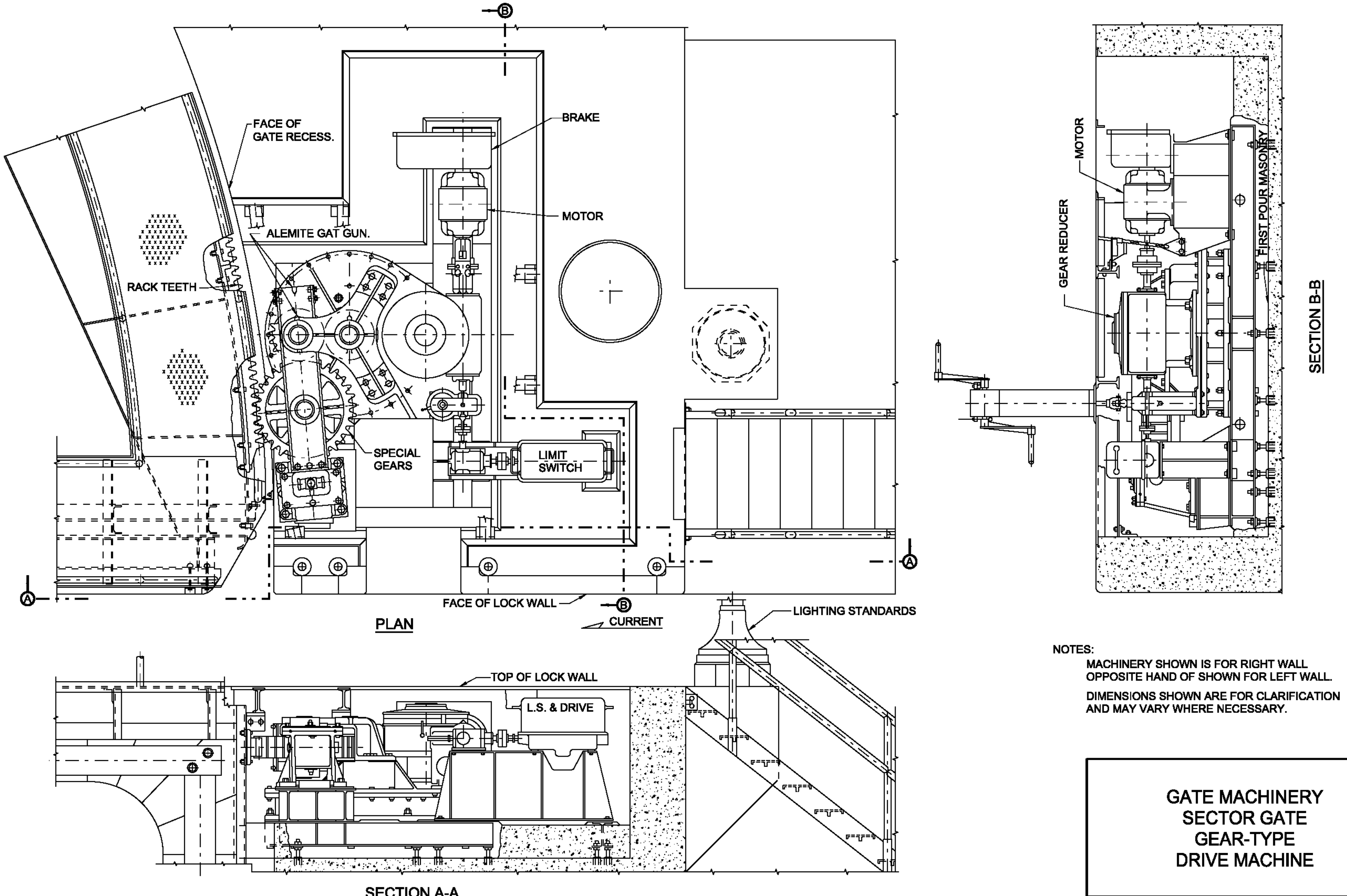
SCALE: 3/4" = 1'-0"



PINTLE ASSEMBLY

SCALE: 1" = 1'-0"

SECTOR GATES
ALTERNATE HINGE AND
PINTLE ASSEMBLES

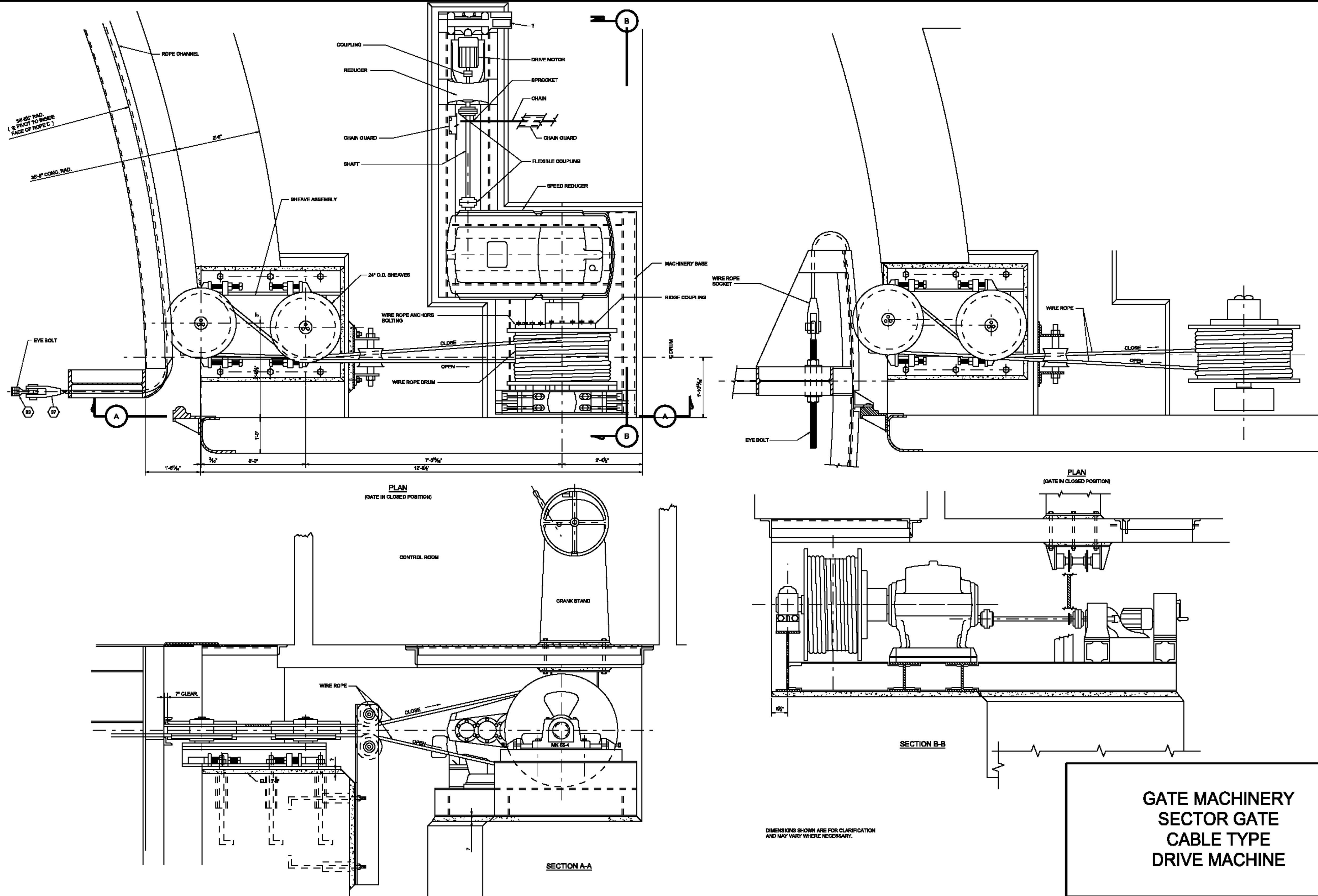


NOTES:

MACHINERY SHOWN IS FOR RIGHT WALL
OPPOSITE HAND OF SHOWN FOR LEFT WALL.

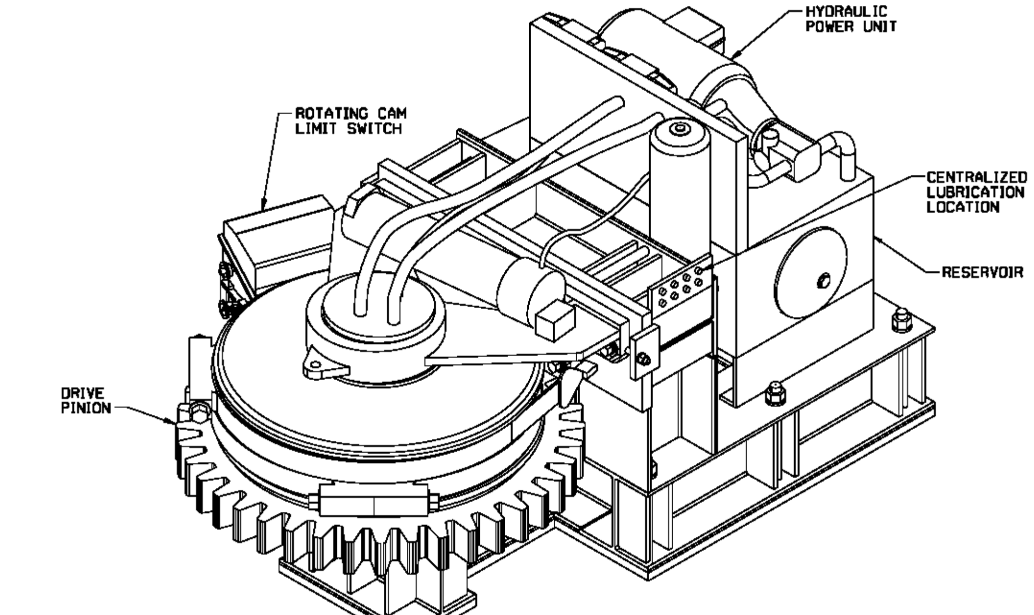
DIMENSIONS SHOWN ARE FOR CLARIFICATION
AND MAY VARY WHERE NECESSARY.

GATE MACHINERY
SECTOR GATE
GEAR-TYPE
DRIVE MACHINE

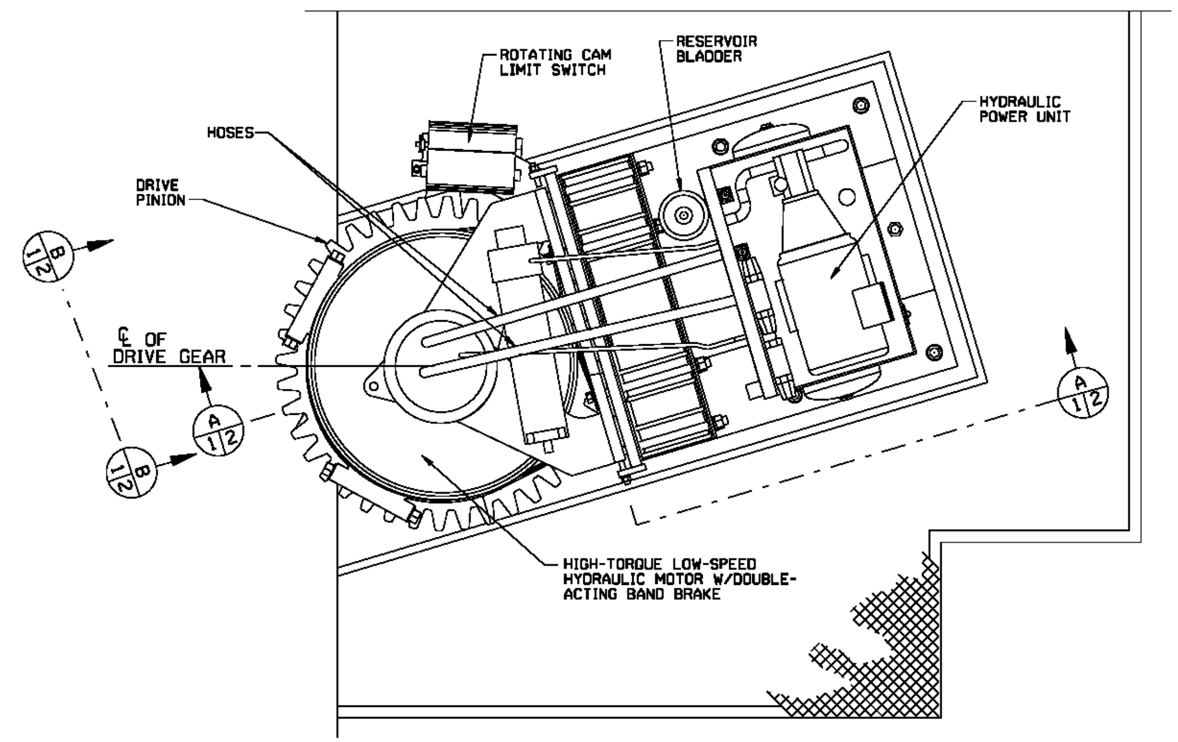


GATE MACHINERY
SECTOR GATE
CABLE TYPE
DRIVE MACHINE

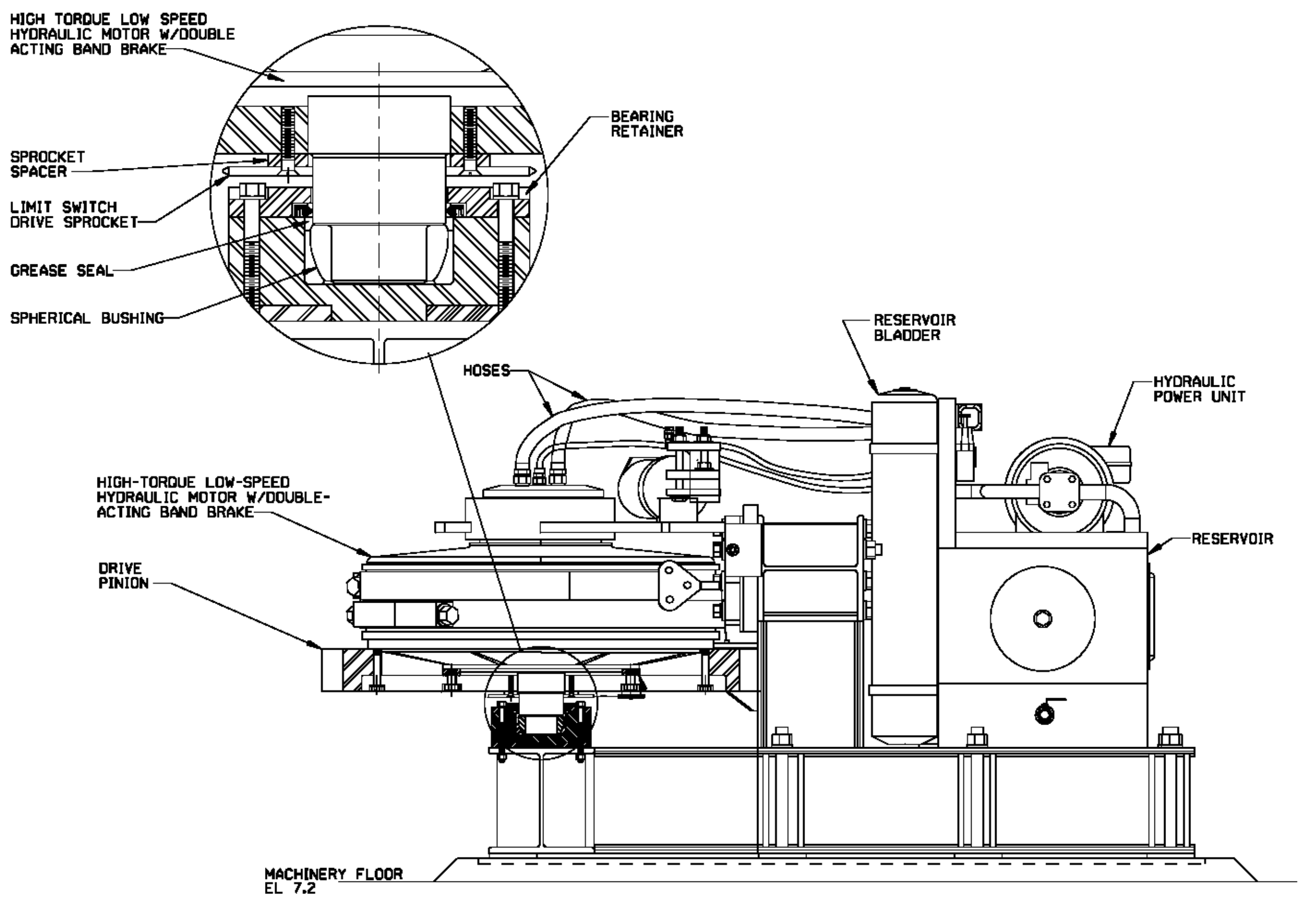
PLATE B-37



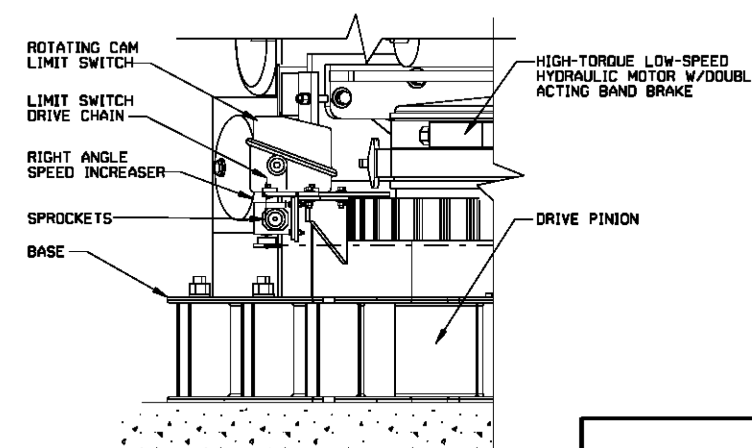
ISOMETRIC VIEW



MACHINERY PIT LAYOUT



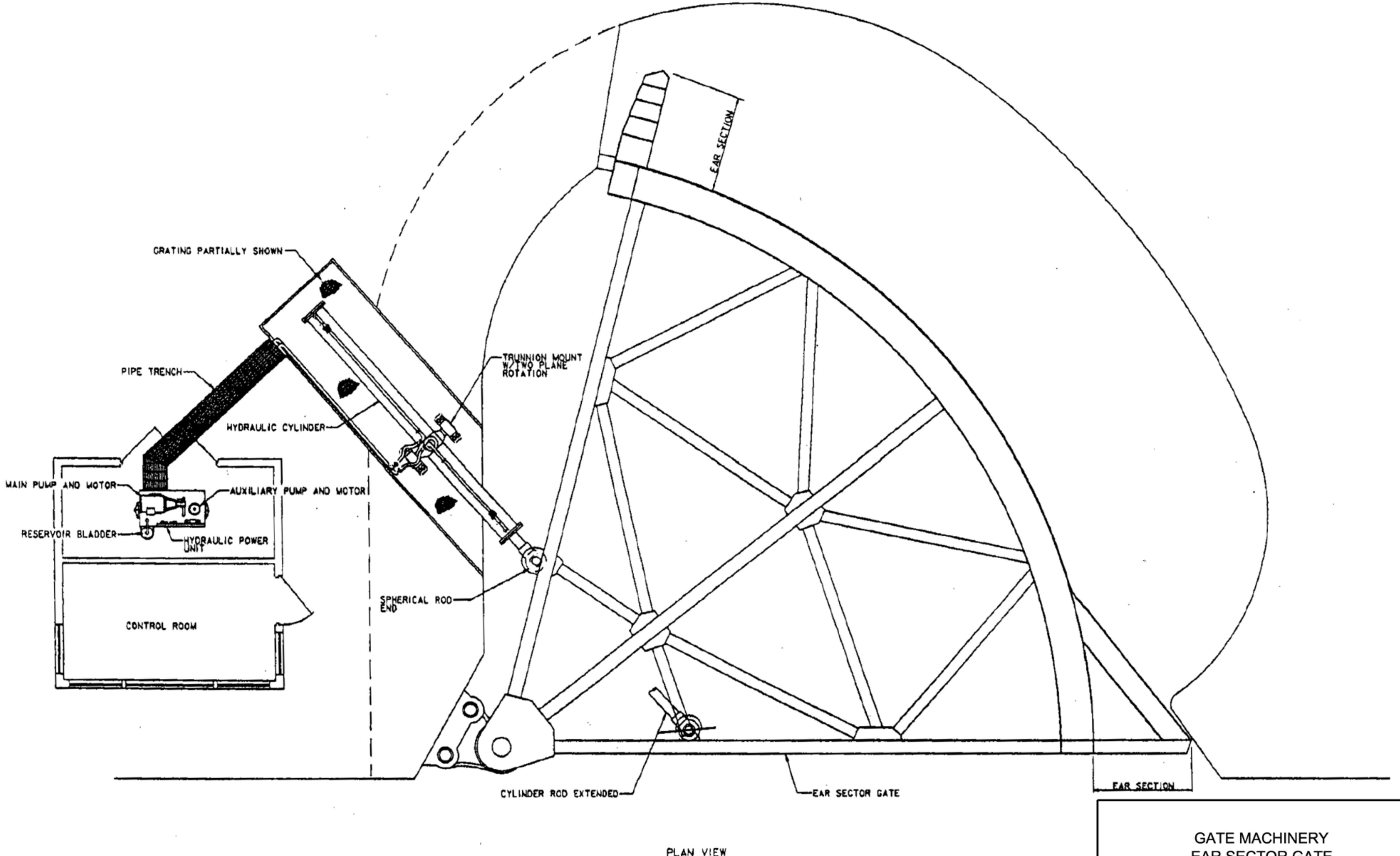
SECTION A/1|2



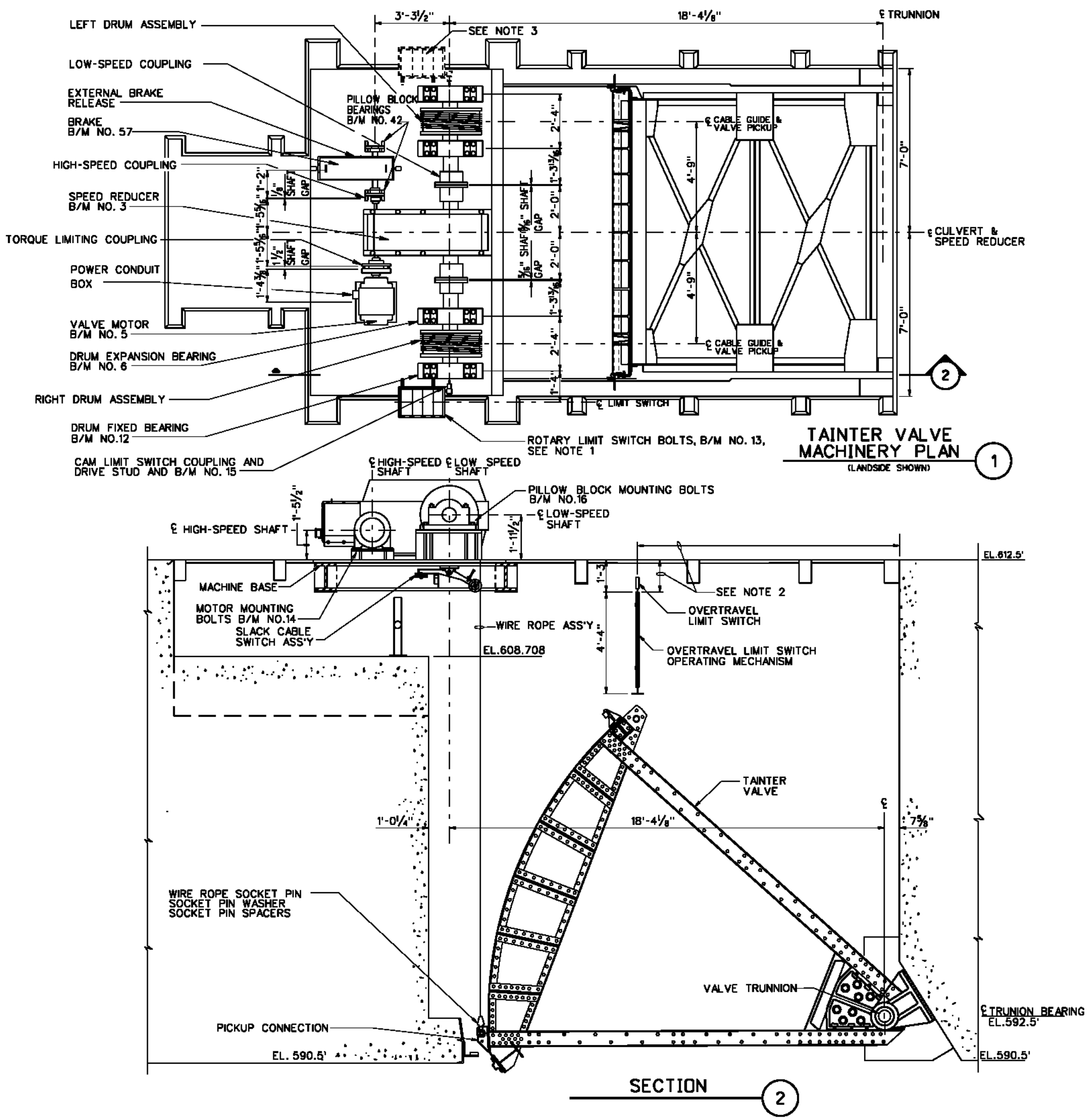
SECTION B/1|2

GATE MACHINERY
SECTOR GATE
RACK AND PINION
DRIVE MACHINERY

PLATE B-38



GATE MACHINERY
EAR SECTOR GATE
DIRECT ACTING HYDRAULIC CYLINDER
DRIVE MACHINERY

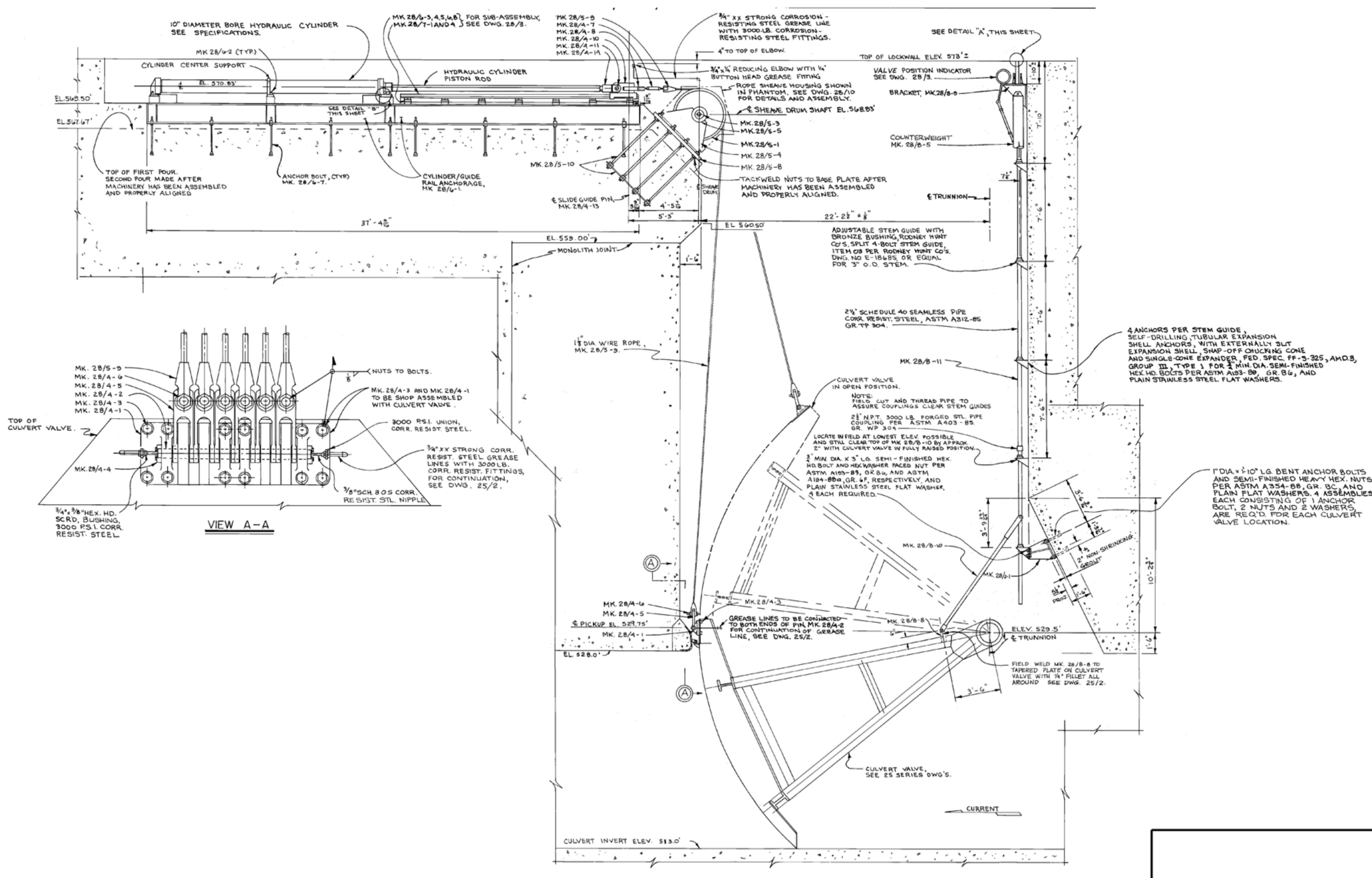


NOTES:

1. THIS DRAWING IS TYPICAL FOR LAND-WALL TAINTER VALVES. INTERMEDIATE WALL TAINTER VALVE MACHINERY UNITS ARE IDENTICAL EXCEPT THE ROTARY LIMIT SWITCH IS TO BE MOUNTED ON THE OPPOSITE SIDE AND ATTACHED TO THE LEFT DRUM SHAFT WITH THE COUPLING AND DRIVE STUD.
2. FIELD-VERIFY LOCATION OF OVERTRAVEL LIMIT SWITCH AFTER DETERMINING NORMAL VALVE OPERATING LIMITS VIA ROTATING CAM LIMIT SWITCH ON LOW-SPEED SHAFT. FIELD-MOUNT NEW LIMIT SWITCH AND OPERATING MECHANISM. SEE SHEET EL80 FOR OVERTRAVEL LIMIT SWITCH.
3. LOCATION OF DASHED ROTARY LIMIT SWITCH SHOWN FOR INTERMEDIATE WALL TAINTER VALVE MACHINERY. LANDSIDE LIMIT SWITCH LOCATION SHALL BE OPPOSITE HAND AS SHOWN.

TAINTER VALVE WIRE ROPE MOTOR DRIVE PLAN AND ELEVATION

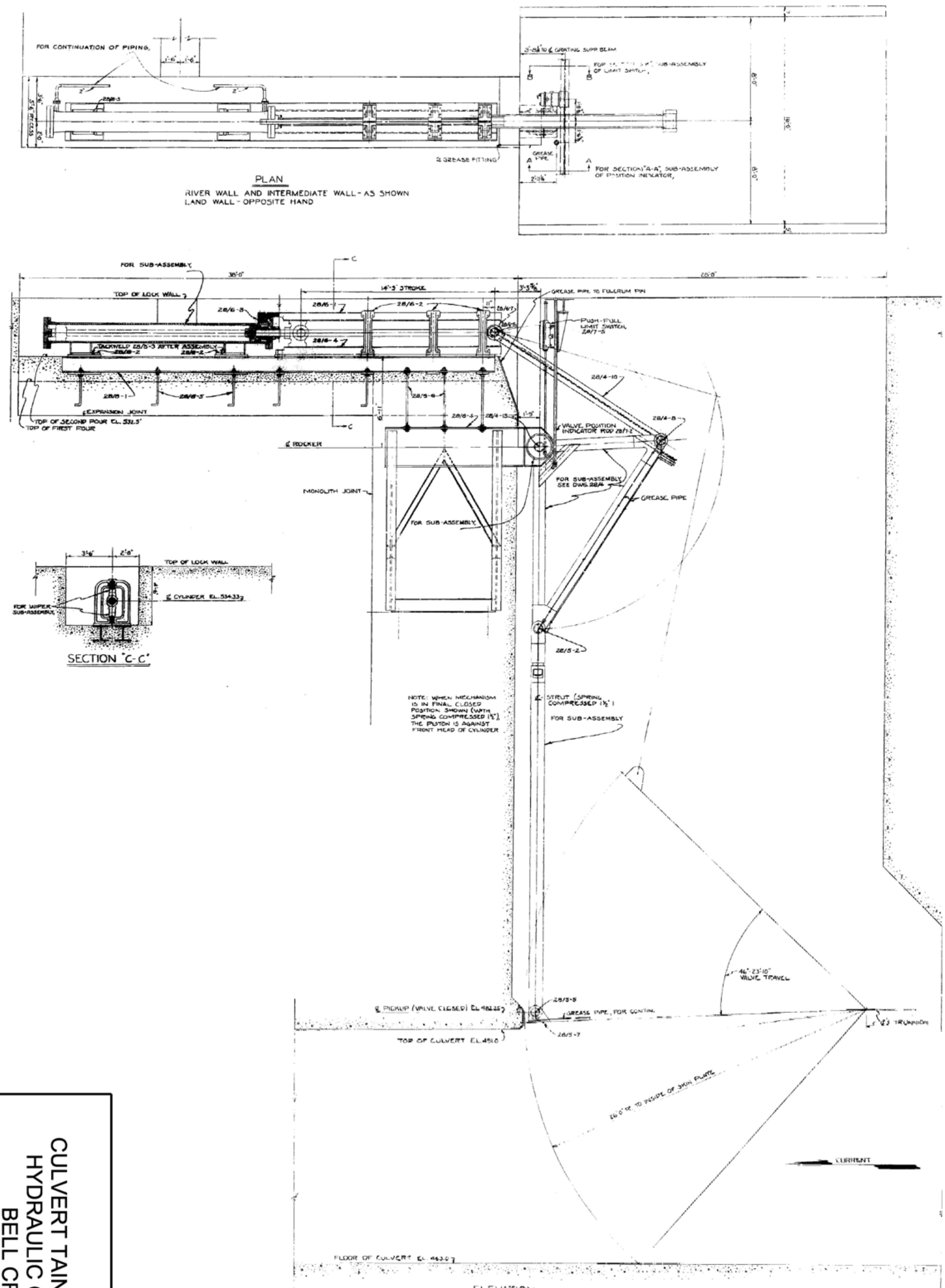
PLATE B-40



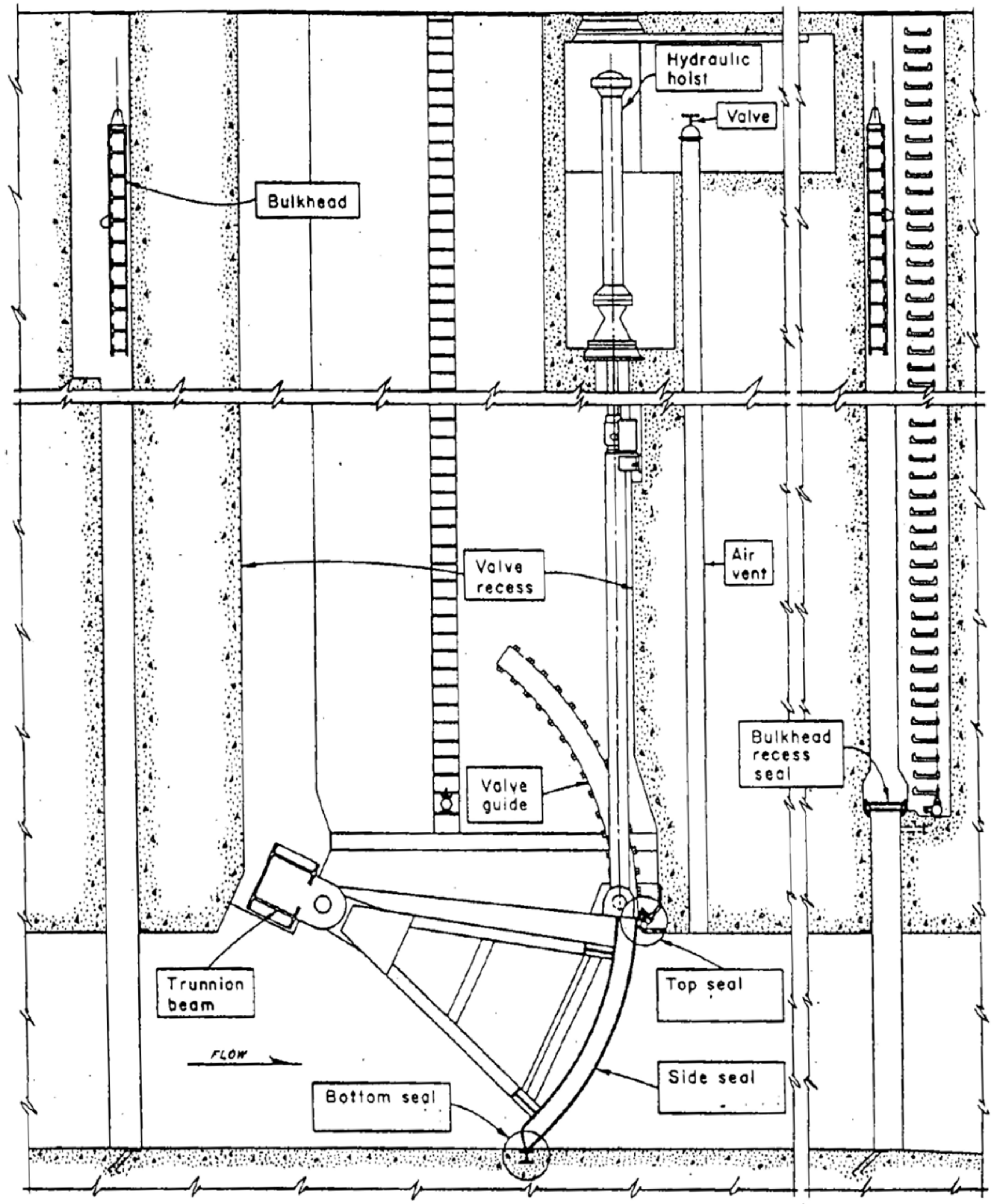
VIEW A-A

SECTIONAL ELEVATION
ASSEMBLY, CULVERT VALVE MACHINERY (4 REQ'D.)

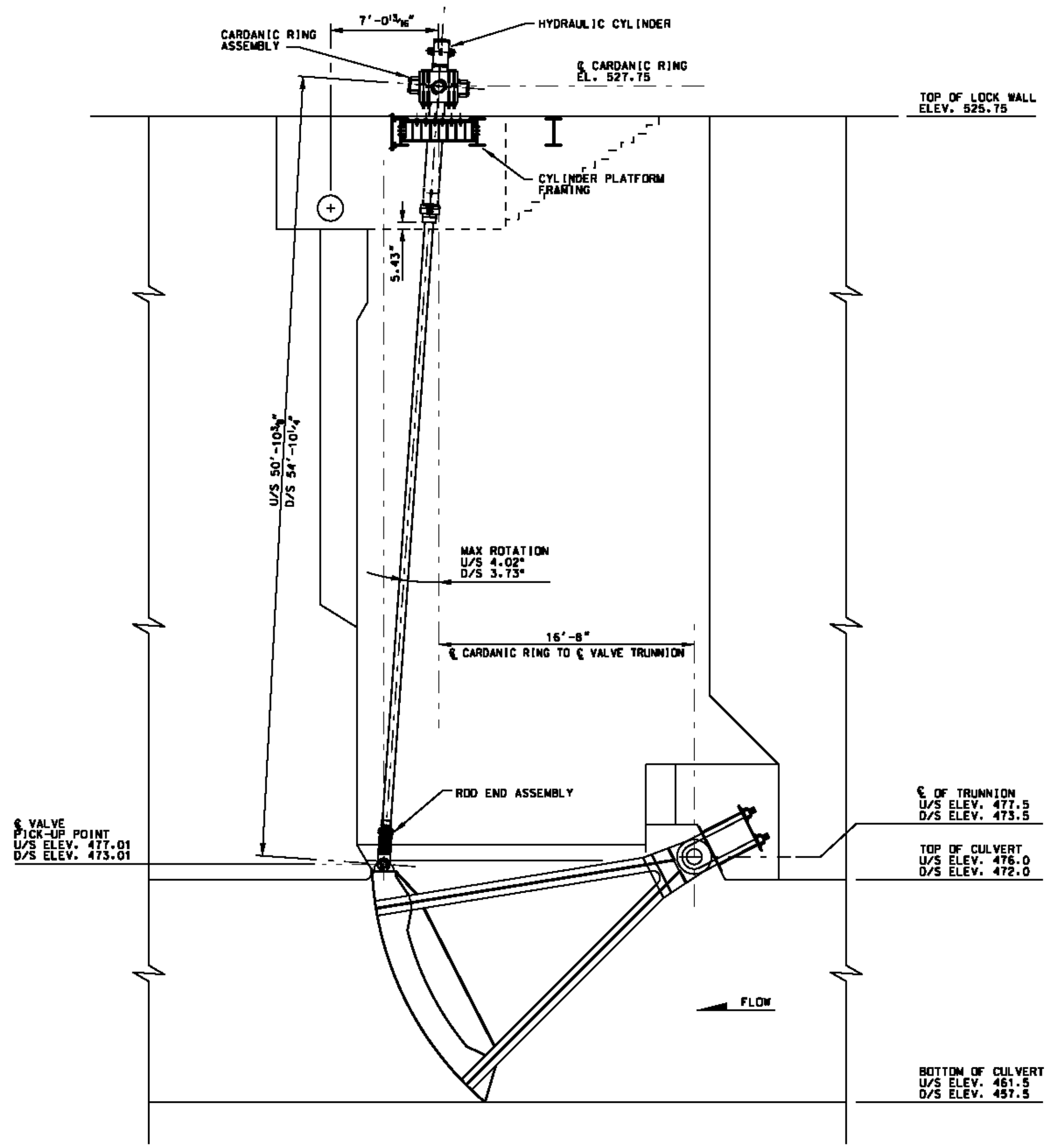
TAINTER VALVE
HYDRAULIC CYLINDER - WIRE ROPE
PLAN & ELEVATION



CULVERT TAINTER VALVE
HYDRAULIC CYLINDER
BELL CRANK
PLAN & ELEVATIONS



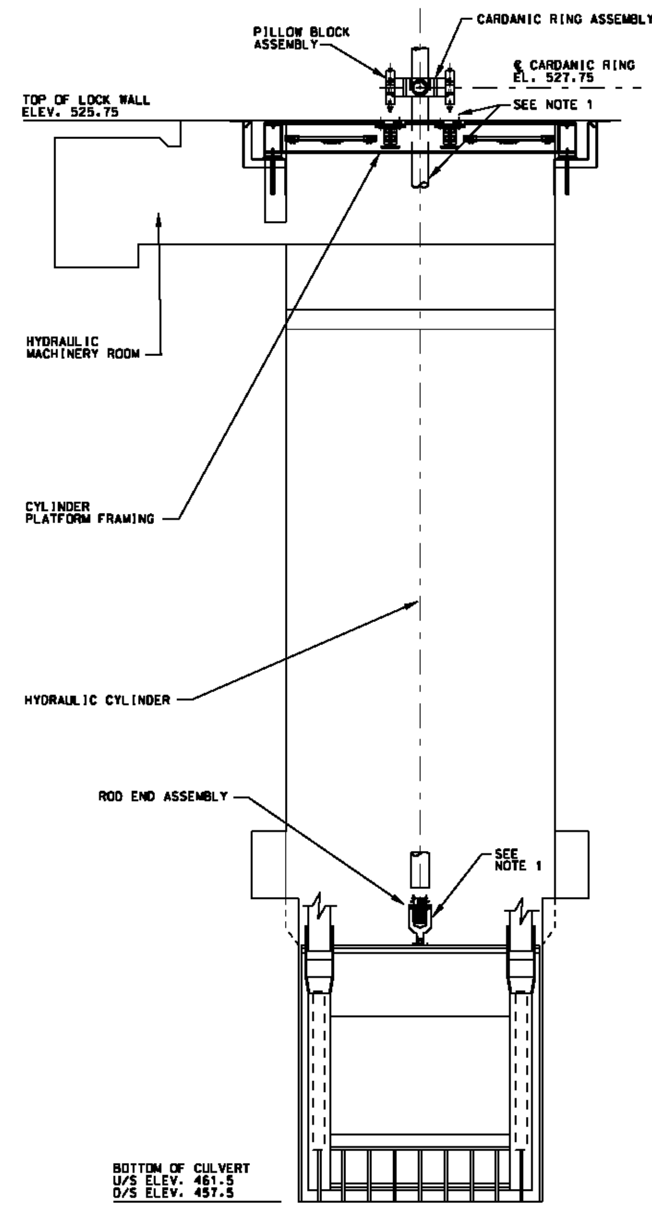
CULVERT TAINTER VALVE
MACHINERY HOIST
ELEVATION



U/S & D/S LANDWALL AND D/S RIVERWALL TAINTER VALVE #1 SHOWN - U/S & D/S TAINTER VALVE #2 RIVERWALL MACHINERY SIMILAR BUT OPPOSITE HAND

U/S TAINTER VALVE ROOM CONDITIONS SHOWN

(A) TAINTER VALVE ELEVATION/SECTION

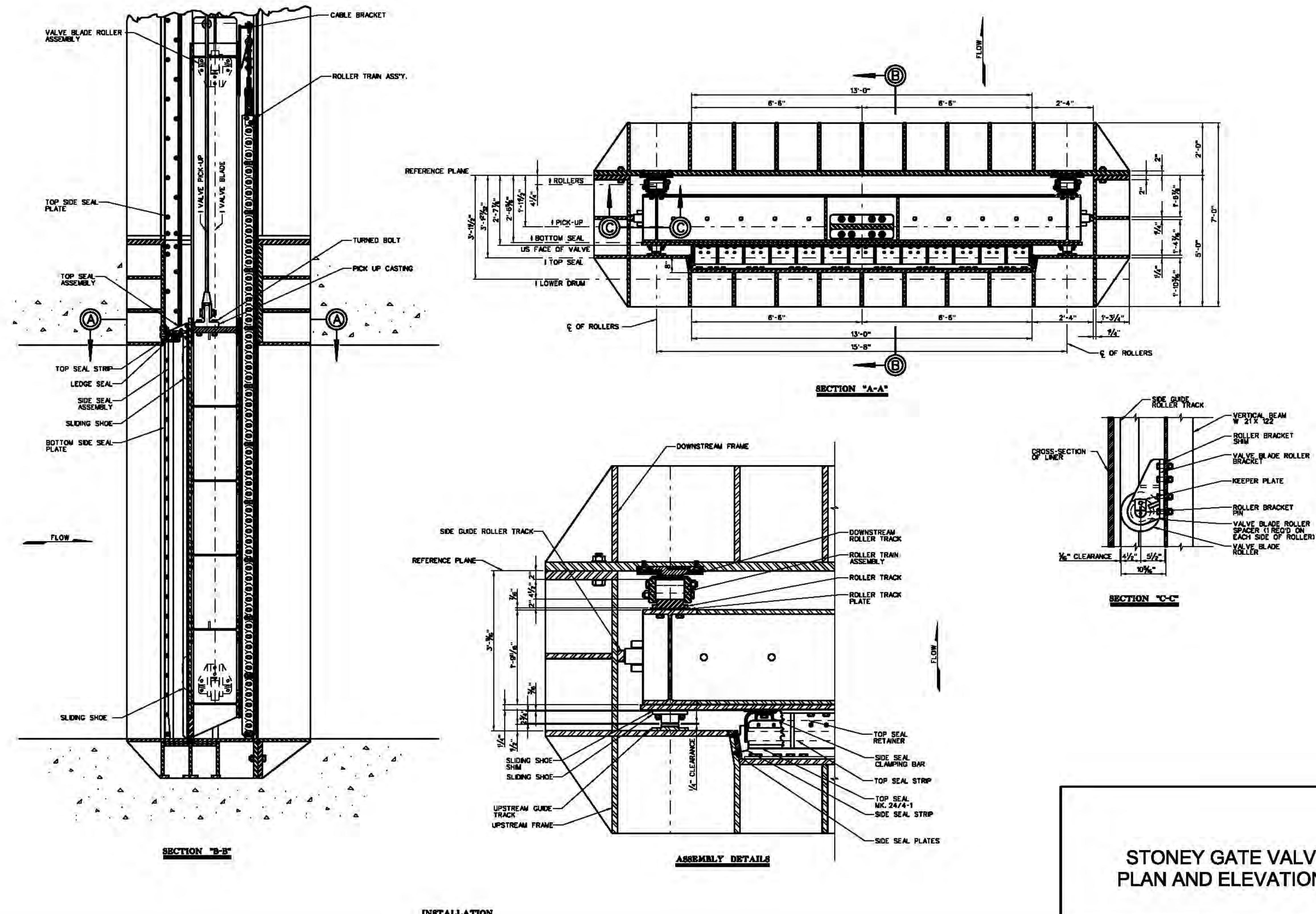


NOTE

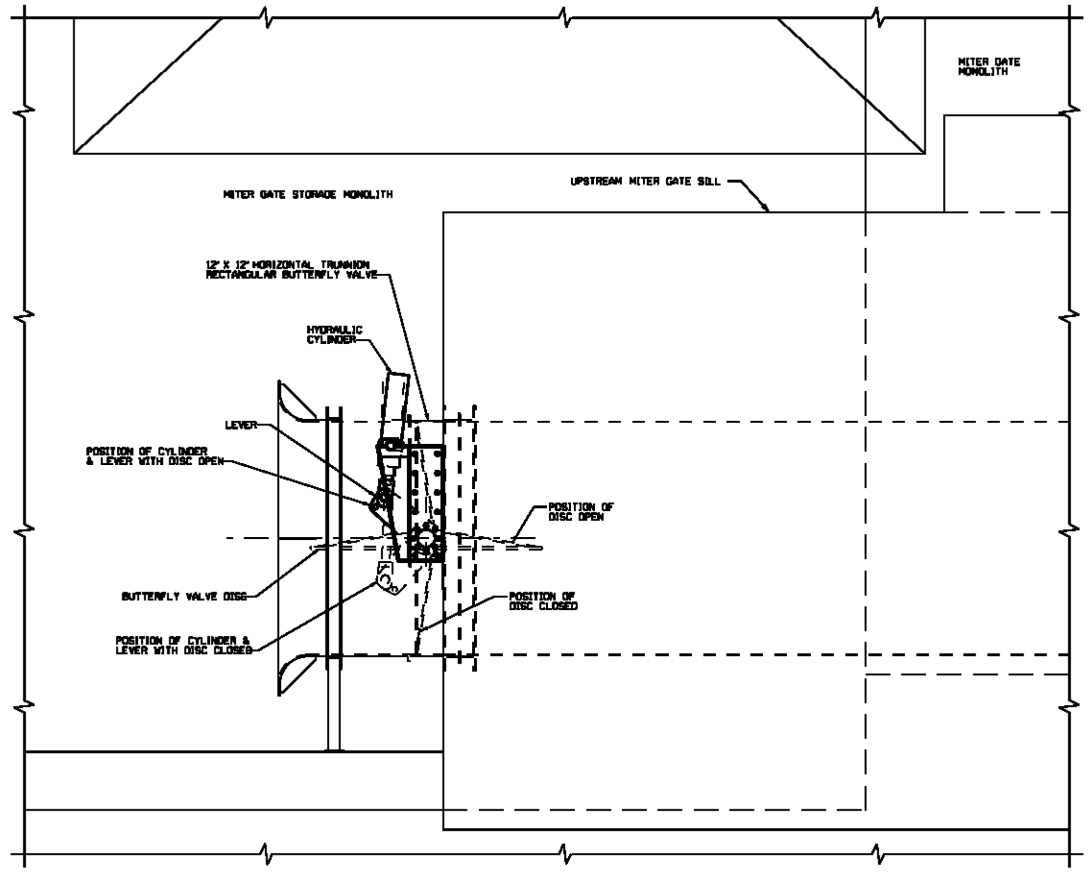
1. CONTRACTOR SHALL INSTALL HYDRAULIC CYLINDER SUPPORT BASE CENTERED AND PLUMB WITH TAINTER VALVE PICK-UP CONNECTION TO ENSURE BINDING IN BEARING AT CONNECTION POINT DOES NOT OCCUR.

(B) TAINTER VALVE SECTION

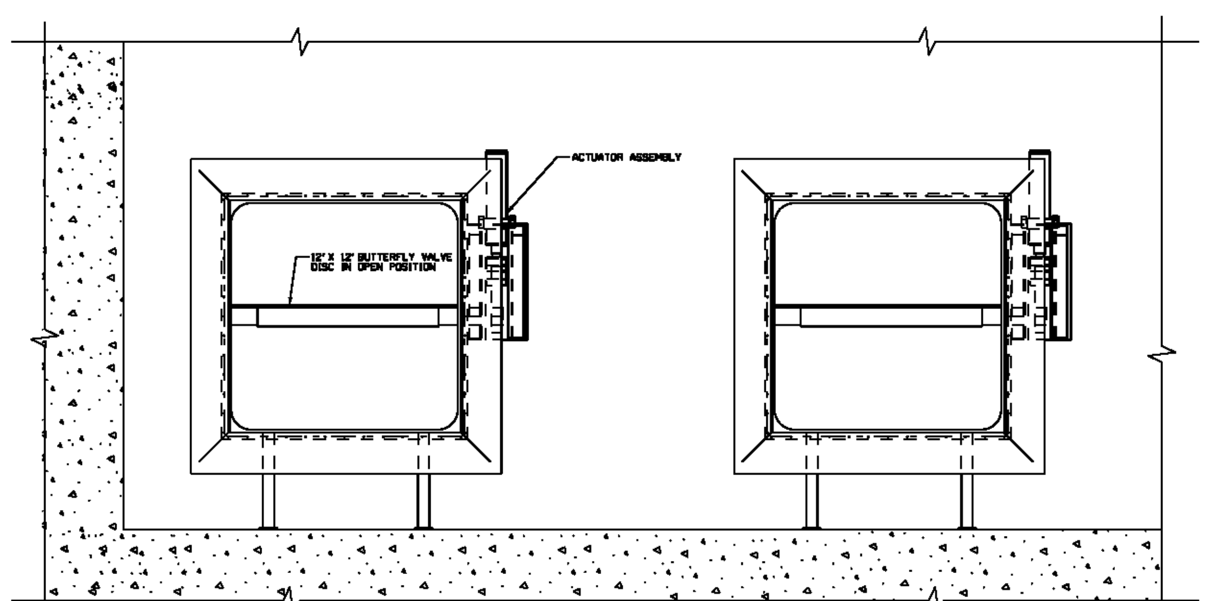
TAINTER VALVE
HYDRAULIC CYLINDER
DIRECT ACTING
ELEVATION



STONEY GATE VALVE
PLAN AND ELEVATIONS

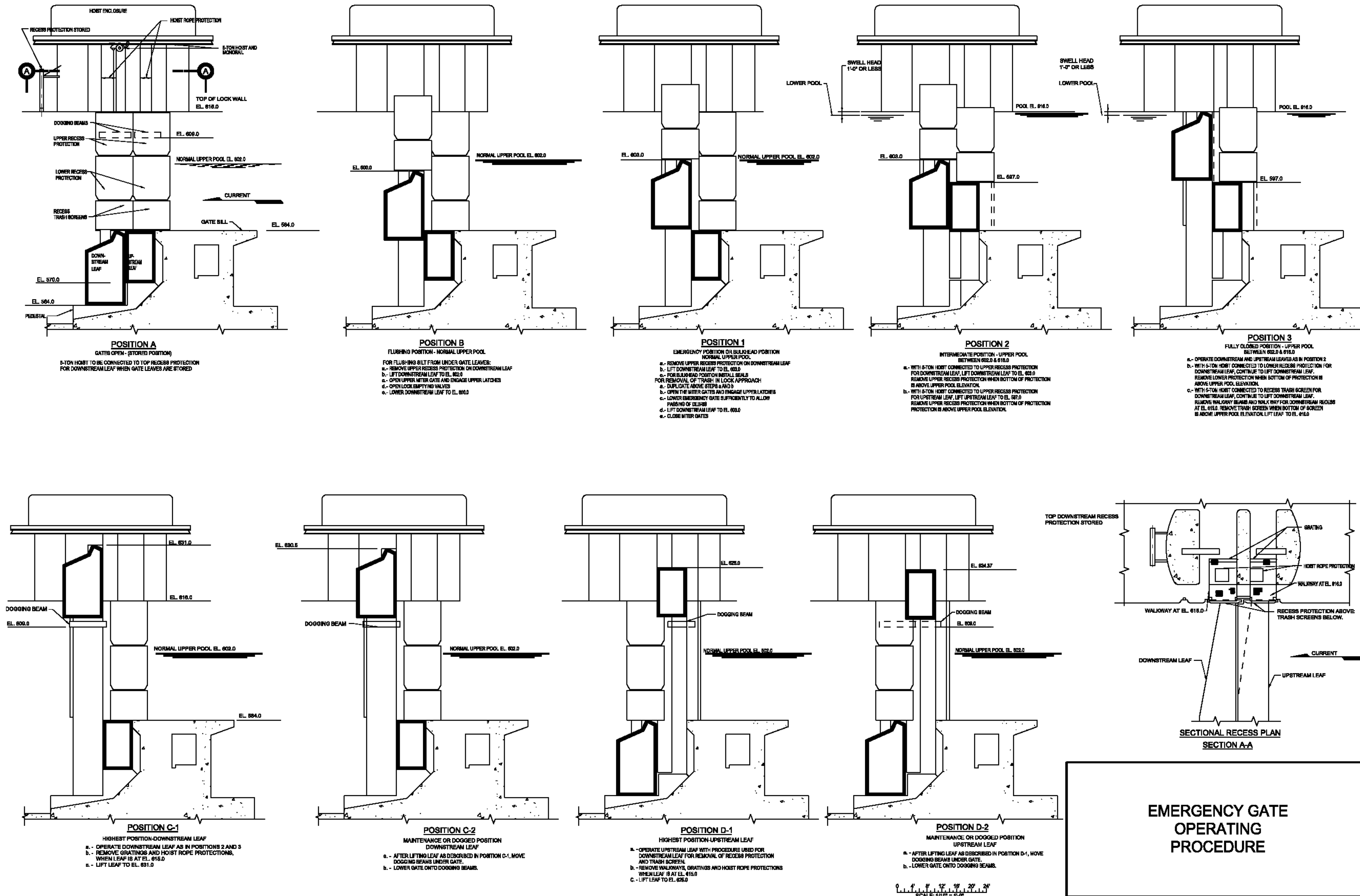


FILL VALVE ELEVATION



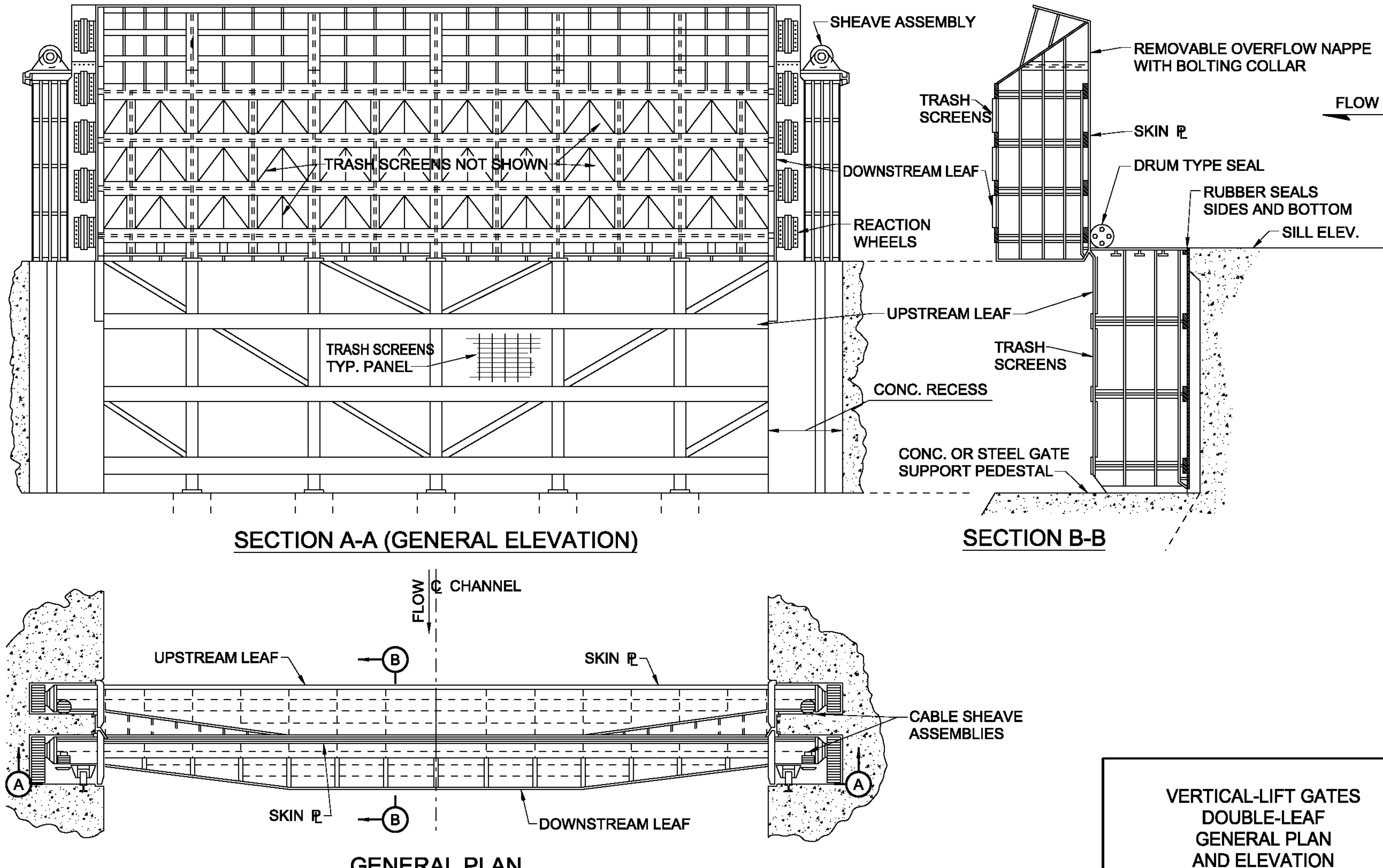
FILL VALVE PARTIAL UPSTREAM ELEVATION

BUTTERFLY VALVE ELEVATIONS

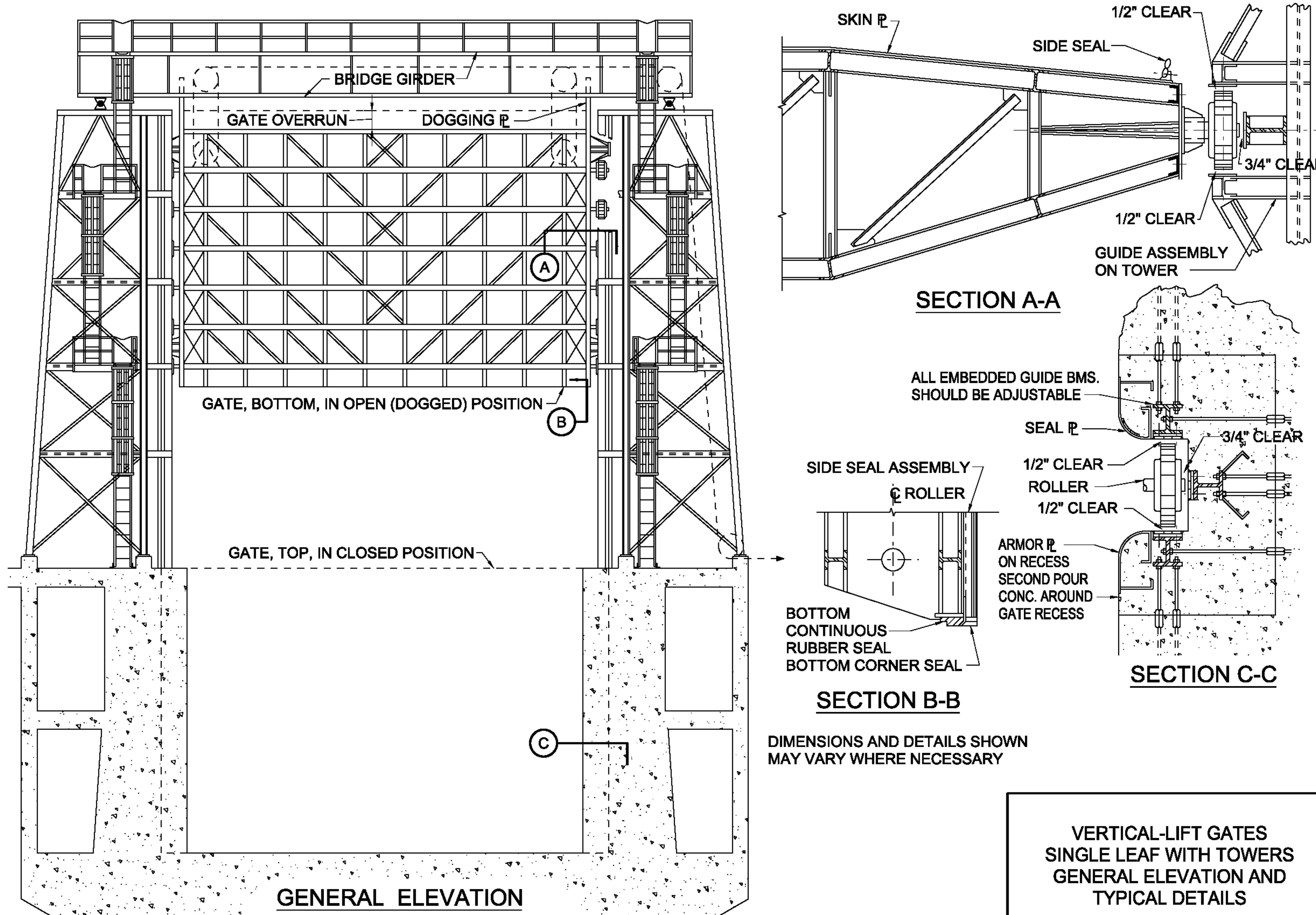


EMERGENCY GATE
OPERATING
PROCEDURE

PLATE B-47

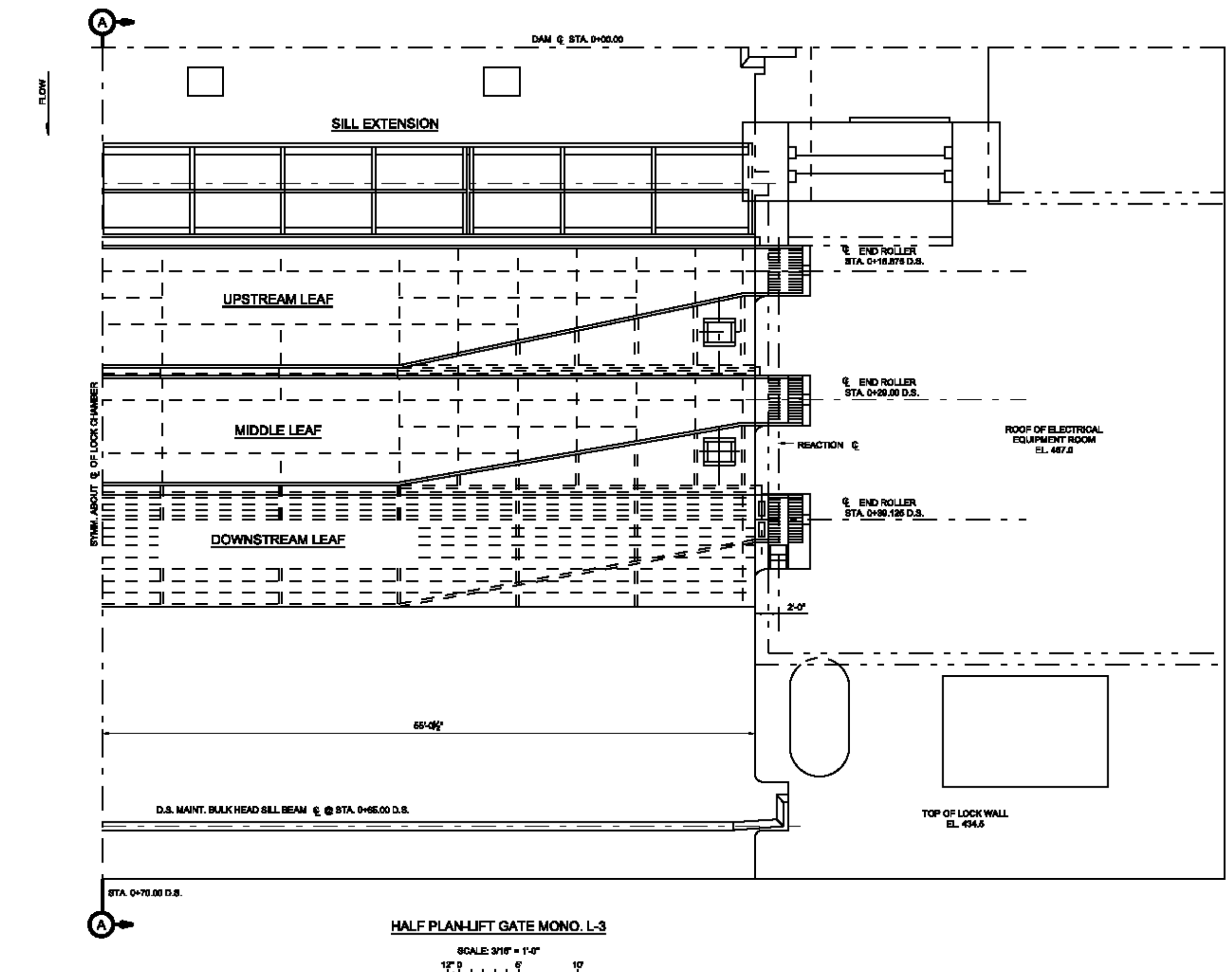


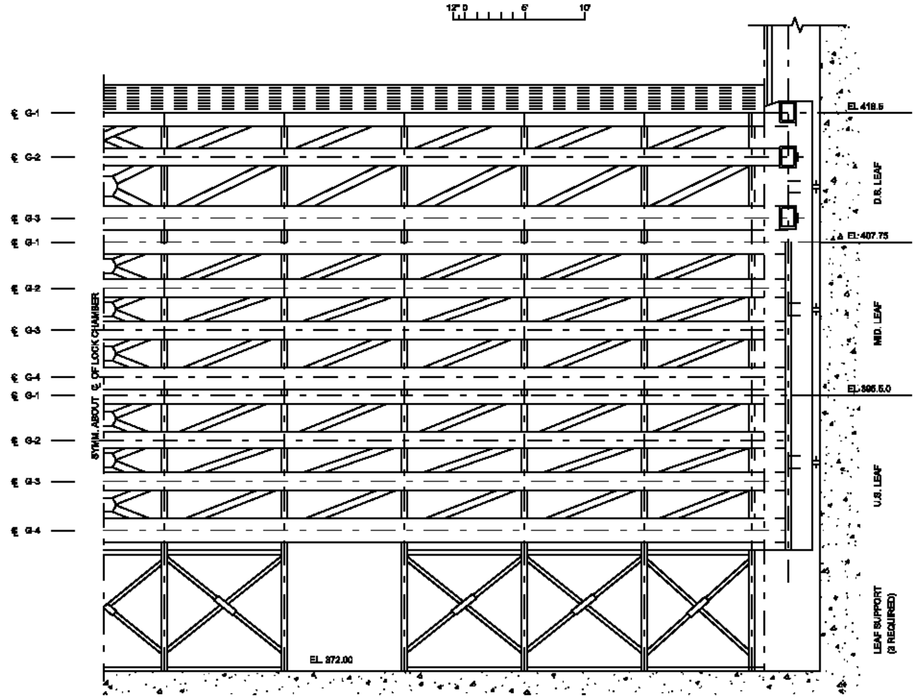
VERTICAL-LIFT GATES
DOUBLE-LEAF
GENERAL PLAN
AND ELEVATION

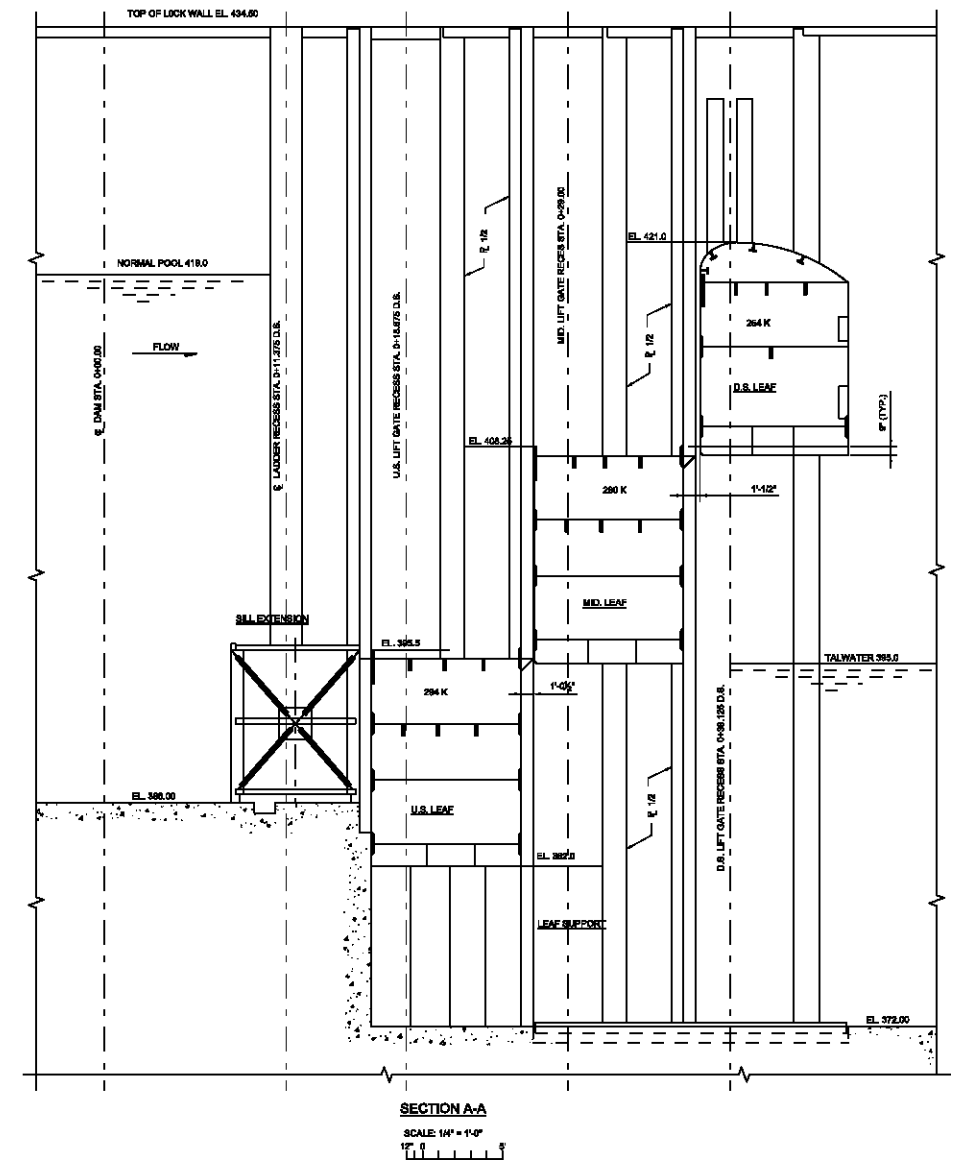


DIMENSIONS AND DETAILS SHOWN MAY VARY WHERE NECESSARY

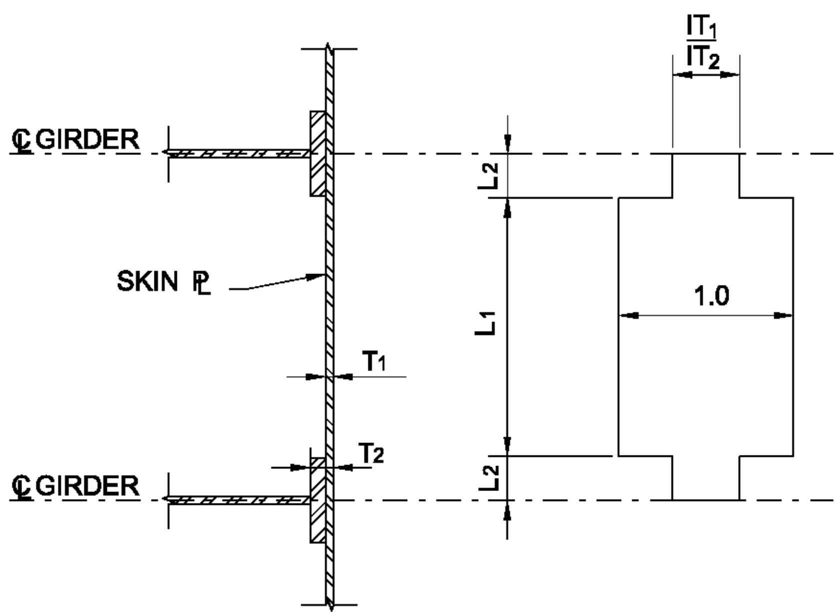
VERTICAL-LIFT GATES
SINGLE LEAF WITH TOWERS
GENERAL ELEVATION AND
TYPICAL DETAILS





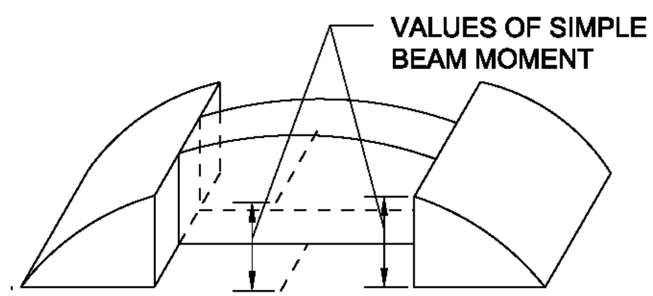


VERTICAL-LIFT GATES
MULTI-LEAF GATES
GENERAL PLAN AND
ELEVATION



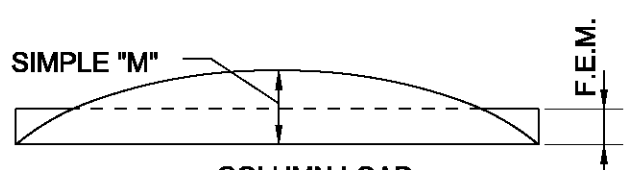
$\frac{1}{EI}$ = 1.0 FOR (CENTER OF ANALOGOUS COLUMN)

COLUMN AREA $A = (l_1 \times 1.0) + 2 \times l_2 \times \frac{It_1}{It_2}$



$\frac{M}{EI}$ DIAGRAM

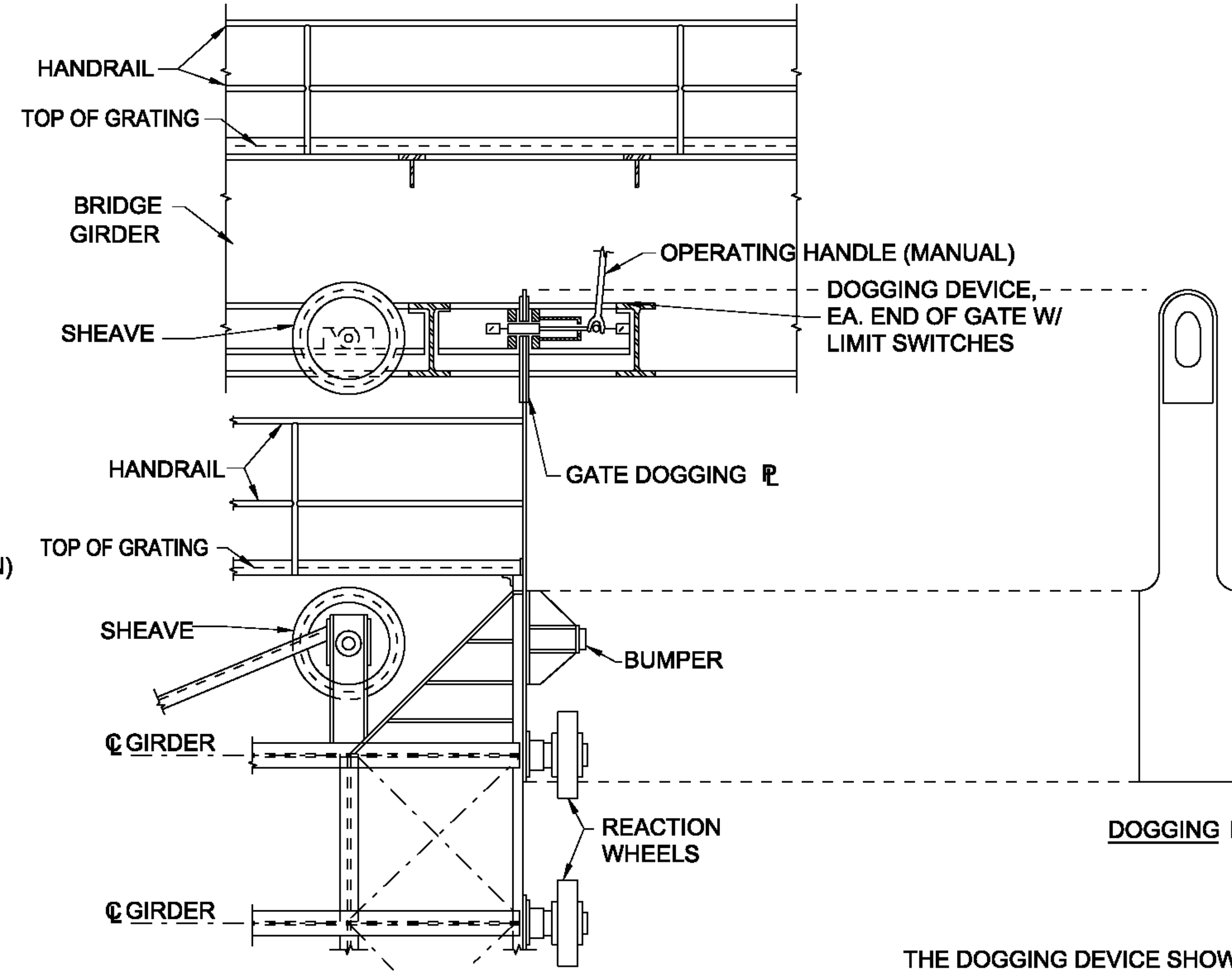
ANALOGOUS COLUMN LOAD IS EQUIVALENT TO THE VOLUME OF THE DIAGRAM SHOWN, USING PARABOLIC CURVES AS THE SHAPE OF THE DIAGRAM.



$\text{F.E.M.} = \frac{\text{COLUMN LOAD}}{\text{COLUMN AREA}}$

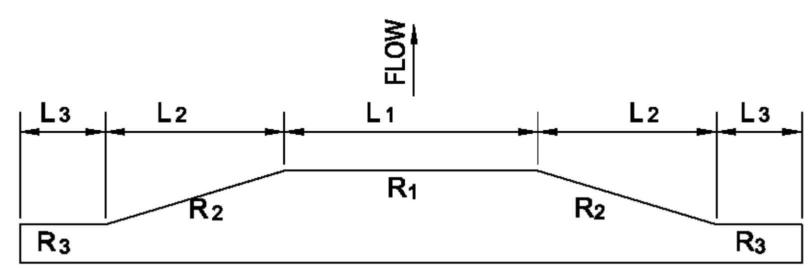
ACTUAL MOMENT = SIMPLE "M" MINUS F.E.M. (MIDSPAN)

SKIN PLATE DESIGN DATA



GATE DOGGING DEVICE

THE DOGGING DEVICE SHOWN IS FOR A TYPICAL SINGLE-LEAF GATE WITH A BRIDGE SPANNING BETWEEN TOWERS. DETAILS SHOWN ARE FOR INFORMATION AND MAY VARY WHERE NECESSARY.

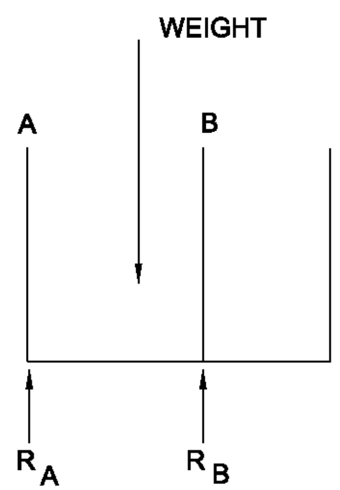


EFFECTIVE RADIUS OF GYRATION $= \frac{(L_1 \times R_1)+(2L_2 \times R_2)+(2L_3 \times R_3)}{L_1 + 2L_2 + 2L_3}$

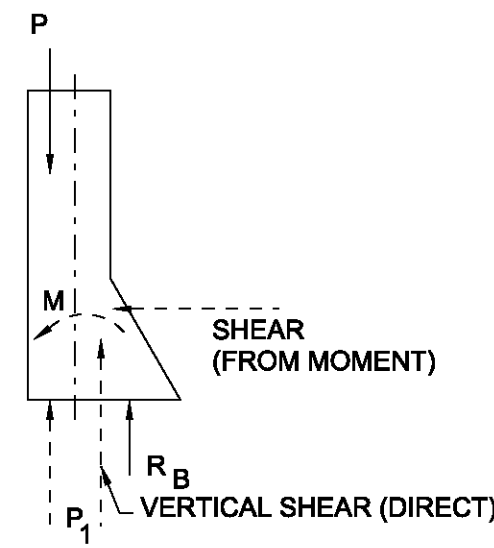
THE VALUES OF "r" FOR THE TAPERED SECTION SHOULD BE CONSIDERED AS THE AVERAGE OF THE MAXIMUM AND MINIMUM OF "r".

THE EFFECTIVE LENGTH SHOULD BE THE TOTAL LENGTH OF THE GIRDER, CENTER TO CENTER OF BEARING GIRDERS.

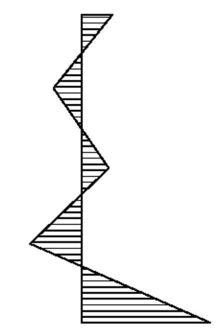
VERTICAL-LIFT GATES
TYPICAL DESIGN DATA



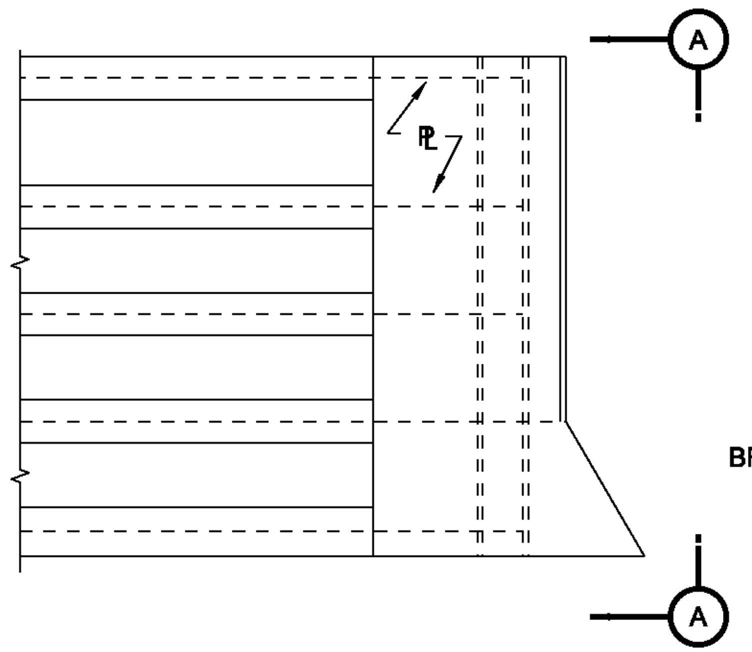
R_A AND R_B ARE REACTIONS
APPLIED TO BRACKETS



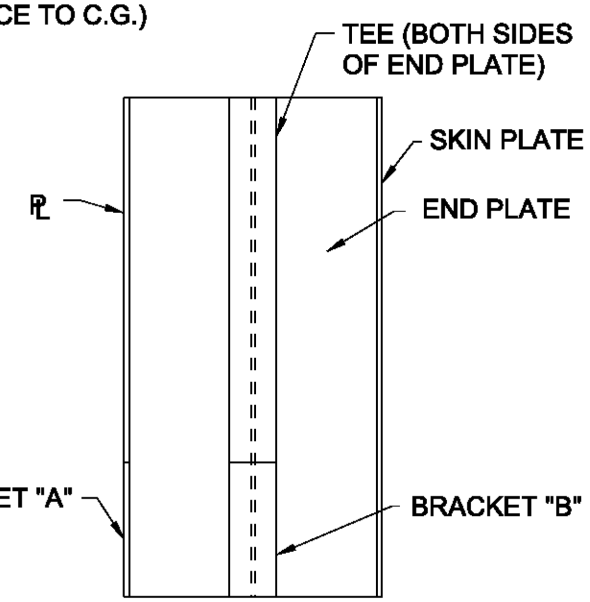
EFFECTIVE COLUMN
REPLACE R_B WITH P AND M
(SYSTEM STILL HAS TO BE ADJUSTED
FOR DISTANCE TO C.G.)



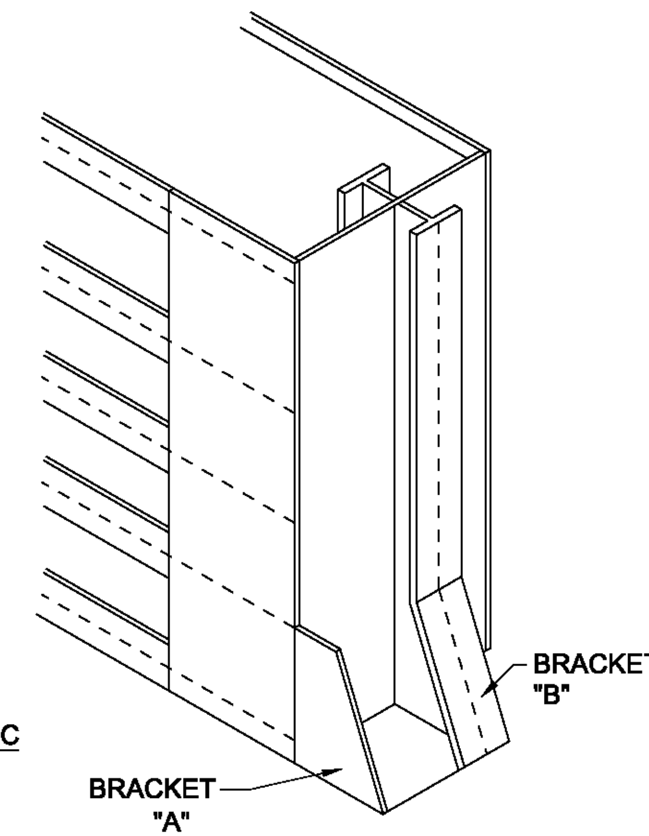
MOMENT DIAGRAM



ELEVATION

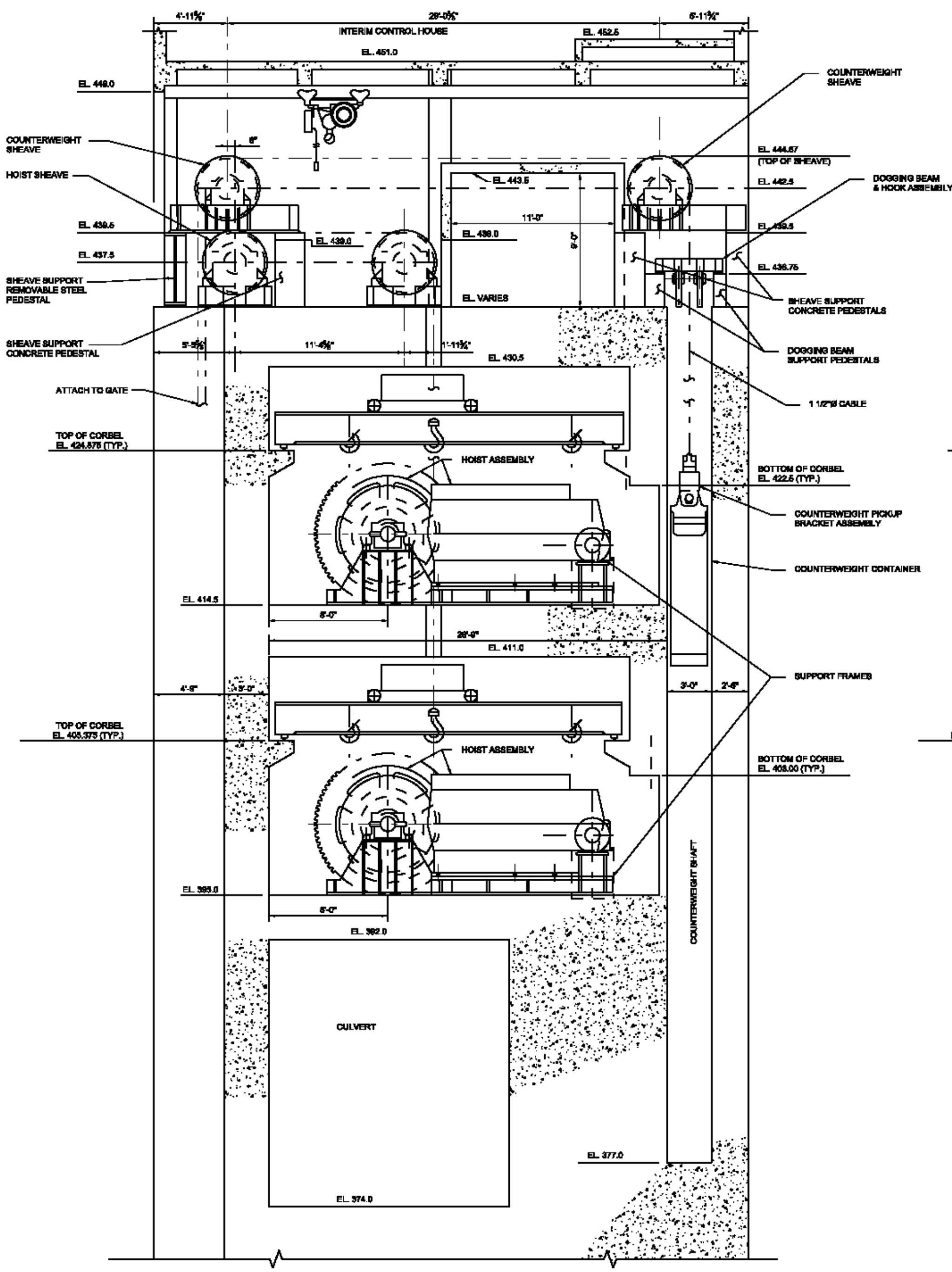


VIEW A - A

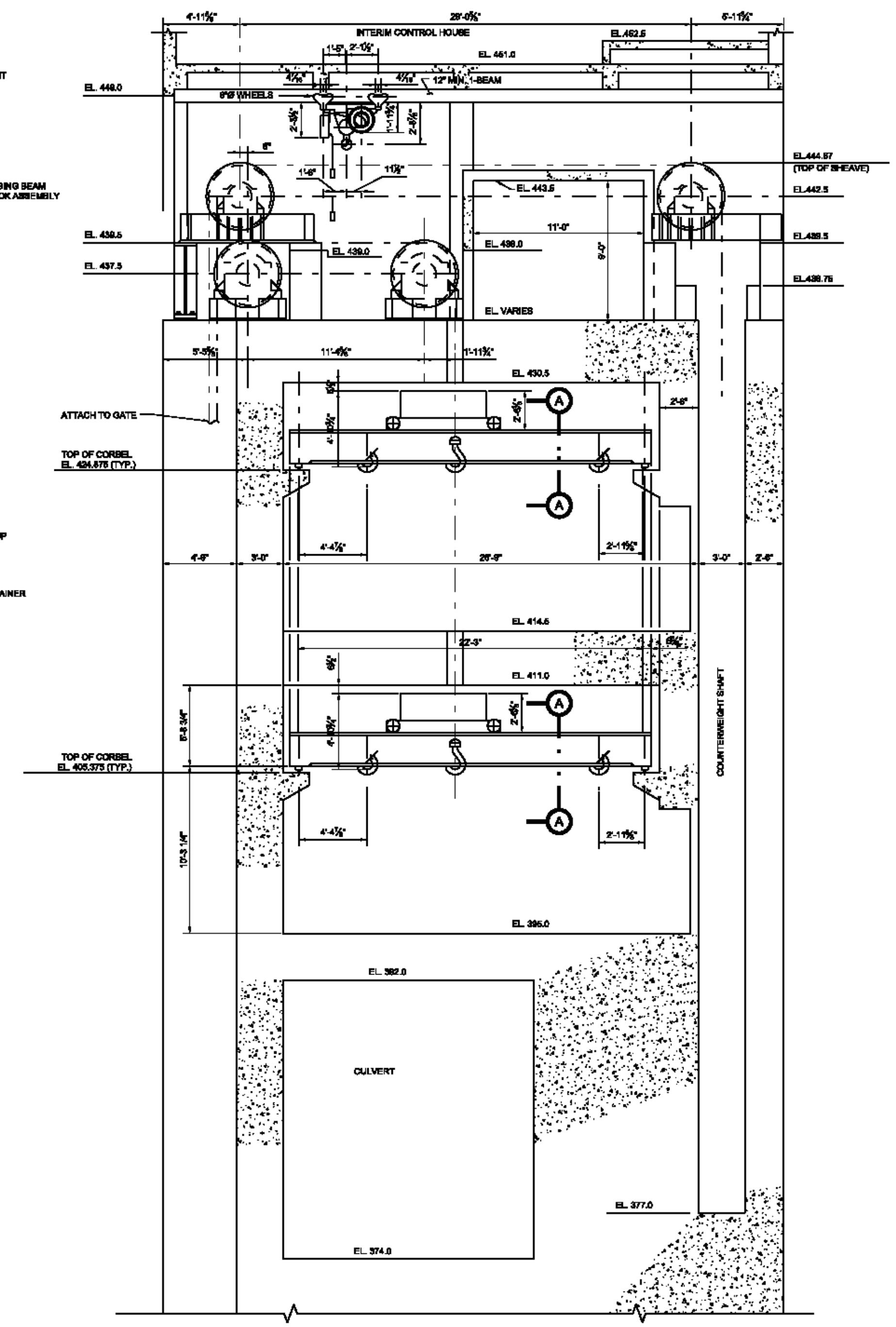


ISOMETRIC

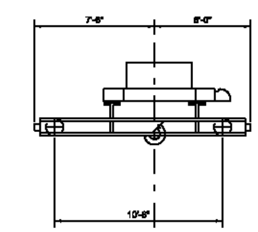
VERTICAL-LIFT GATES
END GIRDERS
SUPPORT BRACKETS
DESIGN DATA



ELEVATION (TYP.)
SCALE: 1/4" = 1'-0"

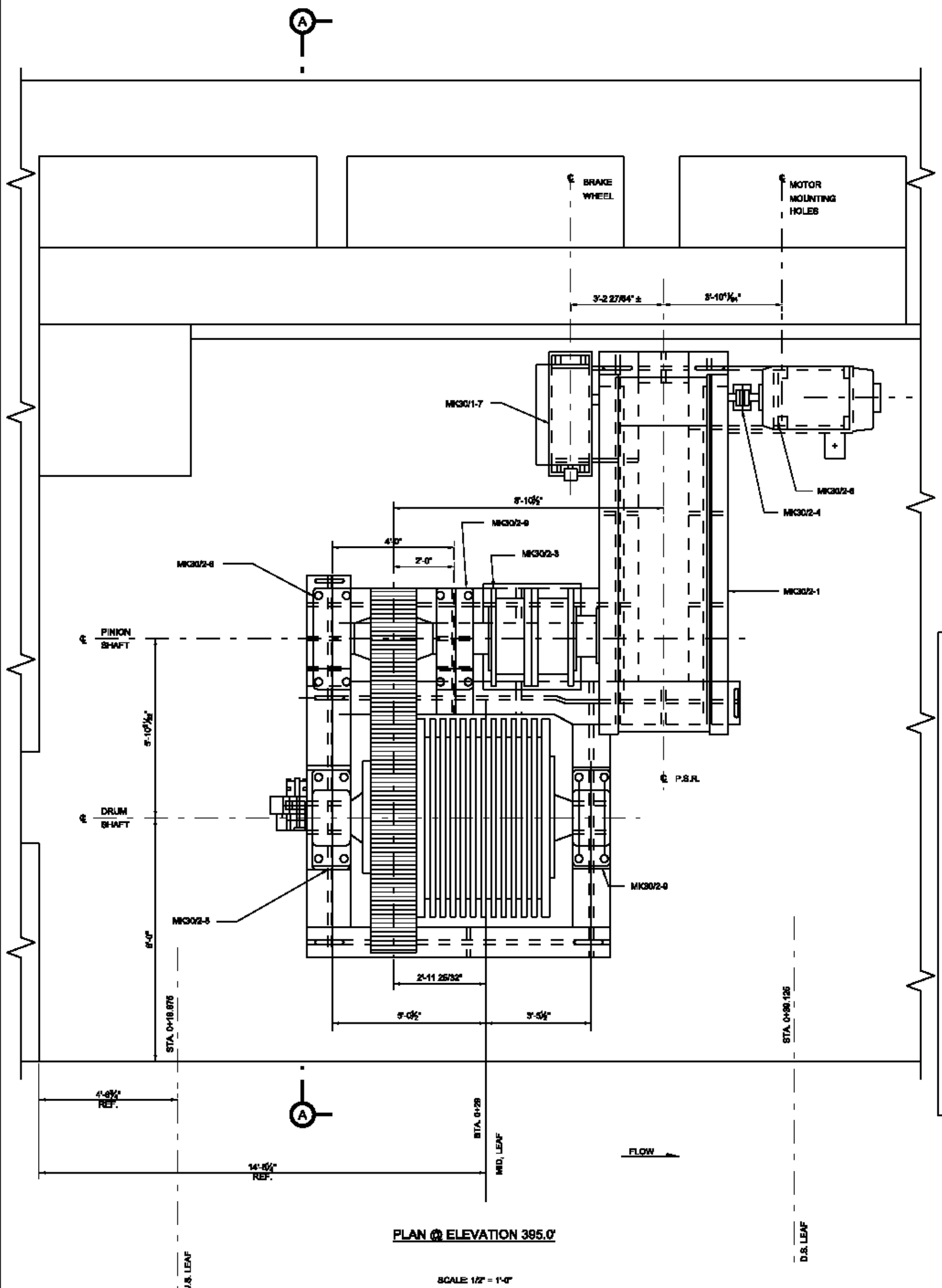


HOIST SYSTEMS ELEVATION (TYP.)
SCALE: 1/4" = 1'-0"



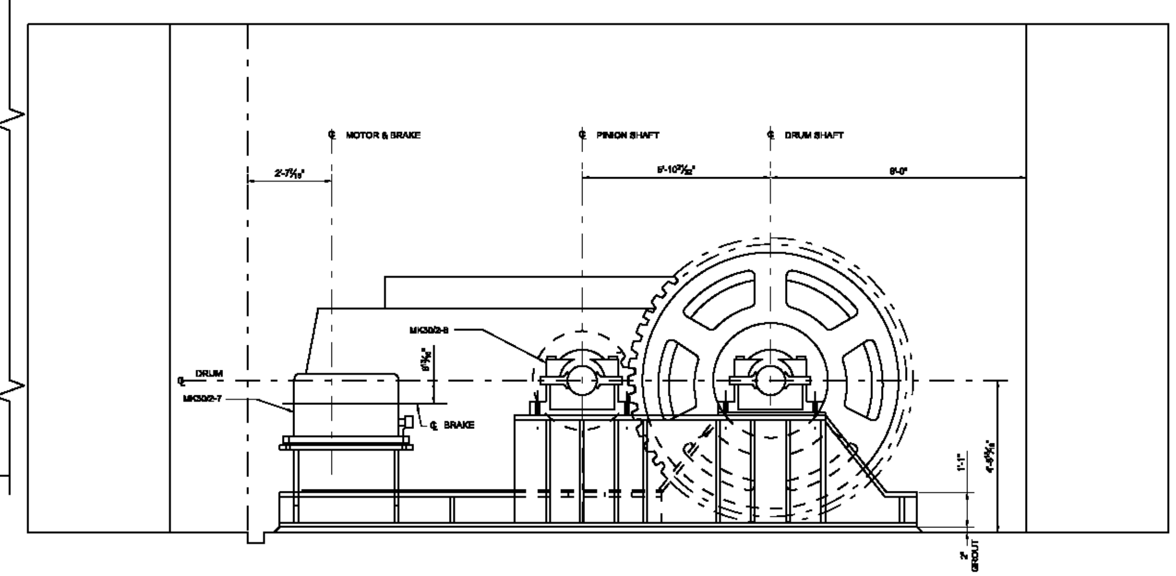
SECTION A-A
SCALE: 1/4" = 1'-0"

GATE MACHINERY
MULTI-LEAF
VERTICAL-LIFT GATES
ELEVATION-HOIST
MECHANISM



PLAN @ ELEVATION 395.0'

SCALE: 1/2" = 1'-0"

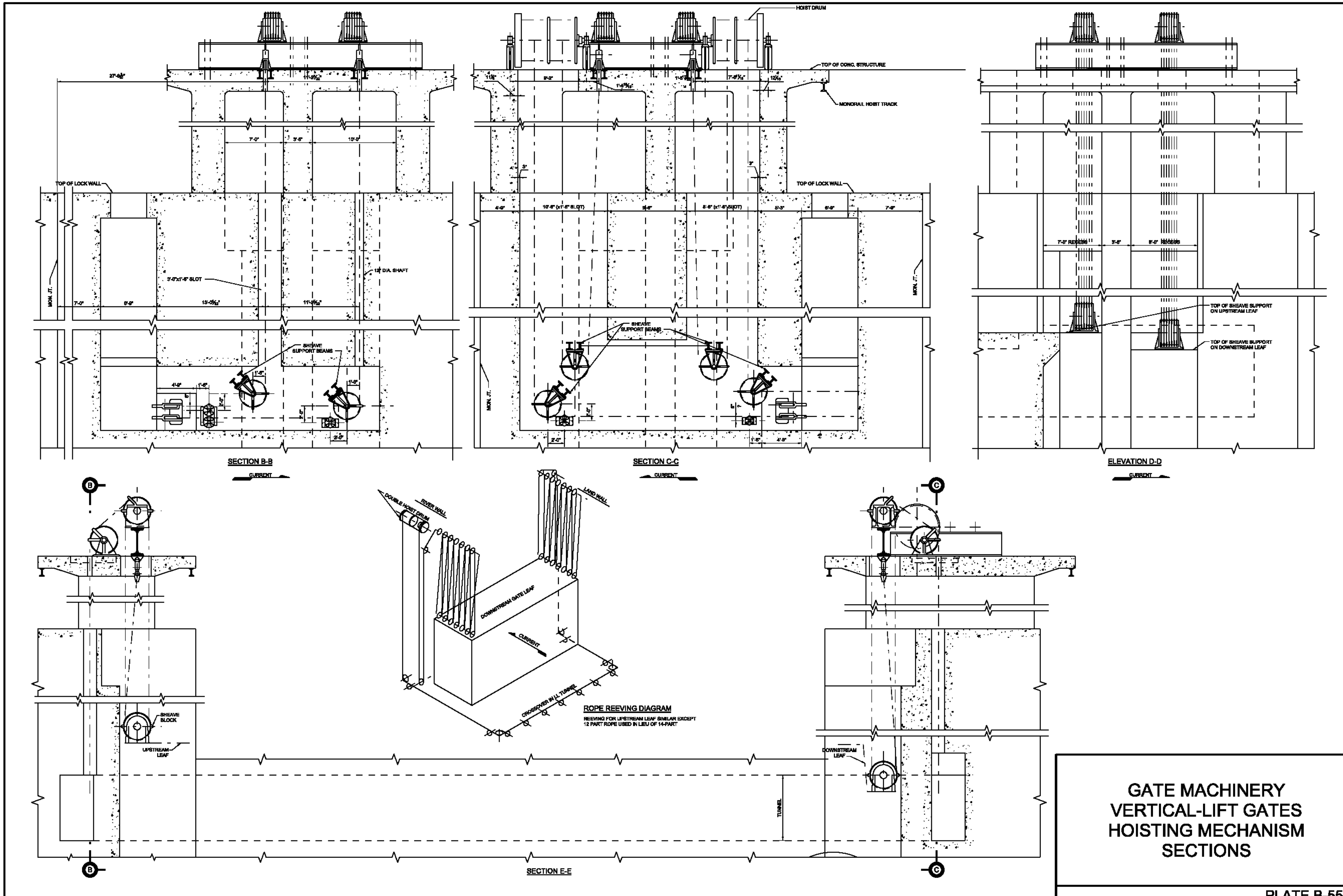


SECTION A-A

SCALE: 1/2" = 1'-0"

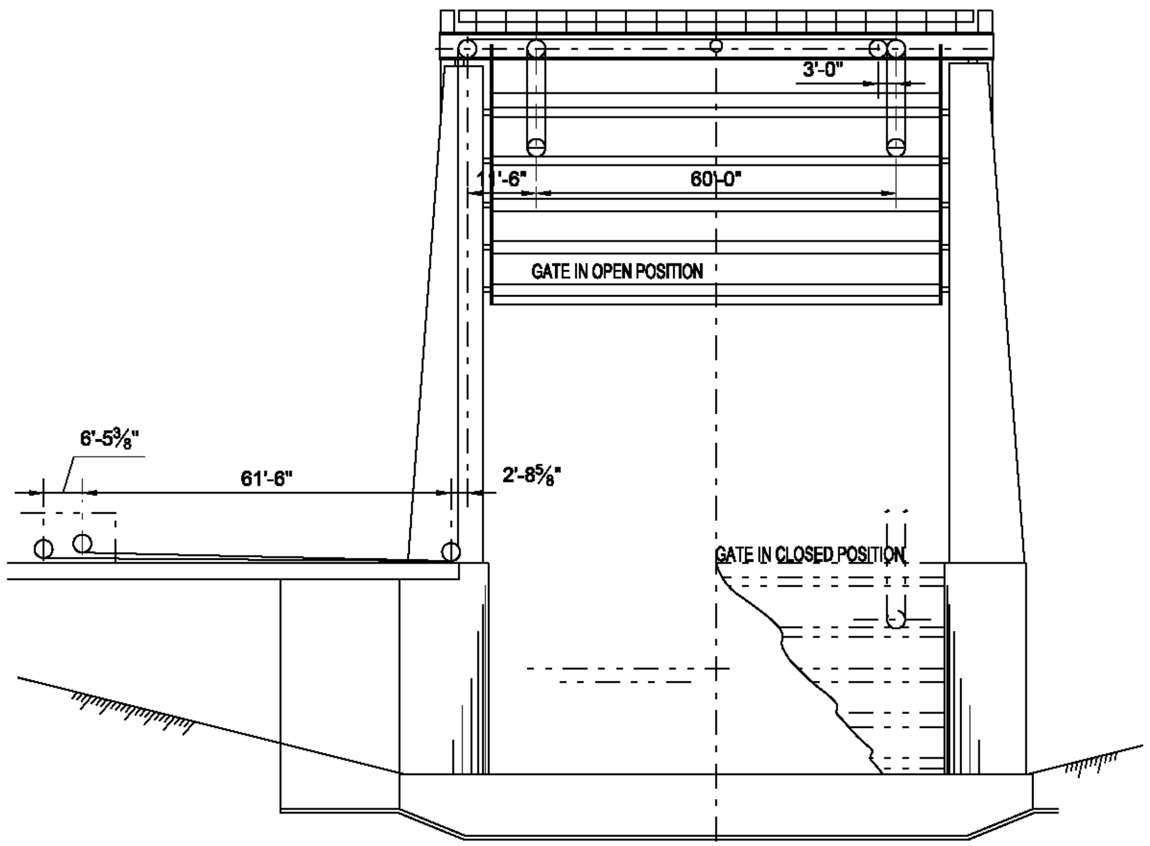
LIST OF PARTS NOT DETAILED			
MARK NO.	NO. REQ.'D.	DESCRIPTION	MATERIAL
MK30/2-1	3	PARALLEL SHAFT REDUCER-SIMILAR AND EQUAL TO "PHILADELPHIA GEAR CORP." UNIT SIZE 215HP4, FIG.7, AS MODIFIED HEREIN, WITH EXTENDED SHAFT TO MK30/2-7. AS REQUIRED.	COMM. GRADE
MK30/2-2	3	PARALLEL SHAFT REDUCER-SIMILAR AND EQUAL TO "PHILADELPHIA GEAR CORP." UNIT SIZE 215HP4, FIG.8, AS MODIFIED HEREIN, WITH EXTENDED SHAFT TO MK30/2-7, AS REQUIRED.	COMM. GRADE
MK30/2-3	6	FLEXIBLE COUPLING-SIMILAR AND AND EQUAL TO "FALK" TYPE G20, SERIES 2160G.	COMM. GRADE
MK30/2-4	6	FLEXIBLE COUPLING-SIMILAR AND AND EQUAL TO "FALK" TYPE G52, SERIES 1025G.	COMM. GRADE
MK30/2-5	2	ELECTRIC MOTOR-75HP, 400-1200 RPM ADJUSTABLE 500 VOLT D.C. -SIMILAR AND EQUAL TO "NORTHWESTERN ELECTRIC" FRAME NO. 508AT.	COMM. GRADE
MK30/2-6	4	ELECTRIC MOTOR-401HP, 400-1200 RPM ADJUSTABLE, 500 VOLT D.C. -SIMILAR AND EQUAL TO "NORTHWESTERN ELECTRIC" FRAME NO. 508AT.	COMM. GRADE
MK30/2-7	6	HOLDING BRAKE, 500-1000 FT-LB., 16" @ WHEEL-SIMILAR AND EQUAL TO "EATON-CUTLER HAMMER" TYPE GH505, SIZE 16.	COMM. GRADE
MK30/2-8	12	PILLOW BLOCK-SIMILAR AND EQUAL TO "TORRINGTON" SDAFS24160, FIXED TYPE, WTIH LOCKNUT NO60 AND LOCKPLATE P60.	COMM. GRADE
MK30/2-9	12	PILLOW BLOCK-SIMILAR AND EQUAL TO "TORRINGTON" SDAFS24160, FLOAT TYPE, WITH LOCKNUT NO60 AND LOCKPLATE P60.	COMM. GRADE
MK30/2-10	22	TUBING-RECTANGULAR 2 X 3 X 0.12" X 1'-7" LONG.	ASTM A511, GR. MT304
MK30/2-11	4	BAR 1/4 X 4 X 4'-0"	ASTM A666, TYPE 304
MK30/2-12	4	L-5 X 3 1/2 X 1/2 X 4'-4 1/2".	ASTM A36
MK30/2-13	44	HEX. HD. CAPSCREW-1/2"-13UNC-2A X 5" LONG WITH NUT AND LOCKWASHER.	ASTM A193, GR. B8 & ASTM A194, GR. 8

GATE MACHINERY
MULTI-LEAF
VERTICAL-LIFT GATES
GENERAL ARRANGEMENT-
HOIST MECHANISM

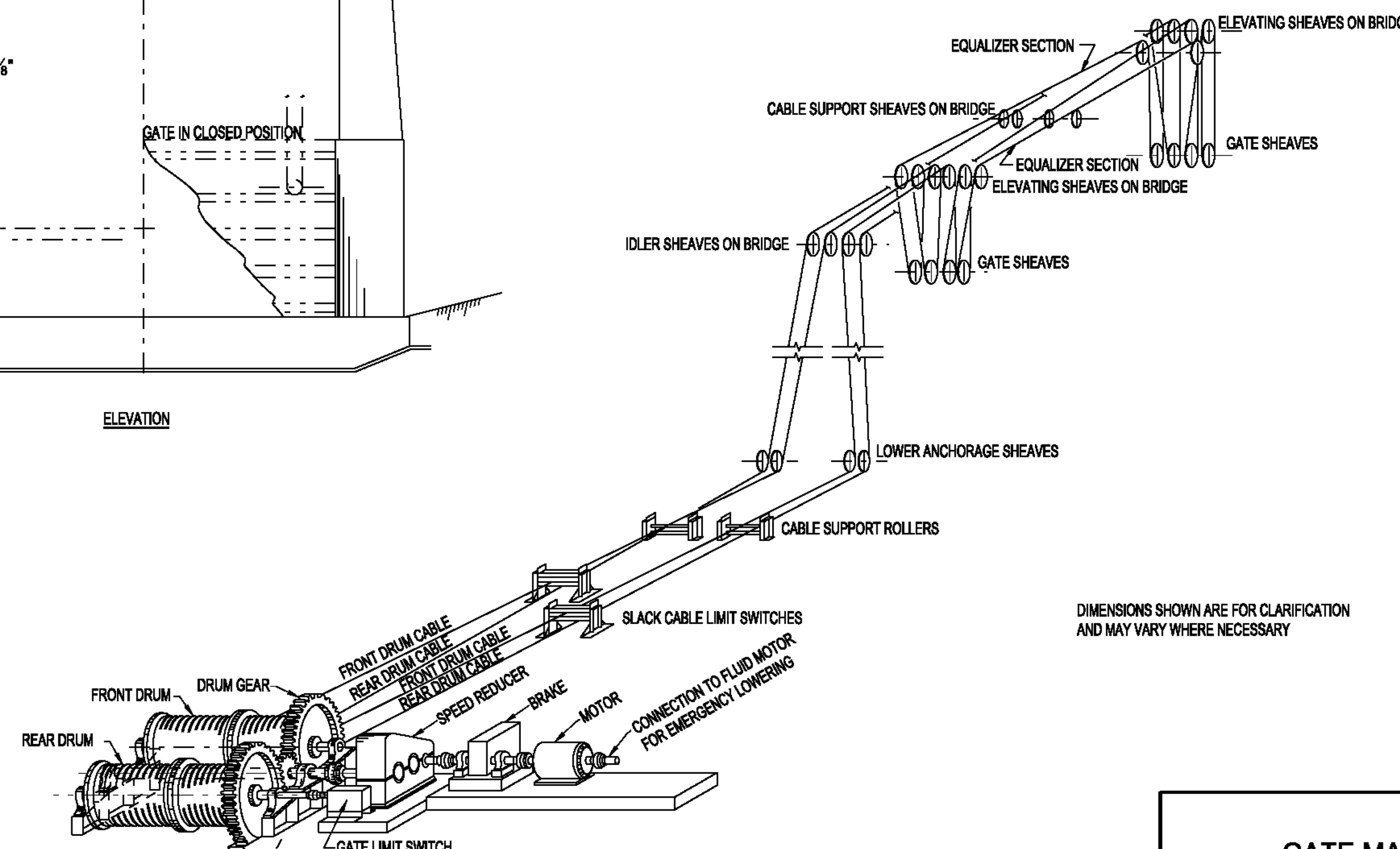


GATE MACHINERY
VERTICAL-LIFT GATES
HOISTING MECHANISM
SECTIONS

PLATE B-55



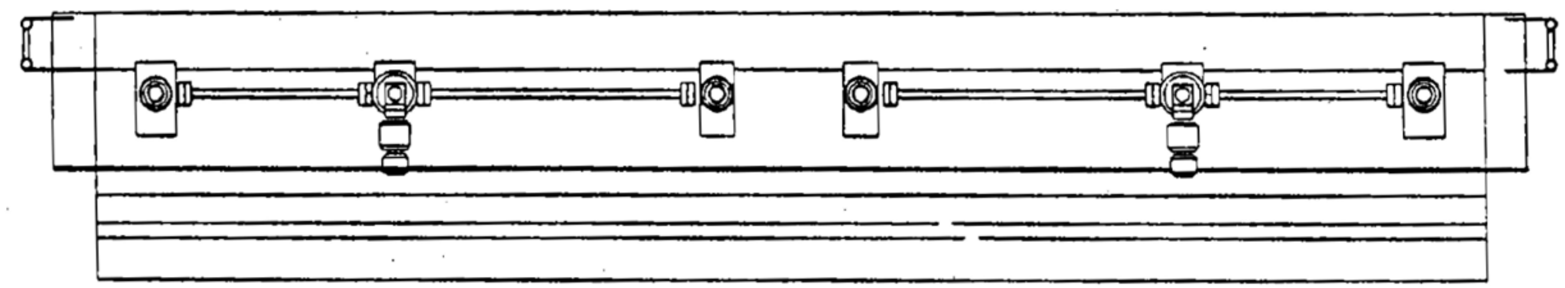
ELEVATION



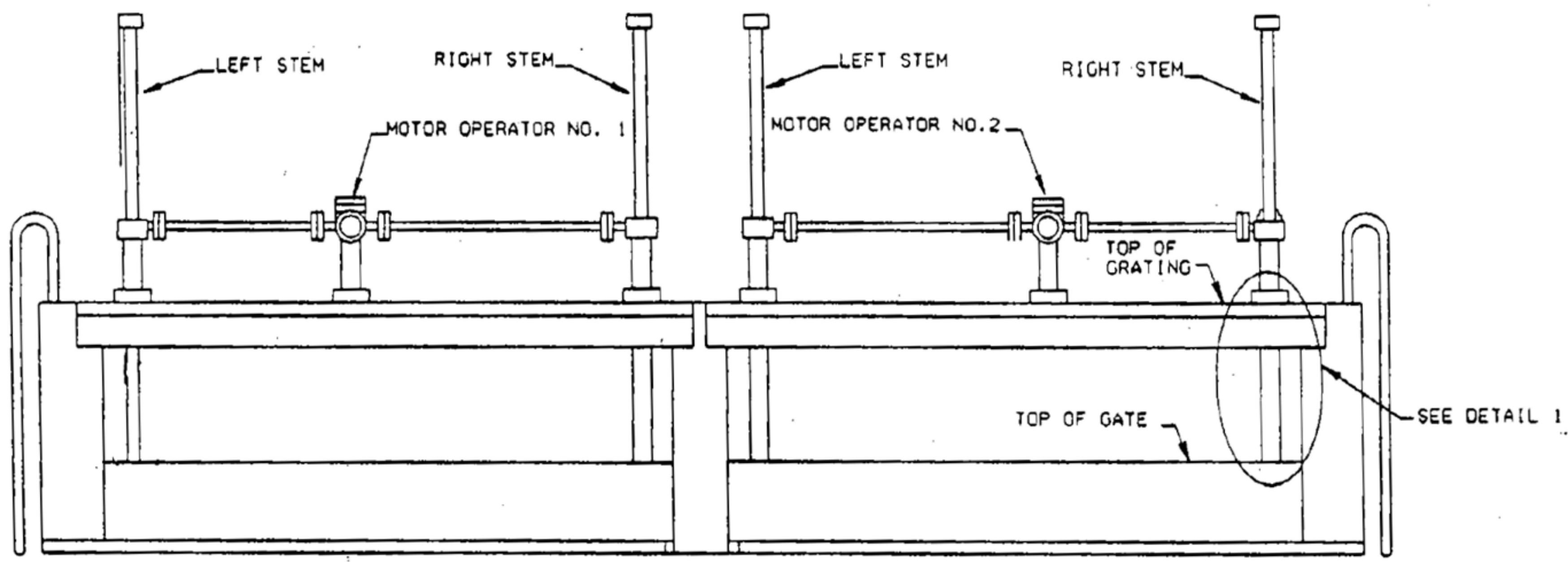
HOIST DRIVE MACHINERY AND REEVING DIAGRAM
NO SCALE

DIMENSIONS SHOWN ARE FOR CLARIFICATION AND MAY VARY WHERE NECESSARY

GATE MACHINERY
VERTICAL-LIFT GATES
HOISTING MECHANISM
ELEVATION



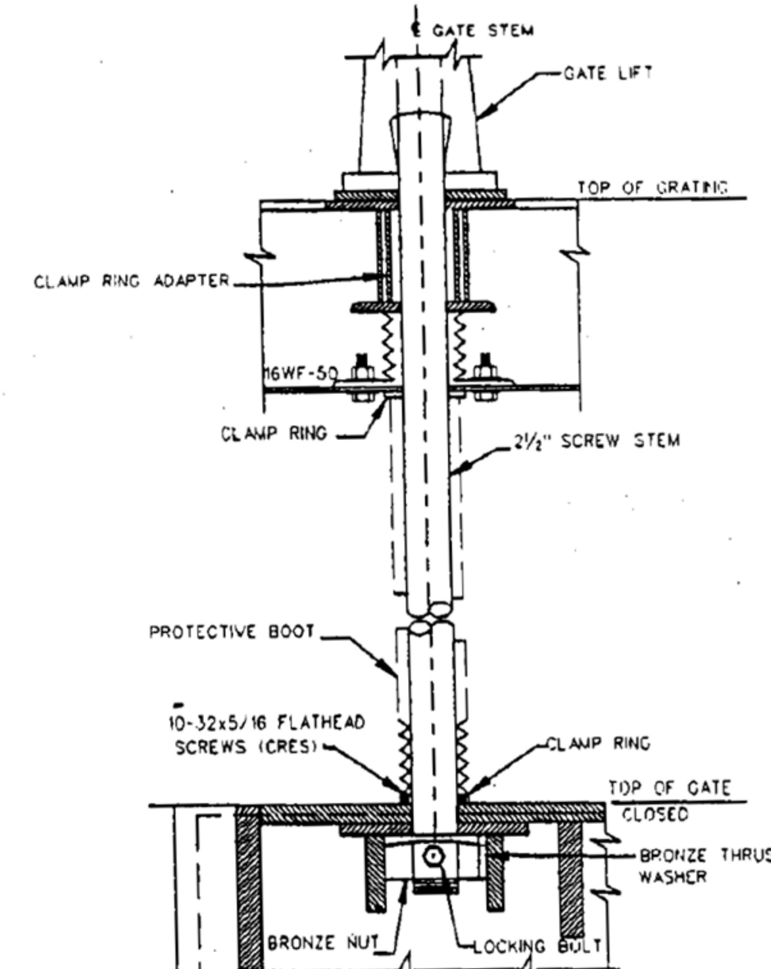
PLAN



ELEVATION

GENERAL ARRANGEMENT

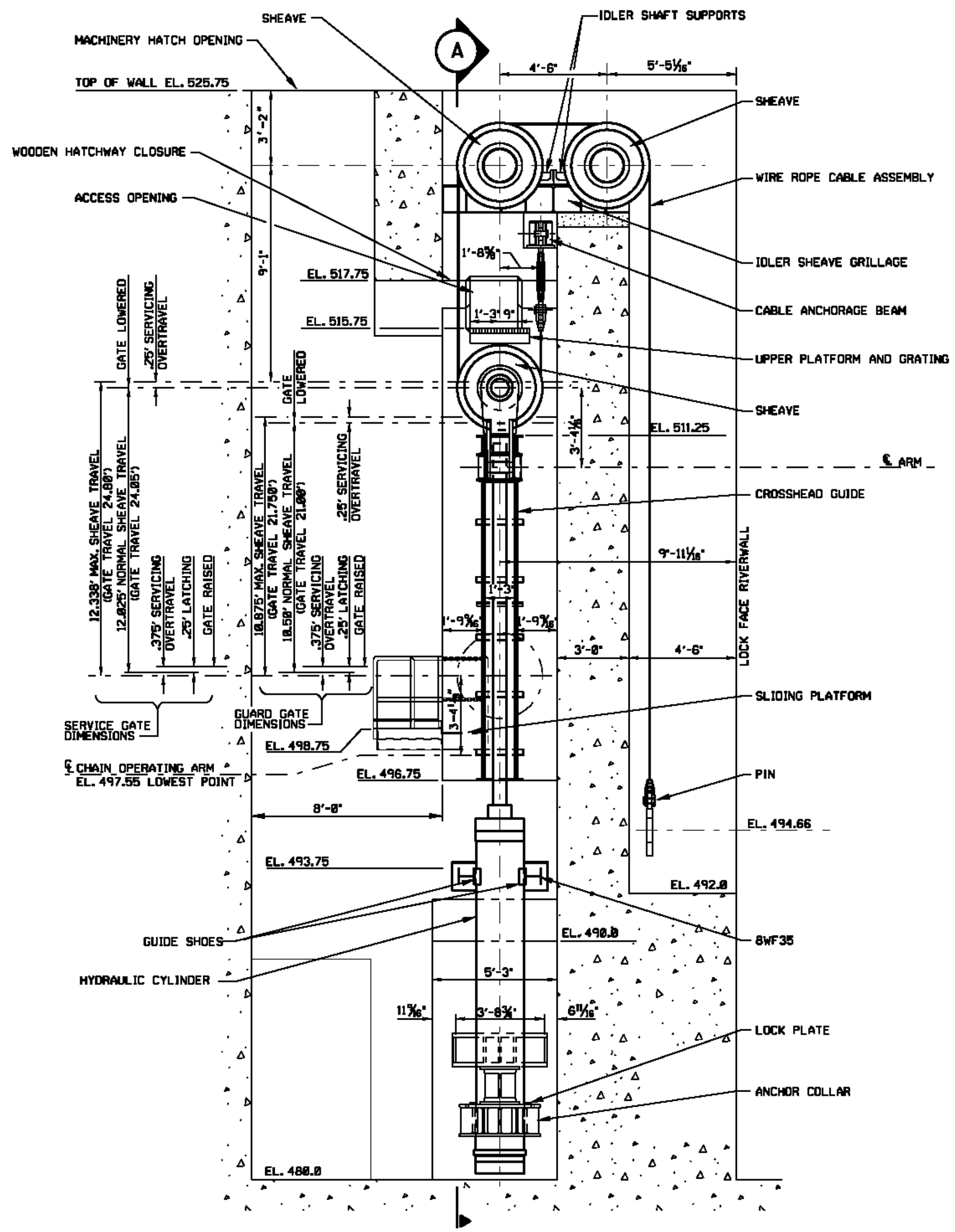
NOT TO SCALE



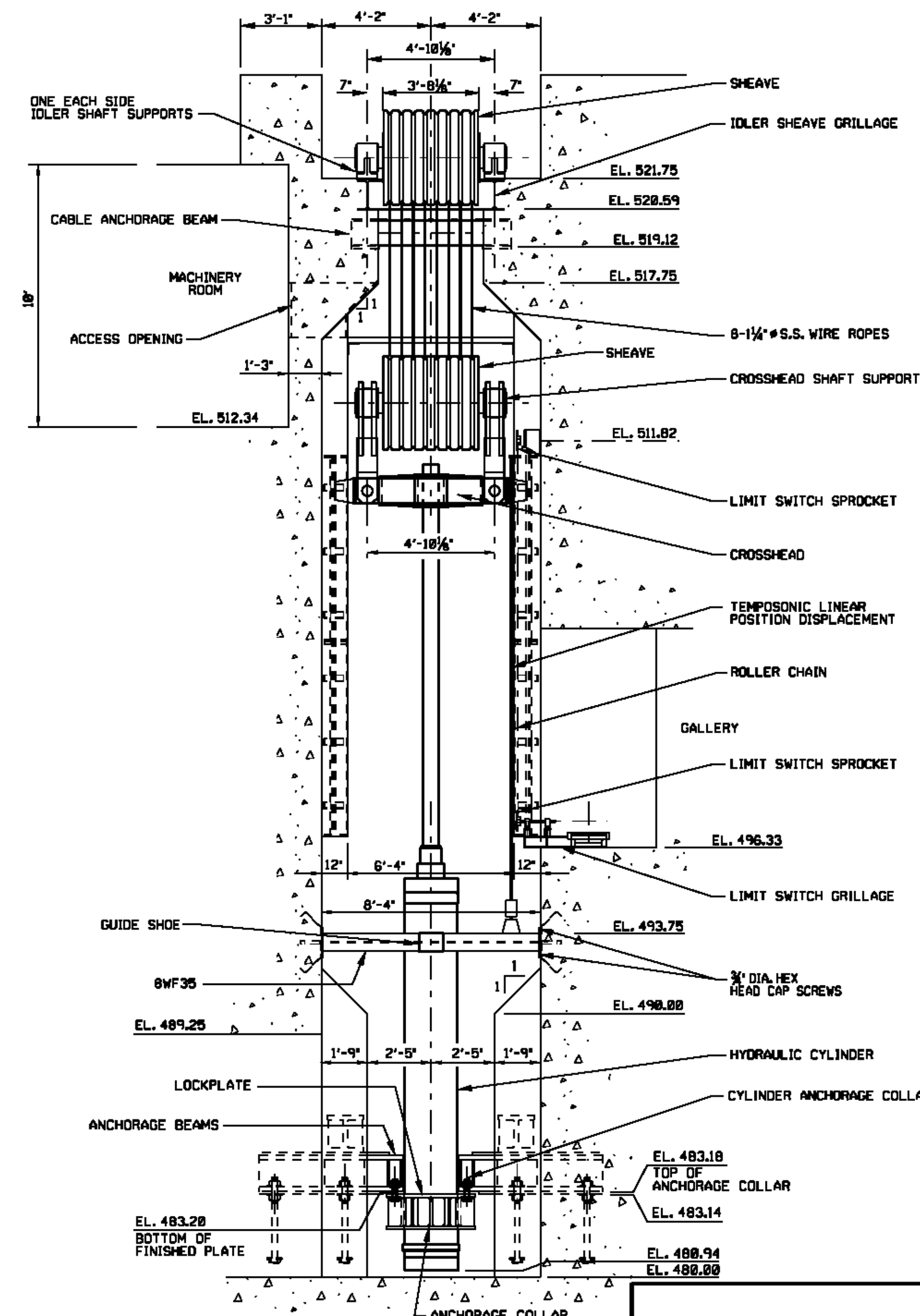
DETAIL "1"

NOT TO SCALE.

SPILLWAY - VERTICAL LIFT GATE
SCREW STEM HOIST
GENERAL ARRANGEMENT

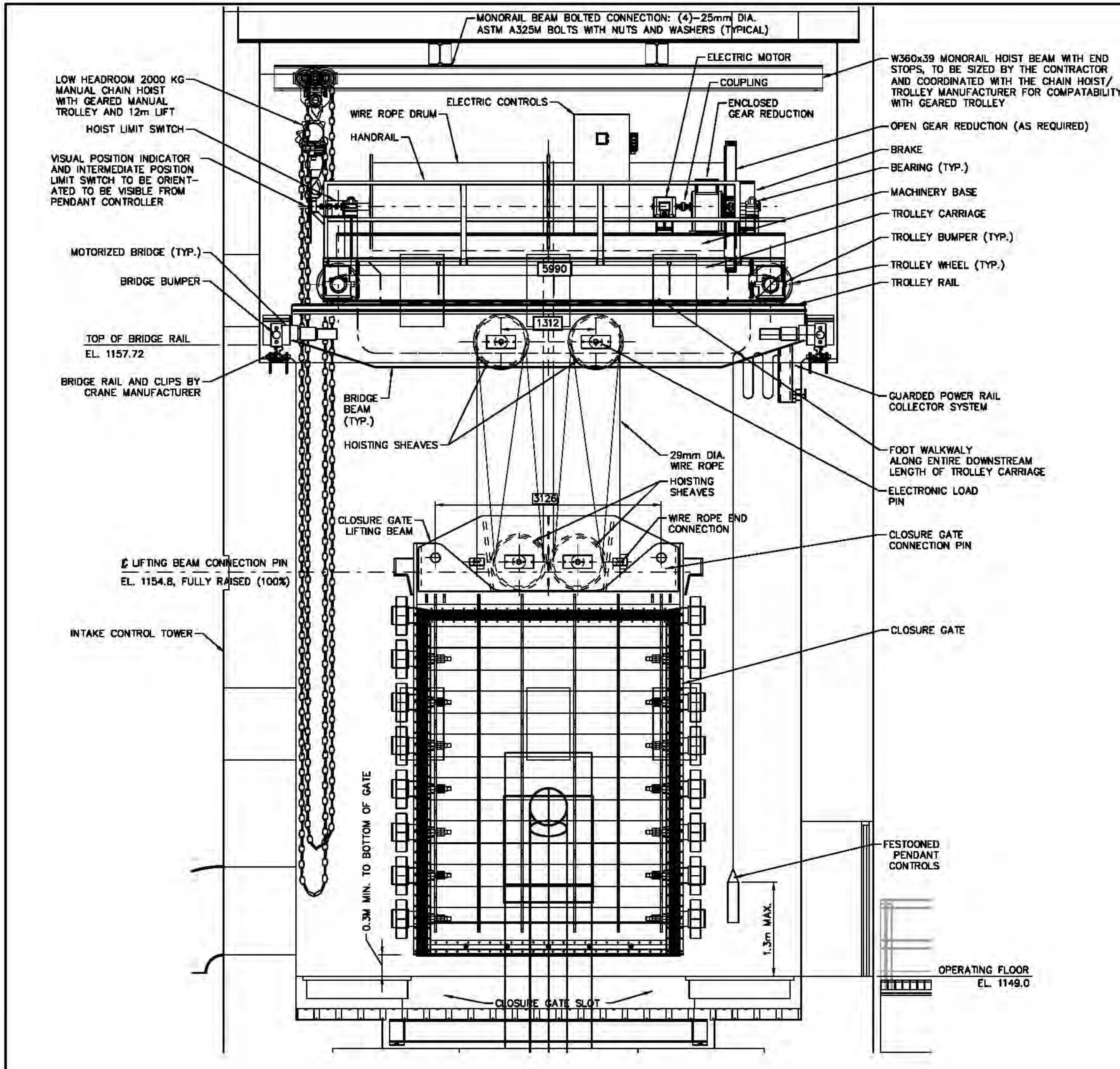


UPPER GATES - SECTION ELEVATION



SECTION A

VERTICAL GATE CYLINDER HOIST DETAIL



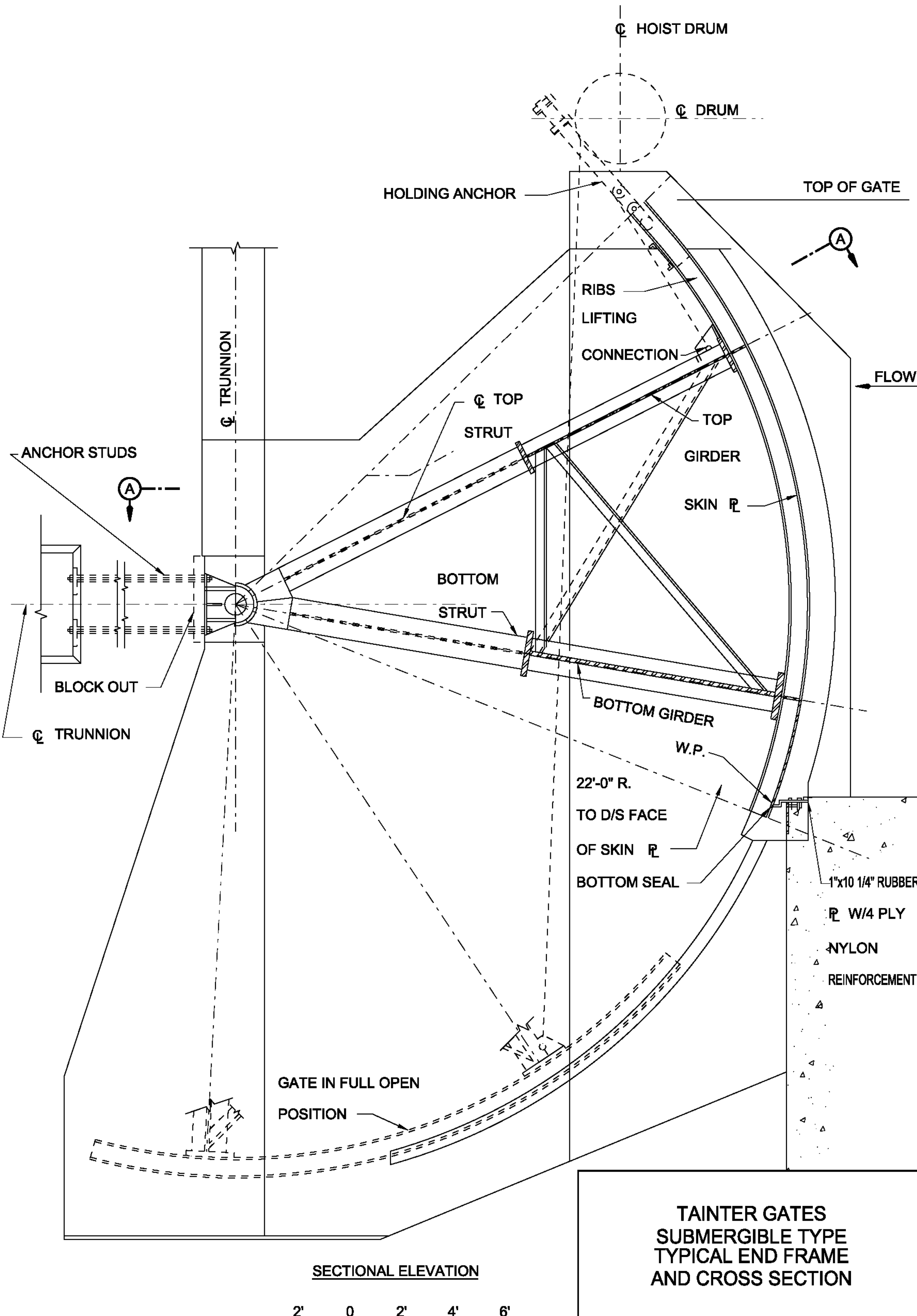
BRIDGE CRANE ELEVATION

NOTES:

1. THE DIMENSIONS SHOWN ARE APPROXIMATE. THE CONTRACTOR SHALL BE RESPONSIBLE TO COORDINATE WITH ALL EQUIPMENT MANUFACTURER'S DIMENSIONS OF FURNISHED EQUIPMENT FOR INSTALLATION AND LAYOUT.
2. ALL DIMENSIONS ENCLOSED IN RECTANGLE ARE DEPENDENT UPON THE OVERHEAD BRIDGE CRANE MANUFACTURER'S DESIGN AND SHALL BE COORDINATED AND IDENTIFIED IN THE SHOP DRAWINGS FOR APPROVAL. SHEAVES SHALL BE STANDARD MANUFACTURER'S PRODUCTS AND SUITABLE FOR THE INTENDED LOADS.
3. THE CONTRACTOR SHALL BE RESPONSIBLE TO ADJUST THE LOCATION OF THE HOISTING SHEAVES AND WIRE ROPE CONNECTIONS FOR THE GATE TO HANG VERTICALLY DURING HOISTING OPERATIONS.
4. THE CRANE MANUFACTURER SHALL DESIGN AND FURNISH A LOAD PIN IN THE CRANE REEVING SYSTEM TO CONTINUOUSLY MONITOR LOAD DURING HOISTING OPERATIONS. THE LOAD PIN SHALL BE EQUIPPED WITH ELECTRIC RELAYS AND INCORPORATED INTO THE CRANE CONTROL CIRCUIT TO TERMINATE CRANE OPERATION IN THE EVENT OF UNDERLOAD (SLACK CABLE) AND OVERLOAD.
5. THE CRANE MANUFACTURER SHALL DESIGN AND FURNISH TWO (2) CLOSURE GATE LIFTING BEAMS SIMILAR AS SHOWN HEREIN. ANY PROPOSED MODIFICATIONS TO THE CLOSURE GATE CONNECTION DIMENSIONS OR LOCATION SHALL BE COORDINATED AND SUBMITTED IN THE SHOP.
6. THE BRIDGE CRANE WIRE ROPE DRUM AND HOISTING SHEAVE SIZES AND LOCATIONS SHALL BE DEPENDENT UPON THE CRANE MANUFACTURER'S DESIGN. MINIMUM BENDING RADIUS FOR THE WIRE ROPE SHALL NOT BE EXCEEDED AND IN ACCORDANCE WITH THE SPECIFICATIONS. THE DRUM LENGTH AND DIAMETER SHALL BE DESIGNED TO REEVE THE ENTIRE LENGTH OF WIRE ROPE NECESSARY TO ACHIEVE THE REQUIRED LIFT AND MAINTAIN SPECIFIED FLEET ANGLES WITHOUT THE WIRE ROPE INTERFERING WITH THE CRANE BRIDGE, CLOSURE GATE SLOT AT THE OPERATING FLOOR, OR ANY PORTION OF THE INTAKE CONTROL TOWER STRUCTURE.
7. FABRICATE AND FURNISH TWO (2) COMPLETE AND FULLY FUNCTIONAL CLOSURE GATE LIFTING BEAM ASSEMBLIES. ONE LIFTING BEAM ASSEMBLY SHALL BE MAINTAINED AS A SPARE AND PREPARED FOR LONG-TERM STORAGE ON SITE.
8. THE CONTRACTOR SHALL FURNISH SUFFICIENT LENGTH OF SPARE WIRE ROPE ON SPOOL(S) TO COMPLETELY REPLACE THE WIRE ROPE FURNISHED WITH THE BRIDGE CRANE. WIRE ROPE SHALL BE PREPARED FOR LONG-TERM STORAGE ON SITE.
9. THE CHAIN HOIST TROLLEY MANUFACTURER SHALL FURNISH A MONORAIL HOIST BEAM SUITABLE FOR THE TROLLEY OPERATION AND SIZED TO PROVIDE A MAXIMUM BEAM DEFLECTION OF 5mm WITH THE DESIGN LOAD LOCATED AT THE END OF MONORAIL BEAM TRAVEL. THE CONTRACTOR SHALL FURNISH AND INSTALL THE MONORAIL HOIST BEAM WITH AN ASTM A325M BOLTED CONNECTION TO THE GATE TOWER ROOF BEAMS. THE CONTRACTOR SHALL DRILL AND INSTALL (8)-25mm DIA. BOLTS, NUTS AND WASHERS THROUGH THE FLANGE INTERFACE (EACH SIDE) OF EACH ROOF BEAM TO ESTABLISH THE CONNECTION. THE BEAMS SHALL BE FIELD DRILLED FOR THE INSTALLATION TO LOCATE THE MONORAIL IN THE APPROXIMATE POSITION AND TO MAXIMIZE ACCESSIBILITY FOR OVERHEAD BRIDGE CRANE MAINTENANCE.

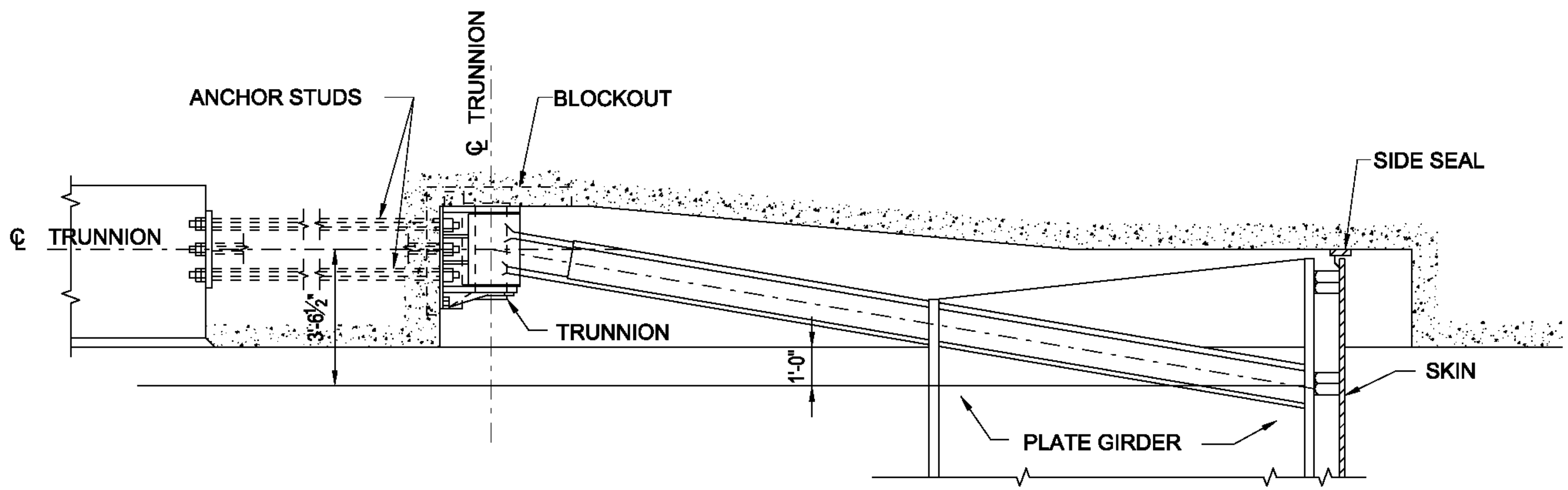
CRANE DESIGN CRITERIA		
RATED CAPACITY	1112 KN	250,000 lbs
HOISTING SPEED	5.0-19.1 mm/s	0.98 – 3.75 ft/min
LIFTING BEAM TRAVEL	61 m	200 ft
LIFTING BEAM UPPER LIMIT – CL PIN	1154.6 m	3784.4 ft
LIFTING BEAM LOWER LIMIT – CL PIN	1095.8 m	3591.5 ft
MAX. HOIST MOTOR POWER	7.4 kw	10 HP
MAX. BRIDGE ENDTRUCK MOTOR POWER	3.0 kw	4 HP
MAX. TROLLEY MOTOR POWER	3.0 kw	4 HP
TROLLEY MAX. TRAVEL	1.43 m	4.69 ft
TROLLEY MAX. SPEED	5.0 -10.0 mm/s	0.98 – 1.97 ft/min
TROLLEY MAX. SPAN	2.07 m	6.8 ft
BRIDGE MAX. TRAVEL	2.49 m	8.18 ft
BRIDGE MAX. SPEED	5.0 – 13.0 mm/s	0.98 – 2.56 ft/min
BRIDGE MAX. SPAN	7.48 m	24.54 ft

TRACTOR GATE DETAIL



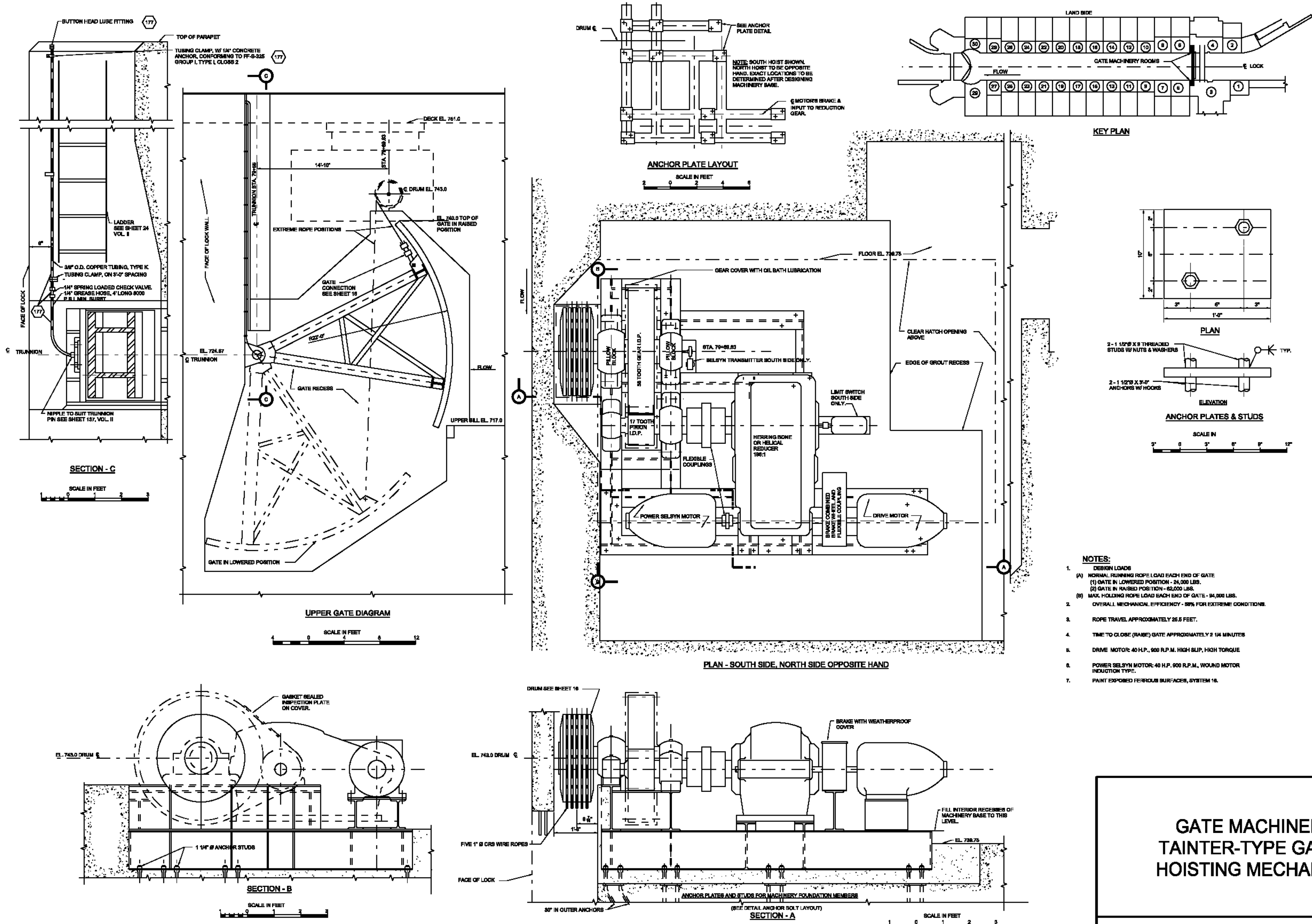
TAINTER GATES
SUBMERGIBLE TYPE
TYPICAL END FRAME
AND CROSS SECTION

PLATE B-60



SECTIONAL PLAN "A"

TAINTER GATES
SUBMERGIBLE TYPE
TYPICAL END FRAME

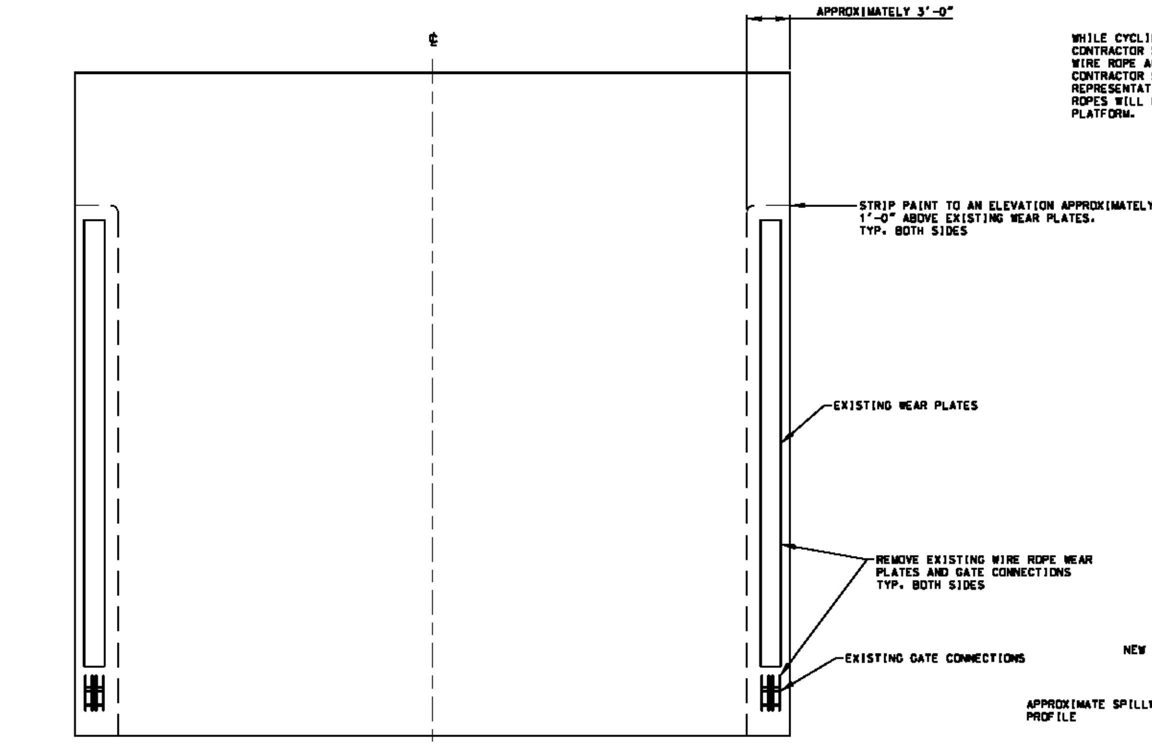


NOTES:

1. DESIGN LOADS
   (A) NORMAL RUNNING ROPE LOAD EACH END OF GATE
       (1) GATE IN LOWERED POSITION - 24,000 LBS.
       (2) GATE IN RAISED POSITION - 62,000 LBS.
   (B) MAX. HOLDING ROPE LOAD EACH END OF GATE - 94,000 LBS.
2. OVERALL MECHANICAL EFFICIENCY - 58% FOR EXTREME CONDITIONS.
3. ROPE TRAVEL APPROXIMATELY 26.5 FEET.
4. TIME TO CLOSE (RAISE) GATE APPROXIMATELY 2 1/4 MINUTES
5. DRIVE MOTOR: 40 H.P., 900 R.P.M. HIGH SLIP, HIGH TORQUE
6. POWER SELSYN MOTOR: 40 H.P. 900 R.P.M., WOUND MOTOR INDUCTION TYPE.
7. PAINT EXPOSED FERROUS SURFACES, SYSTEM 16.

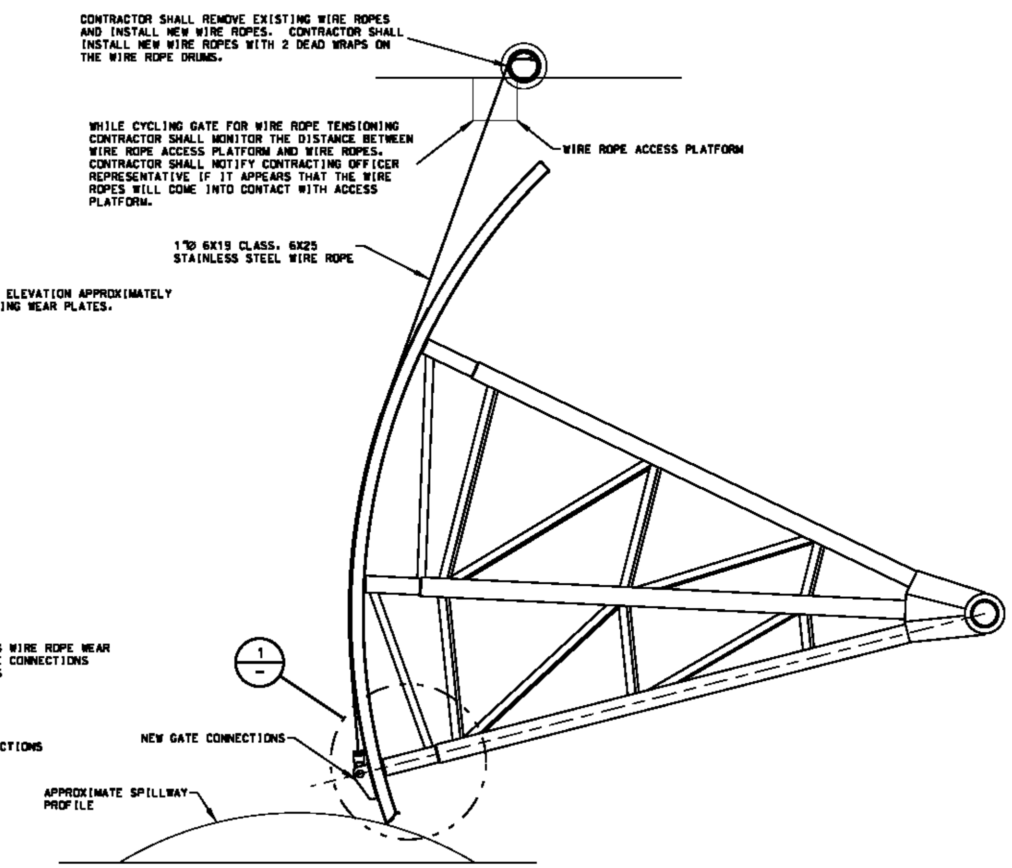
GATE MACHINERY
TAINTER-TYPE GATES
HOISTING MECHANISM

PLATE B-62



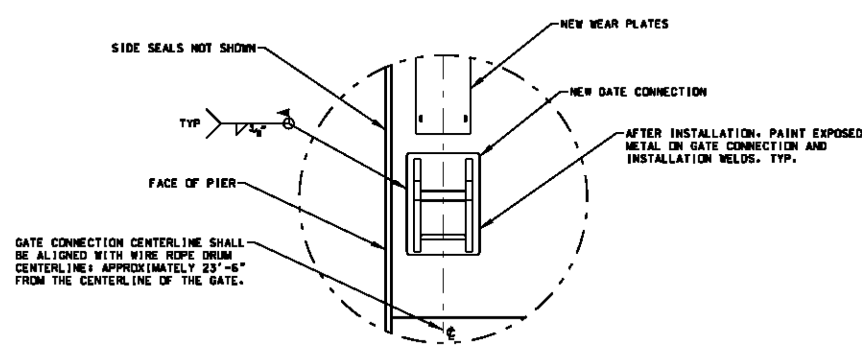
SPILLWAY TAINTER GATE FRONT VIEW

SCALE: 3/16" = 1'-0"



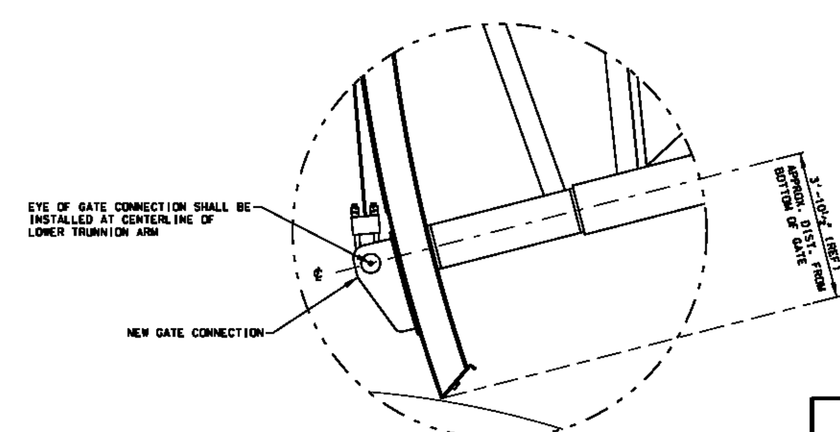
SPILLWAY TAINTER GATE SIDE VIEW

SCALE: 3/16" = 1'-0"



NEW GATE CONNECTION
INSTALLATION DETAIL

SCALE: 1/2" = 1'-0"

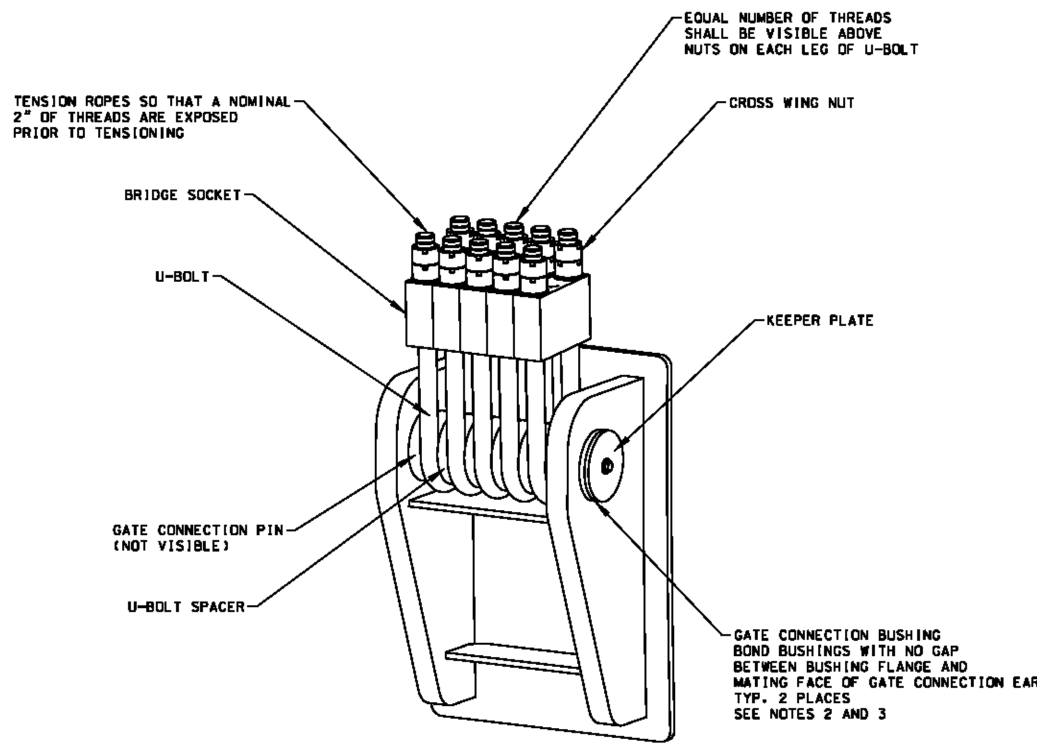


DETAIL 1

SCALE: 1/2" = 1'-0"

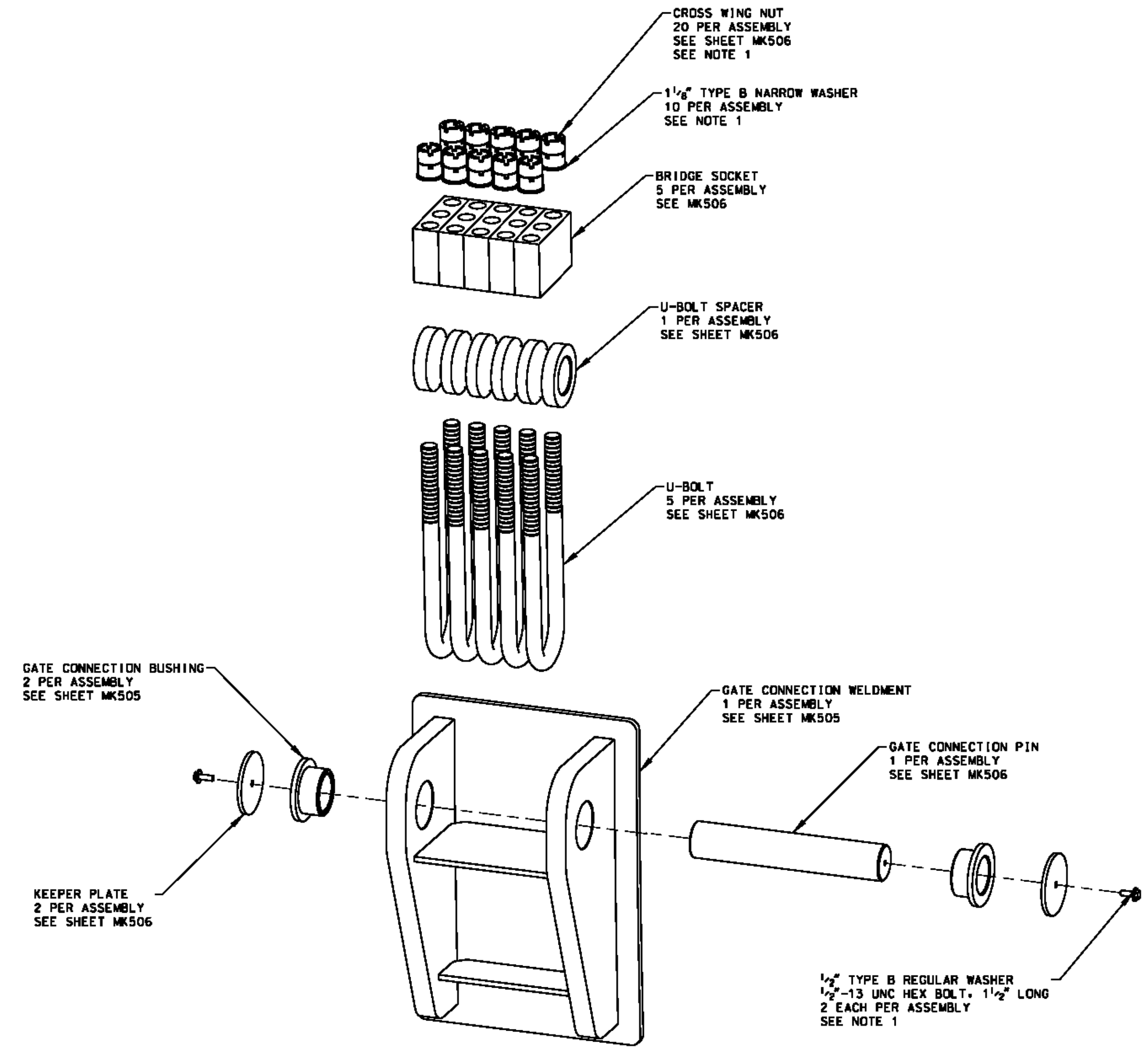
TAINTER GATE
WIRE ROPE
CONNECTION

(SHEET 1 OF 2) PLATE B-63



GATE CONNECTION ASSEMBLY
ISOMETRIC

SCALE: NONE
2 ASSEMBLIES PER GATE
WIRE ROPES AND GATE SKIN PLATE NOT SHOWN



GATE CONNECTION ASSEMBLY
ISOMETRIC-EXPLODED

SCALE: NONE
2 ASSEMBLIES PER GATE
WIRE ROPES AND GATE SKIN PLATE NOT SHOWN

NOTES:

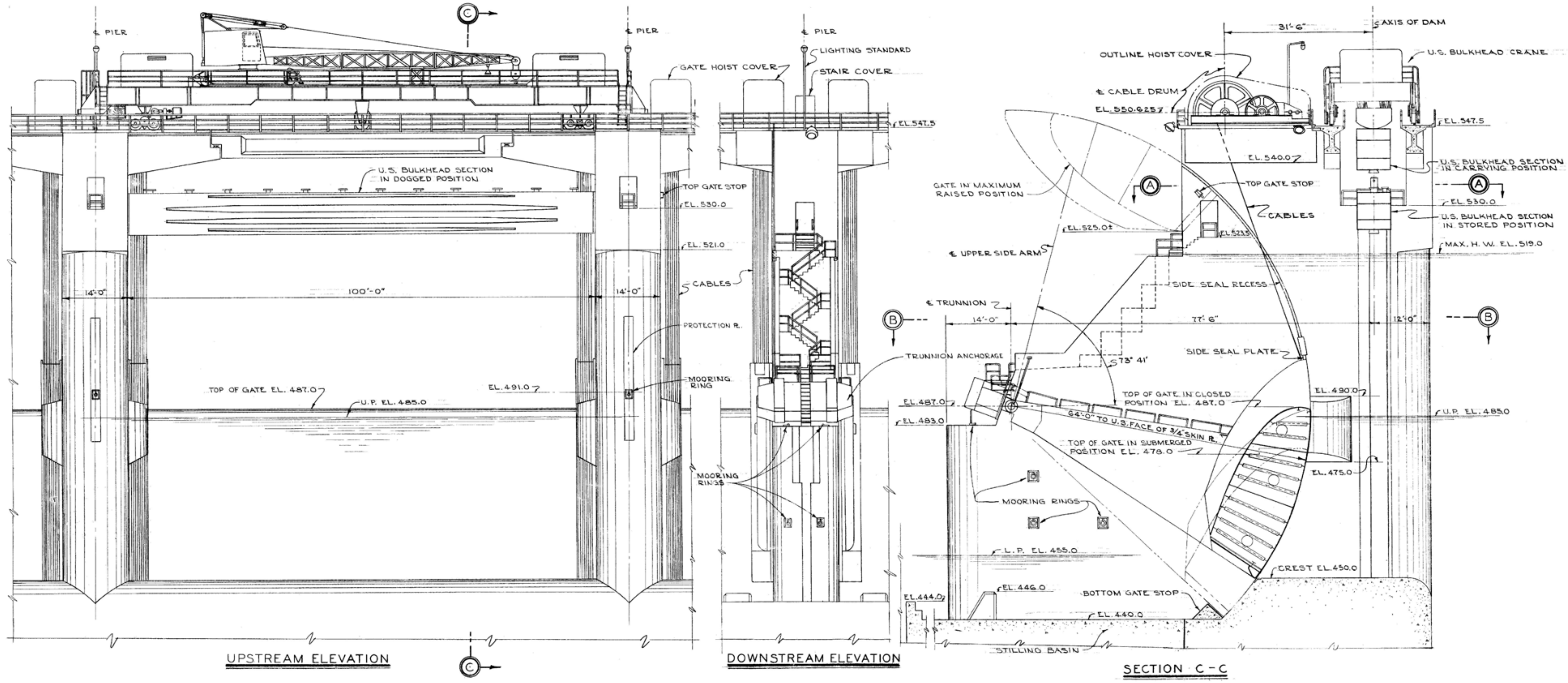
1. MECHANICAL FASTENERS SHALL BE 18-8 OF 300 SERIES STAINLESS STEEL AND INSTALLED USING AN APPROVED ANTI-GALLING COMPOUND.

2. BOND SELF-LUBRICATED BUSHINGS IN PLACE USING HYSOL EA 9309.3NA EPOXY PASTE, OR APPROVED EQUAL, IN ACCORDANCE WITH THE MANUFACTURER'S RECOMMENDATIONS.

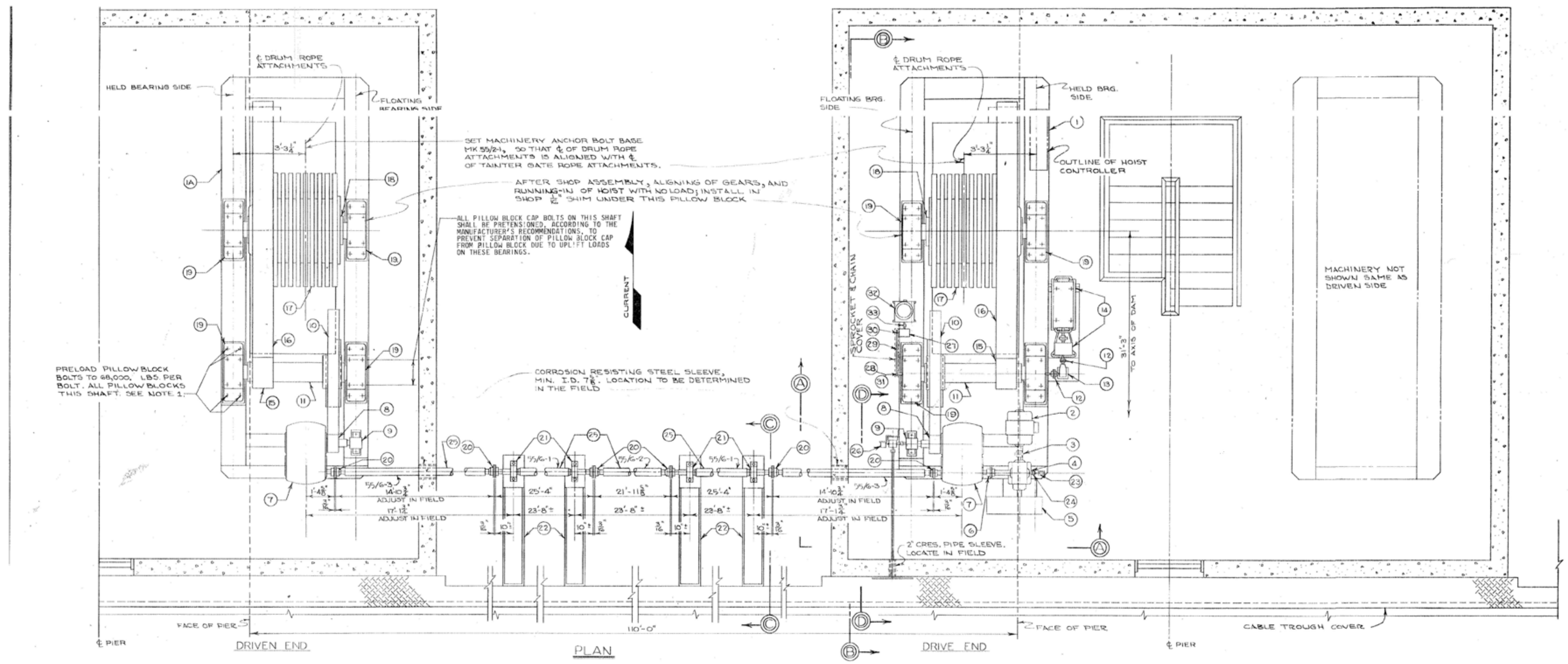
3. USE GATE CONNECTION PIN TO ALIGN GATE CONNECTION BUSHINGS DURING BONDING.

TAINTER GATE
WIRE ROPE
CONNECTION

(SHEET 2 OF 2)



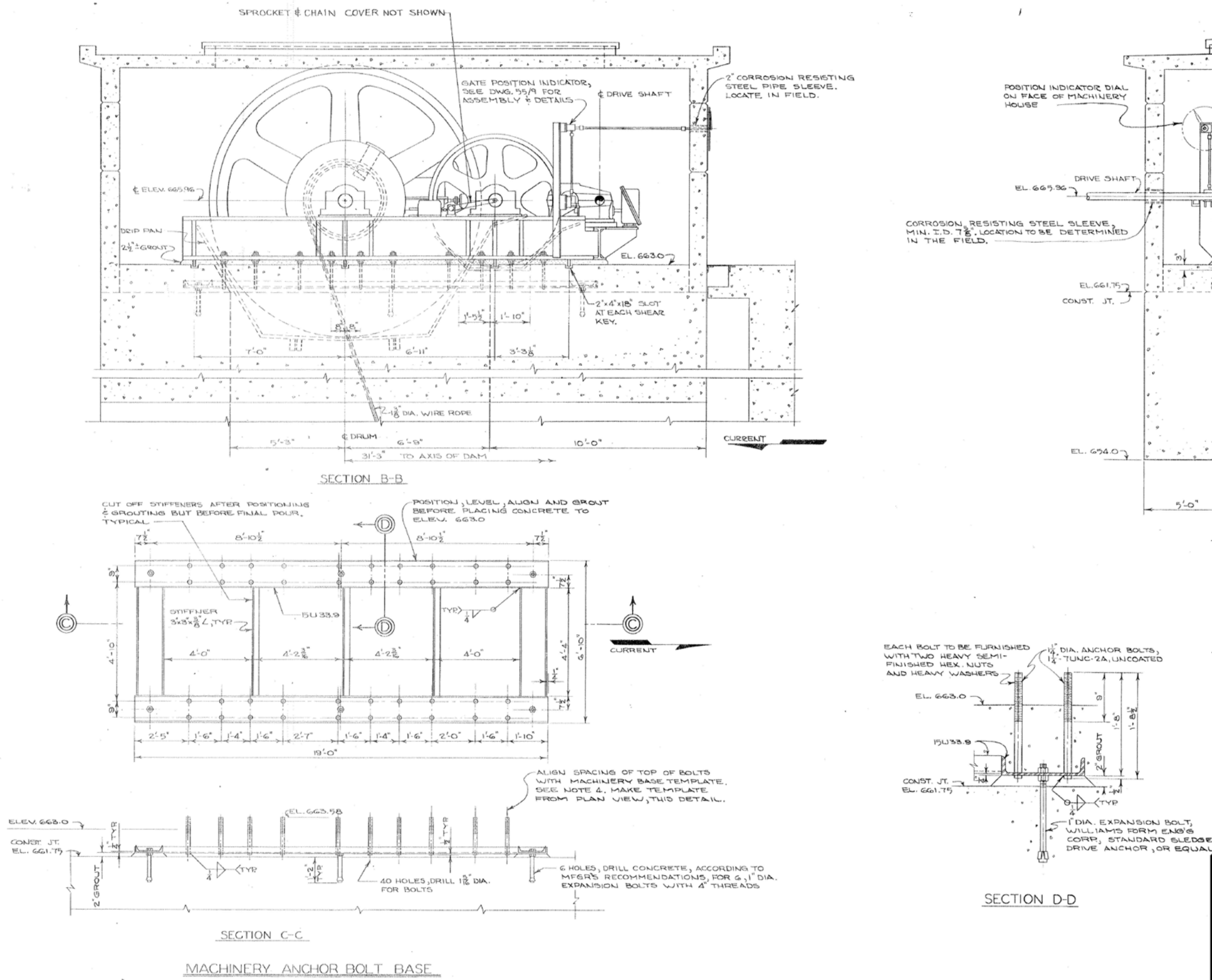
TAINTER GATE MACHINERY
WIRE ROPE - ELECTRIC MOTOR DRIVE
GENERAL ARRANGEMENT



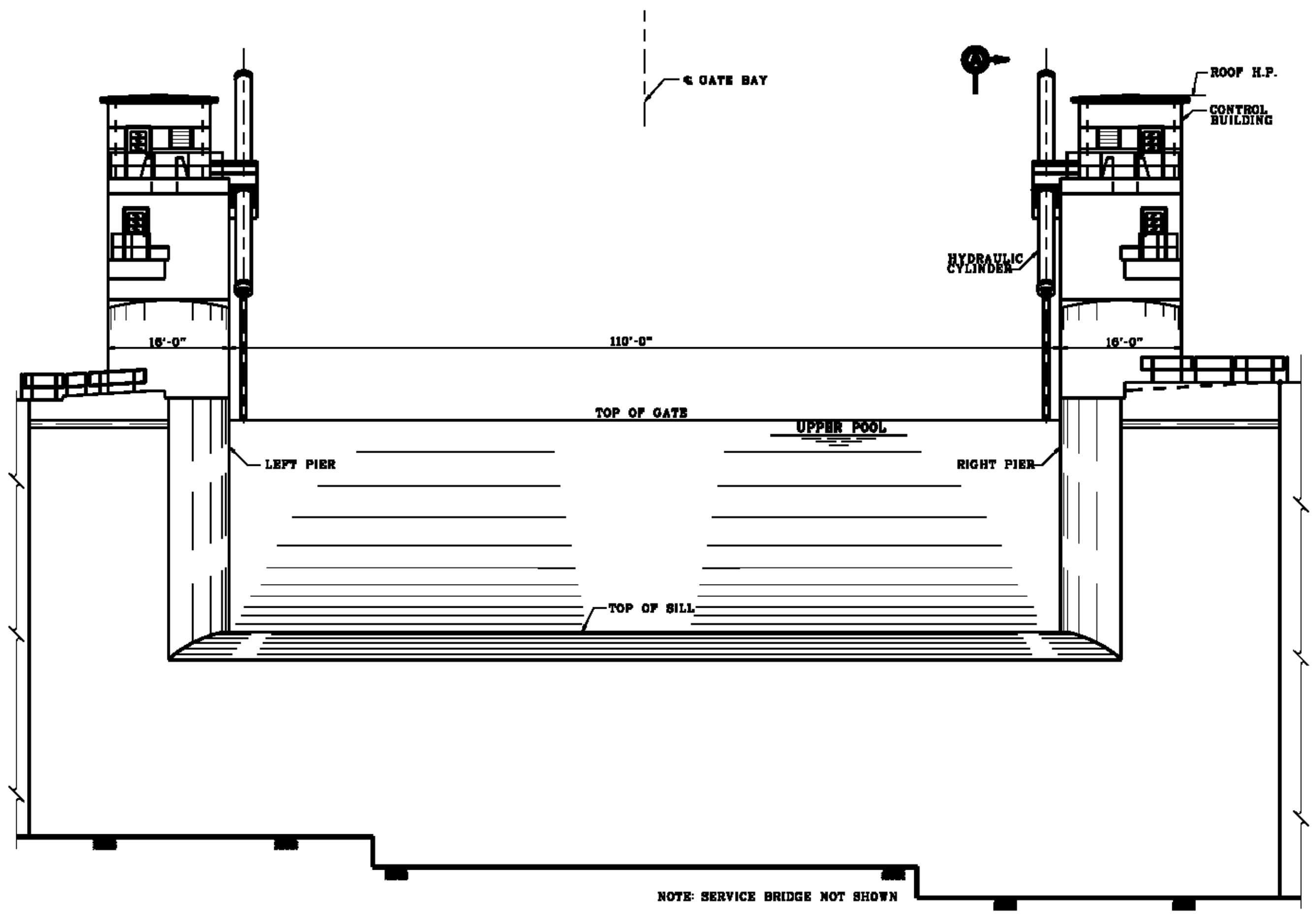
MARK	ITEM	DESCRIPTION
1	MACHINERY BASE - DRIVE END	SEE DWG. NO. 55/7
1A	MACHINERY BASE - DRIVEN END	SEE DWG. NO. 55/8 .
2	HOIST MOTOR	25 HP @ 1800 R.P.M. 3 PHASE, 60 CYCLE, 460 V., 284 TS FRAME , DRIPPROOF, 8 REQ'D.
3	FLEXIBLE COUPLING	PHILADELPHIA GEAR, TYPE K, SIZE 1, OR EQUAL, 8 REQ'D.
4	WORM GEAR REDUCER	10.3 : 1 RATIO - PHILADELPHIA GEAR WORKS' TYPE A, SIZE 675, OR EQUAL, 8 REQ'D. WITH EXTENDED SHAFTS FOR CONNECTING TO 5, 7, 24
5	BRAKE	10" WHEEL 150 LB. FT. TORQUE, CONTINUOUS DUTY WITH WATER-TIGHT ENCLOSURE - WESTINGHOUSE TYPE TMR1035 OR EQUAL, 8 REQ'D.
6	FLEXIBLE COUPLING	PHILADELPHIA GEAR, TYPE K, SIZE 2 1/2, OR EQUAL. 8 REQ'D.
7	DOUBLE REDUCER	PARALLEL SHAFT REDUCER, 31.2:1 RATIO. PHILADELPHIA GEAR WORKS HELICAL REDUCER UNIT 210, OR EQUAL 8 REQ'D. WITH DOUBLE EXTENDED INPUT SHAFT AS PER SPECS, 8 REQD WITH SINGLE STD. INPUT SHAFT, SEE 55/5 FOR OUTPUT SHAFT.
8	REDUCER PINION	SEE DWG. NO. 55/5.
9	PILLOW BLOCK	CAST STEEL WITH SPHERICAL ROLLER BEARINGS, TORRINGTON SAFS-22309, WITH SPLIT STABILIZING RING, AND BRG. NO. 455D23W33, OR EQUAL, 16 REQ'D.
10	SMALL BULL GEAR	SEE DWG. NO. 55/5.
11	LG. BULL GEAR PINION SHAFT	SEE DWG. NO. 55/5.
12	FLEXIBLE COUPLING	BOSTON GEAR FCBB25, OR EQUAL. 16 REQ'D.
13	SPIRAL GEAR SPEED REDUCER	PHILADELPHIA GEAR WORKS TYPE 3403BT, 1.84:1 RATIO, OR EQUAL. 8 REQ'D.
14	LIMIT SWITCH WITH INTERMITTENT MOTION DRIVE UNIT	CUTLER-HAMMER INC'S. BULLETIN 17020, OR EQUAL. 8 REQ'D.
15	LARGE BULL GEAR PINION	SEE DWG. NO. 55/5.

MARK	ITEM	DESCRIPTION
16	LARGE BULL GEAR	SEE DWG. NO. 55/5.
17	ROPE DRUM	SEE DWG. NO. 55/3.
18	LARGE BULL GEAR AND ROPE DRUM SHAFT	SEE DWG. NO. 55/3.
19	PILLOW BLOCK	CAST STEEL WITH SPHERICAL ROLLER BEARINGS, TORRINGTON SDAFS-22238, OR EQUAL, ONE END CLOSED, WITH SPLIT STABILIZING RING AND BRG. NO. 150SD22W33, 64 REQ'D.
20	FLEXIBLE COUPLING	PHILADELPHIA GEAR WORKS, TYPE EK, SIZE 2, OR EQUAL. 48 REQ'D.
21	PILLOW BLOCK	CAST STEEL HOUSING, SELF ALIGNING, SELF LUBRICATING, LINKBELT NO. 3247PT3, OR EQUAL, 32 REQ'D.
22	DRIVE SHAFT SUPPORT	SEE DWG. 55/6.
23	REVOLUTION COUNTER	TOP READING, DURANT MFG. CO. MODEL 5-CS-7-I-L, OR EQUAL. TO SUBTRACT WHEN REVERSED. ROTATION FOR INCREASING NUMBERS WHEN GATE IS RAISED. PROVIDE PEDESTAL ON MACHINERY BASE FOR MOUNTING COUNTER. 8 REQ'D.
24	FLEXIBLE COUPLING	BOSTON GEAR FCBB15, OR EQUAL. 8 REQ'D.
25	DRIVE SHAFT	SEE DWG. 55/6.
26	GEARED-ROTARY LIMIT SWITCH	GENERAL ELECTRIC NO. CR115-E127101, OR EQUAL. 8 REQ'D.
27	WORM GEAR REDUCER	15:1 RATIO, BOSTON GEAR U113 OR EQUAL, 8 REQD.
28	SPROCKET	SINGLE WIDTH, 16 TEETH, NO. 40, 1/2" PITCH. BOSTON GEAR HKSB16-1 OR EQUAL, 8 REQD.
29	CHAIN	SINGLE WIDTH, NO. 40, 1/2" PITCH. BOSTON GEAR CO. OR EQUAL, 8 REQD.
30	SPROCKET	SINGLE WIDTH, 18 TEETH, NO. 40, 1/2" PITCH. BOSTON GEAR HKSB18-1 OR EQUAL, 8 REQD.
31	EXTENDED SHAFT	SEE DWG. NO. 55/5.
32	SELSYN SYSTEM TRANSMITTER	SEE GATE HOIST ELECTRICAL EQUIPMENT SCHEDULE, DWG. 55/15.
33	COUPLING	MARK 55/9-13, DWG. 55/9.

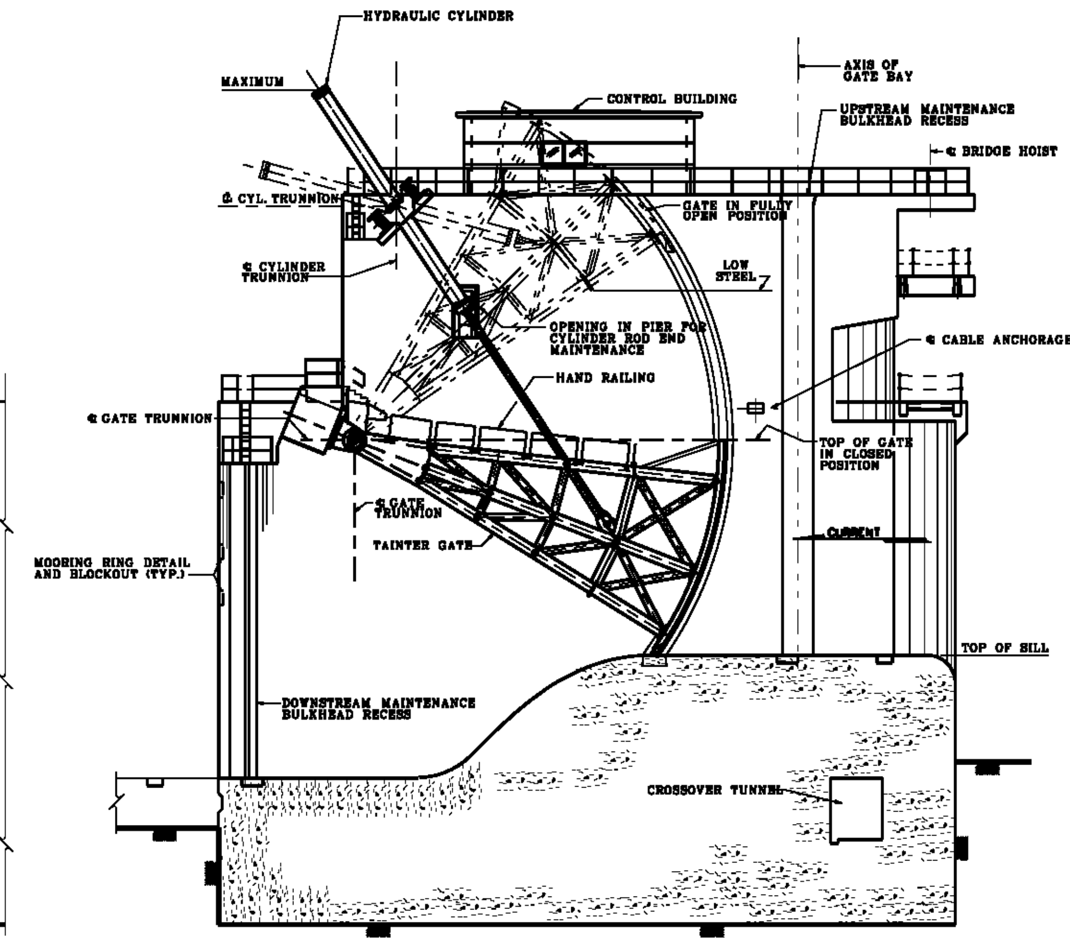
TAINTER GATE MACHINERY
WIRE ROPE - ELECTRIC MOTOR DRIVE
ASSEMBLY - PLAN



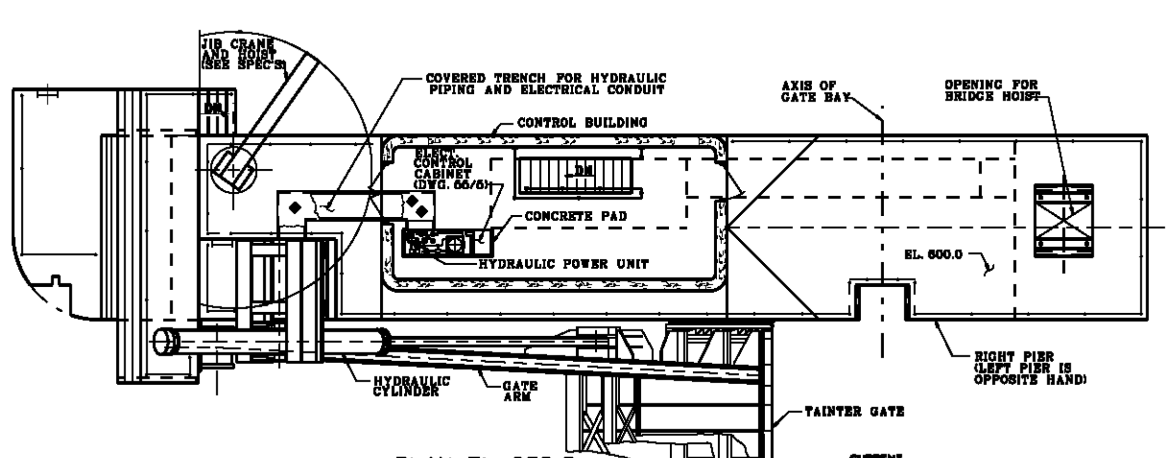
TAINTER GATE MACHINERY
WIRE ROPE - ELECTRIC MOTOR DRIVE
ASSEMBLY - ELEVATIONS



UPSTREAM ELEVATION

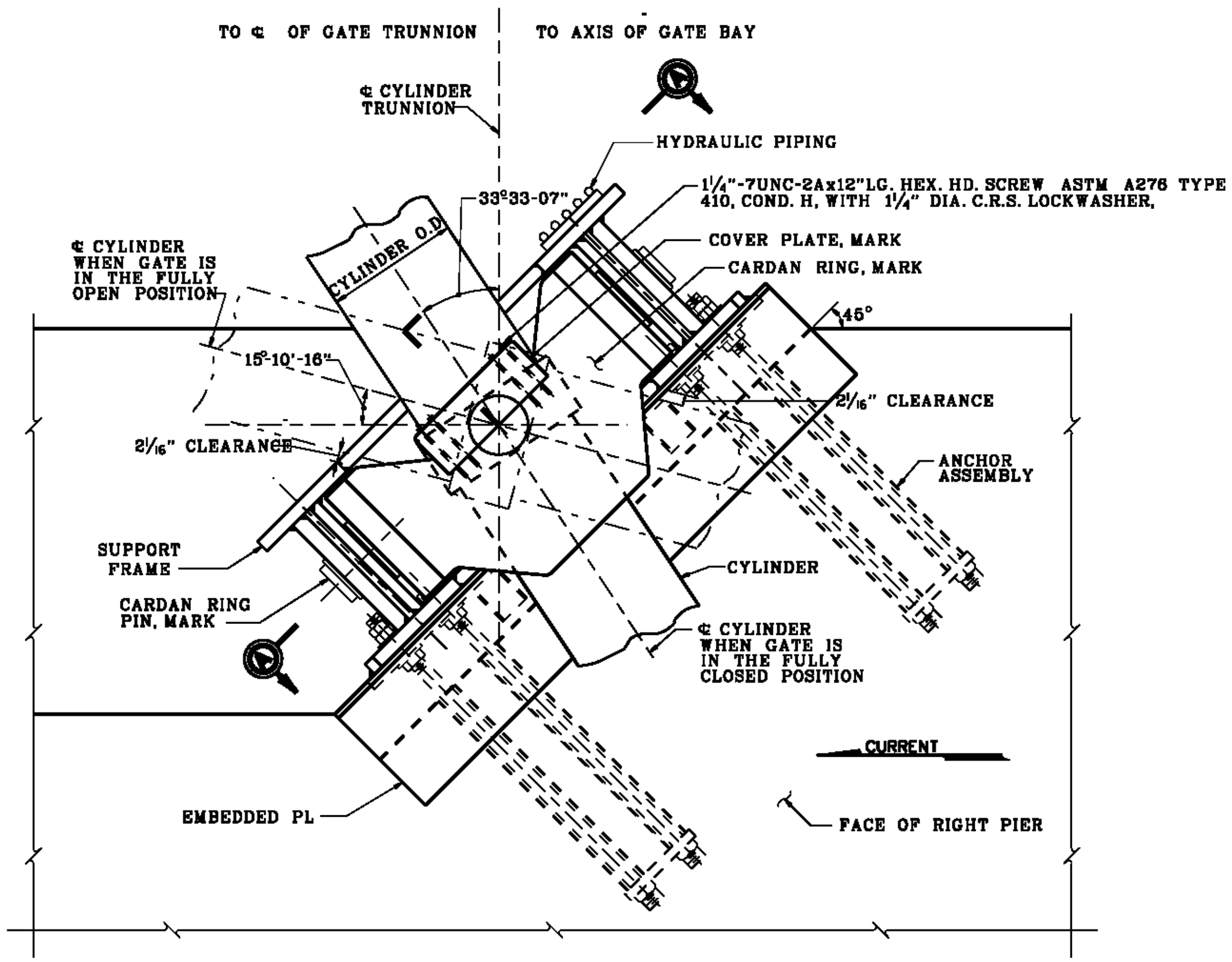


SECTION A-A

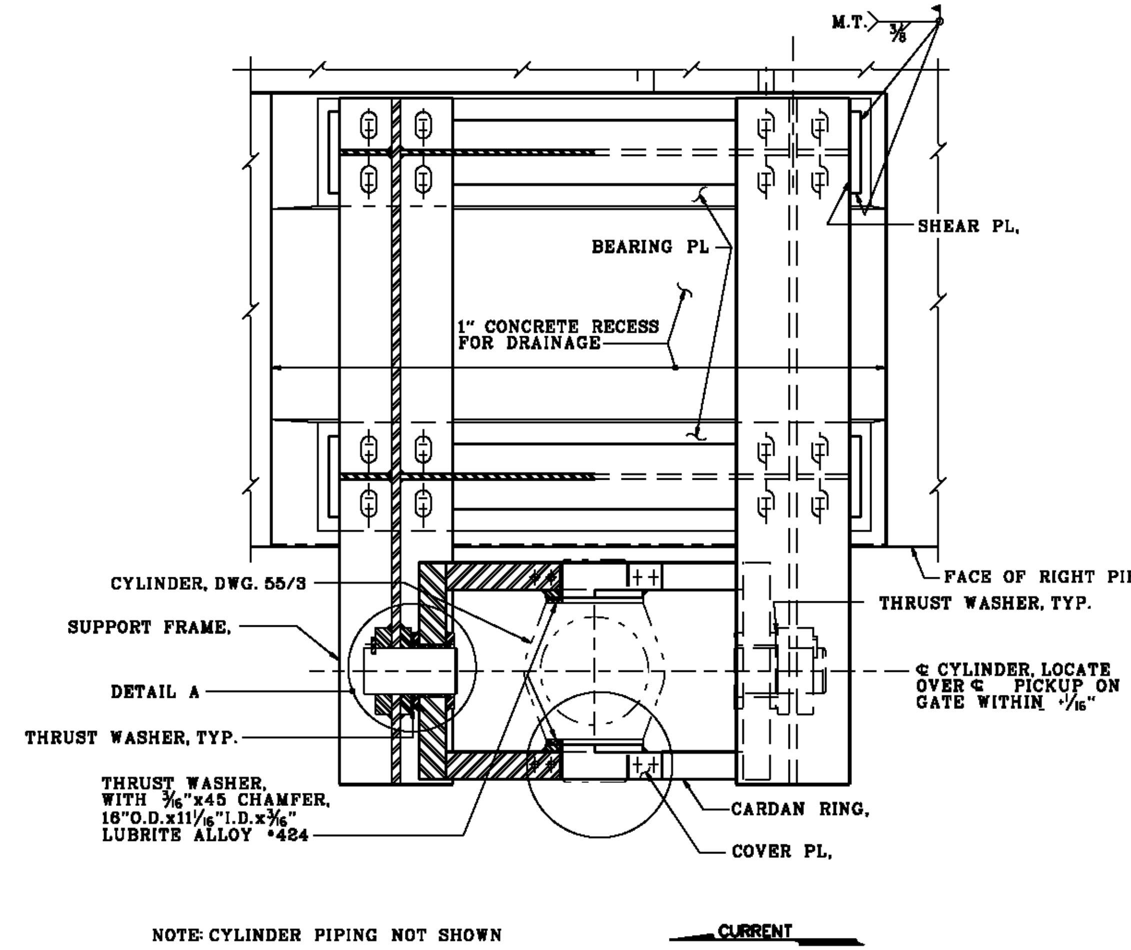


PLAN EL. 603.0

TAINTER GATE MACHINERY
HYDRAULIC CYLINDER
GENERAL ARRANGEMENT

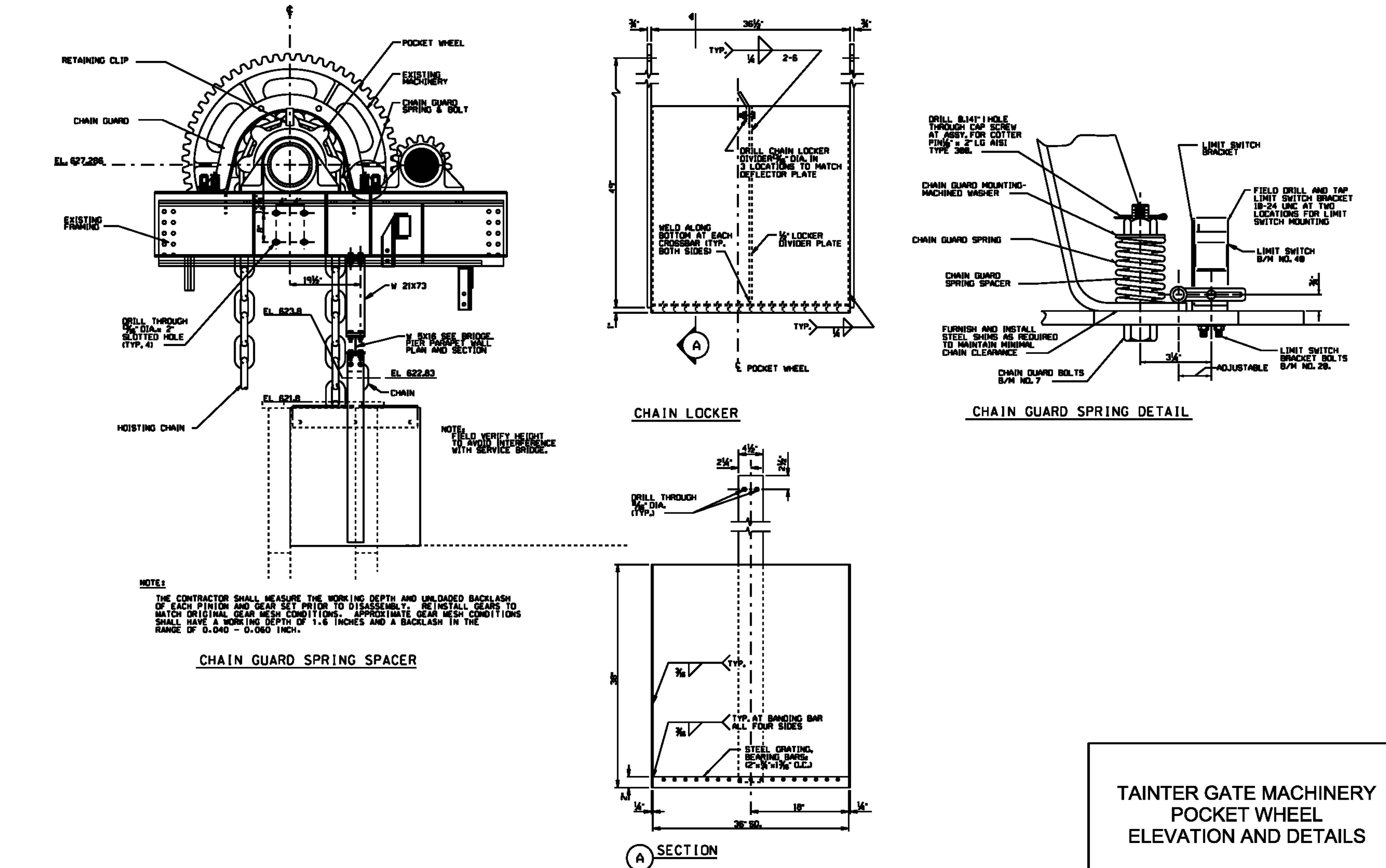


ELEVATION



SECTION A - A

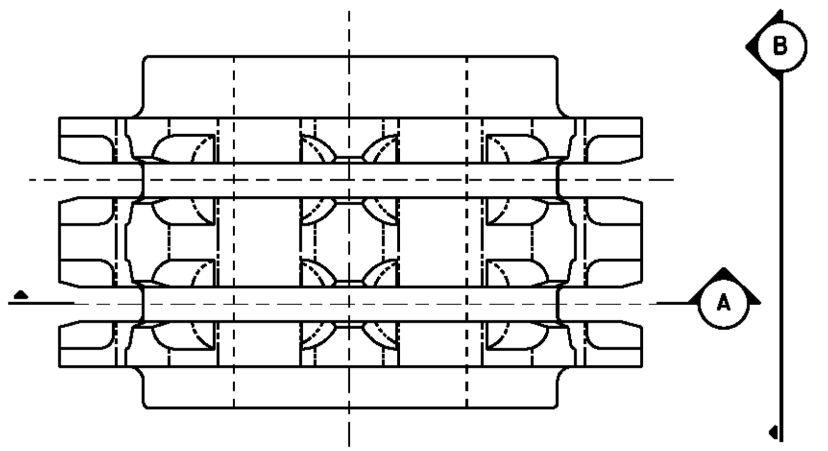
TAINTER GATE MACHINERY
HYDRAULIC CYLINDER
MOUNTING ARRANGEMENT



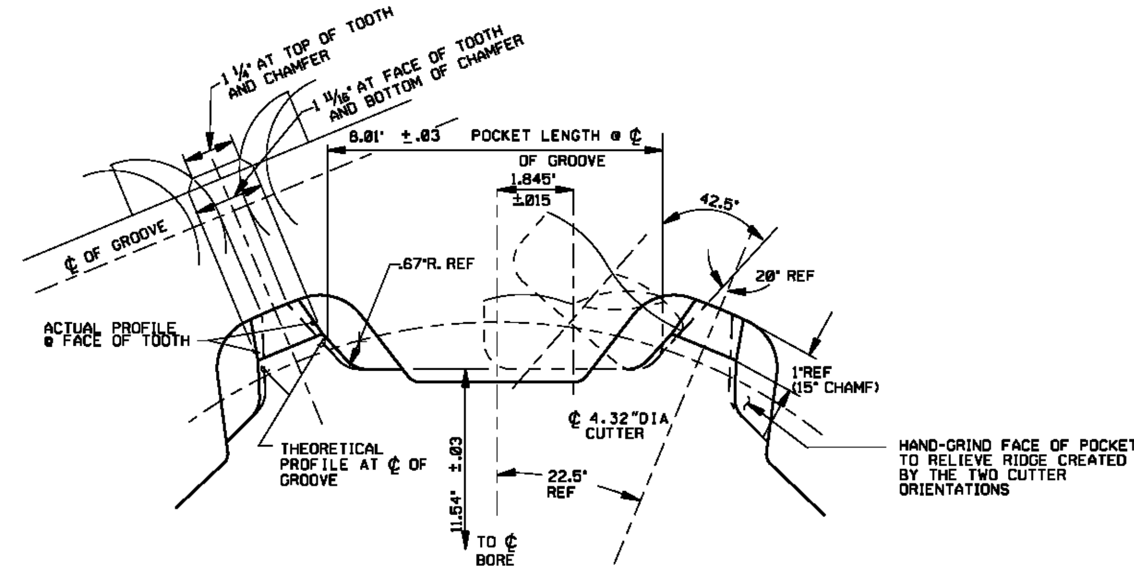
TAINTER GATE MACHINERY
POCKET WHEEL
ELEVATION AND DETAILS

SHEET 1 OF 3

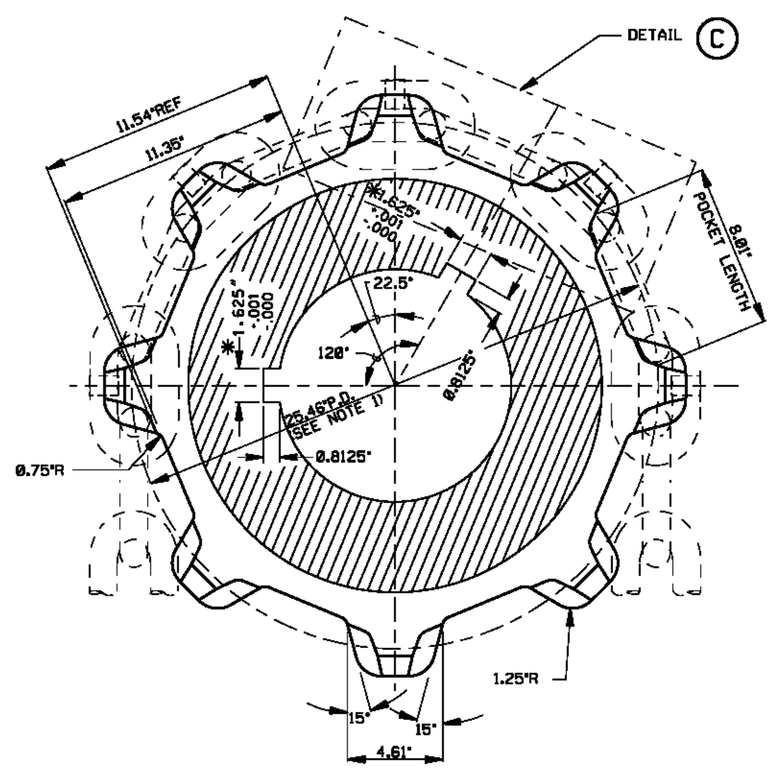
PLATE B-70



POCKET WHEEL PLAN VIEW

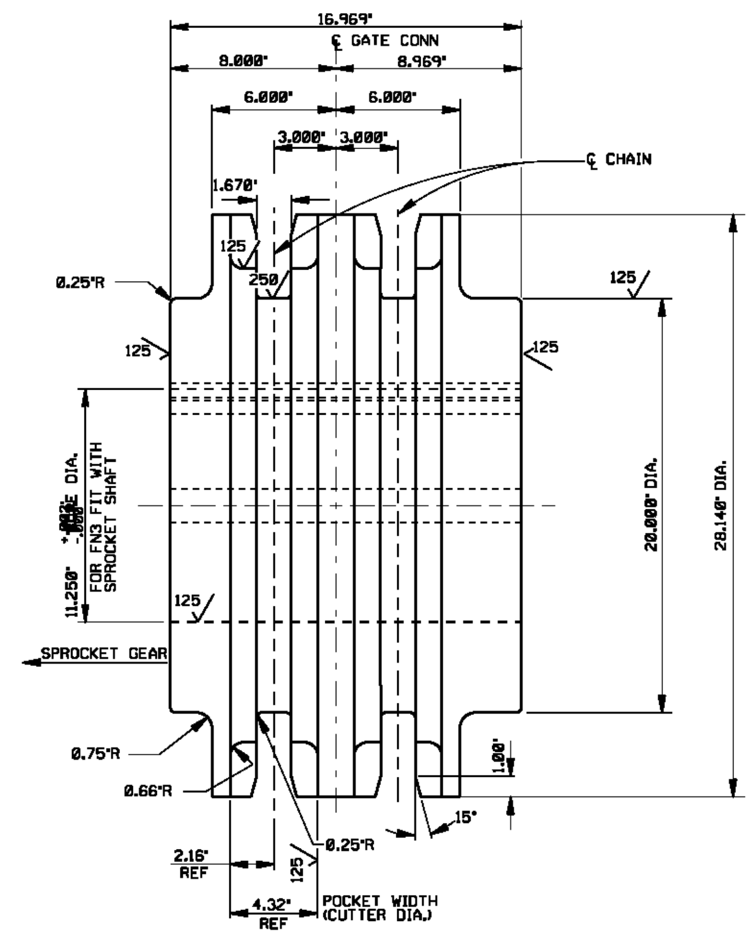


C DETAIL



✱ NOTE:
THE CONTRACTOR SHALL VERIFY THE KEYWAY DIMENSION OF THE EXISTING SPROCKET SHAFT GEARS PRIOR TO FABRICATION OF THE NEW POCKET WHEEL. NOTIFY THE C.O.R. IMMEDIATELY OF ANY DISCREPANCIES FROM THE KEYWAY DIMENSION SHOWN. THE NEW POCKET WHEEL SHALL BE FABRICATED TO HAVE ALL KEYWAYS MATCH THE EXISTING KEYWAYS OF THE SPROCKET GEAR SHAFT.

A SECTION



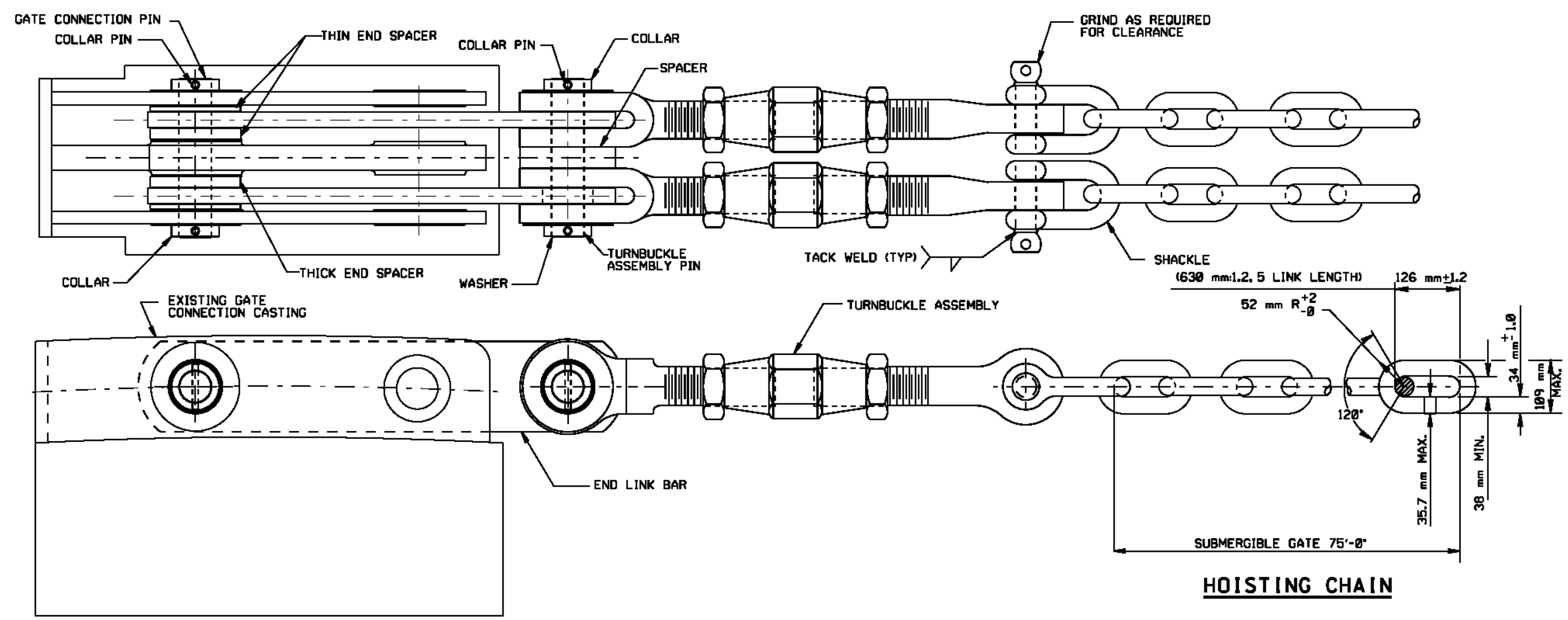
B END VIEW

NOTE

1. 25.46" IS THE THEORETICAL PITCH DIAMETER BASED ON THE CHAIN GEOMETRY. THE POCKETS ARE MACHINED TO GIVE A LOOSE FIT WHICH RESULTS IN AN ACTUAL PITCH DIAMETER OF 25.31" BASED ON THE POCKET WHEEL GEOMETRY.

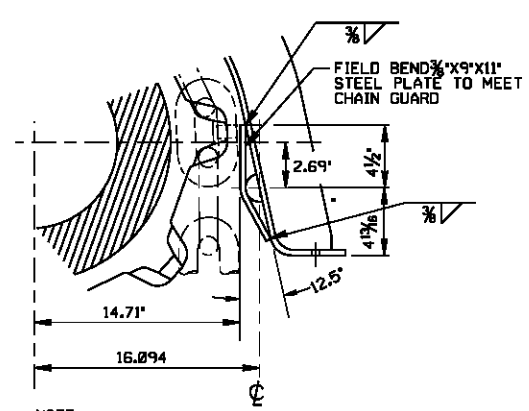
TAINTER GATE MACHINERY
POCKET WHEEL
ELEVATION AND DETAILS

SHEET 2 OF 3 PLATE B-71



GATE CONNECTION ASSEMBLY

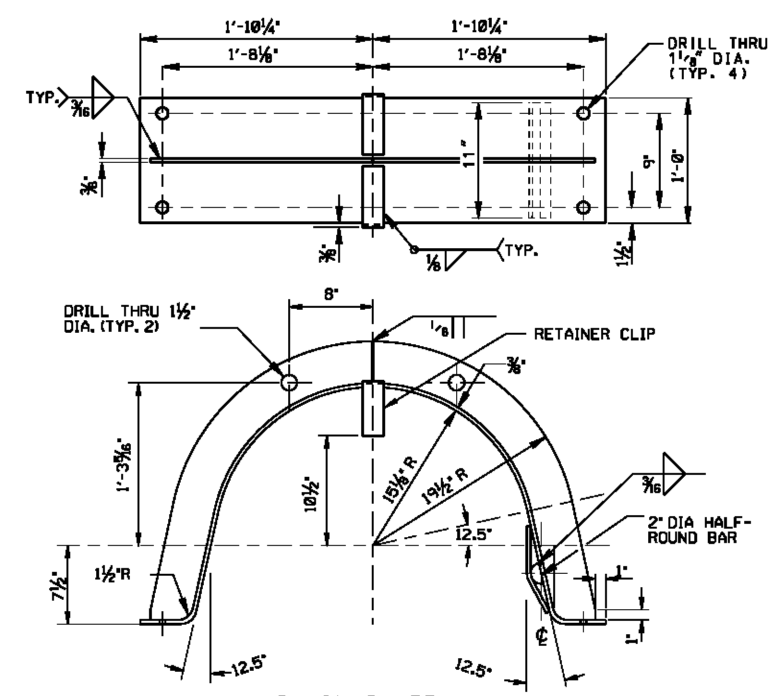
HOISTING CHAIN



NOTE:

CONTRACTOR SHALL FIELD INSTALL STEEL PLATE AND HALF ROUND TO PROVIDE MINIMAL CLEARANCE BETWEEN CHAIN AND STEEL PLATE. DIMENSIONS PROVIDED IN CHAIN GUARD DETAIL ARE BASED ON THE THEORETICAL PITCH DIAMETER OF 25.46".

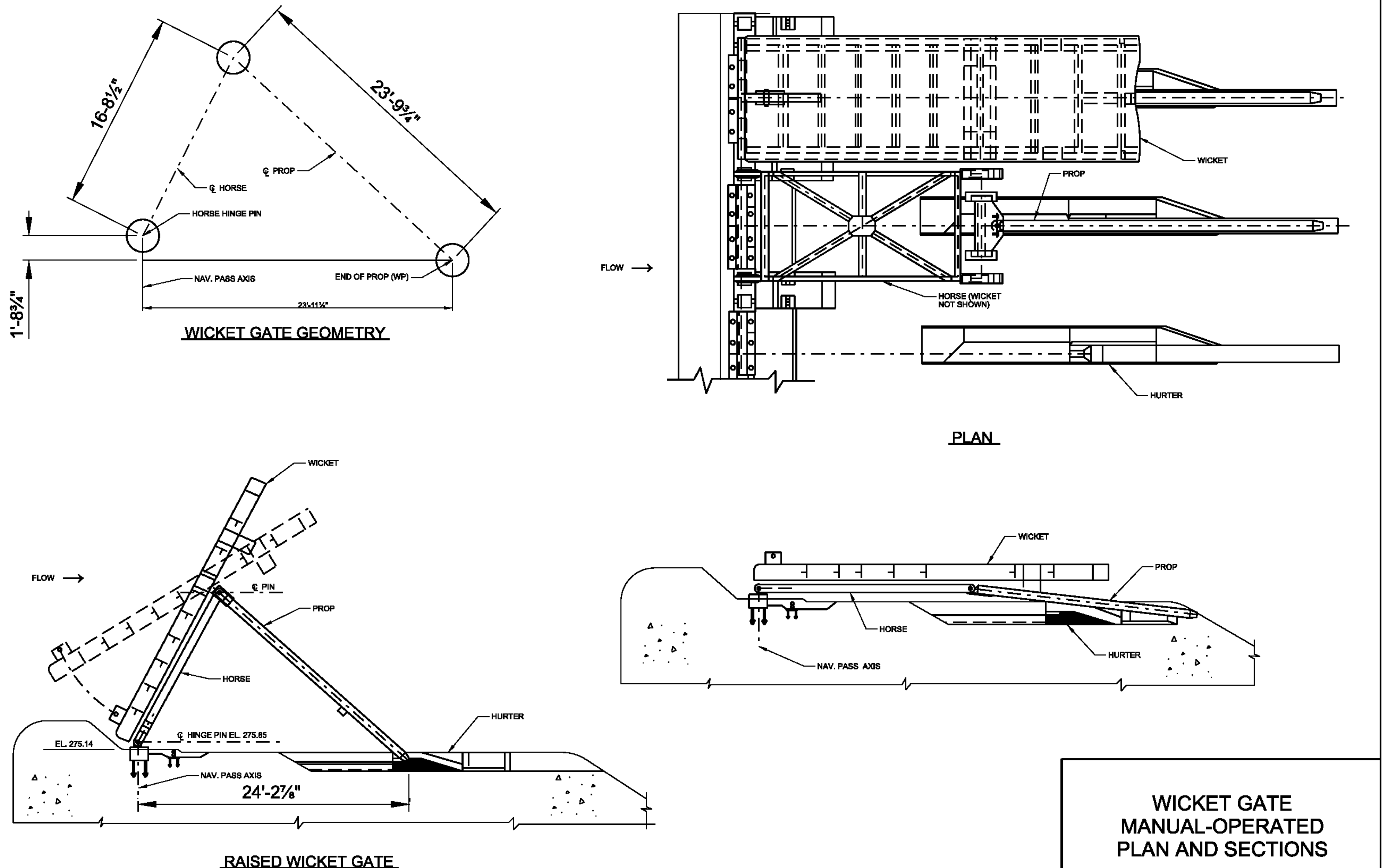
CHAIN GUIDE PLATE



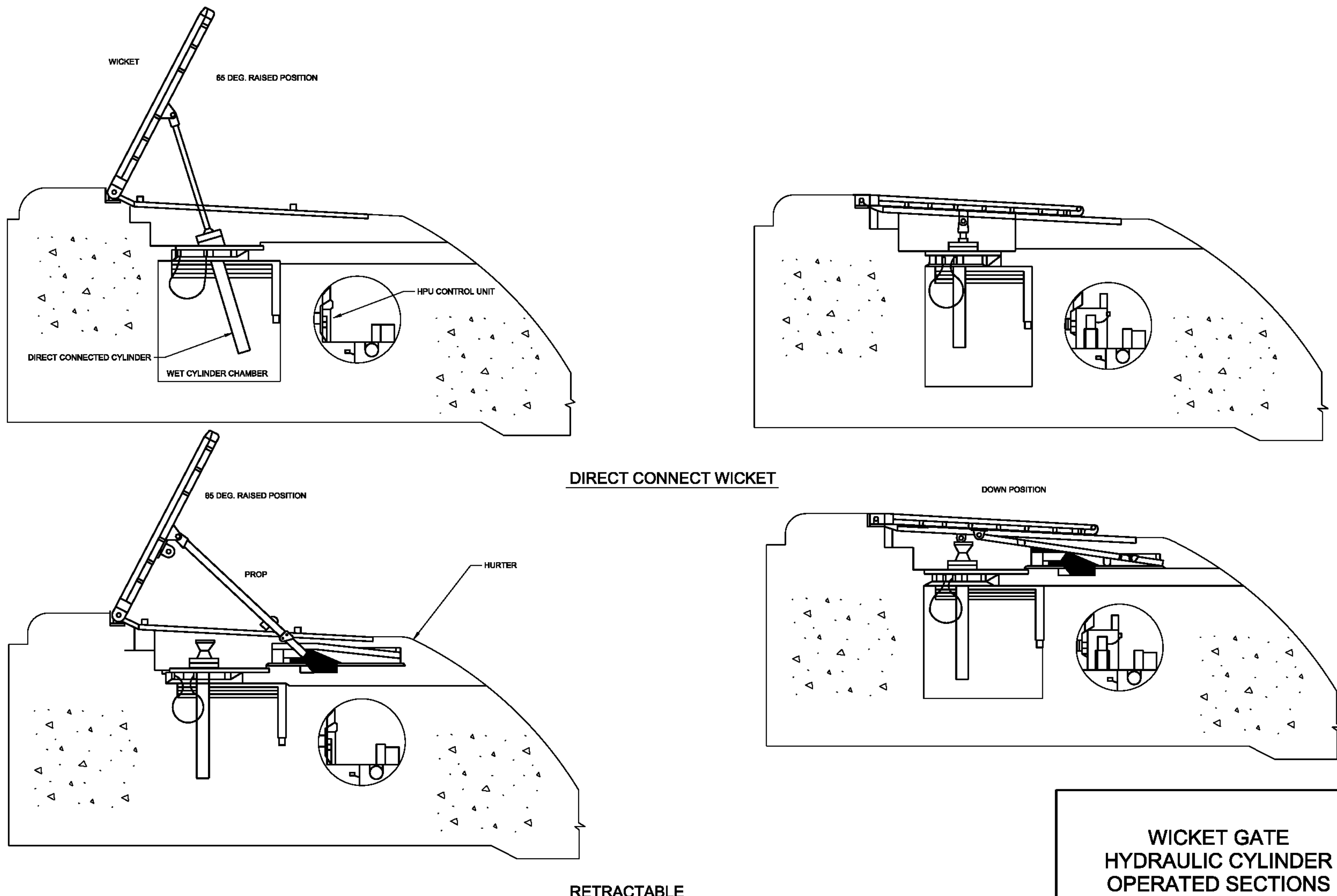
CHAIN GUARD

TAINTER GATE MACHINERY
POCKET WHEEL
GATE CONNECTION
AND DETAILS

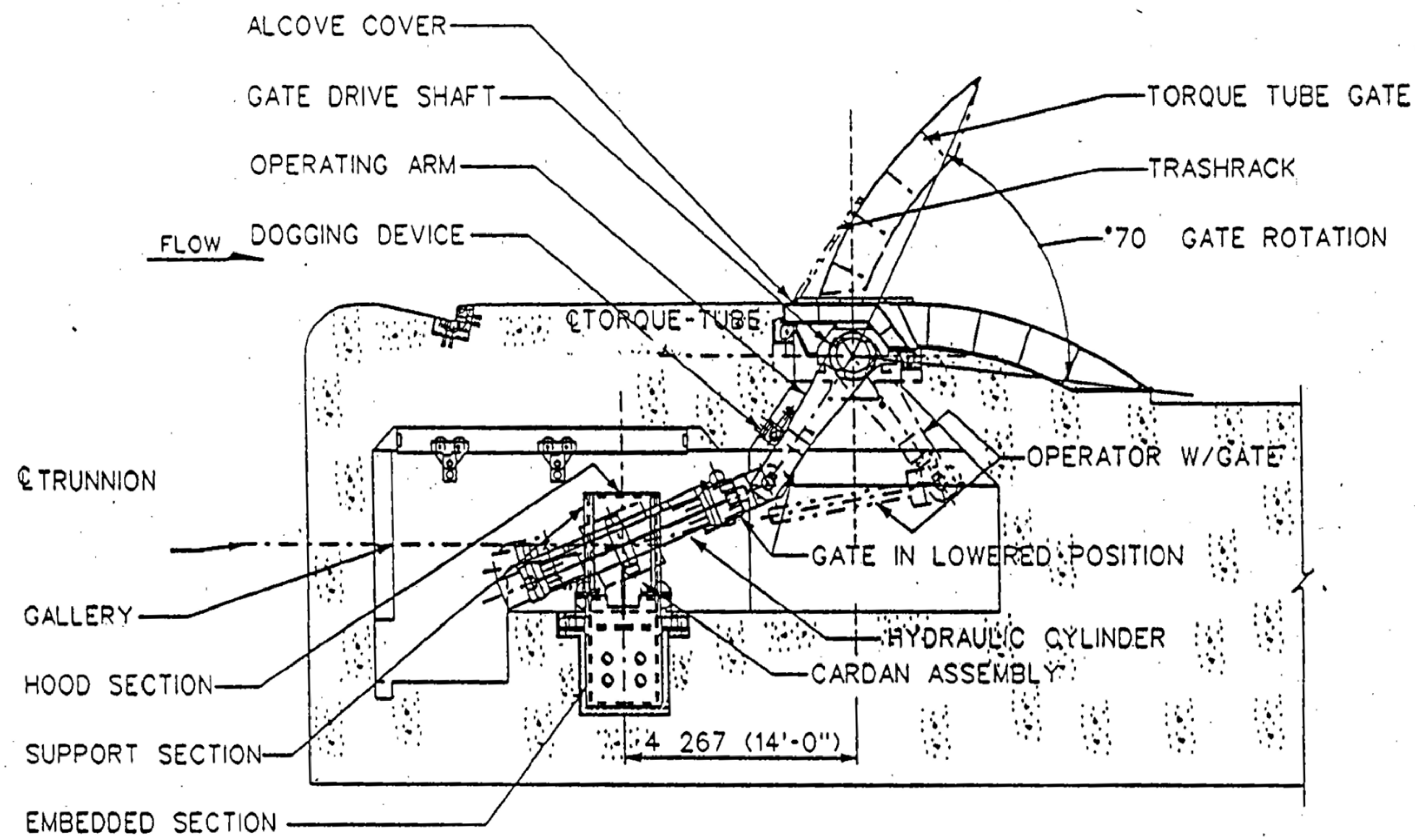
SHEET 3 OF 3 | PLATE B-72



WICKET GATE
MANUAL-OPERATED
PLAN AND SECTIONS

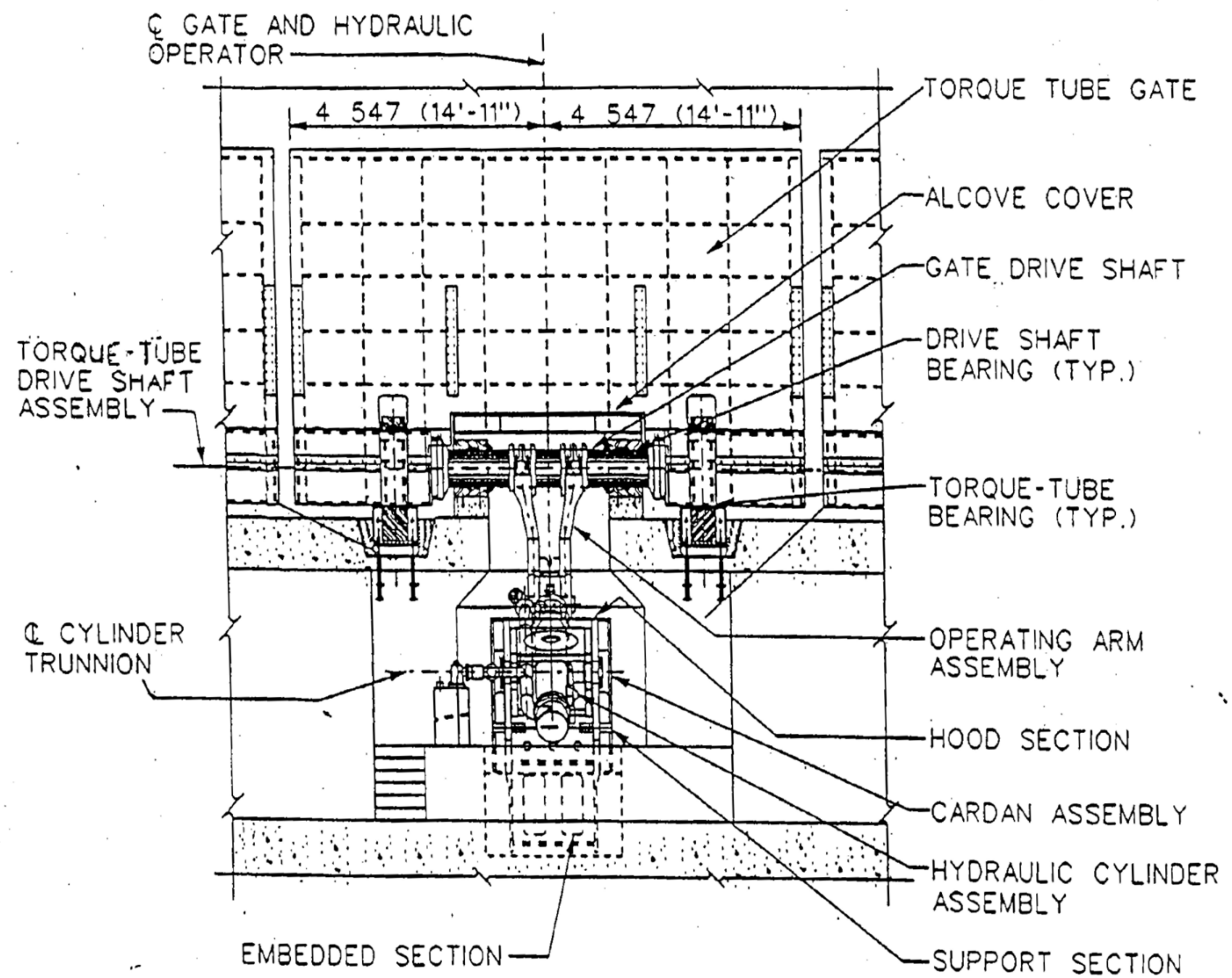


WICKET GATE
HYDRAULIC CYLINDER
OPERATED SECTIONS



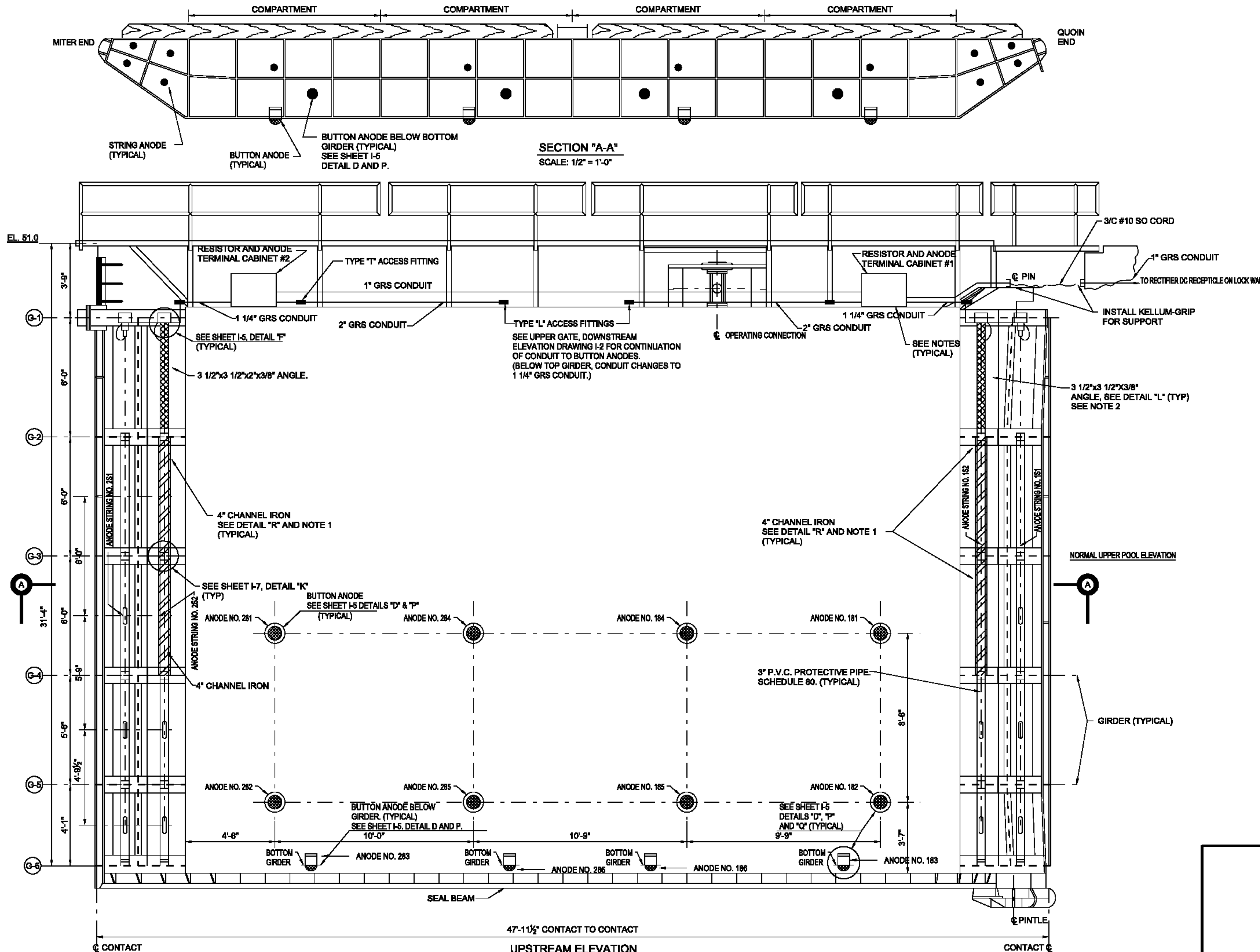
TORQUE TUBE GATE

HINGED CREST GATE
TORQUE TUBE GATE - ELEVATION



TORQUE TUBE GATE

HINGED CREST GATE
TORQUE TUBE GATE - PLAN



NOTES:

1. FOR THE FIRST COMPARTMENTS LOCATED ABOVE AND BELOW THE NORMAL UPPER POOL ELEVATION AS SHOWN ON "UPSTREAM ELEVATION" VIEW INSTALL 4" CHANNEL IRONS IN FRONT OF ANODE STRING NOS. 1S2 AND 2S2. WELD CHANNEL TO INSIDE OF GIRDER FLANGE.

2. FOR ANODE STRING NOS. 1S2 AND 2S2, INSTALL ANGLE IRON FROM THE UNDERSIDE OF THE TOP GIRDER TO THE TOPSIDE OF THE NEXT GIRDER BELOW NORMAL POOL ELEVATION (G-4). FOR CLARITY, ANGLE IRONS ARE SHOWN ONLY IN THE TOP-MOST COMPARTMENTS ON THE "UPTREAM ELEVATION" VIEW.

3. FOR 1-1/4" CONDUIT RUNS FOR BUTTON ANODES, REFER TO SHEET I-5, DETAILS "C" AND "P". A MINIMUM OF 2 CLAMPS SHALL BE INSTALLED BETWEEN EACH GIRDER.

4. OFFSET ANGLE IRONS IN TOP-MOST COMPARTMENT IN THE FIELD IF NECESSARY TO AVOID CONFLICT WITH CLEVIS AND INSULATOR. THIS NOTE ALSO APPLIES TO COMPARTMENTS WITH ONLY CABLE AND PVC.

5. IF CONDUITS ENTER RESISTOR AND ANODE TERMINAL CABINETS FROM BOTTOM, TYPE "L" ACCESS FITTINGS MAY BE USED IN LIEU OF 90-DEGREE BENDS.

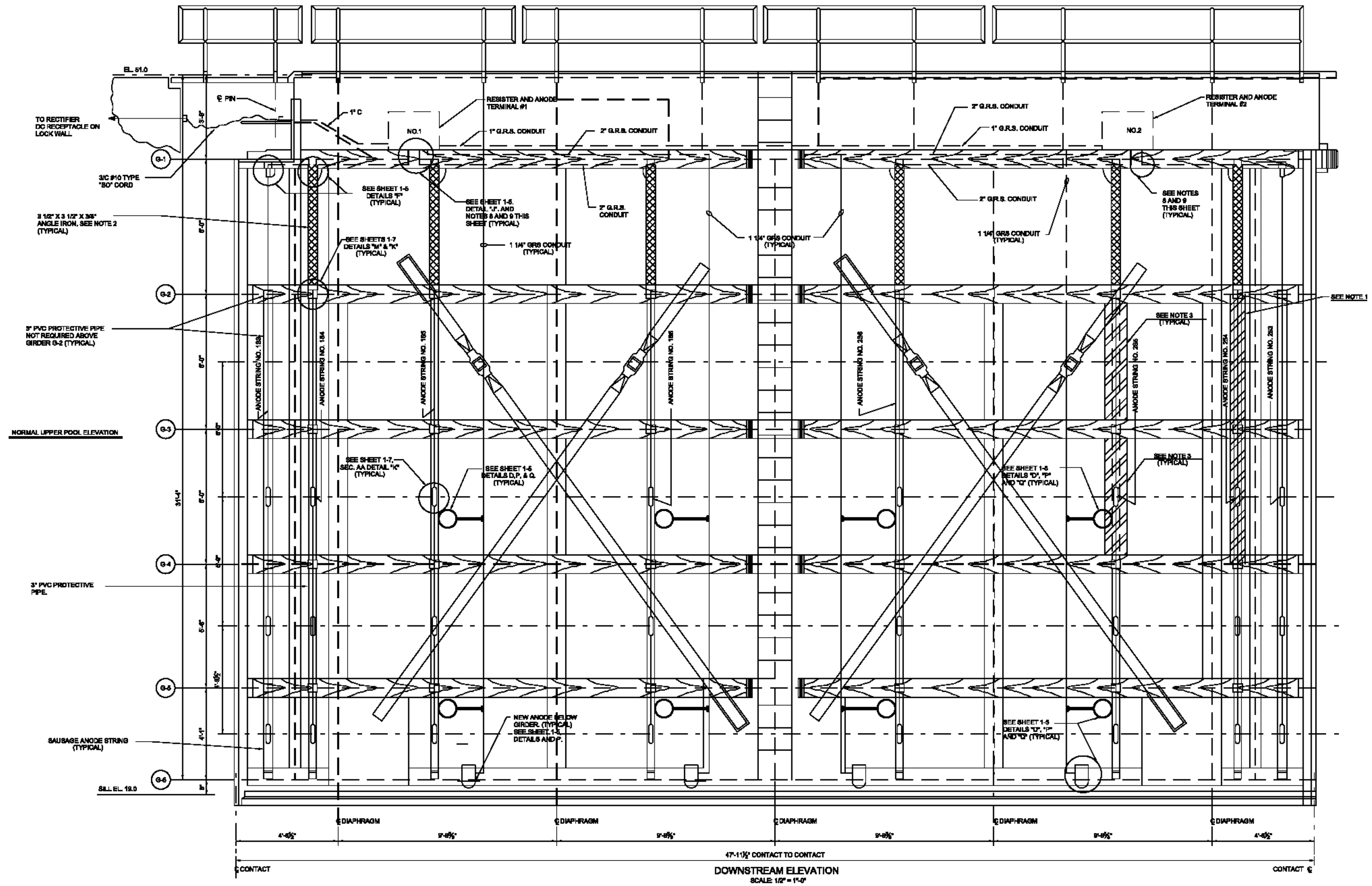
DESIGNERS INSTRUCTIONS:

1. DIMENSIONS AND GIRDER NUMBERS (G-1, G-2, ETC.) FOR THE STRUCTURE SHOWN ON THIS STANDARD DRAWING ARE FOR REFERENCE AND INFORMATION ONLY TO ASSIST THE DESIGNERS. DIMENSIONS WILL VARY FROM STRUCTURE TO STRUCTURE DEPENDING UPON THE WIDTH AND LIFT OF THE NAVIGATION LOCK; THEREFORE, THE DESIGNER IS RESPONSIBLE FOR PROVIDING THE NECESSARY ACTUAL STRUCTURAL DIMENSIONS.

2. THE GATE LEAF CONFIGURATIONS (WIDTH, LIFT, ETC.) OF THE UPSTREAM OR DOWNSTREAM FACE WILL DETERMINE THE GIRDER NUMBERS AND DIMENSIONS.

3. DIMENSIONS FOR ANODE SPACINGS SHOULD ALSO BE BASED UPON CALCULATIONS TO BE MADE BY THE DESIGNER. THROWING POWER OF THE ANODES MUST BE CONSIDERED TO INSURE MEETING THE NACE PROTECTION CRITERIA OF A MINIMUM OF MINUS 850 MILL VOLTS (COMPENSATED) AND 100 MILL VOLTS POLARIZATION DECAY.

GUIDE PLAN
LOCK CATHODIC PROTECTION
UPPER MITER GATE
UPSTREAM ELEVATION



NOTES:

1. FOR THE FIRST COMPARTMENTS LOCATED ABOVE AND BELOW THE NORMAL UPPER POOL ELEVATION AS SHOWN ON "DOWNSTREAM ELEVATION" VIEW . INSTALL 4" CHANNEL IRONS IN FRONT OF ANODE STRING NOS. 154 AND 254. FOR CLARITY. ONLY FOUR-INCH CHANNEL IRONS FOR ONE (1) STRING ANODE ARE SHOWN IN THIS VIEW. WELD CHANNEL TO INSIDE OF GIRDER FLANGE.

2. FOR ANODE STRING NOS. 154, 155, 156, 254, 255, AND 256, INSTALL ANGLE IRON FROM THE UNDERSIDE OF THE TOP GIRDER TO THE TOPSIDE OF THE NEXT GIRDER BELOW NORMAL POOL ELEVATION (G-4).

3. FOR ANODE STRING NOS. 155, 156, 255, AND 256. INSTALL 6" CHANNEL IRONS IN FRONT OF STRING ANODES FOR THE FIRST COMPARTMENT LOCATED ABOVE AND BELOW THE NORMAL UPPER POOL ELEVATION. WELD CHANNEL TO INSIDE OF GIRDER FLANGE.

THE REMAINING CHANNEL IRONS ARE NOT SHOWN ON THIS DRAWING FOR CLARITY.

4. FOR 1-1/4" CONDUIT RUNS FOR BUTTON ANODES, REFER TO SHEET 1-5, DETAILS "C" AND "P". A MINIMUM OF 2 CLAMPS SHALL BE INSTALLED BETWEEN EACH GIRDER.

5. OFFSET ANGLE IRONS IN TOP MOST COMPARTMENTS IN THE FIELD IF NECESSARY TO AVOID CONFLICT WITH CLEVIS AND INSULATOR. THIS NOTE ALSO APPLIES TO COMPARTMENTS WITH ONLY CABLE AND PVC.

6. SEE SHEET 1-5, DETAIL "Q" FOR TYPICAL GATE SECTION VIEW.

7. ANODES ARE TO BE NUMBERED IN SEQUENCE AS SHOWN.

8. ALL CONDUIT PENETRATIONS OF GIRDER G-1 SHALL BE WATERTIGHT IN ACCORDANCE WITH SHEET 1-5, DETAIL J.

9. TYPE "L" ACCESS FITTINGS MAY BE USED IN LIEU OF 90-DEGREE BENDS FOR ENTRY INTO RESISTOR AND ANODE TERMINAL CABINETS.

DESIGNERS INSTRUCTIONS:

1. DIMENSIONS AND GIRDER NUMBERS (G-1, G-2, ETC.) FOR THE STRUCTURE SHOWN ON THIS STANDARD DRAWING ARE FOR REFERENCE AND INFORMATION ONLY TO ASSIST THE DESIGNERS. DIMENSIONS WILL VARY FROM STRUCTURE TO STRUCTURE DEPENDING UPON THE WIDTH AND LIFT OF THE NAVIGATION LOCK; THEREFORE, THE DESIGNER IS RESPONSIBLE FOR PROVIDING THE NECESSARY ACTUAL STRUCTURAL DIMENSIONS.

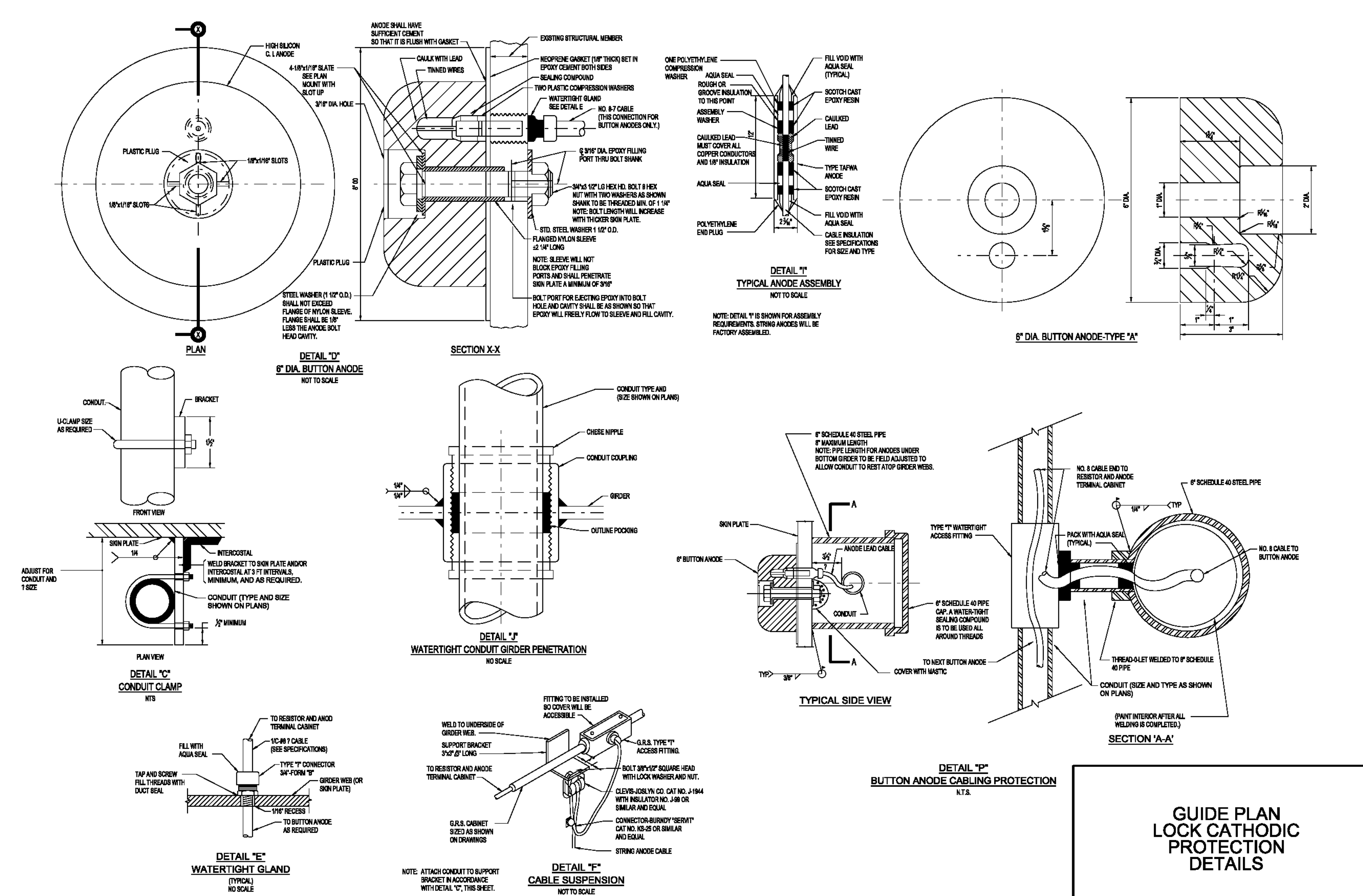
2. THE GATE LEAF CONFIGURATIONS (WIDTH, LIFT, ETC.) OF THE UPSTREAM OR DOWNSTREAM FACE WILL DETERMINE THE GIRDER NUMBERS AND DIMENSIONS.

3. DIMENSIONS FOR ANODE SPACINGS SHOULD ALSO BE BASED UPON CALCULATIONS TO BE MADE BY THE DESIGNER. THROWING POWER OF THE ANODES MUST BE CONSIDERED TO INSURE MEETING THE NACE PROTECTION CRITERIA OF A MINIMUM OF MINUS 850 MILL VOLTS (COMPENSATED) AND 100 MILL VOLTS POLARIZATION DECAY.

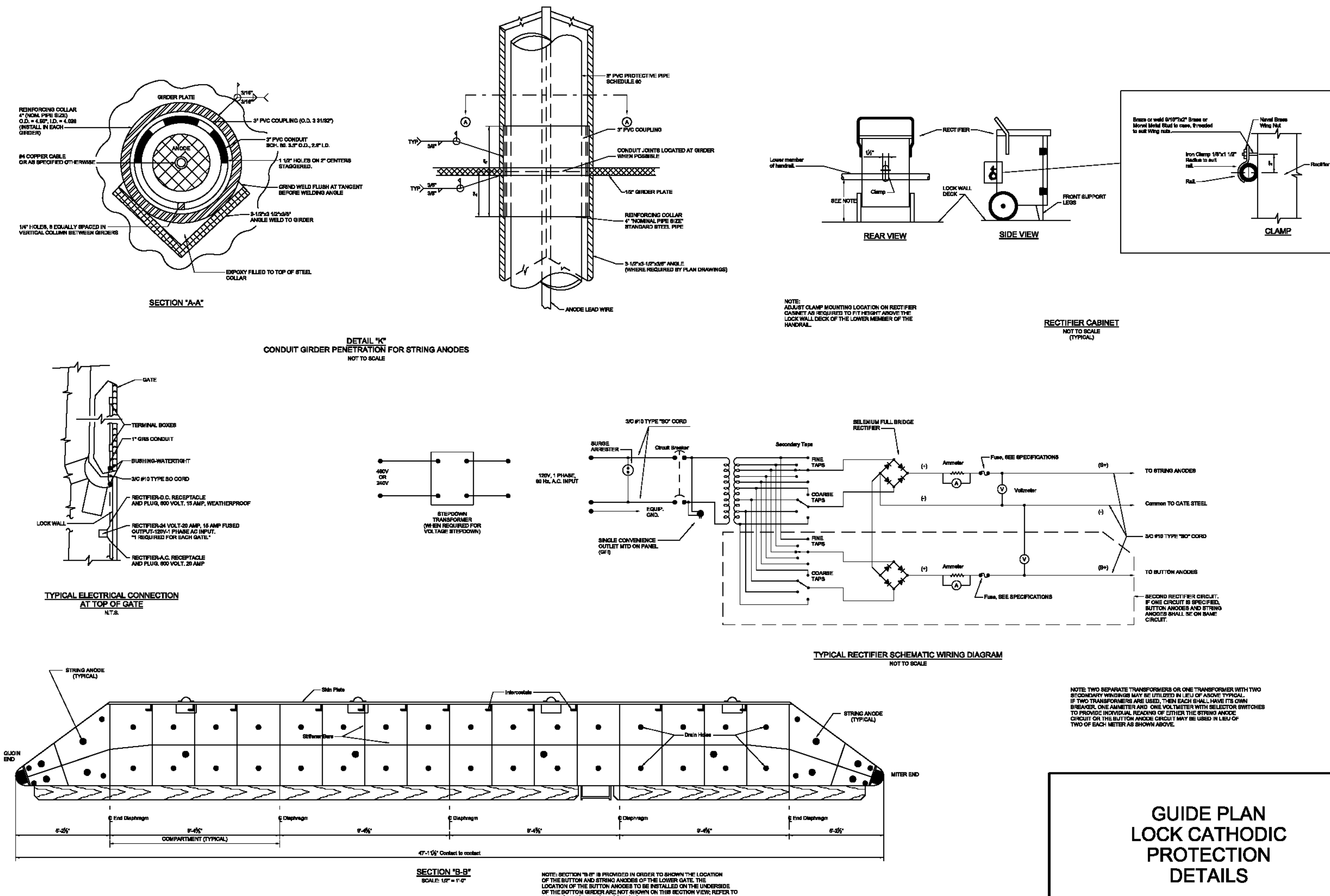
SCALE: 1/2" = 1'-0"

GUIDE PLAN
LOCK CATHODIC
PROTECTION
UPPER MITER GATE
DOWNSTREAM ELEVATION

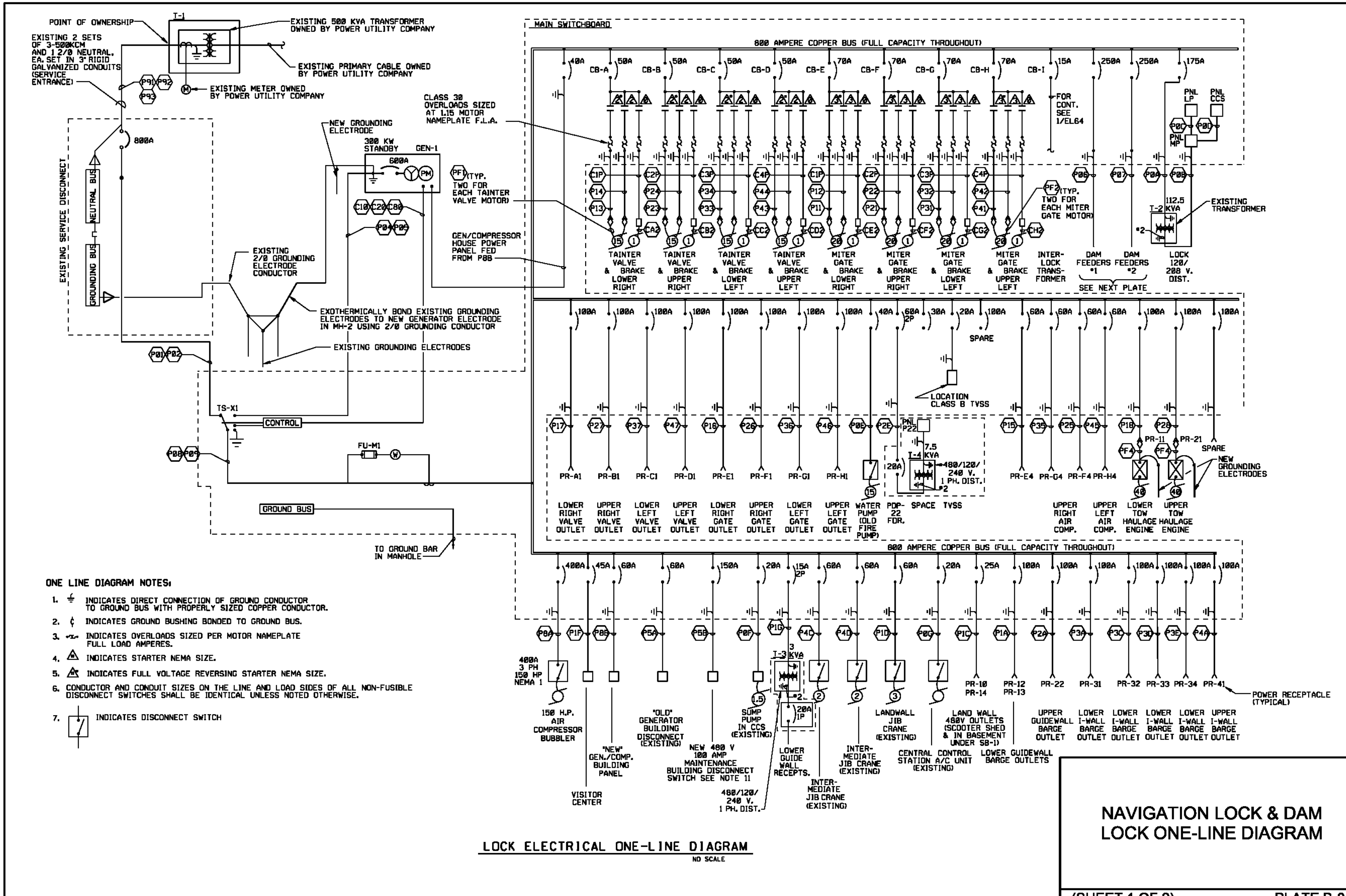
PLATE B-78



GUIDE PLAN
LOCK CATHODIC
PROTECTION
DETAILS

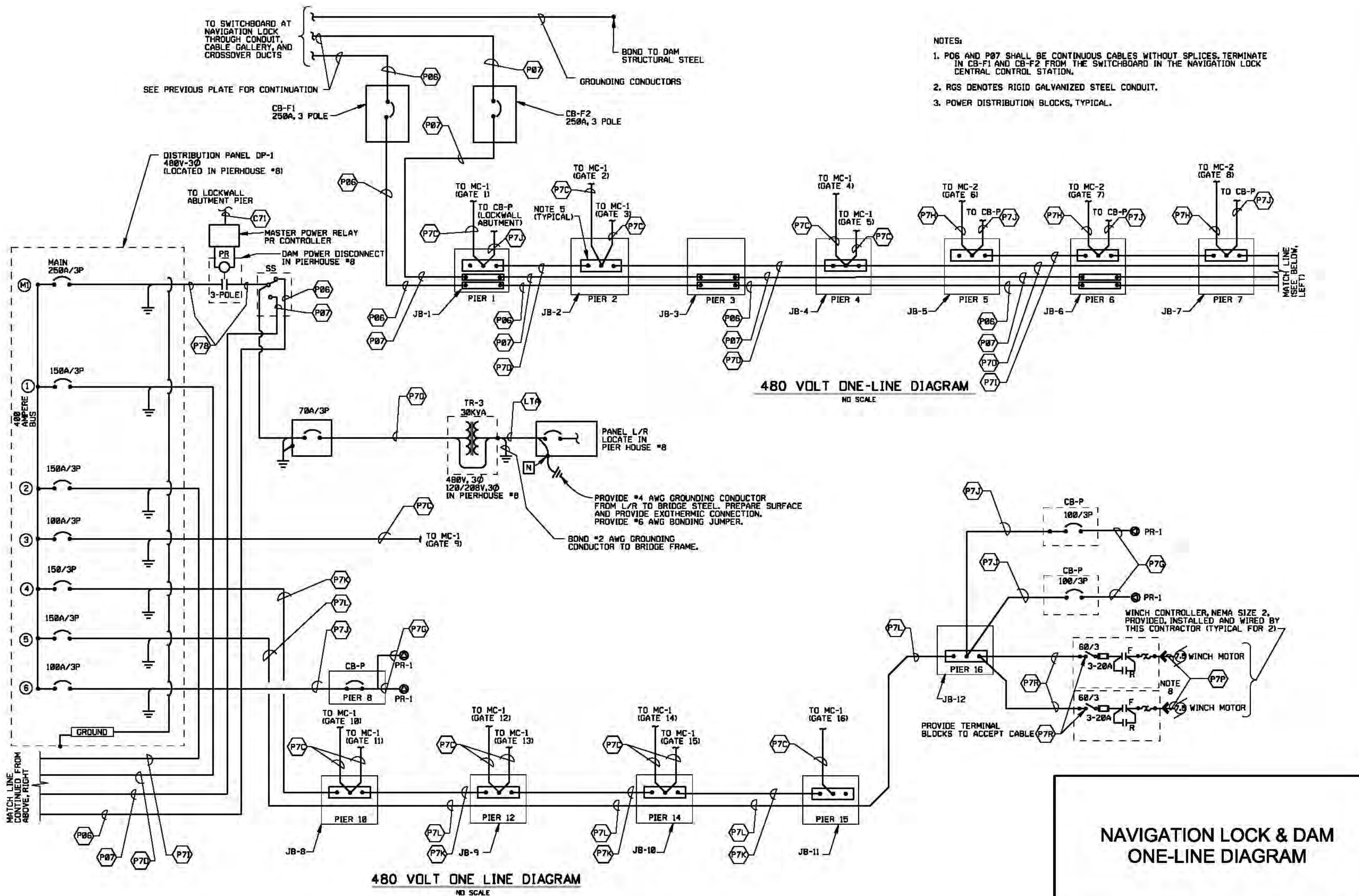


GUIDE PLAN LOCK CATHODIC PROTECTION DETAILS



NAVIGATION LOCK & DAM
LOCK ONE-LINE DIAGRAM

(SHEET 1 OF 2) PLATE B-81



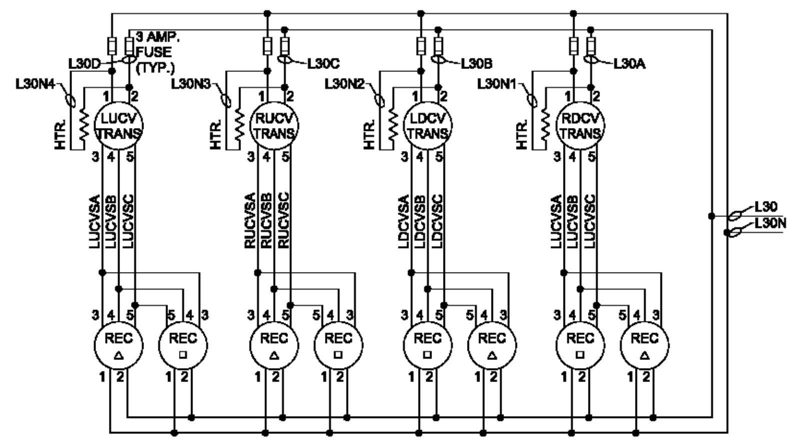
NOTES:

1. P06 AND P07 SHALL BE CONTINUOUS CABLES WITHOUT SPLICES. TERMINATE IN CB-F1 AND CB-F2 FROM THE SWITCHBOARD IN THE NAVIGATION LOCK CENTRAL CONTROL STATION.
2. RGS DENOTES RIGID GALVANIZED STEEL CONDUIT.
3. POWER DISTRIBUTION BLOCKS, TYPICAL.

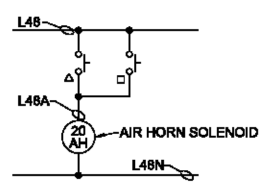
NAVIGATION LOCK & DAM ONE-LINE DIAGRAM

(SHEET 2 OF 2) PLATE B-82

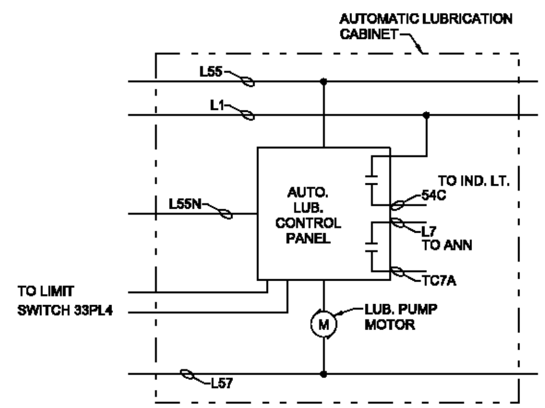
43NB BACK-UP/NORMAL TRANSFER SWITCH			
CONTACT NUMBER	SWITCH POSITION		CIRCUIT
	BACK-UP	NORMAL	
1-2		X	ULV-OPEN
3-4		X	ULV-CLOSE
5-6		X	URV-OPEN
7-8		X	URV-CLOSE
9-10		X	DLV-OPEN
11-12		X	DLV-CLOSE
13-14		X	DRV-OPEN
15-16		X	DRV-CLOSE
17-18		X	ULG-OPEN
19-20		X	ULG-CLOSE
21-22		X	URG-OPEN
23-24		X	URG-CLOSE
25-26		X	DLG-OPEN
27-28		X	DLG-CLOSE
29-30		X	DRG-OPEN
31-32		X	DRG-CLOSE
33-34		X	ULG-LATCH
35-36		X	URG-LATCH
37-38		X	DLG-LATCH
39-40		X	DRG-LATCH
41-42		X	L2
43-44		X	LW-RATE CNTRL.
45-46	X		LW-RATE CNTRL.
47-48		X	LW-20LI
49-50		X	LW-20L2
51-52		X	LW-20L3
53-54		X	LW-20L4
55-56		X	RW-RATE CNTRL.
57-58	X		RW-RATE CNTRL.
59-60		X	RW-20RI
61-62		X	RW-20R2
63-64		X	RW-20R3
65-66		X	RW-20R4



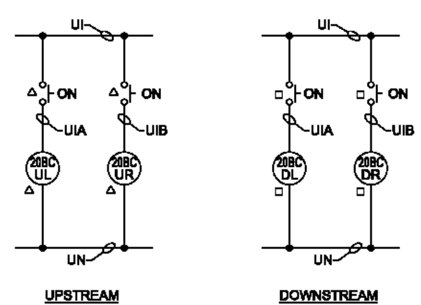
CULVERT VALVE SELSYN CONTROL



AIR HORN



DOWNSTREAM LEFT GATE LUB SYSTEM
(OTHER 3 GATES LUB SYSTEM SIMILAR)

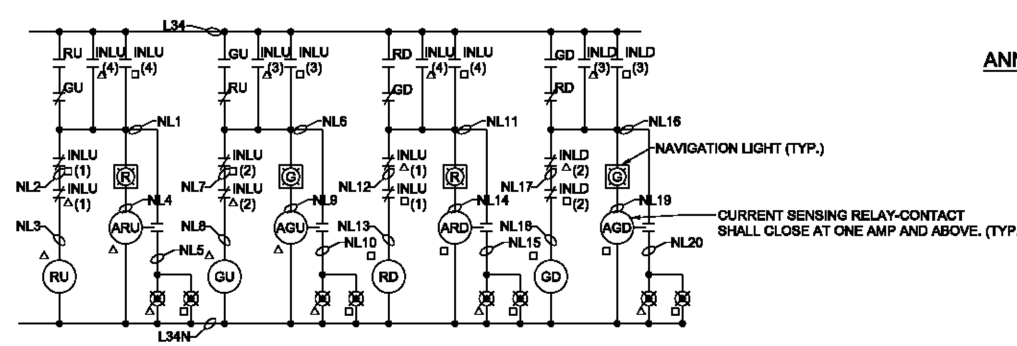


BUBBLER CONTROL

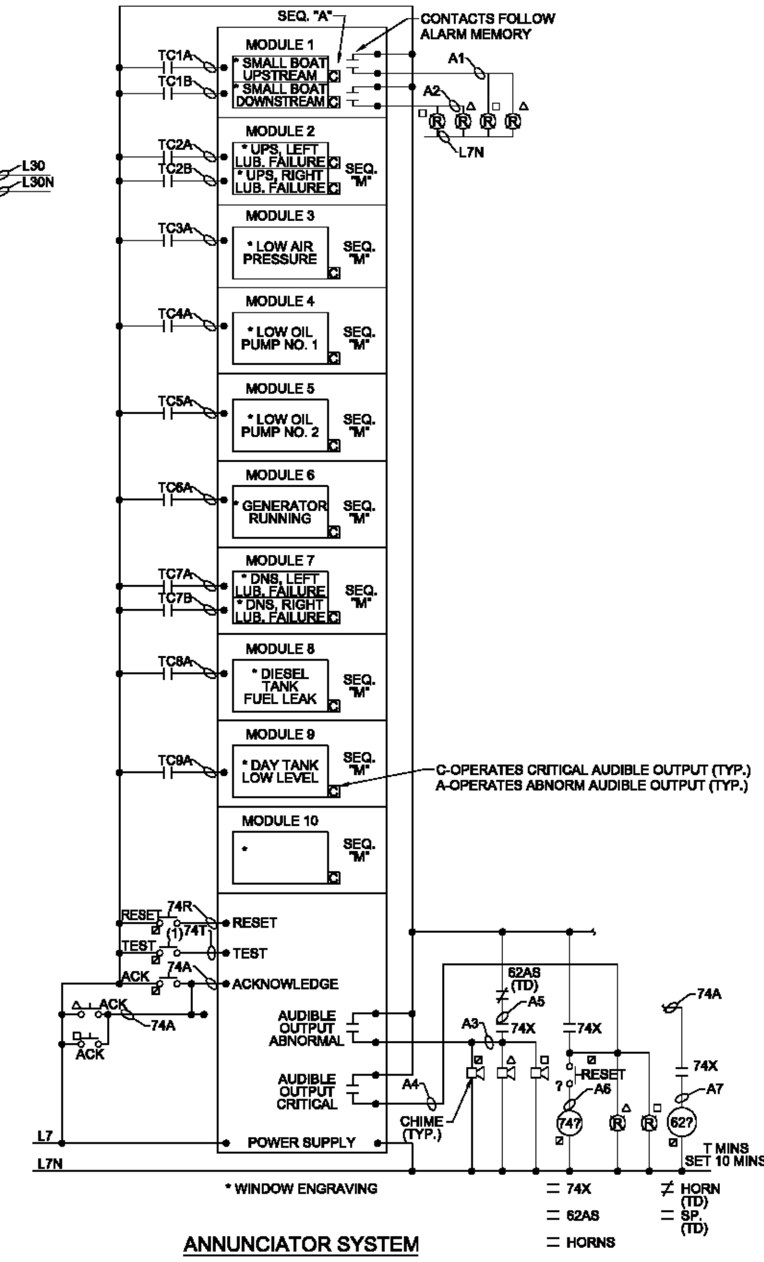
UPSTREAM
NAVIGATIONAL SIGNAL LIGHT
SELECTOR PUSH BUTTON

DEVICE	CONTACT NO.	POSITION FRONT VIEW					
		GREEN		OFF		RED	
		F	D	F	D	F	D
INLU	1	X	-	X	-	X	X
	2	X	X	X	-	X	-
	3	-	X	-	-	-	-
	4	-	-	-	-	-	X

DOWNSTREAM DEVICE "INLD" SIMILAR
X = DENOTES CONTACTS CLOSED
F = FREE; D = DEPRESSED



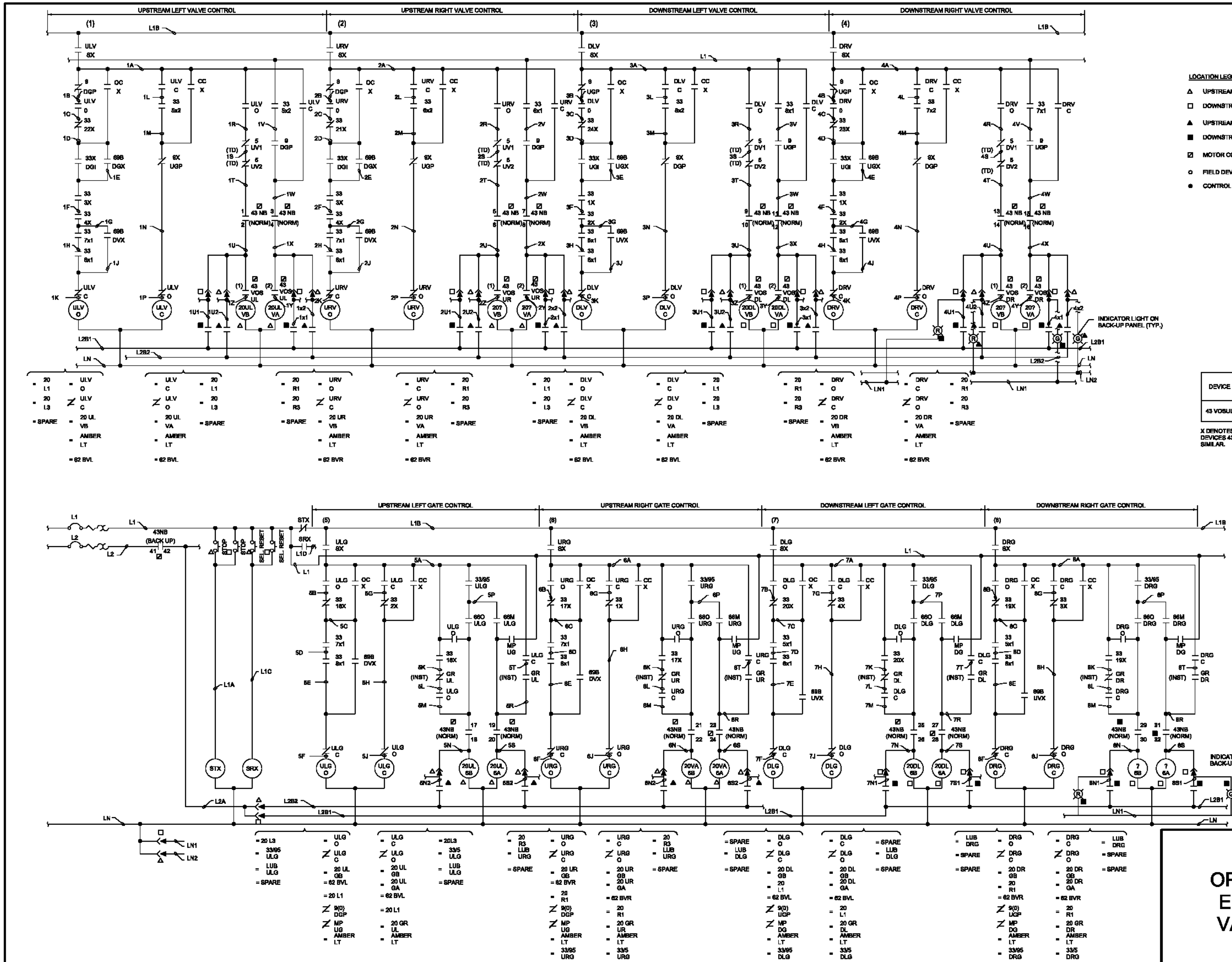
NAVIGATIONAL SIGNAL LIGHT CONTROL



ANNUNCIATOR SYSTEM

LOCK GATES AND
OPERATING EQUIPMENT
ELECTRICAL SYSTEMS
CONTROL SCHEMATICS

PLATE B-83



LOCATION LEGEND

- △ UPSTREAM CONTROL BOOTH
- □ DOWNSTREAM CONTROL BOOTH
- ▲ UPSTREAM BACK-UP PANEL
- ■ DOWNSTREAM BACK-UP PANEL
- ⧄ MOTOR CONTROL CENTER
- ○ FIELD DEVICE
- ● CONTROL STATION

UPSTREAM LEFT VALVE
OUT OF SERVICE
SELECTOR SWITCH

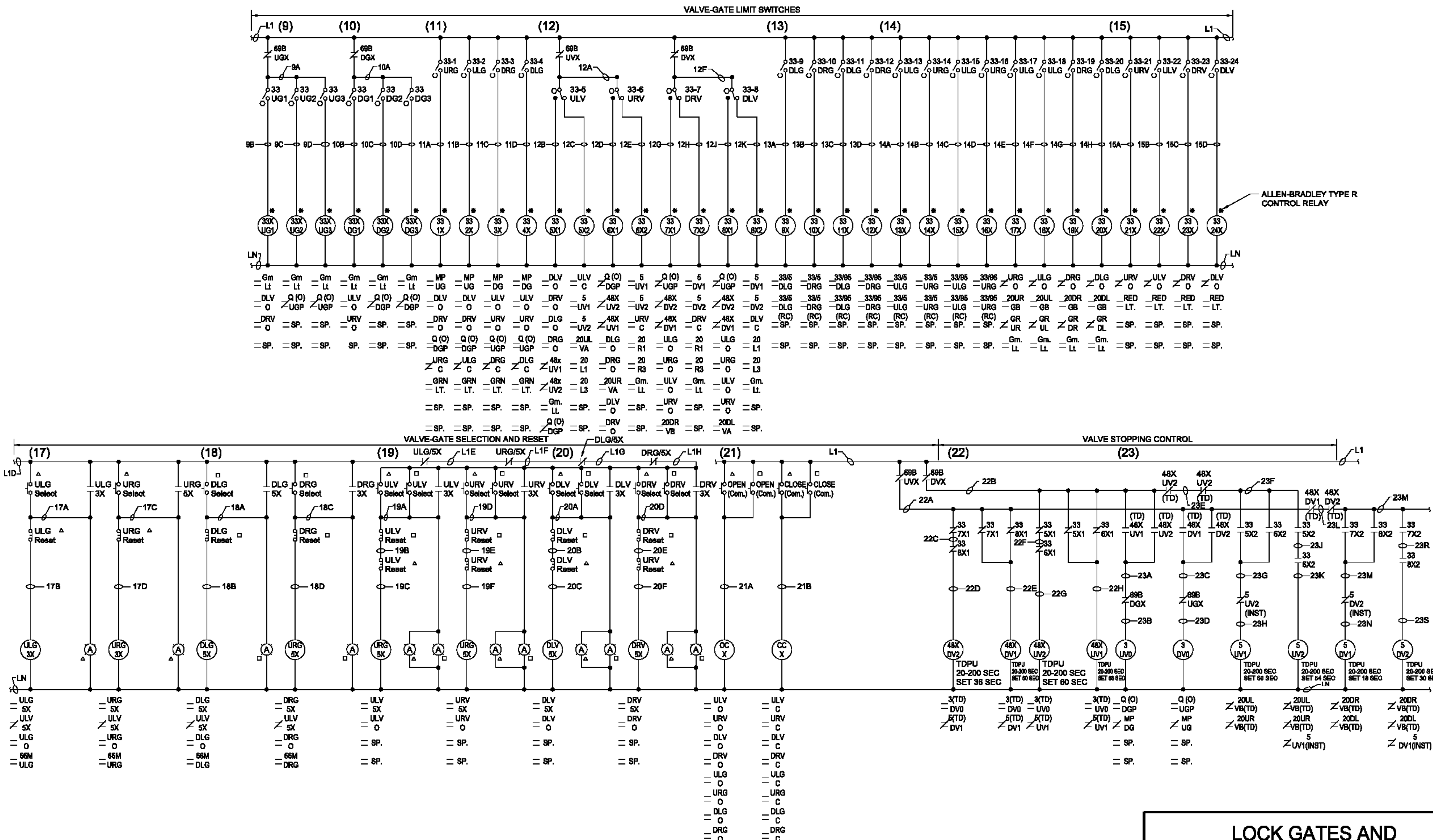
DEVICE	CONTACT NO.	POSITION FRONT VIEW	
		OUT OF SERVICE	NORMAL
43 VOSUL	1		X
	2		X

X DENOTES CONTACTS CLOSED.
DEVICES 43VOSUR, 43VOSDL AND 43VOSDR SIMILAR.

NOTE:
UNLESS OTHERWISE NOTED ALL EQUIPMENT, THIS DRAWING, IS LOCATED IN THE RELAY CABINET.

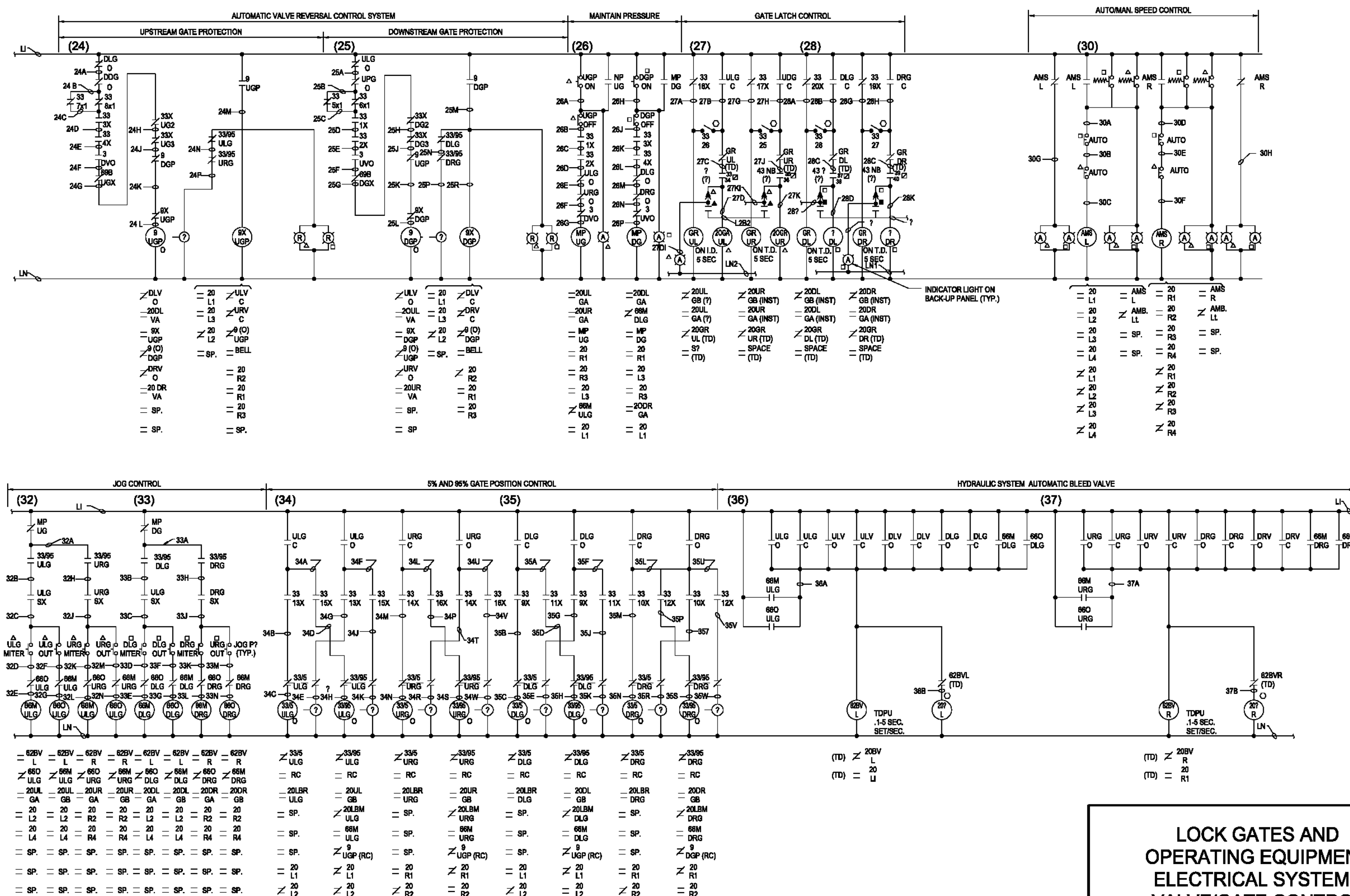
LOCK GATES AND
OPERATING EQUIPMENT
ELECTRICAL SYSTEMS
VALVE/GATE CONTROL
SCHEMATIC

(SHEET 1 OF 5) PLATE B-84



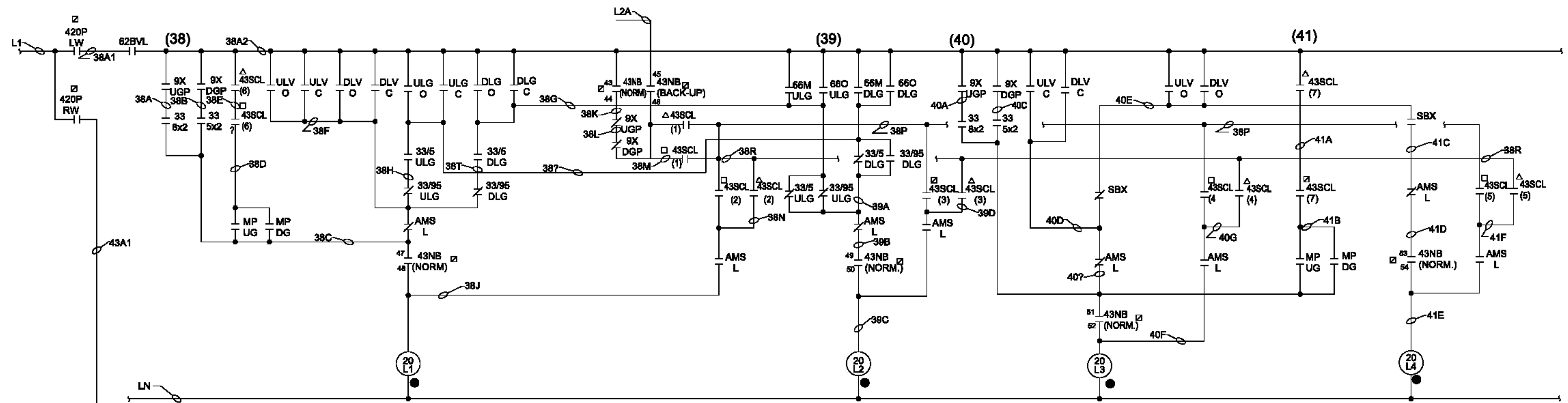
LOCK GATES AND OPERATING EQUIPMENT ELECTRICAL SYSTEMS VALVE/GATE CONTROL SCHEMATIC

(SHEET 2 OF 5) PLATE B-85

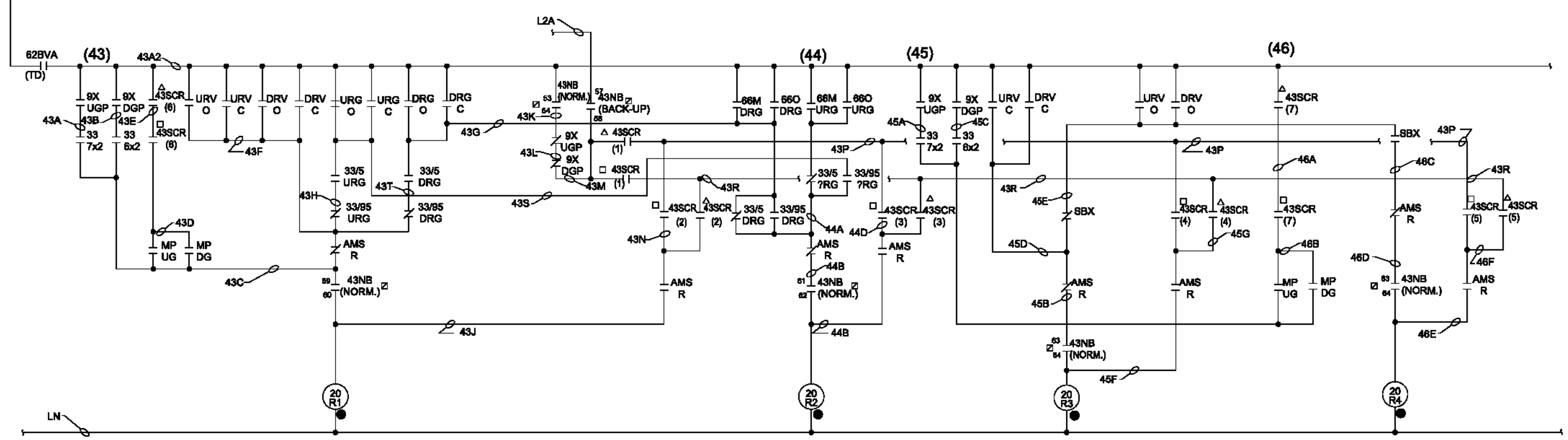


LOCK GATES AND
OPERATING EQUIPMENT
ELECTRICAL SYSTEMS
VALVE/GATE CONTROL
SCHEMATIC

(SHEET 3 OF 5) PLATE B-86



LEFT WALL HYDRAULIC PUMPING RATE CONTROL (OIL PUMP NO. 2)

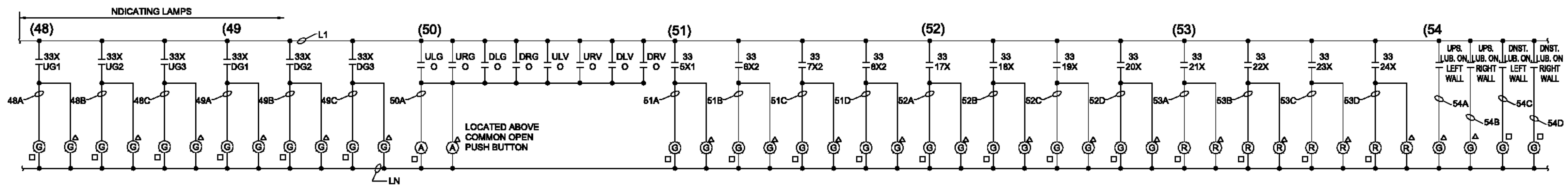


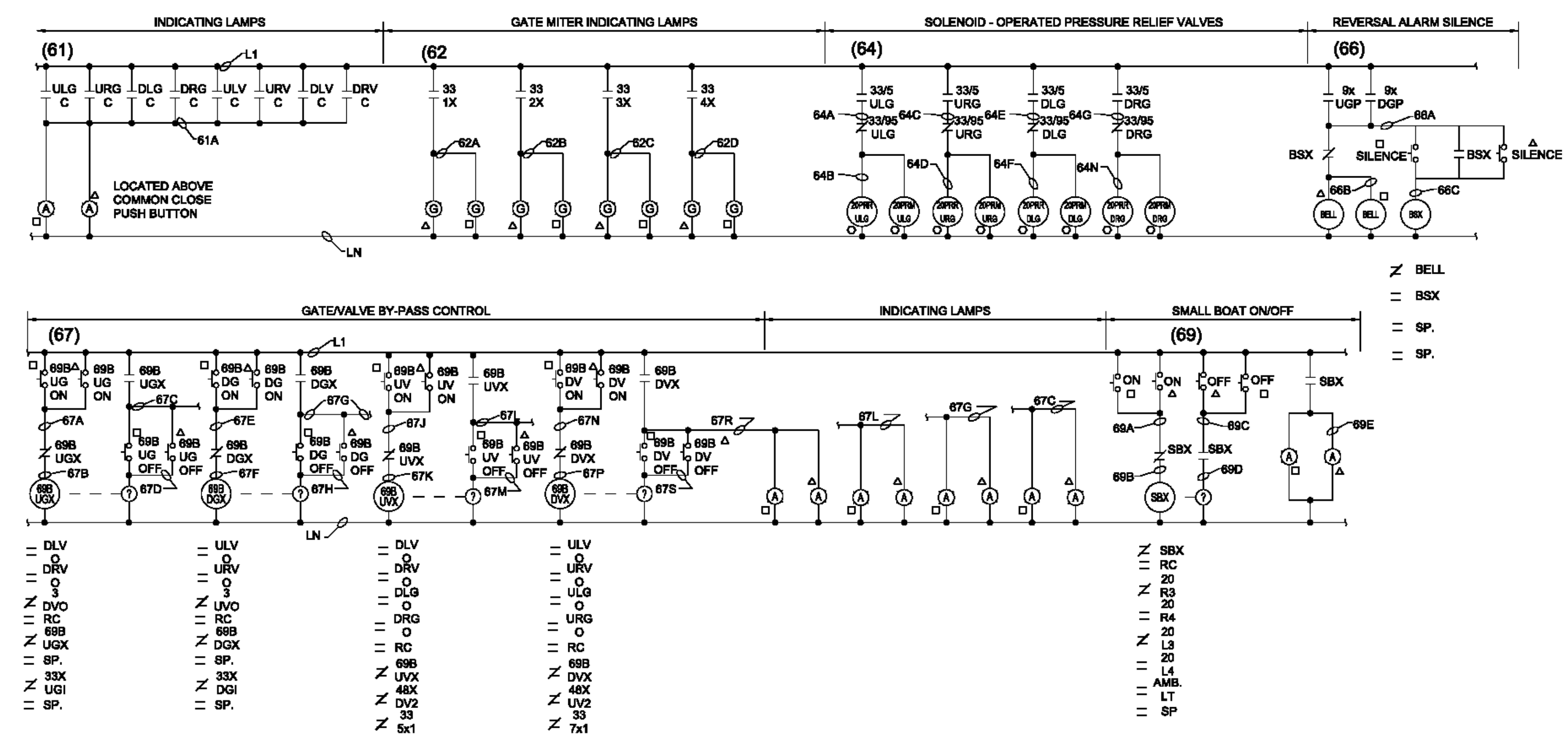
RIGHT WALL HYDRAULIC PUMPING RATE CONTROL (OIL PUMP NO. 1)

43SCR	RIGHT WALL GATE-VALVE SPEED CONTROL SWITCH		GATE FAST	GATE SLOW	NEUT	VALVE SLOW	VALVE FAST	BLACK PISTOL GRIP HANDLE ALL POSITIONS MAINTAINED
		1			X			
		2	X			X	X	
		3		X				
		4					X	
		5				X		
		6			X			
		7			X			
		8			X			

43SCL SIMILAR
X DENOTES CONTACTS CLOSED

LOCK GATES AND
OPERATING EQUIPMENT
ELECTRICAL SYSTEMS
VALVE/GATE CONTROL
SCHEMATIC

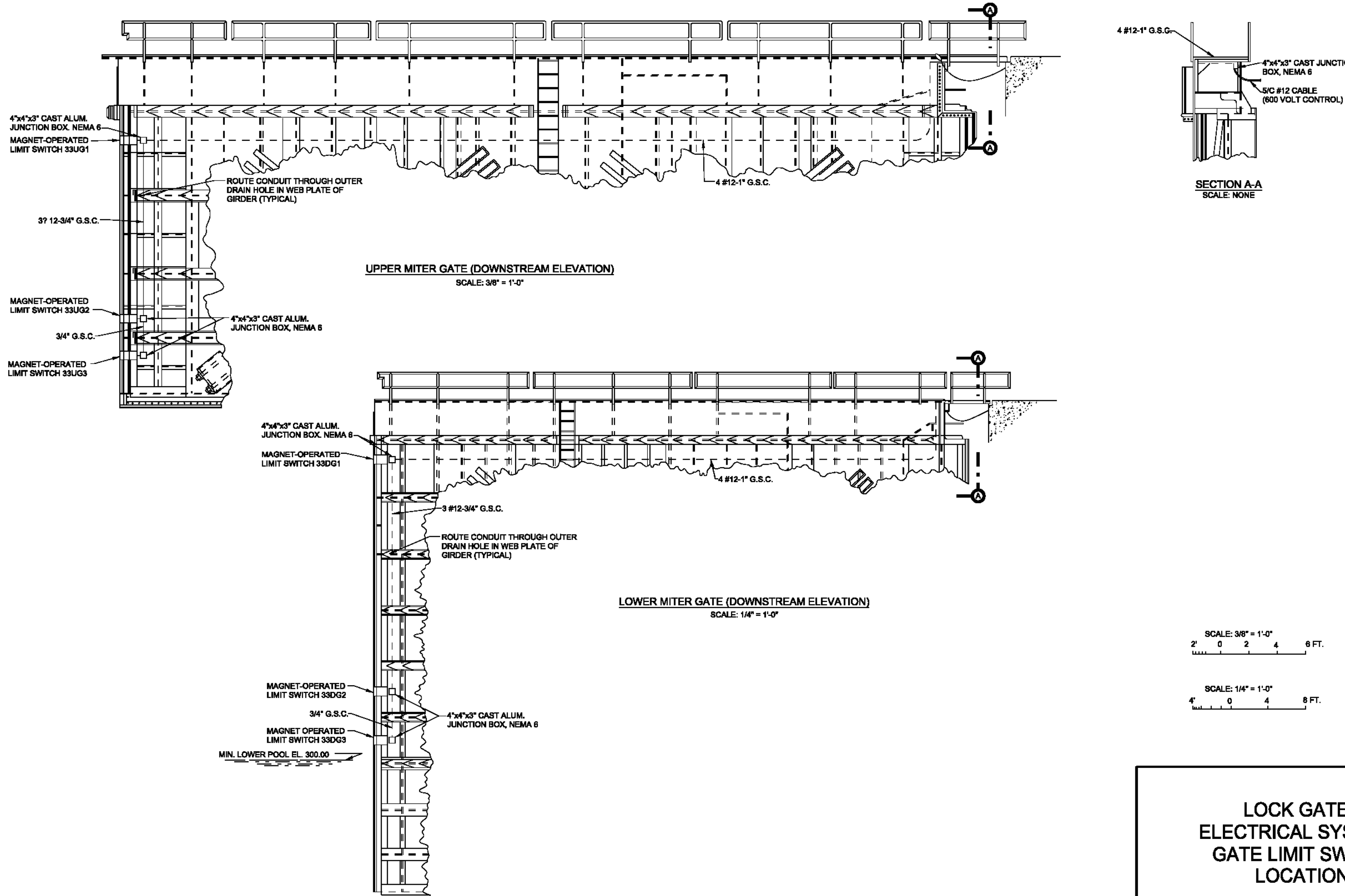




LIMIT SWITCH SCHEDULE			
LIMIT SWITCH	LOCATION	CONTACT OPERATION	REMARKS
33-1	URG MACH. RECESS	CLOSES WHEN GATE REACHES MITER POSITION.	MAGNET OPERATED
33-2	ULG MACH. RECESS	CLOSES WHEN GATE REACHES MITER POSITION.	MAGNET OPERATED
33-3	DRG MACH. RECESS	CLOSES WHEN GATE REACHES MITER POSITION.	MAGNET OPERATED
33-4	DLG MACH. RECESS	CLOSES WHEN GATE REACHES MITER POSITION.	MAGNET OPERATED
33-5	ULV MACH. RECESS	CLOSES WHEN VALVE IS FULLY CLOSED.	MAGNET OPERATED
33-6	URV MACH. RECESS	CLOSES WHEN VALVE IS FULLY CLOSED.	MAGNET OPERATED
33-7	DRV MACH. RECESS	CLOSES WHEN VALVE IS FULLY CLOSED.	MAGNET OPERATED
33-8	DLV MACH. RECESS	CLOSES WHEN VALVE IS FULLY CLOSED.	MAGNET OPERATED
33-9	DLG MACH. RECESS	CLOSES WHEN GATE IS 5% OUT OF THE RECESS.	MAGNET OPERATED
33-10	DRG MACH. RECESS	CLOSES WHEN GATE IS 5% OUT OF THE RECESS.	MAGNET OPERATED
33-11	DLG MACH. RECESS	CLOSES WHEN GATE IS 95% OUT OF THE RECESS.	MAGNET OPERATED
33-12	DRG MACH. RECESS	CLOSES WHEN GATE IS 95% OUT OF THE RECESS.	MAGNET OPERATED
33-13	ULG MACH. RECESS	SAME AS 33-9.	MAGNET OPERATED
33-14	URG MACH. RECESS	SAME AS 33-9.	MAGNET OPERATED
33-15	ULG MACH. RECESS	SAME AS 33-11.	MAGNET OPERATED
33-16	URG MACH. RECESS	SAME AS 33-11.	MAGNET OPERATED
33-17	URG GATE RECESS	CLOSES WHEN GATE IS FULLY OPEN.	MAGNET OPERATED
33-18	ULG GATE RECESS	CLOSES WHEN GATE IS FULLY OPEN.	MAGNET OPERATED
33-19	DRG GATE RECESS	CLOSES WHEN GATE IS FULLY OPEN.	MAGNET OPERATED
33-20	DLG GATE RECESS	CLOSES WHEN GATE IS FULLY OPEN.	MAGNET OPERATED
33-21	URV MACH. RECESS	CLOSES WHEN VALVE IS FULLY OPEN.	MAGNET OPERATED
33-22	ULV MACH. RECESS	CLOSES WHEN VALVE IS FULLY OPEN.	MAGNET OPERATED
33-23	DRV MACH. RECESS	CLOSES WHEN VALVE IS FULLY OPEN.	MAGNET OPERATED
33-24	DLV MACH. RECESS	CLOSES WHEN VALVE IS FULLY OPEN.	MAGNET OPERATED
33-25	URG LATCH	CLOSES WHEN GATE LATCH IS CLEAR.	MAGNET OPERATED
33-26	ULG LATCH	CLOSES WHEN GATE LATCH IS CLEAR.	MAGNET OPERATED
33-27	DRG LATCH	CLOSES WHEN GATE LATCH IS CLEAR.	MAGNET OPERATED
33-28	DLG LATCH	CLOSES WHEN GATE LATCH IS CLEAR.	MAGNET OPERATED
33-DG1	DG TOP OF GATE	CLOSES WHEN GATES ARE MITERED.	MAGNET OPERATED
33-DG2	DG MIDDLE OF GATE	CLOSES WHEN GATES ARE MITERED.	MAGNET OPERATED
33-DG3	DG BOTTOM OF GATE	CLOSES WHEN GATES ARE MITERED.	MAGNET OPERATED
33-UG1	UG TOP OF GATE	CLOSES WHEN GATES ARE MITERED.	MAGNET OPERATED
33-UG2	UG MIDDLE OF GATE	CLOSES WHEN GATES ARE MITERED.	MAGNET OPERATED
33-UG3	UG BOTTOM OF GATE	CLOSES WHEN GATES ARE MITERED.	MAGNET OPERATED
33PL1	URG MACH. RECESS	CLOSES WHEN GATE IS 50% OUT OF THE RECESS.	MAGNET OPERATED
33PL2	ULG MACH. RECESS	CLOSES WHEN GATE IS 50% OUT OF THE RECESS.	MAGNET OPERATED
33PL3	DRG MACH. RECESS	CLOSES WHEN GATE IS 50% OUT OF THE RECESS.	MAGNET OPERATED
33PL4	DLG MACH. RECESS	CLOSES WHEN GATE IS 50% OUT OF THE RECESS.	MAGNET OPERATED

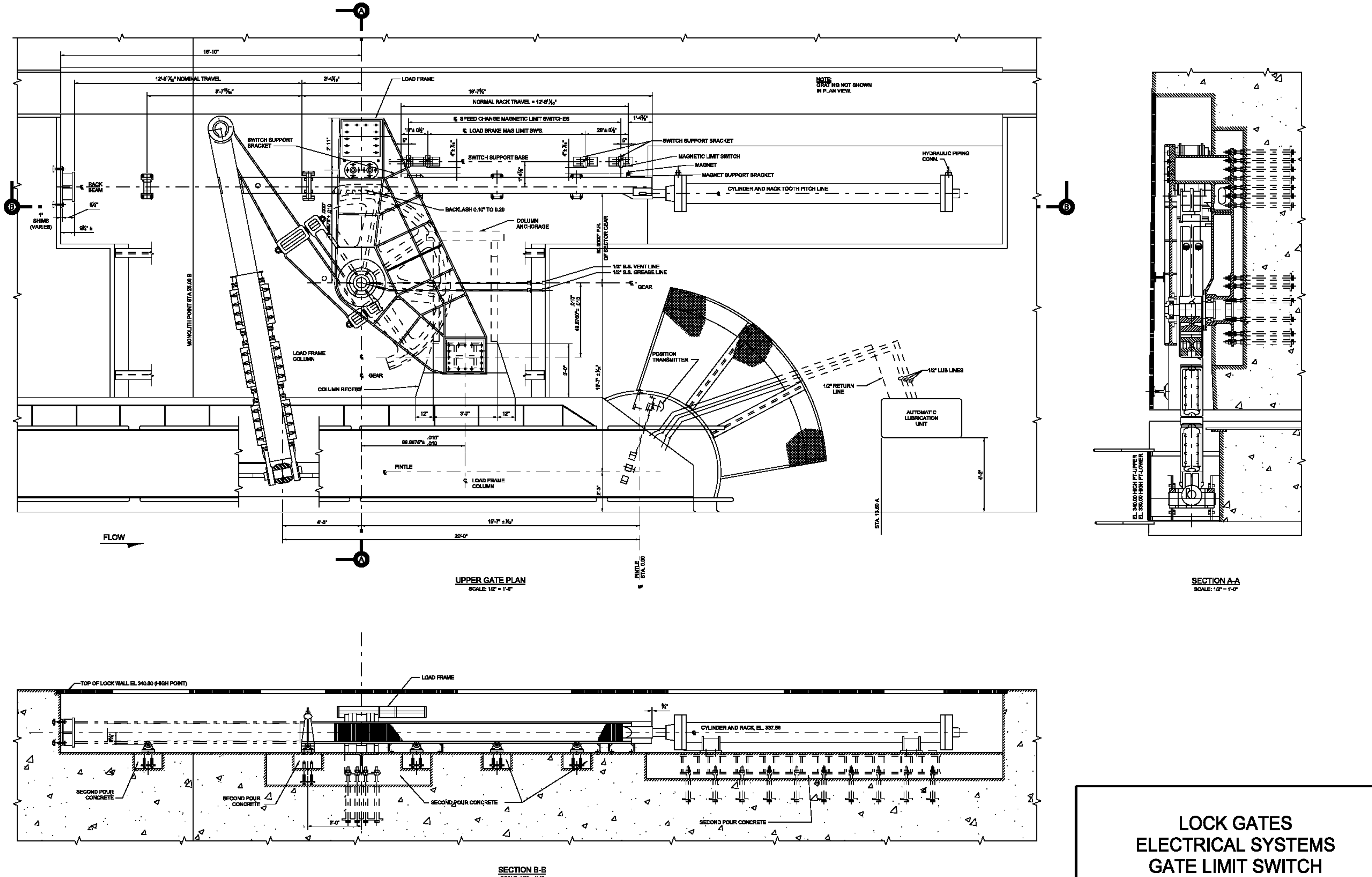
LOCK GATES AND OPERATING EQUIPMENT ELECTRICAL SYSTEMS VALVE/GATE CONTROL SCHEMATIC

(SHEET 5 OF 5) PLATE B-88



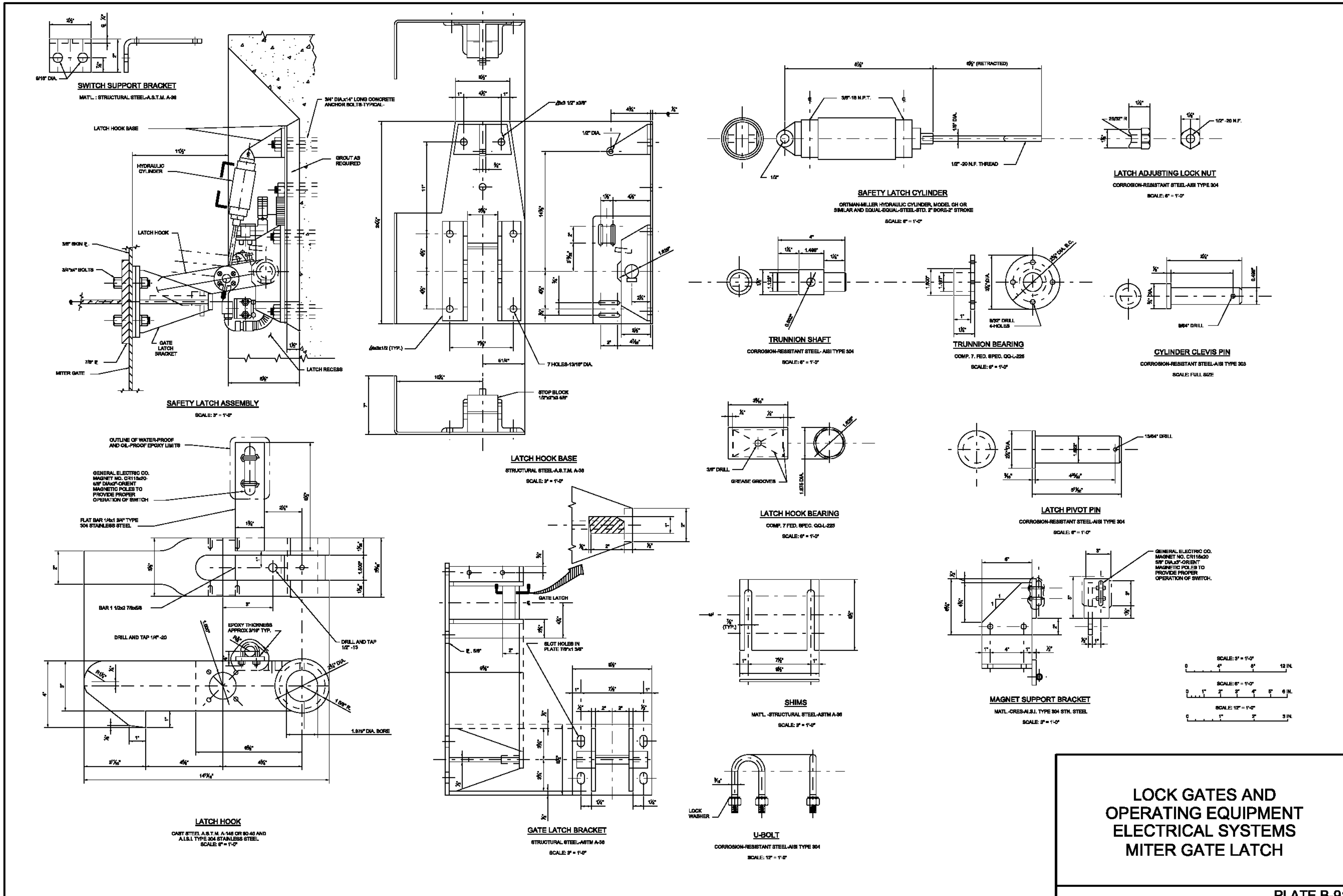
LOCK GATES
ELECTRICAL SYSTEMS
GATE LIMIT SWITCH
LOCATION

(SHEET 1 OF 2) PLATE B-89



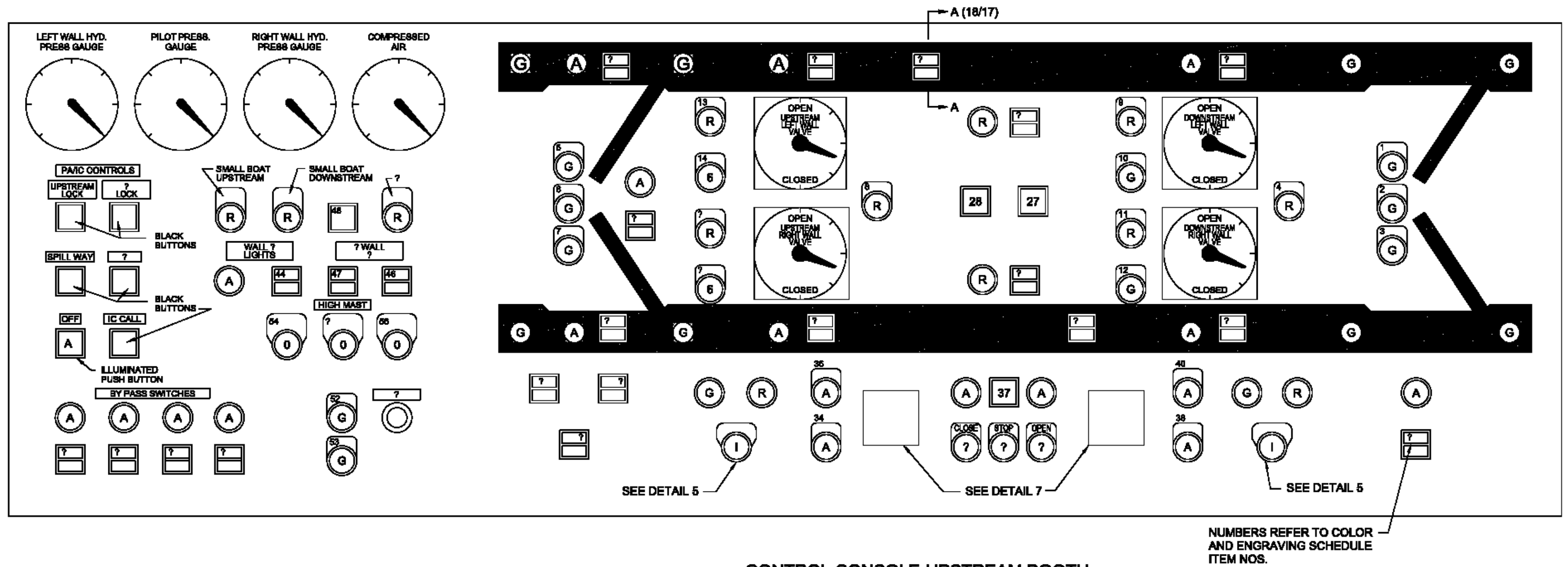
LOCK GATES
ELECTRICAL SYSTEMS
GATE LIMIT SWITCH
LOCATION

(SHEET 2 OF 2) PLATE B-90

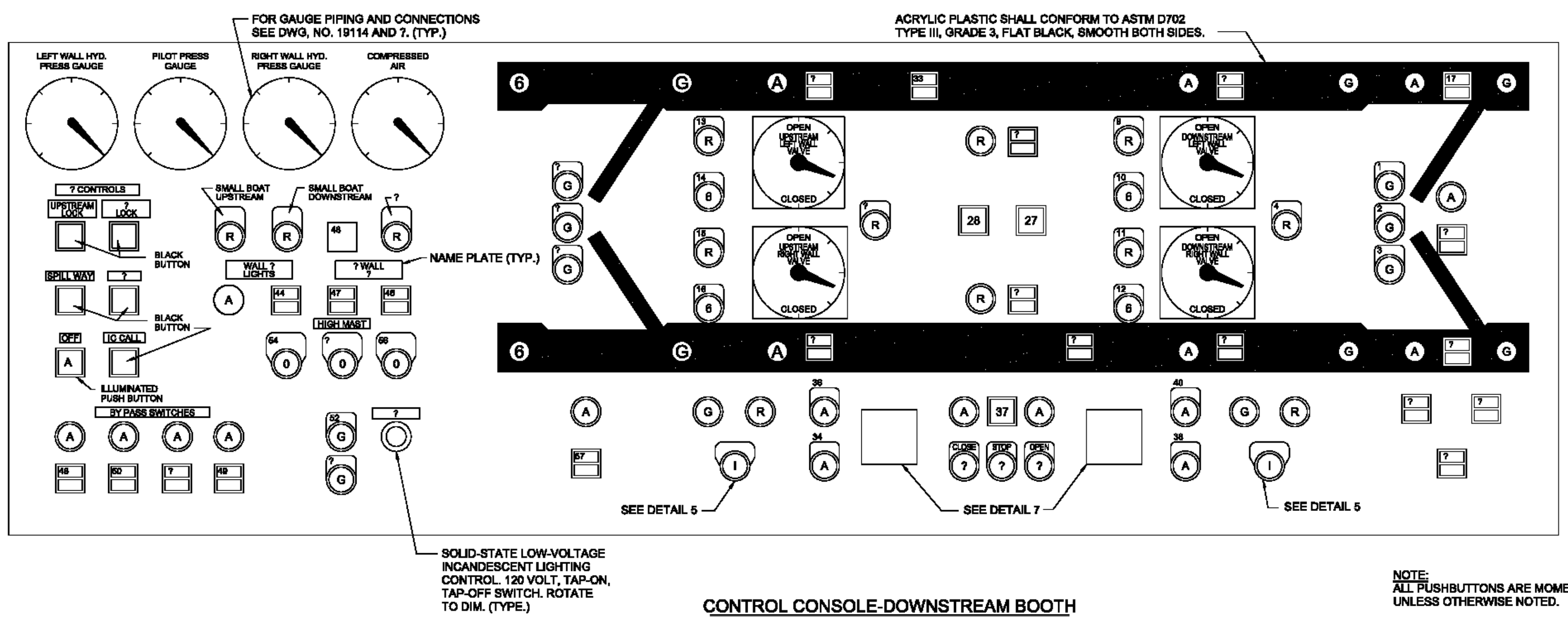


LOCK GATES AND
OPERATING EQUIPMENT
ELECTRICAL SYSTEMS
MITER GATE LATCH

PLATE B-91

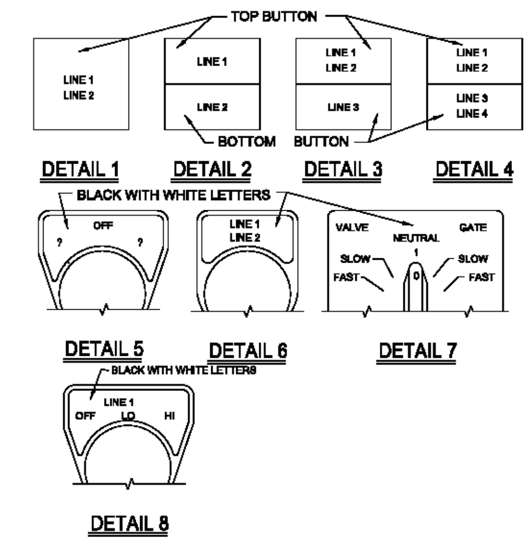


CONTROL CONSOLE-UPSTREAM BOOTH
SCALE: NONE



CONTROL CONSOLE-DOWNSTREAM BOOTH
SCALE: NONE

NOTE:
ALL PUSHBUTTONS ARE MOMENTARY UNLESS OTHERWISE NOTED.



COLOR AND ENGRAVING SCHEDULE							
ITEM NO.	DETAIL	ENGRAVING				BUTTON	COLOR
		LINE 1	LINE 2	LINE 3	LINE 4	TOP	BOTTOM
1	6	DNST. GATE	TOP MITER				
2	6	DNST. GATE	DNST. GATE				
3	6	DNST. GATE					
4	6	DNST.	DNST. GATE				
5	6	UPST. GATE	TOP MITER				
6	6	UPST. GATE	TOP MITER				
7	6	UPST. GATE	TOP MITER				
8	6	UPST.					
9, 11	6	DNST.	OPEN				
10, 12	6	DNST.	CLO.				
13, 15	6	UPST.	OPEN				
14, 16	6	UPST.	CLO.				
17, 18	3	DNST. GATE	SELECT	RESET		YELLOW	WHITE
19, 20	3	UPST. GATE	SELECT	RESET		YELLOW	WHITE
21, 22	3	DNST.	SELECT	RESET		RED	WHITE
23, 24	3	UPST.	SELECT	RESET		RED	WHITE
25, 26	3		START	STOP		RED	GREEN
27	1		SILENCE			BLACK	
28	1	AIR	HORN			ORANGE	
29, 30	3		ON	OFF		ORANGE	GREEN
31, 32	3	DNST. GATE	MITER	OUT		YELLOW	BLACK
33	4		LEFT WALL		? WALL	BLUE	BLACK
34	6	LEFT WALL	MANUAL				
35	2	AUTO	MANUAL			ORANGE	GREEN
36	6	LEFT WALL	AUTO				
37	1		RESET			ORANGE	
38	6	WALL	MANUAL				
39	2	AUTO	MANUAL			ORANGE	GREEN
40	6	WALL	AUTO				
41, 42	3	UPST. GATE	MITER	OUT		YELLOW	BLACK
43	4		LIFT WALL		RT. WALL	BLUE	BLACK
44	2	ON	OFF			ORANGE	GREEN
45	1		ACK			BLACK	
46	3		ON	OFF		ORANGE	GREEN
47	3		ON	OFF		ORANGE	GREEN
48	3		GATE	OFF		ORANGE	GREEN
49	3		GATE	OFF		ORANGE	GREEN
50	3		VALVE	OFF		ORANGE	GREEN
51	3		VALVE	OFF		ORANGE	GREEN
52	6	LEFT WALL	LUB ON				
53	6	RIGHT WALL	LUB ON				
54	8	UPSTREAM					
55	8	MIDDLE					
56	8	DOWNSTREAM					
57	3	SMALL	BOAT	OFF		WHITE	WHITE

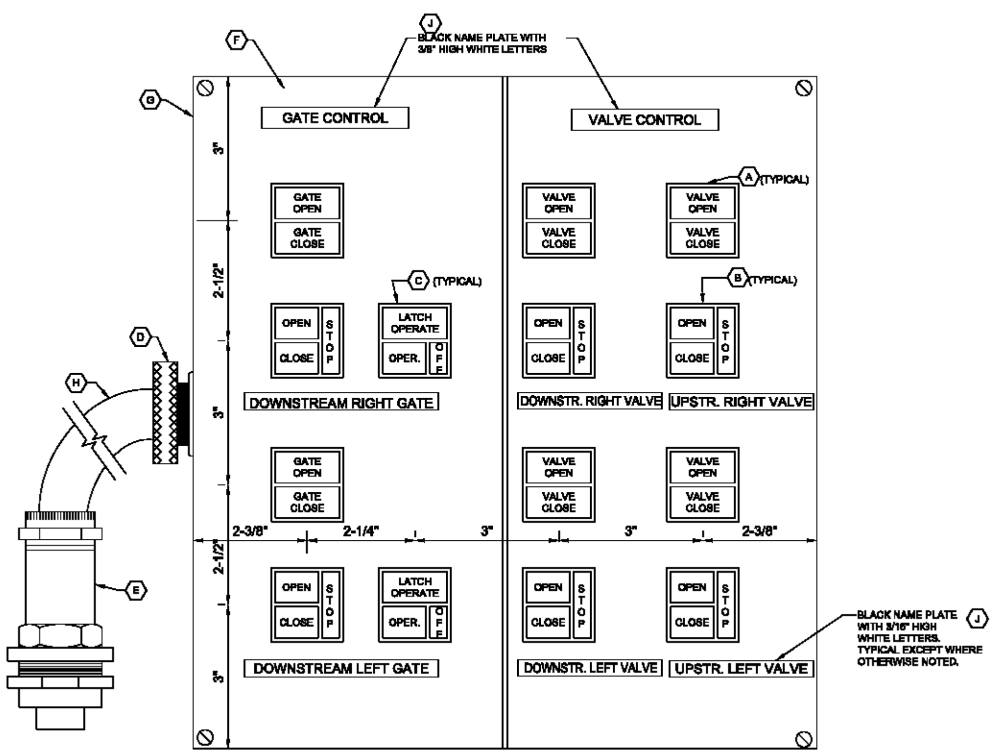
LOCK GATES AND OPERATING EQUIPMENT ELECTRICAL SYSTEMS CONTROL CONSOLE LAYOUT

MATERIAL LIST

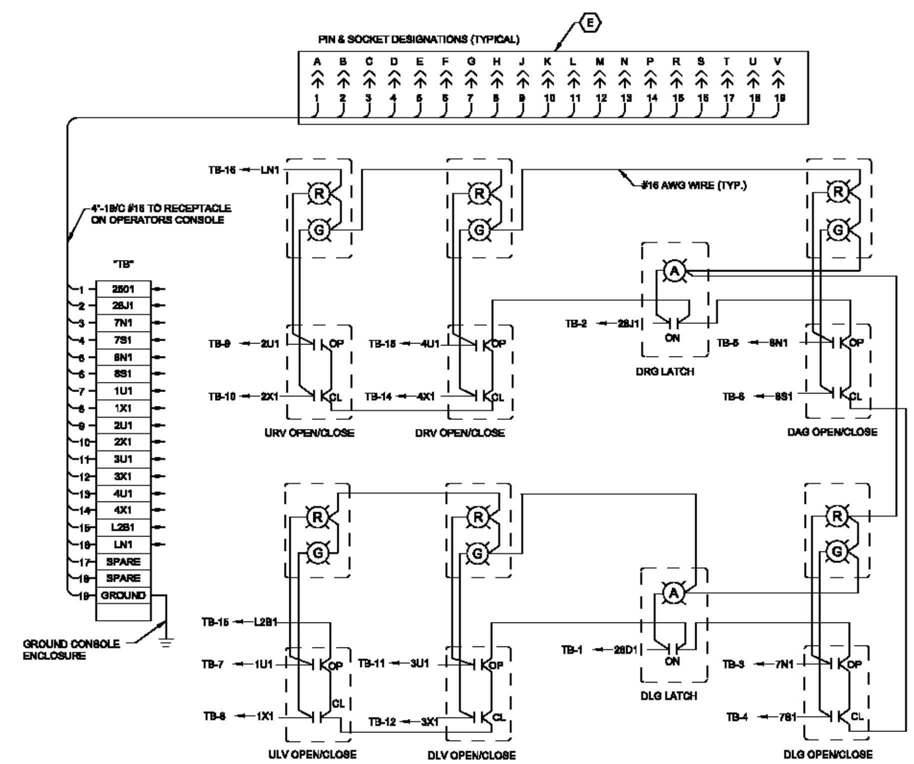
ITEM	DESCRIPTION	MANUFACTURER	CATALOG NO.	NO. REQ'D
A	DUAL INDICATING LIGHT UNIT TRANSFORMER TYPE RED LENS TO READ "VALVE OPEN" OR "GATE OPEN" AND GREEN LENS TO READ "VALVE CLOSE" OR "GATE CLOSE" AS SHOWN	CUTTER HAMMER	E3DCA INDICATOR E3DKGIO RED LENS E3DKG20 GREEN LENS	6 6 6
B	TWO-BUTTON OPERATOR WITH BLACK RELEASE BAR MAINTAINED CONTACT RED BUTTON MARKED "OPEN" AND GREEN BUTTON MARKED "CLOSE". 2 N.O. CONTACTS. OPEN AND CLOSED BUTTONS ARE MECHANICALLY INTERLOCKED. STOP RELEASES DEPRESSED BUTTON		E3QAG OPERATOR KC200 RED BUTTON KC300 GREEN BUTTON E3DKLA1 CONTACT BLOCKS E3DKTG FULL SHROUD	6 6 6 12 6
C	SINGLE-BUTTON OPERATOR WITH "OFF" RELEASE BAR AND TRANSFORMER TYPE INDICATING LIGHT. AMBER LENS TO READ "LIGHT OPERATE" YELLOW BUTTON "OPER" RED BUTTON "OFF"	CUTTER HAMMER	E300G OPERATOR E3DKG30 AMBER LENS E3DKC400 YELLOW BUTTON E3DKLAI CONTACT BLOCK (1 N.O.) E3DNTG FULL SHROUD	2 2 2 2 2
D	WATERTIGHT CABLE CONNECTION		CGB 296	1
E	CABLE PLUG WITH MATCHING SQUARE-FLANGED RECEPTACLE WITH WIRE-SEALED GROMMETS AND DUST CAP 19-WIRE, 19-POLE	ITT CANNON ELECTRIC	CV3456W22-14PN PLUG CV3450W22-14SN RECEPT	1 1
F	TERMINAL BLOCK IS 20 TERMINALS WITH MARKER STRIP IMPRINTED WITH 1/8" LETTERS AS SHOWN ON CONSOLE WIRING DIAGRAM MOUNT INSIDE CONSOLE INSULATED TERMINALS ON WIRES.	TRO CINCH CONNECTORS AMP	345-11-20-001, BLOCK 364-11-20-010, MARKER 320619 WIRE TERMINALS	1 1 34

MATERIAL LIST

ITEM	DESCRIPTION	MANUFACTURER	CATALOG NO.	NO. REQ'D
G	1/4"Hx13"Wx4-3/4"D ENCLOSURE W/BLANK HINGED COVER NEOPRENE GASKETED 14 GAUGE ALUMINUM 6061-T6 ALL SEAMS CONTINUOUSLY WELDED, RUBBER GROMMENTS ON METAL SURFACE. GROMMENTS TO BE SECURED TO ENCLOSURE WITH SCREWSAND NUTS.			1
H	19/C = 16 CABLE WITHxL POLYETHELYNE INSULATION OVER INDIVIDUAL CONDUCTORS WITH OVERALL NEOPRENE JACKET	GENERAL ELECTRIC	S1-58109	10 FT
J	ENGRAVED NAMEPLATE BAKELITE OR AN EQUIVALENT PLASTIC WITH 3/32 " INCH MINIMUM THICKNESS FASTENED TO CONSOLE WITH TWO CORROSION-RESISTANT ROUNDHEAD SCREWS.			8

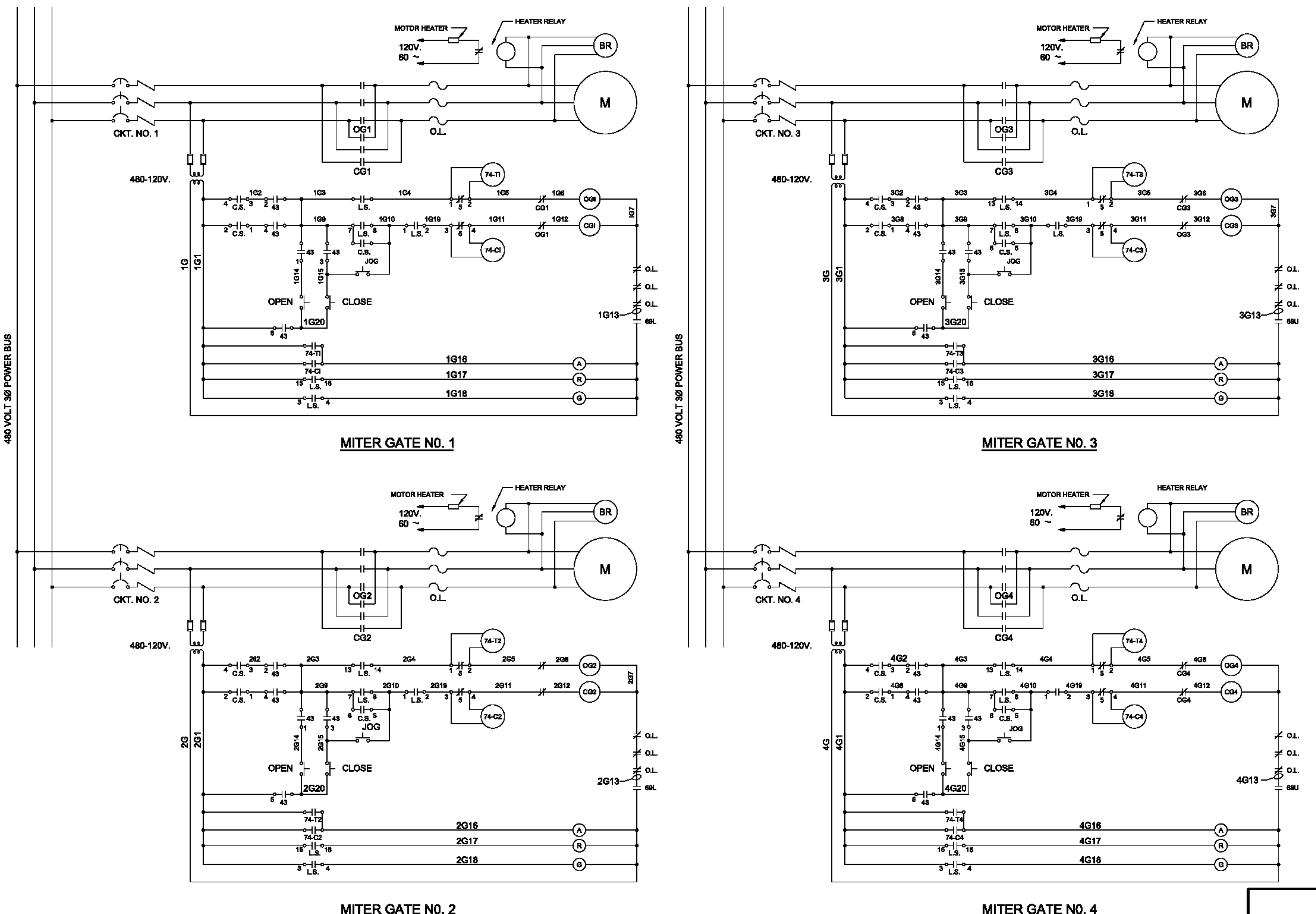


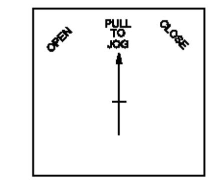
DOWNSTREAM PORTABLE BACKUP CONSOLE
SCALE: 3/4" = 1'-0"



PORTABLE BACKUP CONSOLE
WIRING DIAGRAM

LOCK GATES AND
OPERATING EQUIPMENT
ELECTRICAL SYSTEMS
BACKUP CONTROL
CONSOLE





ESCUTCHEON PLATE
SPRING RETURN TO OFF.

CONTACT		POSITION			
JOG-PULL OUT CLOSE ← ↑ → OPEN		CLOSE	OFF	OPEN	PULL
	1-2	X			X
	3-4			X	
	5-6				X

MITER GATE CONTROL SWITCH
DEVICE C.S.

ELEMENT	STUD NO.	DUTY	CLOSED — OPEN
A	1-2	EXTREME LIMIT CLOSED	
	3-4	INDICATING LAMP	
	5-6	MASTER INTER LOCK	
B	7-8	STOP	
	9-10	SPARE	
	11-12	SPARE	
C	13-14	TRAVEL LIMIT OPEN	
	15-16	INDICATING LAMP	
	17-18	SPARE	

MITER GATE LIMIT SWITCH
DEVICE L.S.
TRAVELING NUT TYPE

CONTACT		POSITION		
LEVER END		COMP.	NORMAL	TENSION
1—┤├—2	1-2	X	X	
3—┤├—4	3-4		X	X

MITER GATE
STRUT STRESS LIMIT SWITCH
DEVICE NO. 5

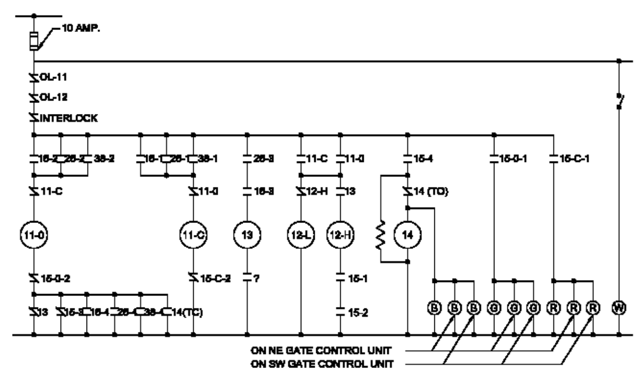
LEGEND

C.S.	MITER GATE CONTROL SWITCH
CGI	CLOSE CONTRACTOR
OGI	OPEN CONTRACTOR
O.L.	MOTOR OVERLOAD
5	STRUT STRESS LIMIT SWITCH
43	LOCAL-REMOTE PERMISSIVE SWITCH
69L	PERMISSIVE SWITCH LOWER BOOTH
69U	PERMISSIVE SWITCH UPPER BOOTH
74C1	ALARM RELAY-COMPRESSION
74T1	ALARM RELAY-TENSION

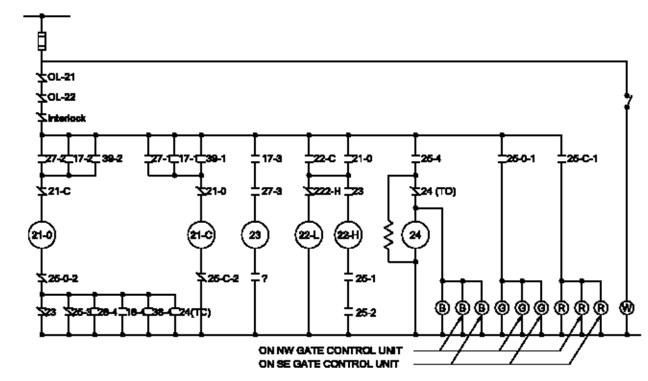
LOCK GATES
ELECTRICAL SYSTEMS
CONTROL SCHEMATIC
MITER GATES
SINGLE-SPEED MOTOR

CONTACT NO.	HANDLE IN NORMAL POSITION			HANDLE PULLED OUT			HANDLE PUSHED IN		
	CLOSE	NEUTRAL	OPEN	CLOSE	NEUTRAL	OPEN	CLOSE	NEUTRAL	OPEN
1	X			X			X		
2			X			X			X
3	X	X	X	X	X	X			
4				X	X	X			

CONTACT DIAGRAM FOR GATE CONTROL SWITCHES 16, 26, 36, 46, 17, 27 37, 47, 18, 38, 19, AND 39

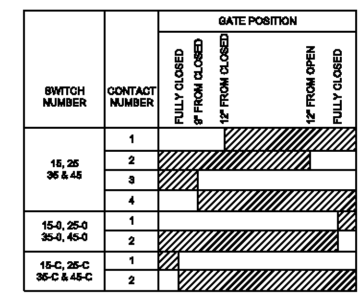


NORTHWEST GATE



NORTHEAST GATE

GATE CONTROL SCHEMATIC DIAGRAMS

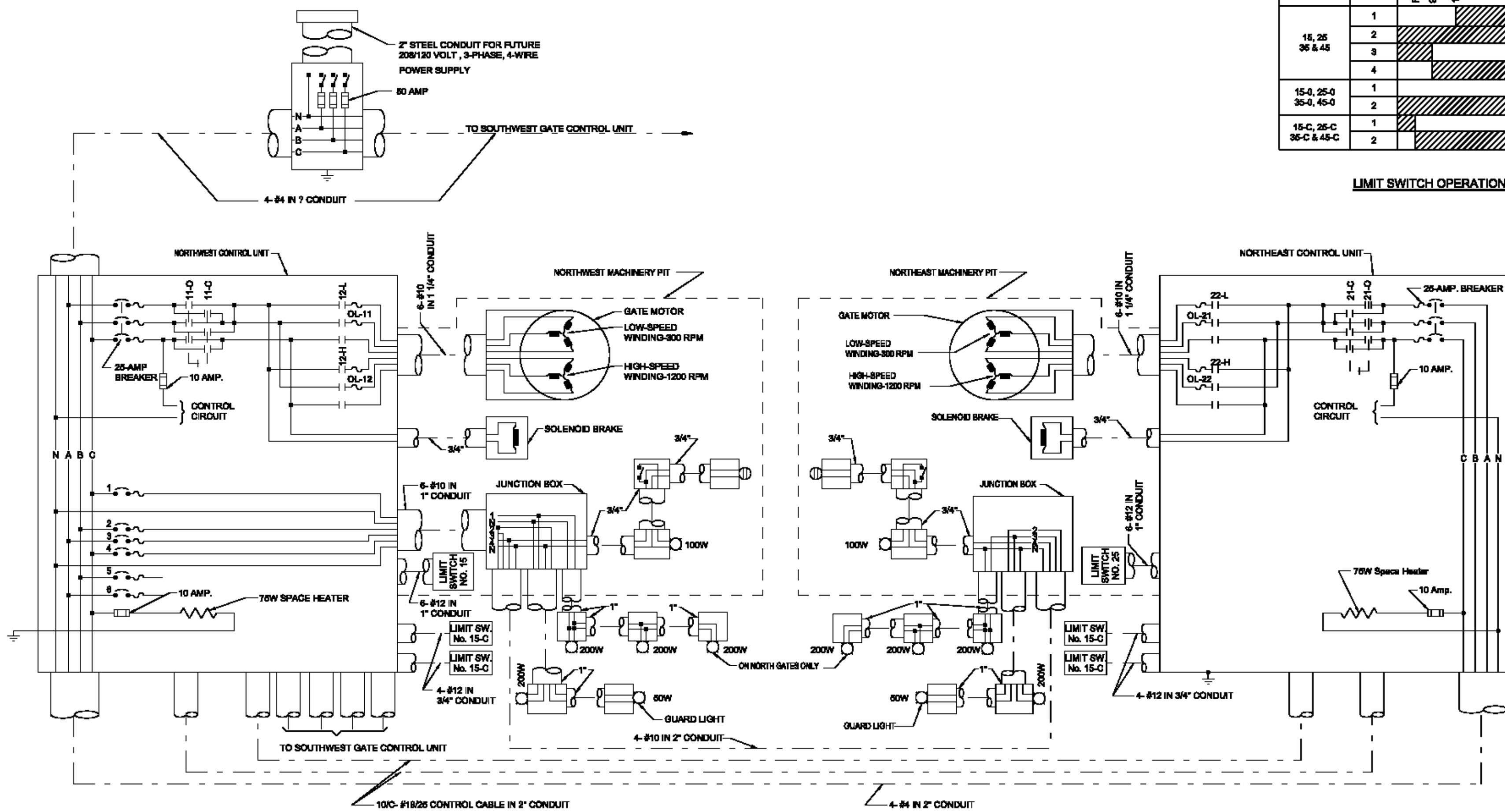


LIMIT SWITCH OPERATION

CONTROL UNIT EQUIPMENT DESIGNATION			
SYMBOL	QUAN	DESCRIPTION	
11-O	1	CONTACTOR-OPEN	NEMA SIZE/MAGNETIC REVERSING CONTACTOR MECHANICALLY INTERLOCKED
11-C	1	CONTACTOR-CLOSE	
21-O	1	CONTACTOR-OPEN	
21-C	1	CONTACTOR-CLOSE	
31-O	1	CONTACTOR-OPEN	
31-C	1	CONTACTOR-CLOSE	
41-O	1	CONTACTOR-OPEN	
41-C	1	CONTACTOR-CLOSE	
12-L	1	CONTACTOR-LOW SPEED	NEMA SIZE/ MAGNETIC CONTRACTOR
22-L	1		
32-L	1		
42-L	1		
12-H	1	CONTACTOR-HIGH SPEED	
22-H	1		
32-H	1		
42-H	1		
13	1	CONTROL RELAY	
23	1		
33	1		
43	1		
14	1	TIME DELAY RELAY	
24	1		
34	1		
44	1		
15	1	RELAY LIMIT SWITCH	
25	1		
35	1		
45	1		
15-O	1	LIMIT SWITCH-OPEN	
25-O	1		
35-O	1		
45-O	1		
15-C	1	LIMIT SWITCH-CLOSE	
25-C	1		
35-C	1		
45-C	1		
16	1	CONTROL SWITCH	
26	1		
36	1		
46	1		
17	1	CONTROL SWITCH	
27	1		
37	1		
47	1		
18	1	CONTROL SWITCH	
38	1		
19	1	CONTROL SWITCH	
39	1		

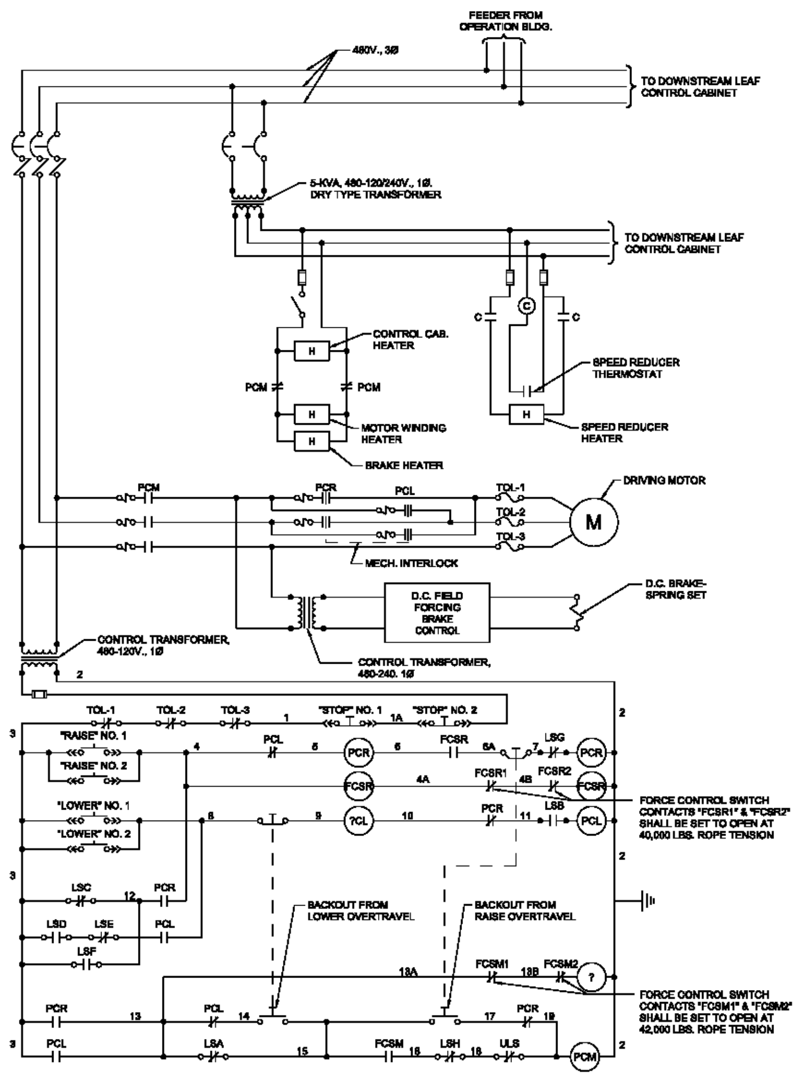
NOTES:

DESIGNATION NUMBERS FROM 10-20 REFER TO NORTHWEST GATE.
DESIGNATION NUMBERS FROM 20-30 REFER TO NORTHEAST GATE.
DESIGNATION NUMBERS FROM 30-40 REFER TO SOUTHEAST GATE.
DESIGNATION NUMBERS FROM 40-50 REFER TO SOUTHEAST GATE.
LIGHTING CIRCUIT BREAKERS ARE 15-AMP, SINGLE-POLE.
WIRING FOR SOUTH GATES IS IDANTICAL TO NORTH GATES WIRING, EXCEPT AS OTHERWISE NOTED.
ALL WIRE IS NO. 12 A.W.G. UNLESS OTHERWISE NOTED.

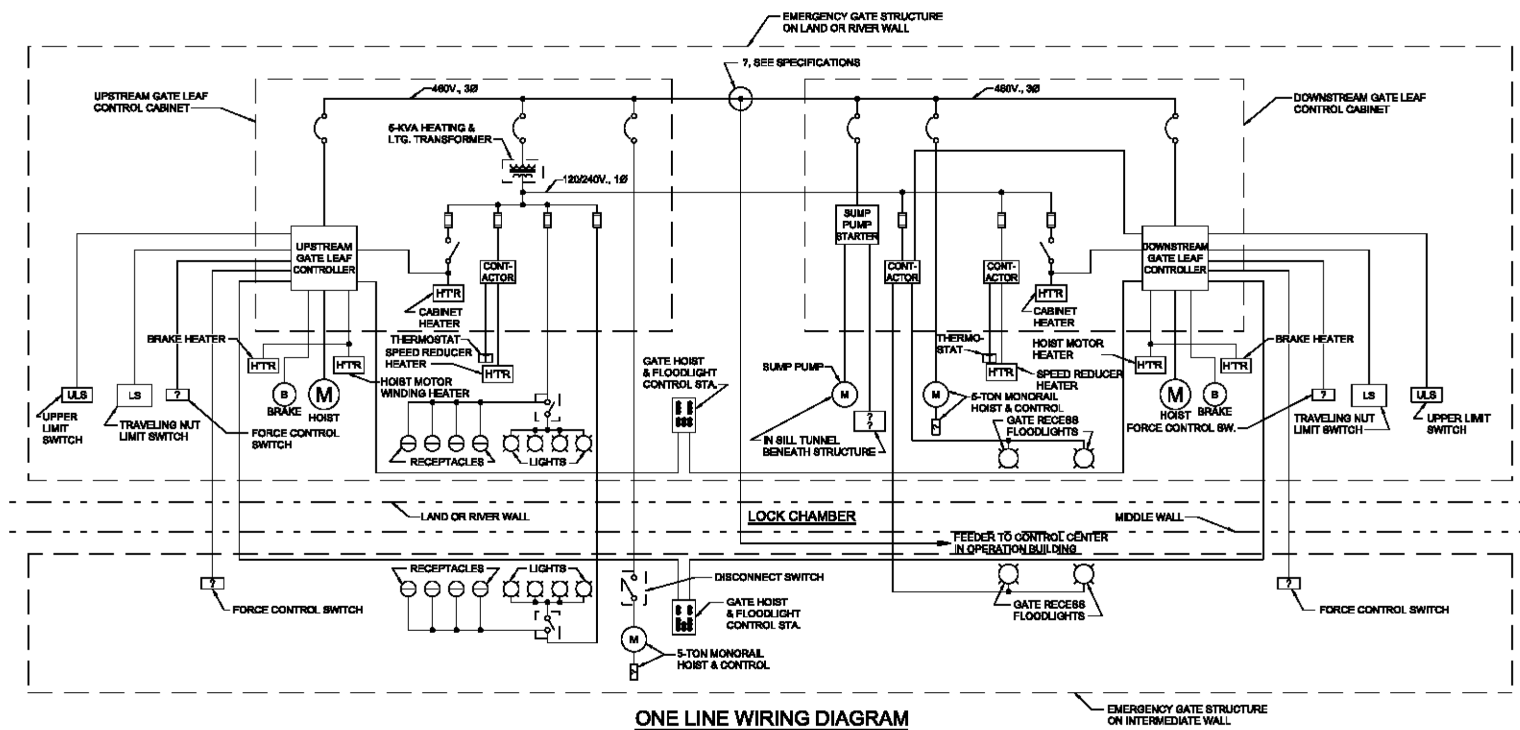


NORTH GATES
POWER AND LIGHT WIRING DIAGRAM

LOCK GATES
ELECTRICAL SYSTEMS
CONTROL SCHEMATIC
MITER GATES
TWO-SPEED MOTOR



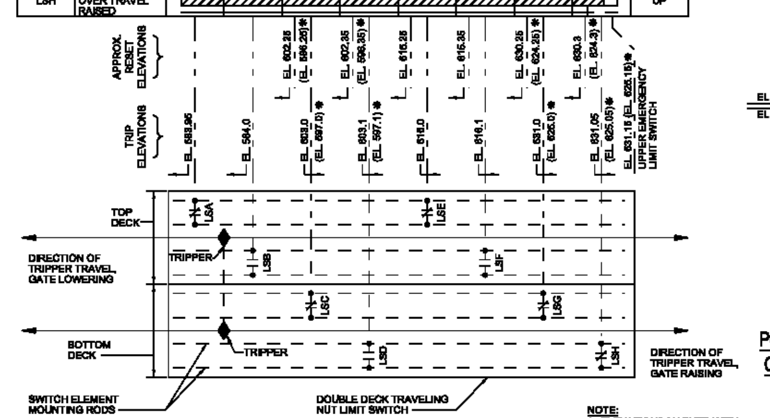
BASIC WIRING DIAGRAM
ONE GATE LEAF



ONE LINE WIRING DIAGRAM
ONE EMERGENCY GATE

SEQUENCE OF OPERATION

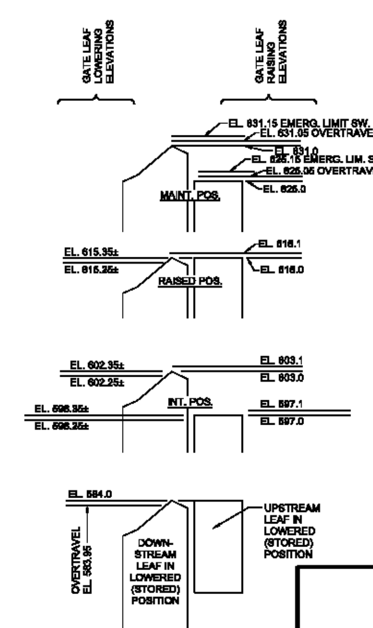
CONTACT DESIG-NATION	CIRCUIT DUTY	GATE POSITION: O'TRAVEL LOWERED	LOWERED	INT. POS. (STOP-UP) (START-DN)	INT. POS. (START-UP) (STOP-DN)	RAISED ? (STOP-UP) (START-DN)	RAISED POS. (START-UP) (STOP-DN)	MAINT. POS. STOP-UP	O'TRAVEL RAISED	DIRECTION OF TRAVEL TO ? SWITCH
LSA	STOP MOTOR, OVER TRAVEL LOWERED									DOWN
LSB	STOP MOTOR, GATE LOWERED									DOWN
LSC	STOP MOTOR (UP) START MOTOR (DN)									UP
LSD	START MOTOR (UP) STOP MOTOR (DN)									UP
LSE	STOP MOTOR (UP) START MOTOR (DN)									UP
LSF	START MOTOR (UP) STOP MOTOR (DN)									UP
LSG	STOP MOTOR (UP)									UP
LSH	STOP MOTOR, OVER TRAVEL RAISED									UP



ARRANGEMENT OF ELEMENTS

LIMIT SWITCHES "LS"

NOTE:
* ELEVATIONS MARKED WITH ASTERISKS REPRESENT UPSTREAM LEAF ELEVATIONS IN THE "INTER-MEDIATE" MAINTENANCE" AND "OVERTRAVEL RAISED" GATE POSITIONS.



POSITION OF GATE LEAVES FOR CONTROL CIRCUIT OPERATION

EACH LEAF OPERATES INDEPENDENTLY

ELECTRICAL SYMBOLS

	P.B., MOM. CONTACT, N.O.
	P.B., MOM. CONTACT, N.C.
	MAINT. CONTACT, S.P.S.T.
OR	LIMIT OR FLOAT SW. CONTACT, N.O.
OR	DITTO ,N.C.
	N.O. CONTACT
	N.C. CONTACT
	CONTACTOR OR RELAY COIL
	AIR CKT. BREAKER WITH THERMAL MAGNETIC TRIP
	BLOWOUT COIL
1 2 3 4	TERMINAL BLOCKS
	MECHANICAL INTERLOCK
	FUSE
	MOLDED CONNECTORS (MALE & FEMALE) NO. OF POLES AS REQUIRED
	SPARE CONDUCTOR, END TAPED

DESCRIPTION OF CONTROL SYSTEM

1. SCHEME OF OPERATION

A. GENERAL THE SCHEME OF OPERATION OF THE CONTROL SYSTEM FOR EACH LEAF OF THE EMERGENCY GATE SHALL BE AS INDICATED BY THE BASIC WIRING DIAGRAM AND AS SPECIFIED BELOW.

B. CONTROL POINTS. THE OPERATION OF THE GATE LEAF HOIST MOTOR SHALL BE CONTROLLED BY:

(1) A PUSH BUTTON TYPE CONTROL STATION ON EACH LOCK WALL, NEAR THE EMERGENCY GATE LEAF, WITH MOMENTARY CONTACTS DESIGNATED "RAISE," "LOWER," AND "STOP".

(2) A PUSH BUTTON TYPE CONTROL STATION, LOCATED ON THE DOOR OF THE CONTROL CABINET. WITH MOMENTARY CONTACTS DESIGNATED "BACKOUT RAISE OVERTRAVEL" AND "BACKOUT LOWER OVERTRAVEL".

C. OPERATING FEATURES.

(1) MOMENTARY ACTUATION OF THE "RAISE" AND "LOWER" CONTACTS OF EITHER OF THE CONTROL STATIONS SHALL PROVIDE SEAL-IN OPERATION OF THE CONTROL CIRCUIT.

(2) LIMIT SWITCHES WITH CONTACTS ACTUATED BY THE MOVEMENT OF THE OPERATING ELEMENTS OF THE GATE HOIST MACHINE, SHALL AUTOMATICALLY CONTROL, AS INDICATED, THE SEQUENCE AND THE STOPPING OF THE OPERATIONS INITIATED MANUALLY AT A CONTROL STATION.

(3) MOMENTARY ACTUATION OF THE "STOP" CONTACT OF EITHER OF THE CONTROL STATIONS DURING ANY OPERATION SHALL DE-ENERGIZE THE CONTROL CIRCUIT, STOP THE MOTOR AND SET THE BRAKE.

2. SEQUENCE OF OPERATION

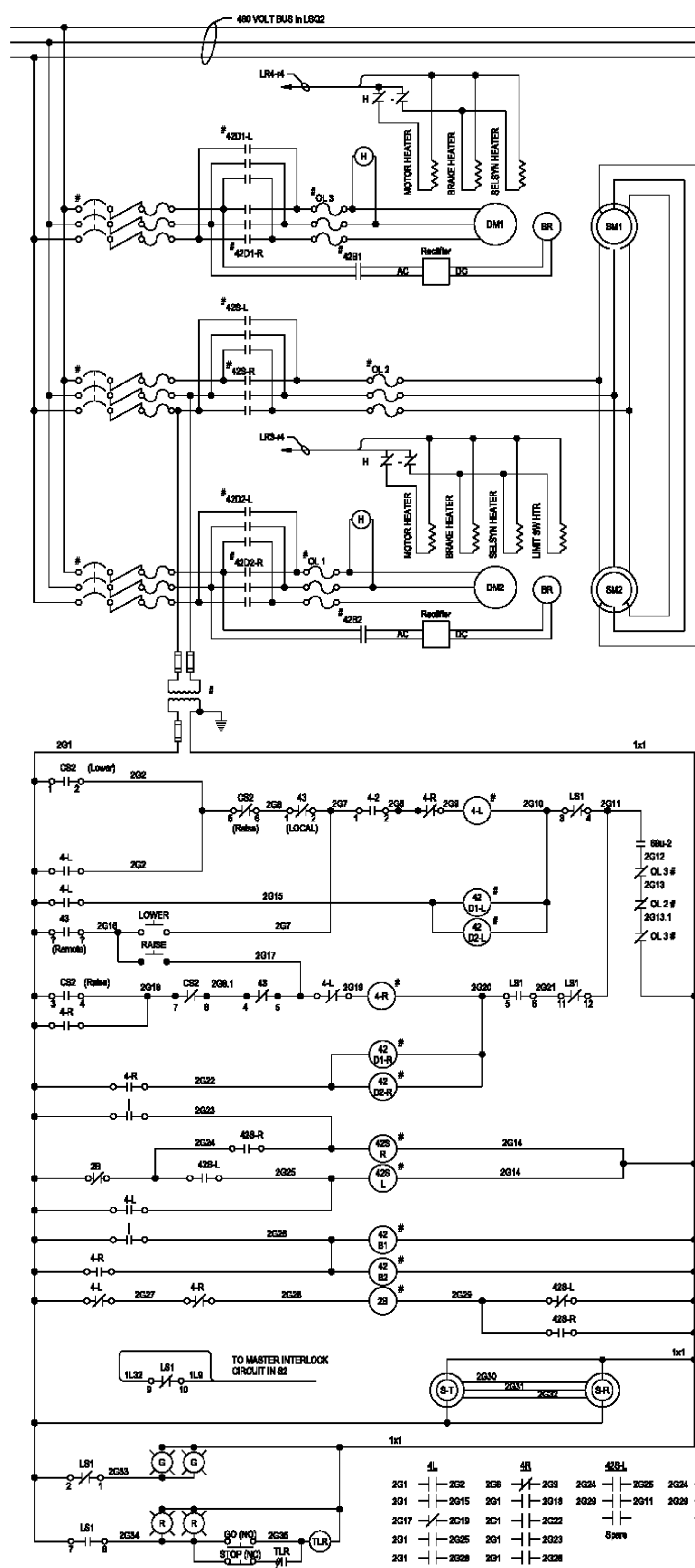
A. ? WHEN THE "RAISE" CONTACT OF EITHER CONTROL STATION IS ACTUATED, HELD MOMENTARILY, AND RELEASED, THE HOIST MOTOR SHALL START AND RUN UNTIL THE GATE HAS RAISED APPROXIMATELY 20FT TO THE INTERMEDIATE POSITION, WHERE THE LIMIT SWITCH SHALL AUTOMATICALLY DE-ENERGIZE THE CONTROL CIRCUIT, STOP THE MOTOR, AND SET THE BRAKE. IF THE GATE IS AT THE INTERMEDIATE POSITION AND THE "RAISE" CONTACT IS PUSHED, HELD UNTIL THE GATE HAS RAISED AT LEAST 0.1 FT, AND RELEASED, THE GATE MOTOR SHALL CONTINUE TO RUN UNTIL THE GATE REACHES THE RAISED POSITION WHERE THE LIMIT SWITCH SHALL AUTOMATICALLY DE-ENERGZE THE CONTROL CIRCUIT, STOP THE MOTOR, AND SET THE BRAKE. BY THE SAME MEANS THE GATE MAY BE MOVED TO THE MAINTENANCE POSITION.

B. LOWERING. WHEN THE "LOWER" CONTACT OF EITHER CONTROL STATION IS ACTUATED, HELD, AND RELEASED, THE HOIST MOTOR SHALL START AND RUN UNTIL THE GATE REACHES THE RAISED POSITION WHERE THE LIMIT SWITCH SHALL AUTOMATICALLY DE-ENERGIZE THE CONTROL CIRCUIT, STOP THE MOTOR, AND SET THE BRAKE. FROM THE RAISED POSITION, DEPRESSING THE " LOWER" CONTACT UNTIL THE GATE HAS LOWERED AT LEAST 0.1 FT SHALL ALLOW THE GATE TO LOWER TO THE INTERMEDIATE POSITION WHERE THE LIMIT SWITCH SHALL AUTOMATICALLY DE-ENERGIZE THE CONTROL CIRCUIT, STOP THE MOTOR, AND SET THE BRAKE. BY THE SAME MEANS THE GATE MAY BE MOVED TO THE LOWERED POSITION.

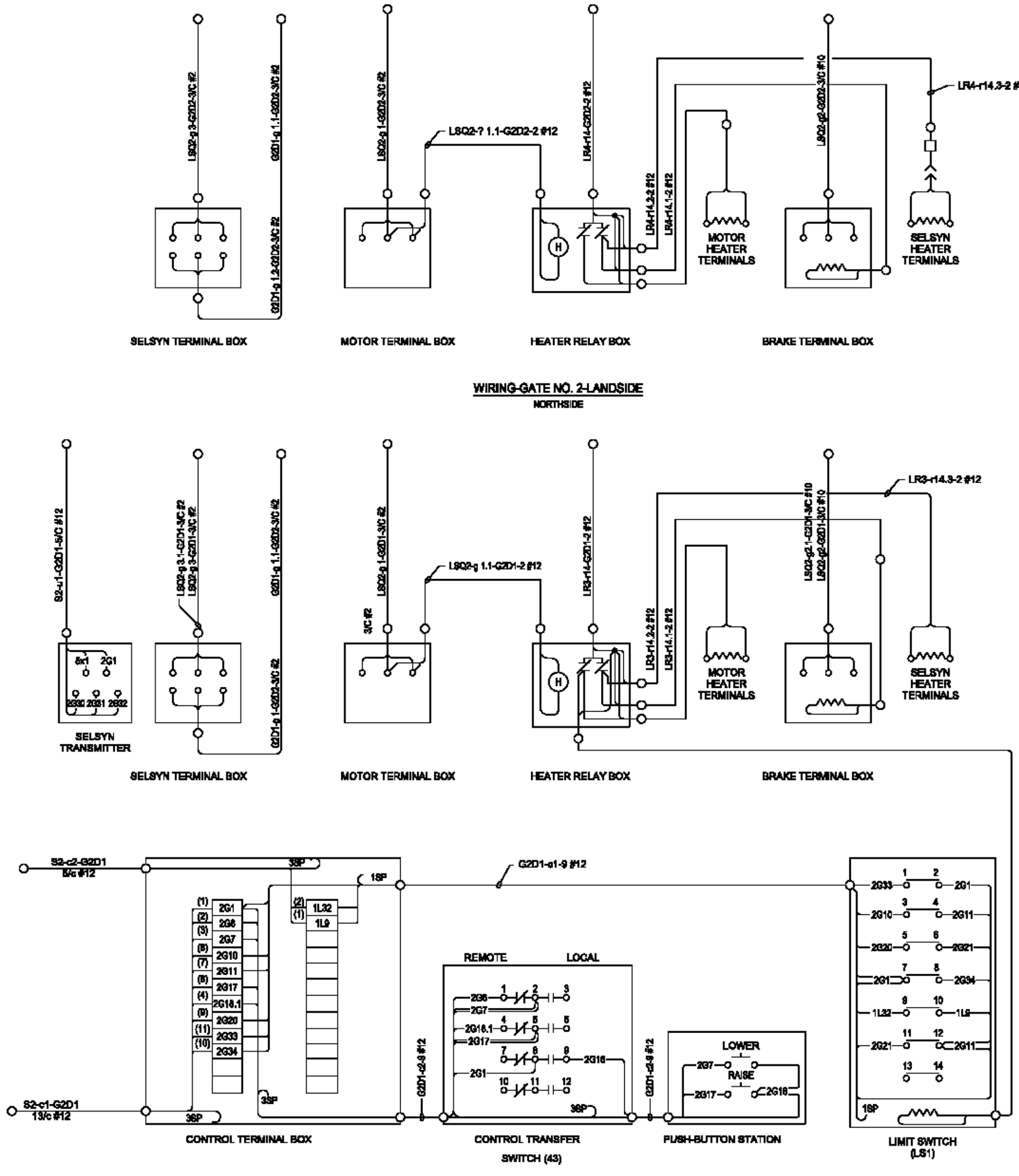
C. BACKOUT FROM OVERTRAVEL. IF, DURING NORMAL OPERATION, THE GATE SHOULD FAIL TO STOP AFTER REACHING THE MAINTENANCE OR LOWERED POSITION, THE OVERTRAVEL LIMIT SWITCH CONTACTS SHALL STOP THE HOIST MOTOR AFTER OVERTRAVEL BY DE-ENERGIZING THE "MAIN" OR "LINE" CONTACTOR OF THE CONTROLLER. MOREOVER, IF THE GATE SHOULD ? TO STOP UPON REACHING THE OVERTRAVEL RAISED POSITION, THE UPPER EMERGENCY LIMIT SWITCH CONTACTS SHALL STOP THE HOIST MOTOR BY DE-ENERGIZING THE "MAIN" OR "LINE" CONTACTOR TO OPERATE THE GATE AFTER AN OVERTRAVEL, THE APPROPRIATE BACKOUT SWITCH CONTACT SHALL BE HELD CLOSED TO PERMIT OPERATION OF THE GATE ONLY IN THE DIRECTION AWAY FROM THE OVERTRAVEL POSITION BY MEANS OF EITHER THE "RAISE" OR "LOWER" PUSHBUTTON ON EITHER CONTROL STATION.

D. BACKOUT FROM ROPE OVERLOAD IF, DURING NORMAL RAISING OPERATION, EXCESSIVE ROPE IS REACHED, FORCE CONTROL SWITCH CONTACTS SHALL STOP THE HOIST MOTOR BY DE-ENERGIZING THE "RAISE" CONTACTOR. IF THE "RAISE" CONTACTOR SHOULD FAIL TO OPEN, ROPE TENSION WILL INCREASE FURTHER UNTIL THE OTHER FORCE CONTROL SWITCH CONTACTS SHALL STOP THE HOIST MOTOR BY DE-ENERGIZING THE "MAIN" CONTACTOR. TO LOWER THE GATE AFTER ROPE OVERLOAD THE "BACKOUT RAISE OVERTRAVEL" PUSHBUTTON SHALL BE HELD CLOSED TO PERMIT LOWERING OF THE GATE BY MEANS OF THE "LOWER" PUSHBUTTON ON EITHER CONTROL STATION.

LOCK GATES
VERTICAL-LIFT GATE
ELECTRICAL SYSTEMS
CONTROL SCHEMATIC
EMERGENCY TYPE



POWER AND CONTROL SYSTEM



WIRING-GATE NO. 2-LANDSIDE
NORTHSIDE

WIRING-GATE NO. 2-RIVERSIDE
SOUTHSIDE

DESCRIPTION OF CONTROL SYSTEM

1. SCHEME OF OPERATION-THE SCHEME OF OPERATION SHALL BE AS INDICATED BY THE SCHEMATIC CONTROL DIAGRAM AND AS DESCRIBED BELOW.

2 CONTROL POINTS-THE OPERATION OF THE GATE SHALL BE CONTROLLED BY A CONTROL SWITCH CS2 OF THE SPRING RETURN CONTACT TYPE ON CONTROL STAND NO. 2. THE SWITCH SHALL HAVE TWO POSITIONS DESIGNATED "RAISE" AND "LOWER". THE GATE SHALL ALSO BE CONTROLLED FROM THE SOUTH GATE MACHINERY ROOM BY MEANS OF A TWO-POSITION PUSH-BUTTON STATION WITH MOMENTARY CONTACTS DESIGNATED "RAISE" AND "LOWER". THE CONTROL SHALL BE TRANSFERRED FROM ONE POINT TO THE OTHER BY MEANS OF TRANSFER SWITCH 43, ALSO IN THE SOUTH MACHINERY ROOM.

3. CONTROL SEQUENCE IN LOWERING GATE-OVERLOAD AND INTERLOCK RELAYS ARE NORMALLY CLOSED. THE CONTROL SWITCH CS2 IS TURNED MOMENTARILY TO THE "LOWER" POSITION. CLOSURE OF THE CONTROL SWITCH CS2 CONTACTS ENERGIZES CONTROL RELAY 4-L. RELAY 4-L CONTACTS CLOSE ? ENERGIZE, (1) CONTACTOR 42S-L FOR THE POWER SELSYN MOTORS (2) THE DRIVE MOTOR BREAK CONTACTORS RELEASING THE BRAKES AND (?) THE DRIVE MOTORS CONTROLLERS 42D1-L AND 42D2-L.

4. CONTROL SEQUENCE IN STOPPING GATE-GATE MOVEMENT IS HALTED EITHER BY PULLING THE CONTROL SWITCH CS2 WHICH OPENS THE CIRCUIT OR BY THE LIMIT SWITCH LS1 WHICH OPENS AT A PREDETERMINED LIMIT OF TRAVEL AND DE-ENERGIZES THE CONTROL CIRCUIT STOPPING THE DRIVE MOTORS AND SETTING THE BRAKER. WHEN 4L IS DE-ENRGIZED, TIME DELAY RELAY ? IS ENERGIZED AND THROUGH IT TIME DELAY OPENING CONTACTS MAINTAINS THE POWER SUPPLY TO THE SELSYN MOTORS FOR A SHORT TIME. ? CONTACTS THEN OPEN AND DE-ENERGIZE CONTACTOR 42S-L.

5. CONTROL SEQUENCE IN RAISING GATE-THE SEQUENCE IS SIMILAR TO THAT FOLLOWED IN LOWERING THE GATE EXCEPT THAT "RAISE" DEVICES OPERATE INSTEAD OF "LOWER" DEVICES.

6. CONTROL SEQUENCE IN STOPPING GATE-THE SEQUENCE IS SIMILAR TO STOPPING THE GATE DESCRIBED ABOVE EXCEPT THAT "RAISE" DEVICES" OPERATE INSTEAD OF "LOWER" DEVICES.

7. THE CONTROLS ARE SHOWN FOR REMOTE OPERATION AND GATE RAISED.

DEVICE DESIGNATIONS

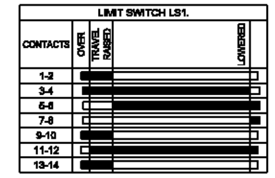
4-L	CONTROL RELAY-LOWER
4-R	CONTROL RELAY-RAISE
42D1-L, 42D2-L	DRIVE MOTOR CONTROLLER-LOWER
42D1-R, 42D2-R	DRIVE MOTOR CONTROLLER-RAISE
42S-L	POWER SELSYN LINE CONTACTORS-LOWER
42S-L	POWER SELSYN LINE CONTACTORS-RAISE
?	CONTROL TRANSFER SWITCH
?	TIME-DELAY OPENING RELAY
BR	BRAKE
CS-3	CONTROL SWITCH-STAND NO. 2
DM	DRIVE MOTOR
G	GREEN LIGHT
H	HEATER RELAY
?	INTERLOCK RELAY
LSQ2	CONTROL CENTER
?	LIMIT SWITCH
?	OVERLOAD DEVICE
G	RED LIGHT
?	POWER SELSYN MOTOR
TB	TERMINAL BOX
ST	SELSYN TRANSMITTER
SR	SELSYN RECEIVER
?	MAKE CONTACTOR
?	SELECTOR SWITCH, CONTROL ? (SEE SHT. 114)
4-2	INTERLOCK RELAY
?	MAKE CONTACTOR
TLR	TRAFFIC LIGHT RELAY

#Located in LSQ2

CONTROL TRANSFER SWITCH 43

CONTRACT	LOCAL	REMOTE
1-2		X
2-3	X	
4-5		X
5-6	X	
8-9	X	

LIMIT SWITCH LS1.



CONTROL SWITCH OPERATION
CS2-GATE 2
(SPRING RETURN TO NORMAL)

CONTACT	LOWER	NORMAL	RAISE	PULL TO STOP
1-2	X			
3-4			X	
5-6	X	X	X	
7-8	X	X	X	

Lateral contacts no spring return to normal.

LOCK GATES
TAINTER TYPE
CONTROL DIAGRAM

APPENDIX C

Sample Calculations – Index

Miter Gates, Panama Canal Linkage (Sheets 1‑13)

Miter Gate Torque Curves

Miter Gates, Direct Connected Cylinder (Sheets 1‑10)

Vertical Lift Gates, Determination of Loads for Upstream Leaf (Sheets 1‑9)

Vertical Lift Gates, Determination of Loads for Downstream Leaf (Sheets 1‐19)

Vertical Lift Gates, Design of Wheels (Sheets 1‑8)

APPENDIX C

Sample Calculations – Cover Sheet

The following sample problems are provided to show methodology used to determine gate hoist operating loads, and component size and strength. The examples show one approach, other approaches and supporting references may also be appropriate.

1. Tainter gate - Electric Motor Wire Rope Hoist. This example provides a design analysis for a wire rope type tainter gate hoist. It shows the relationship among the various load conditions that the hoist may experience during gate operation, both normal and extreme. Sizing calculations are provided for the wire rope, gear boxes, shafting, and open gearing.

1A. Tainter Gate Loading Unbalanced Load Criteria. This analysis presents different loading criteria for tainter gate hoists. Tainter gate machinery that includes a single motor and drum hoists on each end of the tainter gate is analyzed. An analysis is provided for loads that are split evenly (50%/50%), split 70%/30%, and for loading of 100% on one drum hoist unit.

2. Sector Gate Machinery Design. Two separate calculations are provided. One example provides typical calculations for determining a sector gate's closing pintle torque for a reverse head. Reverse head torque is obtained from WES model study presented in WES Report H-70-2. Closing torque is composed of hydrodynamic forces acting on the nose of the gate, and hinge and pintle friction that is composed of friction from gate's weight and hydrostatic head.

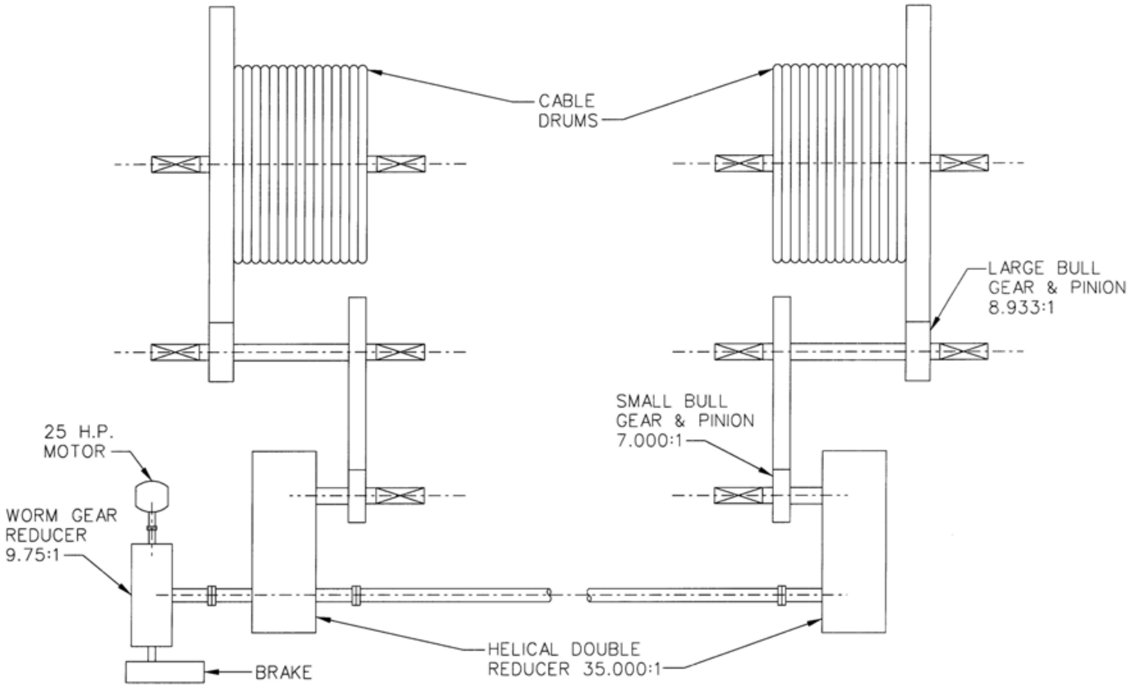
2a. The second example is from a 56 foot sector gate project in New Orleans. Both the direct head and reverse head calculations are provided. Calculations are done using actual hydrostatic head against the gate rather than using WES Report H-70-2.

3. Round Link Chain - Grooved Drum and Pocket Wheel. This example provides a detailed design analysis for a round link chain grooved drum and pocket wheel tainter gate hoist.

1. Tainter Gate - Electric Motor Wire Rope Hoist. The following is an example of a wire rope hoist design for a tainter gate. It is not intended to be a comprehensive analysis of all components, but an example sizing some of the major components for the normal and stalled motor load cases.

List of Selected Variables

CP_{BG} = Large Bull Gear Circular Pitch, p. 8
CP_{BGP} = Large Bull Gear Pinion Circular Pitch, p. 8
DP_{SG} = Small Bull Gear Diametral Pitch, p. 12
DP_{SGP} = Small Bull Gear Pinion Diametral Pitch, p. 12
DR = Drum Radius, p. 8
E_{BG} = Bull Gear & Pinion Mesh Efficiency, p. 6
E_{DB} = Drumshaft Bearing Efficiency, p. 6
E_{IB} = Intermediate Shaft Roller Bearings Efficiency, p. 6
E_{RD} = Rope to Drum Efficiency, p. 6
E_{WGR} = Worm Gear Reducer Efficiency, p. 6
F_{maxa} = Max. Available Rope Pull for 1/2 Gate (At Rated Motor Load), p.4
F_{maxn} = Max. Req'd Rope Pull for 1/2 Gate (Normal Load), p. 4
HDR = Helical Double Reducer Ratio, p. 6
ML_{BG} = Max. Large Bull Gear Tooth Load (Normal Gate Load), p. 8
ML_{SG} = Max. Small Bull Gear Tooth Load (Normal Gate Load), p. 12
M_p = Motor Horsepower, p. 3
NWR = Number of Wire Ropes per Side, p. 4
PD_{BG} = Large Bull Gear Pitch Diameter, p. 8
PD_{BGP} = Bull Gear Pinion Pitch Diameter, p. 8
PD_{SG} = Small Bull Gear Pitch Diameter, p. 12
PD_{SGP} = Small Bull Gear Pinion Pitch Diameter, p. 12
RPS = Stalled Load Rope Pull, Force Divided Equally, p. 4
RPS_{70} = Stalled Load Rope Pull, Forces Split 70/30, p. 5
RPS_{100} = Stalled Load Rope Pull, Forces Split 100/0, p. 5
SBG = Small Bull Gear & Pinion Ratio, p. 5
SL_{BG} = Large Bull Gear Stalled Tooth Load, Forces Divided Equally, p. 8
SL_{BG70} = Large Bull Gear Stalled Tooth Load, Forces Split 70/30, p. 8
SL_{BG100} = Large Bull Gear Stalled Tooth Load, Forces Split 100/0, p. 9
SL_{SG} = Small Bull Gear Stalled Tooth Load, Forces Divided Equally, p. 12
SL_{SG70} = Small Bull Gear Stalled Tooth Load, Forces Split 70/30, p. 12
SL_{SG100} = Small Bull Gear Stalled Tooth Load, Forces Split 100/0, p. 13
S_m = Motor Speed, p. 3
W_{BG} = Large Bull Gear Weight, p. 16
W_{BGS} = Large Bull Gear Shaft Weight, p. 16
W_{cable} = Cable Weight, p. 16
W_{Drum} = Drum Weight, p. 16
WGR = Worm Gear Ratio, p. 6
WR = Wire Rope Breaking Strength, p. 4



Machinery Arrangement

Forces :=

Forces for 1/2 Gate, or the evenly divided normal working forces for each side of the gate.

Gate Opening (ft)	Req'd Rope Pull (kips)	Avail. Rope Pull, Rated hp, 9% Slip (kips)	Avail. Rope Pull, 115% Rated hp, 2.5% Slip (kips)
0	359.9	349.8	375.4
2	357.5	345.5	370.7
4	354.7	341.4	366.4
6	351.7	337.5	362.2
8	348.5	333.8	358.2
10	345.1	330.3	354.5
12	341.5	326.9	350.9
16	338.5	320.5	344
20	335.1	314.6	337.7
25	308	307.8	330.4
30	302.7	301.5	323.6
35	296.4	295.6	317.2
40	288	290	311.3
45	275.2	283.9	304.7
50	257.7	277.5	297.9
55	236.1	271.3	291.2
60	212.6	265.2	284.6
65	189.6	259.2	278.1
70	177.4	253.7	272.3
72	188.8	251.9	270.4
73	208.1	251.2	269.5
73.5	226.1	250.9	269.3
74	255.4	250.7	269

Design Loads - Criteria to determine tainter gate machinery loads can be found in EM 1110-2-2105. Req'd Rope Pull is the rope pull required to hoist lift the gate through its full range. It includes forces due to the weight of the gates, trunnion friction, seal friction, and hydraulic forces. The Available Rope Pull is the rope pull produced by the rated torque of the motor at an assumed value of maximum slip of 9%. The motor was selected to provide at least the required rope pull through the full range using the 1.15 service factor of the motor and assuming 2.5% slip.

Motor Horsepower

$M_p := 25\text{hp}$

Locked Rotor Torque is Considered to be 280% of Rated Torque

Motor Speed, 9% Slip

$$S_m := 1638 \cdot \frac{1}{min}$$

Max. Req'd Rope Pull for 1/2 Gate (Normal Load)

$$F_{maxn} := Forces_{0,1} \cdot 10^3 \cdot lbf$$

$$F_{maxn} = 3.599 \times 10^5 \cdot lbf$$

Max. Available Rope Pull for 1/2 Gate (At Rated Motor Load)

$$F_{maxa} := Forces_{0,2} \cdot 10^3 \cdot lbf$$

$$F_{maxa} = 3.498 \times 10^5 \cdot lbf$$

Wire Rope

Assume 1 3/8" dia. 6x30 IWRC Flattened Strand, Regular Lay, Corr. Resistant Steel Rope. Breaking Strength = 165 kips. Wire rope selection should be optimized by an analysis of commercial availability and hoist dimensional constraints.

Wire Rope Breaking Strength

$$WR := 165 \cdot 10^3 lbf$$

Number of Wire Ropes on One Side, Number Selected to Equal at Least Required Rope with a Factor of Safety of 5

$$NWR := 11$$

Wire Rope Factor of Safety, Max. Working Load

$$FS_5 := \frac{WR \cdot NWR}{F_{maxn}}$$

$$FS_5 = 5.043$$

Rope Pull at Motor Stall - 280% of rated torque applied equally to both sides of the gate

$$RPS := 2.8 \cdot F_{maxa}$$

$$RPS = 9.794 \times 10^5 \cdot lbf$$

Percent of Rope Breaking Strength, Stalled Load

$$PBS := \frac{RPS}{WR \cdot NWR}$$

$$PBS = 53.964 \cdot \%$$

Rope Pull at Motor Stall - 280% of rated torque split 70%/30% between each side of the gate

$$RPS_{70} := .7 \cdot 2 \cdot 2.8 \cdot F_{maxa}$$

$$RPS_{70} = 1.371 \times 10^{6} \cdot lbf$$

Percent of Rope Breaking Strength, Stalled Load, 70/30 Split

$$PBS_{70} := \frac{RPS_{70}}{WR \cdot NWR}$$

$$PBS_{70} = 75.549 \cdot \%$$

Number of Wire Ropes Required to Stay Below 70% of Wire Rope Breaking Strength, See Chapter 2

$$NWR_{70} := \frac{RPS_{70}}{WR \cdot .7}$$

$NWR_{70} = 11.872$ Use 12 Wire Ropes on Each Side

Rope Pull at Motor Stall - 280% of rated torque applied 100%/0% to both sides of the gate. The unstalled side is still supporting half of the gate weight, so F_{maxn} is subtracted from the stalled side.

$$RPS_{100} := 2 \cdot 2.8 \cdot F_{maxa} - F_{maxn}$$

$$RPS_{100} = 1.599 \times 10^{6} \cdot lbf$$

Percent of Rope Breaking Strength, Stalled Load, 100/0 Split, 12 Ropes

$$PBS_{100} := \frac{RPS_{100}}{WR \cdot 12}$$

$$PBS_{100} = 80.757 \cdot \%$$

Total Reduction Ratio

Large Bull Gear and Pinion Ratio

$$LBG := \frac{134}{15}$$

$$LBG = 8.933$$

Small Bull Gear and Pinion Ratio

$$SBG := \frac{126}{18}$$

$$SBG = 7$$

Helical Double Reducer Ratio

$$HDR := 35$$

Worm Gear Reducer Ratio

$$WGR := 9.75$$

Total Reduction Ratio

$$TR := LBG \cdot SBG \cdot HDR \cdot WGR$$

$$TR = 2.134 \times 10^{4}$$

Overall Efficiency of Machinery (Assumed)

Rope to Drum Efficiency

$$E_{RD} := .98$$

Drumshaft Brg, Spherical Self-Align., Roller, Efficiency

$$E_{DB} := .97$$

Bull Gear & Pinion Mesh Efficiency

$$E_{BG} := .98$$

Int. Shaft Brgs., Sperical Self-Align., Roller, Efficiency

$$E_{IB} := .97$$

Intermediate Gear and Pinion Mesh Efficiency

$$E_{IG} := .98$$

Outboard Brg., Spherical Self Align., Roller, Efficiency

$$E_{OB} := .98$$

Helical Double Reduer, 35:1 Ratio

$$E_{HR} := .94$$

Worm Gear Reducer, 9.75:1, Efficiency

$$E_{WGR} := .92$$

Total

$$E_{tot} := E_{RD} \cdot E_{DB} \cdot E_{BG} \cdot E_{IB} \cdot E_{IG} \cdot E_{OB} \cdot E_{HR} \cdot E_{WGR}$$

$$E_{tot} = 0.751$$

Worm Gear Reducer

Input Speed

$$S_m = 1.638 \times 10^{3} \cdot \frac{1}{min}$$

Req'd Input Rating

$$R_{WG} := M_p \cdot \frac{F_{maxn}}{F_{maxa}}$$

$$R_{WG} = 25.722 \cdot hp$$

Overload Capacity Required (280%), Sees full torque (before split)

$$R_{WGO} := 2.8 \cdot M_p$$

$$R_{WGO} = 70 \cdot hp$$

Parallel Shaft Reducer

Input Speed

$$S_{PS} := \frac{S_m}{WGR}$$

$$S_{PS} = 168 \cdot \frac{1}{min}$$

Req'd Input Rating

$$R_{PS} := \frac{M_p}{2} \cdot E_{WGR} \cdot \frac{F_{maxn}}{F_{maxa}}$$

$$R_{PS} = 11.832 \cdot hp$$

Overload Capacity Required - 280% of Motor Torque applied equally to both sides of the gate

$$R_{PSO} := \frac{M_p}{2} \cdot E_{WGR} \cdot 2.8$$

$$R_{PSO} = 32.2 \cdot hp$$

Overload Capacity Required - Rope Pull at Motor Stall - 280% of rated torque split 70%/30% between each side of the gate

$$R_{PSO70} := .7 \cdot 2.8 \cdot M_p \cdot E_{WGR}$$

$$R_{PSO70} = 45.08 \cdot hp$$

Brake

Required Torque

$$T_B := 1.5 \cdot \frac{M_p \cdot 5250 \cdot \frac{ft \cdot lbf}{min \cdot hp}}{S_m}$$

$$T_B = 120.192 \cdot ft \cdot lbf$$

Large Bull Gear and Pinion

Drum Radius

$DR := 1.8374ft$

Ratio

$LBG = 8.933$

Gear

Pitch Diameter

$PD_{BG} := 170.62in$

Circular Pitch

$CP_{BG} := 4in$

Number of Teeth

$NT_{BG} := 134$

Pinion

Pitch Diameter

$PD_{BGP} := 19.1in$

Circular Pitch

$CP_{BGP} := 4in$

Number of Teeth

$NT_{BGP} := 15$

Maximum Tooth Load (Normal Load)

$$ML_{BG} := F_{maxn} \cdot \frac{DR}{\frac{PD_{BG}}{2}} \cdot \frac{1}{E_{RD} \cdot E_{DB}}$$

$$ML_{BG} = 9.785 \times 10^{4} \cdot lbf$$

Stalled Tooth Load - Forces Equally Divided

$$SL_{BG} := ML_{BG} \cdot \frac{F_{maxa}}{F_{maxn}} \cdot 2.8$$

$$SL_{BG} = 2.663 \times 10^{5} \cdot lbf$$

Stalled Tooth Load - Forces Divided 70/30

$$SL_{BG70} := .7 \cdot 2 \cdot ML_{BG} \cdot \frac{F_{maxa}}{F_{maxn}} \cdot 2.8$$

$$SL_{BG70} = 3.728 \times 10^{5} \cdot lbf$$

Stalled Tooth Load - Forces Divided 100/0

$$SL_{BG100} := 2 \cdot 2.8 \cdot ML_{BG} \cdot \frac{F_{maxa}}{F_{maxn}} - ML_{BG} \cdot \frac{F_{maxa}}{F_{maxn}}$$

$$SL_{BG100} = 4.375 \times 10^{5} \cdot lbf$$

Pinion Material

ASTM A 291, Cl. 2

$$Sy_p := 70000 \frac{lbf}{in^2}$$

$$Sut_p := 95000 \frac{lbf}{in^2}$$

Pinion Velocity

$$V_p := \frac{S_m}{SBG \cdot HDR \cdot WGR} \cdot PD_{BGP} \cdot \pi$$

$$V_p = 3.429 \cdot \frac{ft}{min}$$

Allowable Stress (Normal Load)

$$S_{allp} := \frac{Sut_p}{5}$$

$$S_{allp} = 1.9 \times 10^{4} \cdot \frac{lbf}{in^2}$$

Allowable Stress (Stalled Load)

$$S_{maxp} := 0.75 \cdot Sy_p$$

$$S_{maxp} = 5.25 \times 10^{4} \cdot \frac{lbf}{in^2}$$

Required Pinion Face Width - Normal Load

$y_p := .111$ For 15 teeth

$$FW_p := \frac{ML_{BG}}{S_{allp} \cdot CP_{BGP} \cdot y_p} \cdot \frac{600 + V_p \cdot \frac{min}{ft}}{600}$$

$$FW_p = 11.666 \cdot in$$

Required Pinion Face Width - Stalled Load, Equally Divided

$$FW_{pmax} := \frac{SL_{BG}}{S_{maxp} \cdot CP_{BGP} \cdot y_p} \cdot \frac{600 + V_p \cdot \frac{min}{ft}}{600}$$

$$FW_{pmax} = 11.489 \cdot in$$

Required Pinion Face Width - Stalled Load, 70/30 Split

$$FW_{pmax70} := \frac{SL_{BG70}}{S_{maxp} \cdot CP_{BGP} \cdot y_p} \cdot \frac{600 + V_p \cdot \frac{min}{ft}}{600}$$

$$FW_{pmax70} = 16.085 \cdot in$$

Bull Gear Material

QQ-S-681d, Cl. 80-50

$$S_{ybg} := 50000 \frac{lbf}{in^2}$$

$$S_{ubg} := 80000 \frac{lbf}{in^2}$$

Allowable Stress - Normal Load, Bull Gear

$$S_{allbg} := \frac{S_{ubg}}{5}$$

$$S_{allbg} = 1.6 \times 10^4 \cdot \frac{lbf}{in^2}$$

Allowable Stress - Stalled Load, Bull Gear

$$S_{maxbg} := 0.75 \cdot S_{ybg}$$

$$S_{maxbg} = 3.75 \times 10^4 \cdot \frac{lbf}{in^2}$$

Required Bull Gear Face Width - Normal Load

$y_{BG} := .163$ For 134 teeth

$$FW_{BG} := \frac{ML_{BG}}{S_{allbg} \cdot CP_{BG} \cdot y_{BG}}$$

$$FW_{BG} = 9.38 \cdot in$$

Required Bull Gear Face Width - Stalled Load, Equally Divided

$$FW_{maxBG} := \frac{SL_{BG}}{S_{maxbg} \cdot CP_{BG} \cdot y_{BG}}$$

$$FW_{maxBG} = 10.891 \cdot in$$

Required Bull Gear Face Width - Stalled Load, 70/30 Split

$$FW_{maxBG70} := \frac{SL_{BG70}}{S_{maxbg} \cdot CP_{BG} \cdot y_{BG}}$$

$$FW_{maxBG70} = 15.248 \cdot in$$

Selected Bull Gear Pinion and Gear Width for Load Divided 70/30 (Greater of Required Pinion and Gear Widths, Rounded Up)

$$FW_{BGs} := 16.25in$$

Bull Gear Pinion Stress with load split 100/0 using face width required for 70/30 load split

$$S_{100p} := \frac{SL_{BG100}}{FW_{BGs} \cdot CP_{BGP} \cdot y_p} \cdot \frac{600 + V_p \cdot \frac{min}{ft}}{600}$$

$$S_{100p} = 6.098 \times 10^4 \cdot \frac{lbf}{in^2}$$

Percent Yield Strength of Bull Gear Pinion, 100/0 split, selected face width

$$P_{s100p} := \frac{S_{100p}}{Sy_p}$$

$$P_{s100p} = 87.117 \cdot \%$$

Bull Gear Stress with load split 100/0 using face width required for 70/30 load split

$$S_{100bg} := \frac{SL_{BG100}}{FW_{BGs} \cdot CP_{BG} \cdot y_{BG}}$$

$$S_{100bg} = 4.129 \times 10^4 \cdot \frac{lbf}{in^2}$$

Percent Yield Strength of Bull Gear, 100/0 split, selected face width

$$P_{s100bg} := \frac{S_{100bg}}{Sy_p}$$

$$P_{s100bg} = 58.988 \cdot \%$$

Small Bull Gear and Pinion

Ratio

$LSG := 7$

Gear

Pitch Diameter

$PD_{SG} := 84in$

Diametral Pitch

$DP_{SG} := 1.5 \cdot \frac{1}{in}$

Number of Teeth

$NT_{SG} := 126$

Pinion

Pitch Diameter

$PD_{SGP} := 12in$

Diametral Pitch

$DP_{SGP} := 1.5 \cdot \frac{1}{in}$

Number of Teeth

$NT_{SGP} := 18$

Maximum Tooth Load (Normal Load)

$$ML_{SG} := F_{maxn} \cdot \frac{DR}{\frac{PD_{BG}}{2}} \cdot \frac{PD_{BGP}}{PD_{SG}} \cdot \frac{1}{E_{RD} \cdot E_{DB} \cdot E_{BG} \cdot E_{IB}}$$

$ML_{SG} = 2.341 \times 10^4 \cdot lbf$

Stalled Tooth Load, Load Equally Divided

$$SL_{SG} := ML_{SG} \cdot \frac{F_{maxa}}{F_{maxn}} \cdot 2.8$$

$SL_{SG} = 6.37 \times 10^4 \cdot lbf$

Stalled Tooth Load, 70/30 Split

$$SL_{SG70} := 2 \cdot .7 ML_{SG} \cdot \frac{F_{maxa}}{F_{maxn}} \cdot 2.8$$

$SL_{SG70} = 8.918 \times 10^4 \cdot lbf$

Stalled Tooth Load, 100/0 Split

$$SL_{SG100} := 2 \cdot 2.8 ML_{SG} \cdot \frac{F_{maxa}}{F_{maxn}} - ML_{SG} \cdot \frac{F_{maxa}}{F_{maxn}}$$

$$SL_{SG100} = 1.046 \times 10^{5} \cdot lbf$$

Pinion Material

ASTM A 291, Cl. 1

$$Sy_{sp} := 50000 \frac{lbf}{in^2}$$

$$Sut_{sp} := 85000 \frac{lbf}{in^2}$$

Pinion Velocity

$$V_{sp} := \frac{S_m}{HDR \cdot WGR} \cdot PD_{SGP} \cdot \pi$$

$$V_{sp} = 15.08 \cdot \frac{ft}{min}$$

Allowable Stress (Normal Load)

$$S_{allsp} := \frac{Sut_{sp}}{5}$$

$$S_{allsp} = 1.7 \times 10^{4} \cdot \frac{lbf}{in^2}$$

Allowable Stress (Stalled Load)

$$S_{maxsp} := 0.75 \cdot Sy_{sp}$$

$$S_{maxsp} = 3.75 \times 10^{4} \cdot \frac{lbf}{in^2}$$

Required Pinion Face Width - Normal Load

$Y_{sp} := .378$ For 18 teeth

$$FW_{sp} := \frac{ML_{SG} \cdot DP_{SGP}}{S_{allsp} \cdot Y_{sp}}$$

$$FW_{sp} = 5.464 \cdot in$$

Required Pinion Face Width - Stalled Load, Equally Divided

$$FW_{spmax} := \frac{SL_{SG} \cdot DP_{SGP}}{S_{maxsp} \cdot Y_{sp}}$$

$$FW_{spmax} = 6.74 \cdot in$$

Required Pinion Face Width - Stalled Load, 70/30 Split

$$FW_{spmax70} := \frac{SL_{SG70} \cdot DP_{SGP}}{S_{maxsp} \cdot Y_{sp}}$$

$$FW_{spmax70} = 9.437 \cdot in$$

Small Bull Gear Material

QQ-S-681d, Cl. 80-40

$$S_{ysg} := 40000 \frac{lbf}{in^2}$$

$$S_{usg} := 80000 \frac{lbf}{in^2}$$

Allowable Stress - Normal Load, Bull Gear

$$S_{allsg} := \frac{S_{ubg}}{5}$$

$$S_{allsg} = 1.6 \times 10^4 \cdot \frac{lbf}{in^2}$$

Allowable Stress - Stalled Load, Bull Gear

$$S_{maxsg} := 0.75 \cdot S_{ysg}$$

$$S_{maxsg} = 3 \times 10^4 \cdot \frac{lbf}{in^2}$$

Required Bull Gear Face Width - Normal Load

$Y_{sbg} := .513$ For 126 teeth

$$FW_{sbg} := \frac{ML_{SG} \cdot DP_{SG}}{S_{allsg} \cdot Y_{sbg}}$$

$$FW_{sbg} = 4.277 \cdot in$$

Required Bull Gear Face Width - Stalled Load, Equally Divided

$$FW_{maxSBG} := \frac{SL_{SG} \cdot DP_{SG}}{S_{maxsg} \cdot Y_{sbg}}$$

$$FW_{maxSBG} = 6.208 \cdot in$$

Required Bull Gear Face Width - Stalled Load, 70/30 Split

$$FW_{maxSBG70} := \frac{SL_{SG70} \cdot DP_{SG}}{S_{maxsg} \cdot Y_{sbg}}$$

$$FW_{maxSBG70} = 8.692 \cdot in$$

Selected Small Bull Gear Pinion and Gear Width for Load Divided 70/30 (Greater of Required Pinion and Gear Widths, Rounded Up)

$$FW_{SBGs} := 9.5in$$

Small Bull Gear Pinion Stress with load split 100/0 using face width required for 70/30 load split

$$S_{100sp} := \frac{SL_{SG100} \cdot DP_{SGP}}{FW_{SBGs} \cdot Y_{sp}}$$

$$S_{100sp} = 4.371 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Percent Yield Strength of Bull Gear Pinion, 100/0 split, selected face width

$$P_{100sp} := \frac{S_{100sp}}{Sy_{sp}}$$

$$P_{100sp} = 87.423 \cdot \%$$

Small Bull Gear Pinion Stress with load split 100/0 using face width required for 70/30 load split

$$S_{100sbg} := \frac{SL_{SG100} \cdot DP_{SG}}{FW_{SBGs} \cdot Y_{sbg}}$$

$$S_{100sbg} = 3.221 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Percent Yield Strength of Bull Gear Pinion, 100/0 split, selected face width

$$P_{100sbg} := \frac{S_{100sbg}}{S_{ysg}}$$

$$P_{100sbg} = 80.521 \cdot \%$$

Large Bull Gear Shaft

Maximum Load Occurs When the Gate is at the Closed Position

Basic Radius of Drum

$$DR = 1.837 \cdot ft$$

Pitch Radius of Large Bull Gear

$$PR_{BG} := \frac{PD_{BG}}{2}$$

$$PR_{BG} = 7.109 \cdot ft$$

Normal Load - Bearing Reactions

Drum Torque

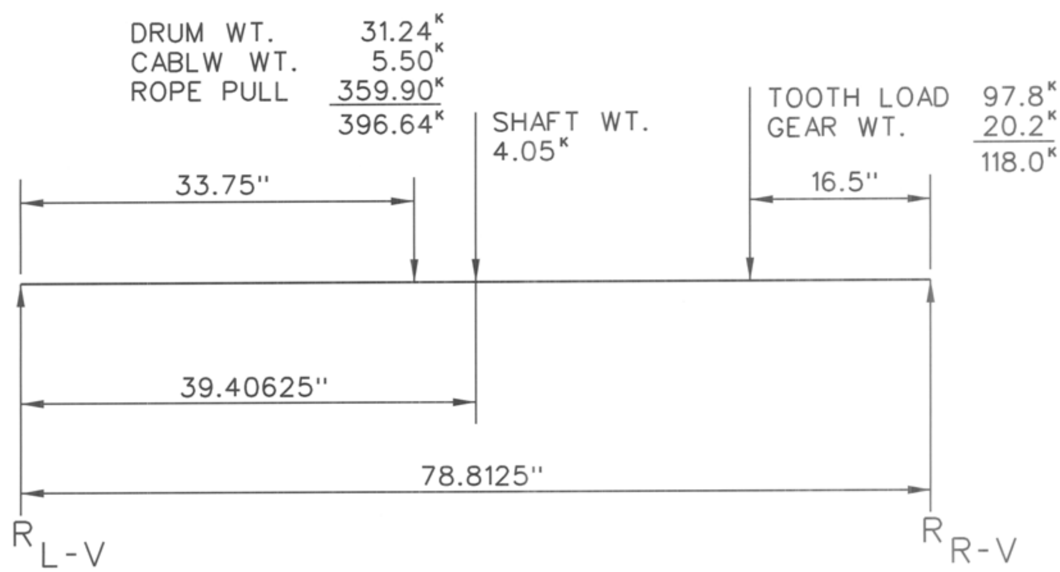
$$DT := F_{maxn} \cdot DR$$

$$DT = 6.613 \times 10^{5} \cdot ft \cdot lbf$$

Large Bull Gear Tangential Tooth Load (Calculated Above)

$$ML_{BG} = 9.785 \times 10^{4} \cdot lbf$$

Vertical Loading, Max. Normal Load



Drum Weight

$$W_{Drum} := 31.24 \cdot 10^{3} \cdot lbf$$

Cable Weight

$$W_{cable} := 5.5 \cdot 10^{3} \cdot lbf$$

Large Bull Gear Weight

$$W_{BG} := 20.2 \cdot 10^{3} \cdot lbf$$

Large Bull Gear Shaft Weight

$$W_{BGS} := 4.05 \cdot 10^{3} \cdot lbf$$

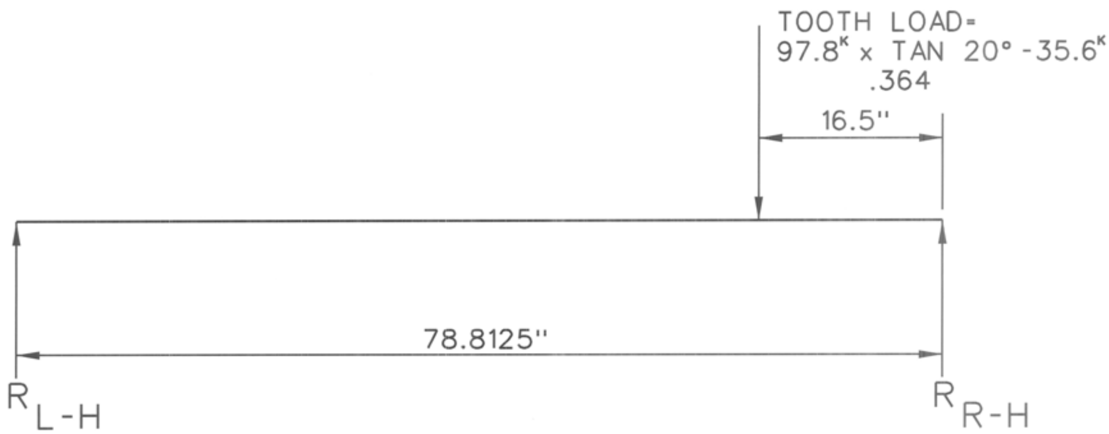
$$R_{RV} := \frac{(F_{maxn} + W_{cable} + W_{Drum})\cdot 33.75in + W_{BGS}\cdot 39.41in + (ML_{BG} + W_{BG})\cdot 62.31i}{78.8125in}$$

$$R_{RV} = 2.652 \times 10^{5}\cdot lbf$$

$$R_{LV} := W_{Drum} + W_{cable} + F_{maxn} + W_{BGS} + ML_{BG} + W_{BG} - R_{RV}$$

$$R_{LV} = 2.535 \times 10^{5}\cdot lbf$$

Horizontal Loading, Max. Normal Load



$$R_{RH} := \frac{ML_{BG}\cdot \tan\left(\frac{20\cdot\pi}{180}\right)\cdot 62.31in}{78.8125in}$$

$$R_{RH} = 2.816 \times 10^{4}\cdot lbf$$

$$R_{LH} := ML_{BG}\cdot \tan\left(\frac{20\cdot\pi}{180}\right) - R_{RH}$$

$$R_{LH} = 7.457 \times 10^{3}\cdot lbf$$

Resultant Loads

$$R_R := \sqrt{R_{RV}^{2} + R_{RH}^{2}}$$

$$R_R = 2.667 \times 10^{5}\cdot lbf$$

$$R_L := \sqrt{R_{LV}^{2} + R_{LH}^{2}}$$

$$R_L = 2.536 \times 10^{5}\cdot lbf$$

Stalled Load Bearing Reactions, Equally Divided

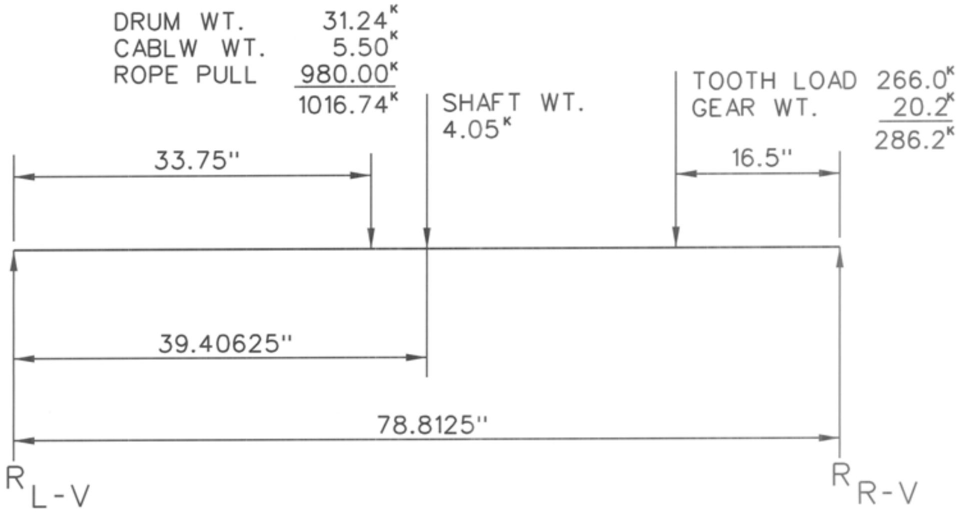
Large Bull Gear Tangential Tooth Load (Calculated Above)

$$SL_{BG} = 2.663 \times 10^{5}\cdot lbf$$

Stalled Load Rope Pull (Calculated Above)

$$RPS = 9.794 \times 10^{5}\cdot lbf$$

Vertical Loading, Stall Load Equally Divided



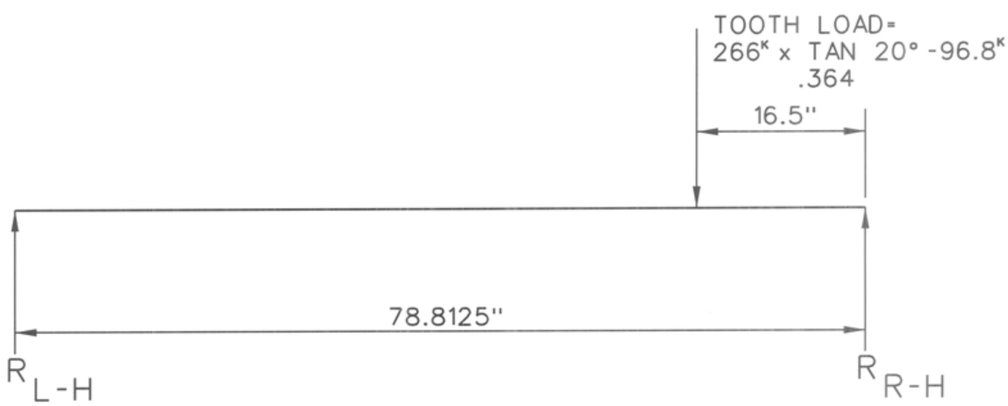
$$R_{RVs} := \frac{\left(W_{Drum} + W_{cable} + RPS\right)\cdot 33.75in + W_{BGS}\cdot 39.4in + \left(SL_{BG} + W_{BG}\right)\cdot 62.31}{78.8125in}$$

$$R_{RVs} = 6.637 \times 10^{5}\cdot lbf$$

$$R_{LVs} := W_{Drum} + W_{cable} + RPS + W_{BGS} + SL_{BG} + W_{BG} - R_{RVs}$$

$$R_{LVs} = 6.43 \times 10^{5}\cdot lbf$$

Horizontal Loading, Stall Load Equally Divided



$$R_{RHs} := \frac{SL_{BG}\cdot \tan\left(20\cdot\frac{\pi}{180}\right)\cdot 62.31in}{78.8125in}$$

$$R_{RHs} = 7.663 \times 10^{4}\cdot lbf$$

$$R_{LHs} := SL_{BG}\cdot \tan\left(20\cdot\frac{\pi}{180}\right) - R_{RHs}$$

$$R_{LHs} = 2.029 \times 10^{4} \cdot lbf$$

Resultant Loads

$$R_{Rs} := \sqrt{R_{RVs}^{2} + R_{RHs}^{2}}$$

$$R_{Rs} = 6.681 \times 10^{5} \cdot lbf$$

$$R_{Ls} := \sqrt{R_{LVs}^{2} + R_{LHs}^{2}}$$

$$R_{Ls} = 6.434 \times 10^{5} \cdot lbf$$

Stalled Load Bearing Reactions, 70/30 Split

Large Bull Gear Tangential Tooth Load (Calculated Above)

$$SL_{BG70} = 3.728 \times 10^{5} \cdot lbf$$

Stalled Load Rope Pull (Calculated Above)

$$RPS_{70} = 1.371 \times 10^{6} \cdot lbf$$

Vertical Loading

$$R_{RVs70} := \frac{\left(W_{Drum} + W_{cable} + RPS_{70}\right) \cdot 33.75in + W_{BGS} \cdot 39.4in + \left(SL_{BG70} + W_{BG}\right) \cdot 62.31}{78.8125in}$$

$$R_{RVs70} = 9.157 \times 10^{5} \cdot lbf$$

$$R_{LVs70} := W_{Drum} + W_{cable} + RPS_{70} + W_{BGS} + SL_{BG70} + W_{BG} - R_{RVs70}$$

$$R_{LVs70} = 8.893 \times 10^{5} \cdot lbf$$

Horizontal Loading

$$R_{RHs70} := \frac{SL_{BG70} \cdot \tan\left(20 \cdot \frac{\pi}{180}\right) \cdot 62.31in}{78.8125in}$$

$$R_{RHs70} = 1.073 \times 10^{5} \cdot lbf$$

$$R_{LHs70} := SL_{BG70} \cdot \tan\left(20 \cdot \frac{\pi}{180}\right) - R_{RHs70}$$

$$R_{LHs70} = 2.841 \times 10^{4} \cdot lbf$$

Resultant Loads

$$R_{Rs70} := \sqrt{R_{RVs70}^{2} + R_{RHs70}^{2}}$$

$$R_{Rs70} = 9.219 \times 10^{5} \cdot lbf$$

$$R_{Ls70} := \sqrt{R_{LVs70}^{2} + R_{LHs70}^{2}}$$

$$R_{Ls70} = 8.898 \times 10^{5} \cdot lbf$$

Stalled Load Bearing Reactions, 100/0 Split

$$R_{RHs100} := \frac{SL_{BG100}}{SL_{BG70}} \cdot R_{RHs70}$$

$$R_{RHs100} = 1.259 \times 10^{5} \cdot lbf$$

$$R_{LHs100} := \frac{SL_{BG100}}{SL_{BG70}} \cdot R_{LHs70}$$

$$R_{LHs100} = 3.334 \times 10^{4} \cdot lbf$$

$$R_{RVs100} := \frac{SL_{BG100}}{SL_{BG70}} \cdot R_{RVs70}$$

$$R_{RVs100} = 1.075 \times 10^{6} \cdot lbf$$

$$R_{LVs100} := \frac{SL_{BG100}}{SL_{BG70}} \cdot R_{LVs70}$$

$$R_{LVs100} = 1.044 \times 10^{6} \cdot lbf$$

Resultant Loads

$$R_{Rs100} := \sqrt{R_{RVs100}^{2} + R_{RHs100}^{2}}$$

$$R_{Rs100} = 1.082 \times 10^{6} \cdot lbf$$

$$R_{Ls100} := \sqrt{R_{LVs100}^{2} + R_{LHs100}^{2}}$$

$$R_{Ls100} = 1.044 \times 10^{6} \cdot lbf$$

Bearing Selection

Max. Normal Load

$$R_{R} = 2.667 \times 10^{5} \cdot lbf$$

Max. Stalled Load, Equally Divided

$$R_{Rs} = 6.681 \times 10^{5} \cdot lbf$$

Max. Stalled Load, 70/30 Split

$$R_{Rs70} = 9.219 \times 10^{5} \cdot lbf$$

Bull Gear Shaft Rotational Speed

$$BG_{s} := \frac{S_{m}}{TR}$$

$$BG_{s} = 0.077 \cdot \frac{1}{min}$$

Use Spherical Self Aligning Roller Bearing

Dynamic Load Rating, From Catalog Data

$$C_d := 383 \cdot 10^3 \cdot lbf$$

Static Load Rating - Considered to be the Load at Which Negligible Permanent Deformation Begins, Bearing Fracture will begin at a Multiple of This Load

$$C_o := 403 \cdot 10^3 lbf$$

Life Expectancy for 90% of Bearings (L_{10} Life), from Mechanical Engineering Design, Shigley and Mischke, 5th Ed.

$$L_B := \frac{\left(\frac{C_d}{R_R}\right)^{\frac{10}{3}} \cdot 10^6}{BG_s \cdot 60 \frac{min}{hr}}$$

$$L_B = 7.255 \times 10^5 \cdot hr$$

Per Cent Static Capacity, Stall Load Equally Divided

$$SC := \frac{R_{Rs70}}{C_o}$$

$$SC = 228.769 \cdot \%$$

Per Cent Static Capacity, Stall Load 70/30 Split

$$SC_{70} := \frac{R_{Rs70}}{C_o}$$

$$SC_{70} = 228.769 \cdot \%$$

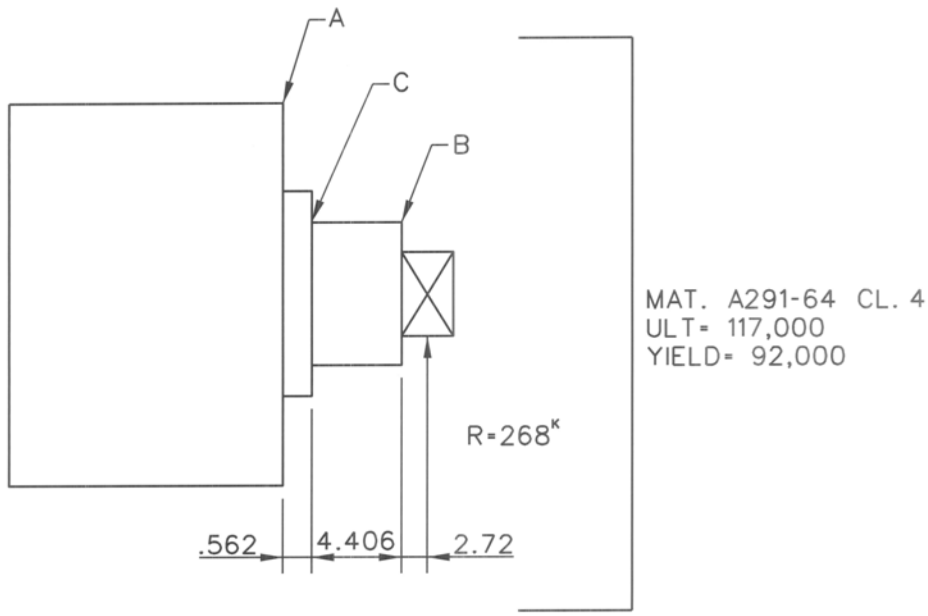
Selected Bearing Fractures at 8x the Static Load Rating

Shaft Diameter at Shaft Ends

Material A291,
Cl.4

$$S_{us} := 117000 \frac{lbf}{in^2}$$

$$S_{ys} := 92000 \frac{lbf}{in^2}$$



Normal Load

At Point A

Max Bearing Reaction - Normal Load

$$R_R = 2.667 \times 10^5 \cdot \mathrm{lbf}$$

Bending Moment

$$M_{As} := R_R \cdot 7.126\mathrm{in}$$

$$M_{As} = 1.901 \times 10^6 \cdot \mathrm{in} \cdot \mathrm{lbf}$$

Allowable Stress - Normal Load

$$S_{alls} := \frac{S_{us}}{5}$$

$$S_{alls} = 2.34 \times 10^4 \cdot \frac{\mathrm{lbf}}{\mathrm{in}^2}$$

Stress Concentration Factor

$$K_m := 1.5$$

Required Shaft Diameter at A

$$D_{sA} := \sqrt[3]{\frac{M_{As} \cdot K_m \cdot 32}{\pi \cdot S_{alls}}}$$

$$D_{sA} = 10.746 \cdot \mathrm{in}$$

At Point B

Bending Moment

$$M_{Bs} := R_R \cdot 2.72in$$

$$M_{Bs} = 7.254 \times 10^5 \cdot in \cdot lbf$$

Required Shaft Diameter at B, shaft equation, Mechanical Engineering Design, Shigley & Mischke, 5th Ed.

$$D_{sB} := \sqrt[3]{\frac{M_{Bs} \cdot K_m \cdot 32}{\pi \cdot S_{alls}}}$$

$$D_{sB} = 7.795 \cdot in$$

At Point C

Bending Moment

$$M_{Cs} := R_R \cdot 6.56in$$

$$M_{Cs} = 1.75 \times 10^6 \cdot in \cdot lbf$$

Required Shaft Diameter at C

$$D_{sC} := \sqrt[3]{\frac{M_{Cs} \cdot K_m \cdot 32}{\pi \cdot S_{alls}}}$$

$$D_{sC} = 10.454 \cdot in$$

Max. Load

Max. Bearing Reaction - Stalled Load, Equally Divided

$$R_{Rs} = 6.681 \times 10^5 \cdot lbf$$

Max. Bearing Reaction - Stalled Load, 70/30 Split

$$R_{Rs70} = 9.219 \times 10^5 \cdot lbf$$

Max. Bearing Reaction - Stalled Load, 100/0 Split

$$R_{Rs100} = 1.082 \times 10^6 \cdot lbf$$

At Point A

Bending Moment - Stalled Load, Equally Divided

$$M_{Asm} := R_{Rs} \cdot 7.126in$$

$$M_{Asm} = 4.761 \times 10^6 \cdot in \cdot lbf$$

Bending Moment - Stalled Load, 70/30 Split

$$M_{Asm70} := R_{Rs70} \cdot 7.126in$$

$$M_{Asm70} = 6.57 \times 10^6 \cdot in \cdot lbf$$

Bending Moment - Stalled Load, 100/0 Split

$$M_{Asm100} := R_{Rs100} \cdot 7.126in$$

$$M_{Asm100} = 7.709 \times 10^{6} \cdot in \cdot lbf$$

Allowable Stress - Stalled Load

$$S_{sm} := 0.75 \cdot S_{ys}$$

$$S_{sm} = 6.9 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Stress Concentration Factor

$$K_{mm} := 1$$

Required Shaft Diameter at A - Stalled Load, Equally Divided

$$D_{sAm} := \sqrt[3]{\frac{M_{Asm} \cdot 32 \cdot K_{mm}}{\pi \cdot S_{sm}}}$$

$$D_{sAm} = 8.891 \cdot in$$

Required Shaft Diameter at A - Stalled Load, 70/30 Split

$$D_{sAm70} := \sqrt[3]{\frac{M_{Asm70} \cdot 32 \cdot K_{mm}}{\pi \cdot S_{sm}}}$$

$$D_{sAm70} = 9.898 \cdot in$$

Selected Diameter at A - Greater of Required Diameter for Normal Load, Stalled Load Equally Divided, and Stalled Load 70/30 Split, Rounded Up

$$D_{sAs} := 10.75in$$

Stress at A for Selected Diameter, 100/0 Stalled Load

$$S_{A} := \frac{M_{Asm100} \cdot 32 \cdot K_{mm}}{\pi \cdot D_{sAs}^{3}}$$

$$S_{A} = 6.321 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Percent of Yield Stress at A for Selected Diameter, 100/0 Stalled Load

$$P_{yA} := \frac{S_{A}}{S_{ys}}$$

$$P_{yA} = 68.708 \cdot \%$$

At Point B

Bending Moment - Stalled Load, Equally Divided

$$M_{Bsm} := R_{Rs} \cdot 2.72in$$

$$M_{Bsm} = 1.817 \times 10^{6} \cdot in \cdot lbf$$

Bending Moment - Stalled Load, 70/30 Split

$$M_{Bsm70} := R_{Rs70} \cdot 2.72in$$

$$M_{Bsm70} = 2.508 \times 10^{6} \cdot in \cdot lbf$$

Bending Moment - Stalled Load, 100/0 Split

$$M_{Bsm100} := R_{Rs100} \cdot 2.72in$$

$$M_{Bsm100} = 2.943 \times 10^{6} \cdot in \cdot lbf$$

Required Shaft Diameter at B - Stalled Load, Equally Divided

$$D_{sBm} := \sqrt[3]{\frac{M_{Bsm} \cdot 32 \cdot K_{mm}}{\pi \cdot S_{sm}}}$$

$$D_{sBm} = 6.449 \cdot in$$

Required Shaft Diameter at B - Stalled Load, 70/30 Split

$$D_{sBm70} := \sqrt[3]{\frac{M_{Bsm70} \cdot 32 \cdot K_{mm}}{\pi \cdot S_{sm}}}$$

$$D_{sBm70} = 7.18 \cdot in$$

Selected Diameter at B - Greater of Required Diameter for Normal Load, Stalled Load Equally Divided, and Stalled Load 70/30 Split, Rounded Up

$$D_{sBs} := 7.875in$$

Stress at B for Selected Diameter, 100/0 Stalled Load

$$S_B := \frac{M_{Bsm100} \cdot 32 \cdot K_{mm}}{\pi \cdot D_{sBs}^{3}}$$

$$S_B = 6.138 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Percent of Yield Stress at B for Selected Diameter, 100/0 Stalled Load

$$P_{yB} := \frac{S_B}{S_{ys}}$$

$$P_{yB} = 66.712 \cdot \%$$

At Point C

Bending Moment, Stalled Load, Equally Divided

$$M_{Csm} := R_{Rs} \cdot 6.56in$$

$$M_{Csm} = 4.383 \times 10^{6} \cdot in \cdot lbf$$

Bending Moment, Stalled Load, 70/30 Split

$$M_{Csm70} := R_{Rs70} \cdot 6.56in$$

$$M_{Csm70} = 6.048 \times 10^{6} \cdot in \cdot lbf$$

Bending Moment - Stalled Load, 100/0 Split

$$M_{Csm100} := R_{Rs100} \cdot 6.56in$$

$$M_{Csm100} = 7.097 \times 10^{6} \cdot in \cdot lbf$$

Required Shaft Diameter at C - Stalled Load, Equally Divided

$$D_{sCm} := \sqrt[3]{\frac{M_{Csm} \cdot 32 \cdot K_{mm}}{\pi \cdot S_{sm}}}$$

$$D_{sCm} = 8.649 \cdot in$$

Required Shaft Diameter at C - Stalled Load, 70/30 Split

$$D_{sCm70} := \sqrt[3]{\frac{M_{Csm70} \cdot 32 \cdot K_{mm}}{\pi \cdot S_{sm}}}$$

$$D_{sCm70} = 9.629 \cdot in$$

Selected Diameter at C - Greater of Required Diameter for Normal Load, Stalled Load Equally Divided, and Stalled Load 70/30 Split, Rounded Up

$$D_{sCs} := 10.5in$$

Stress at C for Selected Diameter, 100/0 Stalled Load

$$S_C := \frac{M_{Csm100} \cdot 32 \cdot K_{mm}}{\pi \cdot D_{sCs}^{\ 3}}$$

$$S_C = 6.245 \times 10^{4} \cdot \frac{lbf}{in^2}$$

Percent of Yield Stress at C for Selected Diameter, 100/0 Stalled Load

$$P_{yC} := \frac{S_C}{S_{ys}}$$

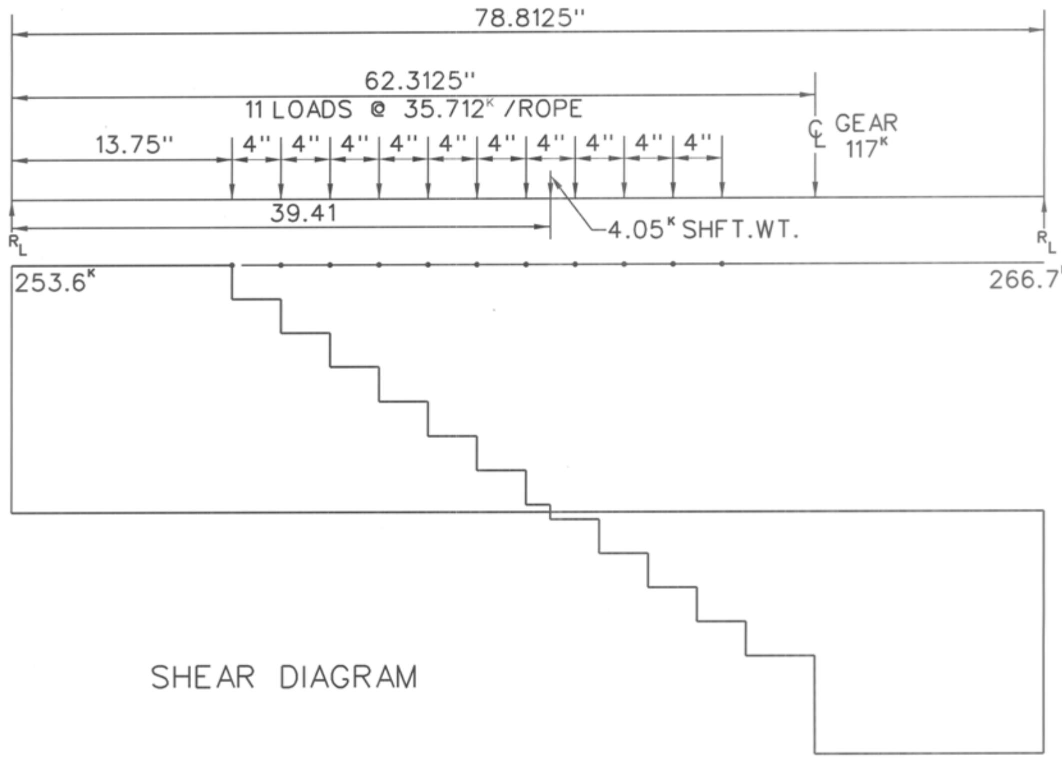
$$P_{yC} = 67.877 \cdot \%$$

Diameter of Bull Gear Shaft at Max. Bending Moment

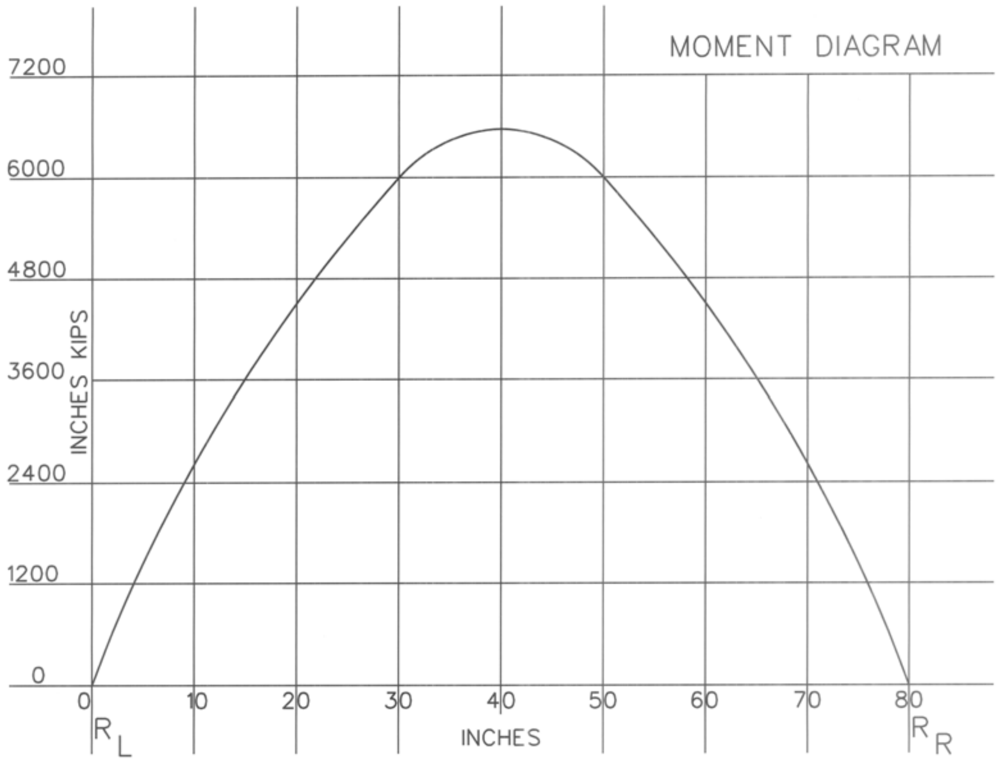
Normal Load

Max. Bending Moment - From Moment Diagram

$$M_{snmax} := 6500 \cdot 10^3 in \cdot lbf$$



Shown for 11 ropes, 12 ropes would result in a similar max. moment



Diameter Required at Max. Bending Moment, Max. Working Load

$$D_{snmax} := \sqrt[3]{\frac{32 \cdot K_m \cdot M_{snmax}}{\pi \cdot S_{alls}}}$$

$$D_{snmax} = 16.191 \cdot in$$

Stalled Load

Max. Bending Moment, Stalled Load Equally Divided

$$M_{ssmax} := M_{snmax} \cdot \frac{F_{maxa}}{F_{maxn}} \cdot 2.8$$

$$M_{ssmax} = 1.769 \times 10^{7} \cdot in \cdot lbf$$

Max. Bending Moment, Stalled Load, 70/30 Split

$$M_{ssmax70} := 0.7 \cdot 2M_{snmax} \cdot \frac{F_{maxa}}{F_{maxn}} \cdot 2.8$$

$$M_{ssmax70} = 2.476 \times 10^{7} \cdot in \cdot lbf$$

Max. Bending Moment, Stalled Load, 100/0 Split

$$M_{ssmax100} := M_{ssmax} \cdot 2 - M_{snmax} \cdot \frac{F_{maxa}}{F_{maxn}}$$

$$M_{ssmax100} = 2.906 \times 10^{7} \cdot in \cdot lbf$$

Allowable Stress - Stalled Load

$$S_{sm} = 6.9 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Diameter Required at Max. Bending Moment, Stalled Load Equally Divided

$$D_{ssmax} := \sqrt[3]{\frac{32 \cdot K_{mm} \cdot M_{ssmax}}{\pi \cdot S_{sm}}}$$

$$D_{ssmax} = 13.771 \cdot in$$

Diameter Required at Max. Bending Moment, Stalled Load, 70/30 Split

$$D_{ssmax70} := \sqrt[3]{\frac{32 \cdot K_{mm} \cdot M_{ssmax70}}{\pi \cdot S_{sm}}}$$

$$D_{ssmax70} = 15.405 \cdot in$$

Diameter Selected at Max. Bending Moment Point, Greater of Required Diameter for Normal Load, Stalled Load Equally Divided, and Stalled Load Split 70/30

$$D_{max} := 16.25 in$$

Stress at Max. Bending Moment Point for Stalled Load Split 100/0 with Selected Diameter

$$S_{max} := \frac{M_{ssmax100} \cdot 32 \cdot K_{mm}}{\pi \cdot D_{max}^{3}}$$

$$S_{max} = 6.898 \times 10^{4} \cdot \frac{lbf}{in^{2}}$$

Percent of Yield Stress at C for Selected Diameter, 100/0 Stalled Load

$$P_{smax} := \frac{S_{max}}{S_{ys}}$$

$$P_{smax} = 74.983 \cdot \%$$

Summary of Component Sizes for Each Load Case

	Required Component Size				
	Max. Working Load	Stall Load Equally Divided	Stall Load 70/30 Split	Selected Dimension Based on Governing Case	% of Yield Stress for Stall Load Split 100/0**
No. of Wire Ropes	11	11	12	12	80.6%***
Large Bull Gear Pinion Face Width	11.666"	11.489"	16.085"	16.25"	87.1%
Large Bull Gear Face Width	9.38"	10.891"	15.248"	16.25"	59.0%
Small Bull Gear Pinion Face Width	5.464"	6.74"	9.437"	9.5"	87.4%
Small Bull Gear Face Width	4.277"	6.208"	8.692"	9.5"	80.5%
Bull Gear Shaft End Diameter at A	10.746"	8.891"	9.898"	10.75"	68.7%
Bull Gear Shaft End Diameter at B	7.795"	6.449"	7.18"	7.875"	66.7%
Bull Gear Shaft End Diameter at C	10.454"	8.649"	9.629"	10.5"	67.9%
Bull Gear Shaft Diameter at Max. Moment	16.191"	13.771"	15.405"	16.25"	75.0%

*Governing Case is Shaded
**Based on Selected Dimension or Number of Wire Ropes
***Percent of Wire Rope Breaking Strength

1A .Tainter Gate Unbalanced Loading for Single Motor Hoist Units:

Discussion:

The EM 2610 criteria requires a factor of safety of 5 with respect to the material ultimate strength for normal loading conditions. The EM criteria also requires overload loading from the maximum torque of the motor to not result in any more than 75% of the yield strength of the material. There have been documented cases of unequal load sharing between sides of a tainter gate hoist. There is a concern that an uneven loading condition could result in a gate jamming between piers. This may result in the maximum motor torque being applied to the hoist system unevenly. It is possible that up to a 100%/0% split load sharing could be experienced at a maximum motor torque loading. However, this is type of loading is extremely unlikely unless a component in the power train after the split between sides of a hoist first experiences a failure. It should also be noted that EM 2105 requires the gate structure to be designed for 100%/0% load sharing between sides of a hoist for both normal loading and maximum motor torque conditions. The EM 2610 criteria for maximum overload conditions now requires using a 70%/30% split for sizing components. The following calculations will investigate unequal hoist loading scenarios for a single motor hoist system.

EM 2610 Criteria Normal Loading and Overload Conditions Assuming Load Evenly Split:

$$\boldsymbol{Normal\ Loading\ Critera}: FLT \ \leq \frac{U_L}{5}\ or\ NTE: 20\%(U_L)$$

&

$$\boldsymbol{Maximum\ motor\ torque\ Critera}: 2.8(FLT) \leq 75\%(Y_S)$$

Where:
FLT = Full Load Torque
Ys = Yield Strength
U_L = Ultimate Tensile Strength

(For these calculations, it will be assumed normal operating load is equal to the FLT of the motor. Also, it is assumed that the maximum motor torque is the locked rotor torque or stall torque equal to 280% of the FLT.)

The loading between sides of a hoist the load is assumed to be equally split. So for each side of a hoist the criteria would be:

$$\boldsymbol{Normal\ Loading\ Critera}: 50\%(FLT) \leq \frac{U_L}{5} or\ NTE: 20\%(U_L)$$

&

$$\boldsymbol{Maximum\ motor\ torque\ Critera}: 50\%(2.8)(FLT) \leq 75\%(Y_S)$$

However, what would happen if a hoist designed for 50%/50% loading experiences uneven loading? Let’s assume a 100%/0% breakdown between the sides of the hoist.

For this condition, the loading on one side of the hoist would be doubled. This results in:

$$\boldsymbol{Normal\ Loading\ Critera}: \ 2x50\%(FLT) = 2x20\%(U_L) = 40\%(U_L)$$
&
$$\boldsymbol{Maximum\ motor\ torque\ Critera}: 2x50\%(2.8)(FLT) = 2(75\%)(Y_S) = 150\%(Y_S)$$

The maximum overload condition or stalled condition applied to one side of the hoist unit could result in 150% of the yield strength of the components. How does this relate to the ultimate strength? To look at this we have to make an assumption for the relation between yield strength and ultimate strength for the materials used. For most steels, the yield strength is approximately 75% of the ultimate($Y_S = 75\%(U_L)$). The most conservative case would be if a very brittle material was used which has a yield at a higher percentage of the ultimate. The worst case steel should have a yield around 90% ($Y_S = 90\%(U_L)$). Using this scenario, let's look at what ultimate strength we would be at:

Assume: $Y_S = 90\%(U_L)$

Then: $100\%(2.8)(FLT) = 2(75\%)(Y_S) = 150\%(Y_S) = 150\%(90\%)(U_L) = 135\%(U_L)$

So, if a hoist designed for 50/50 loading experienced 100% loading on one side, the components downstream of the split between sides of a hoist would be loaded to 40% of ultimate for normal operation. A stalled condition would result in a loading of 150% of yield which should be no higher than 135% of ultimate tensile strength. This loading would result in hoist components breaking. While this is an extremely rare situation it is still assumed to be an unacceptable risk.

<u>EM 2610 Criteria Normal Loading and Overload Conditions Assuming a 70/30 Split:</u>

EM 2610 now requires designing a tainter gate hoist for a 70/30 split between sides of a hoist. Let's look at how this would affect the system:

First let's define this criteria. After the split between sides of a hoist:

$$\boldsymbol{Normal\ Loading\ Critera}: 70\%(FLT) \le \frac{U_L}{5}\ or\ NTE: 20\%(U_L)$$
&
$$\boldsymbol{Maximum\ motor\ torque\ Critera}: 70\%(2.8)(FLT) \le 75\%(Y_S)$$

Now let's assume this hoist designed for a 70%/30% split experiences 100%/0% uneven loading. This would be a 143% increase ($\frac{100}{70} = 1.43$). This would result in:

$$\boldsymbol{Normal\ Loading\ Critera}: \quad 1.43x70\%(FLT) = 1.43x20\%(U_L) = 28.6\%(U_L)$$

&

$$\boldsymbol{Maximum\ motor\ torque\ Critera}: 1.43x70\%(2.8)(FLT) = 1.43(75\%)(Y_S) = 107\%(Y_S)$$

Again let's conservatively assume we have steel with a yield at 90% of the ultimate:

Assume: $Y_S = 90\%(U_L)$

Then:

$$1.43x70\%(2.8)(FLT) = 1.43(75\%)(Y_S) = 107\%(Y_S) = 107\%(90\%)(U_L) = 96.3\%(U_L)$$

So, if a hoist designed for 70/30 breakdown of load experienced 100/0 loading, the components downstream of the split between sides of a hoist would be loaded to 28.6% of ultimate for normal operation. A stalled condition would result in a loading of 107% of yield which should be no higher than 96.3% of ultimate. This value would be less if the yield strength was closer to 75% of the ultimate tensile strength. The designer should select materials and components so that 90% of the ultimate tensile strength is not exceeded.

2. Sector Gate - Machinery Design

Typical calculations for determining closing pintle torque for a reverse head.

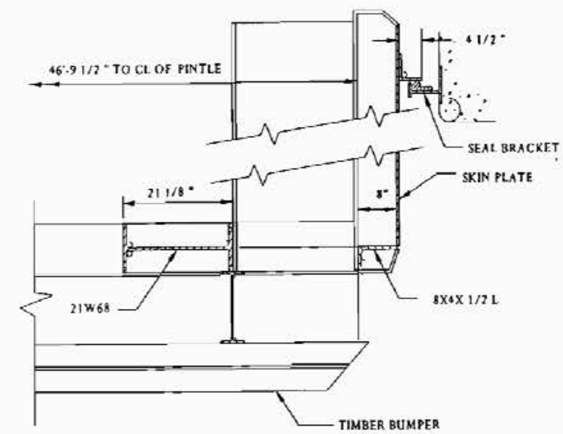
Closing torque is composed of

- Hydrodynamic forces acting on the nose of the gate,
- Hinge and pintle friction - composed of friction from gate's weight and hydrostatic head.

There is no torque due to seal friction because the reverse head lifts the bottom seal. There are no reverse head seals.

Design Conditions: 5' Reverse Head
16' Tailwater

a. Hydrodynamic Torque: To obtain peak pintle torque due to hydrodynamics see WES Report H-70-2 Appendix A, Plate A4, Figure a. Nose of the design gate is as shown below.



NOSE AND SEAL DETAIL
OF THE DESIGN GATE

Projected width of miter beam, skin plate rib and
seal plate bracket of design gate= 21.125" +8" +4.5"= 33.625"

For the design conditions Figure a indicates a hydrodynamic
torque = 200 Ft-Kips.

Figure a was developed for a gate with a radius of 42 feet and total projected width of miter beam, skin plate rib, and seal bracket of 30.375" (see WES Technical Report, H-70-2, Appendix A, Plate A1. The design gate has a gate radius of 46.792' and a projected width of miter beam, skin plate and seal plate bracket of 33.625". To obtain the hydrodynamic torque for the design gate it is necessary to apply correction factors for the gate radius and projected width as follows:

Hydrodynamic torque for the design gate =
200 Ft-Kips(33.625"/30.375") (46.792'/42') = <u>247 Ft-Kips</u>

b. Hinge and Pintle Friction:

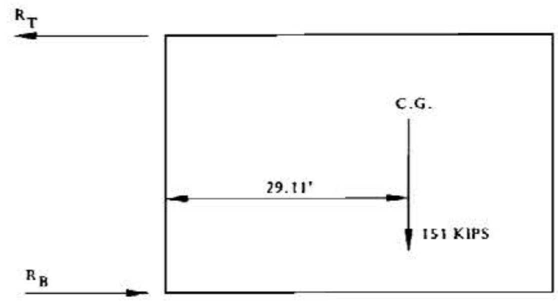
Design Data:

Spherical Hinge Diameter = 12"
Spherical Pintle Diameter = 18"
Gate Weight = 151 kips @ cg 29.11' from vertical hinge & pintle centerline.

Assume:

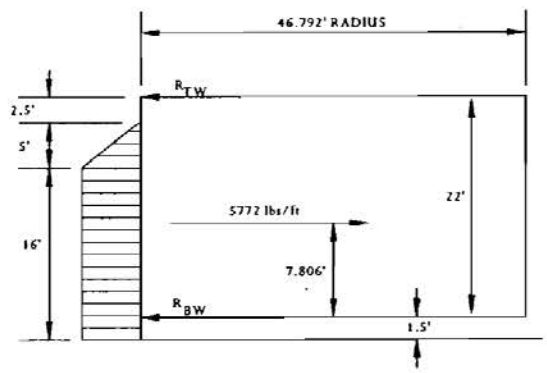
- Vertical dead weight reaction acts at one half of the spherical pintle radius or 4.5".
- Coefficient of friction for bushings = 0.25.
- Static water load and horizontal dead weight reactions act at the hinge and pintle radius

Hinge and pintle load from gate's weight:



$R_T = R_B$ = 151 Kips (29.11') / 22' = 200 Kips

Hinge and pintle load from the hydrostatic head:



53.716' (.5(62.4 lbs/ft^3) (5')^2 + 62.4 lbs/ft^3(5')(16')) = 310.05 Kips

Where 53.716' is the cord length of the skin plate.

Centroid of net static water load is 7.806' up from the pintle, therefore,
Pintle reaction = R_{BW} = 310.5 (22'-7.806')/(22') = 200.33 Kips and

Hinge reaction = R_{TW} = 310.05 Kips - 200.33 Kips = 109.72 Kips.

Net pintle horizontal reaction = -200.33 Kips + 200 Kips = 0.33 Kips

Net hinge horizontal reaction = 200 Kips + 109.72 Kips = 309.72 Kips
Friction from horizontal reactions
= 0.25 (9/12)' (.33 Kips) + 0.25 (6/12)' (309.72 Kips) = <u>38.78 Ft-Kips</u>

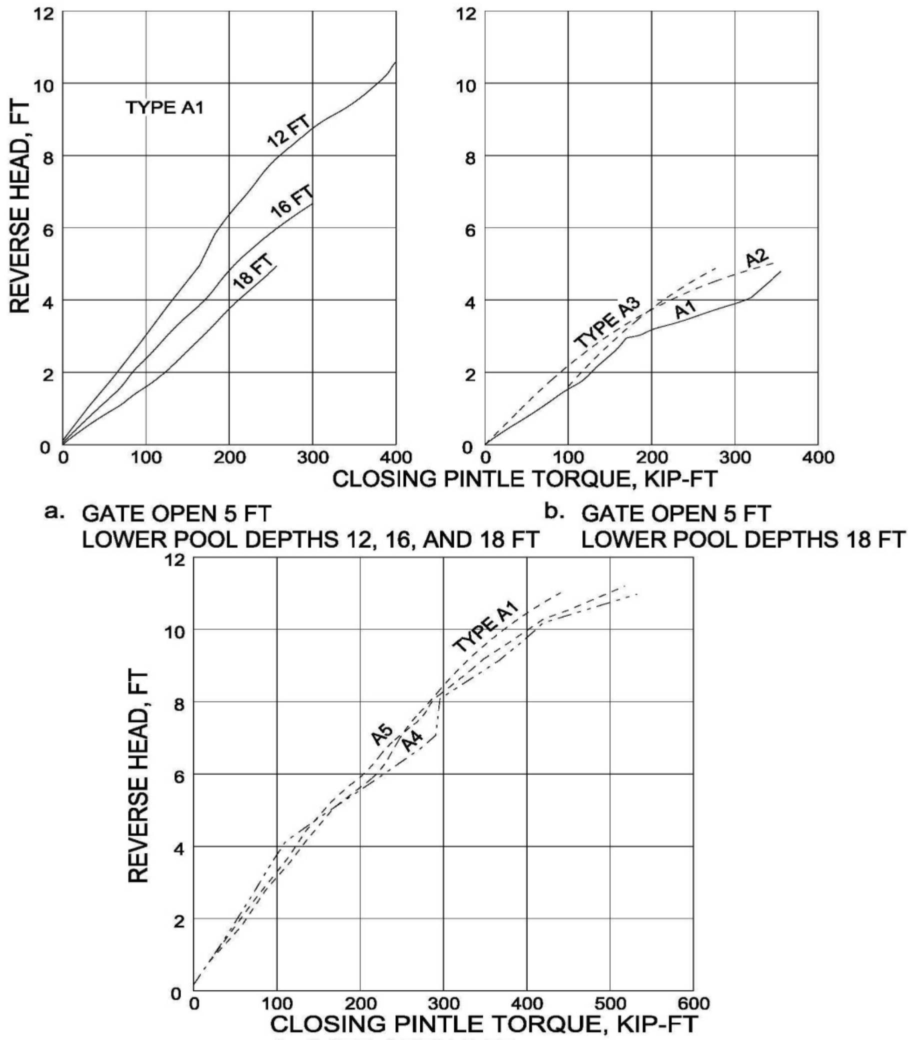
Friction from vertical reaction = 0.25 (4.5/12)' (151 Kips) = <u>14.16 Ft-Kips</u>

Total calculated torque = 247 Ft-Kips + 38.78 Ft-Kips + 14.16 Ft-Kips = <u>299.94 Ft-Kips</u>

Applying a service factor of 1.5, the design operating torque =

1.5 (299.94 Ft-Kips) = <u>**449.9 Ft-Kips**</u>

EFFECTS OF REVERSE HEAD ON CLOSING PINTLE TORQUE



a. GATE OPEN 5 FT
LOWER POOL DEPTHS 12, 16, AND 18 FT

b. GATE OPEN 5 FT
LOWER POOL DEPTHS 18 FT

C. GATE OPEN 7 FT
LOWER POOL DEPTH 12 FT

FROM WES OPERATING
FORCES ON SECTOR GATES
WES REPORT H-70-2,
UNDER REVERSE HEADS.
APPENDIX A, PLATE A4

Section 2A : 56' Sector Floodgate Mechanical Gate Operating Machinery & Hinge and Pintle Bearings Computations

Calculations provided for both direct head and reverse head case.

The gate will be gear driven with a gear rack gear mounted to the gate's skin plate. A pinion gear mounted to a high torque low speed motor will drive the rack.

Maximum operating cases were considered:

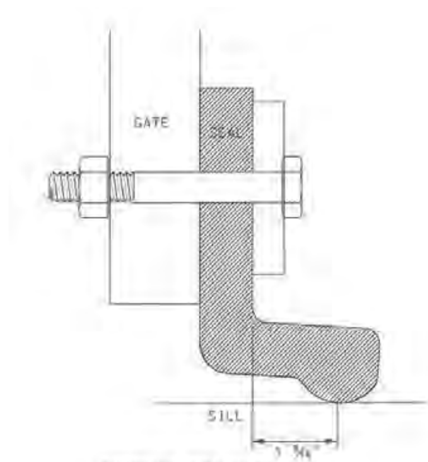
Case 1…………. 4' Direct head, Elevations 2.5 and -1.5 at a gate rotational speed of 6 degrees per minute

Case 2…………. 5' Reverse head, Elevations -1.5 and 3.5 at a gate rotational speed of 6 degrees per minute

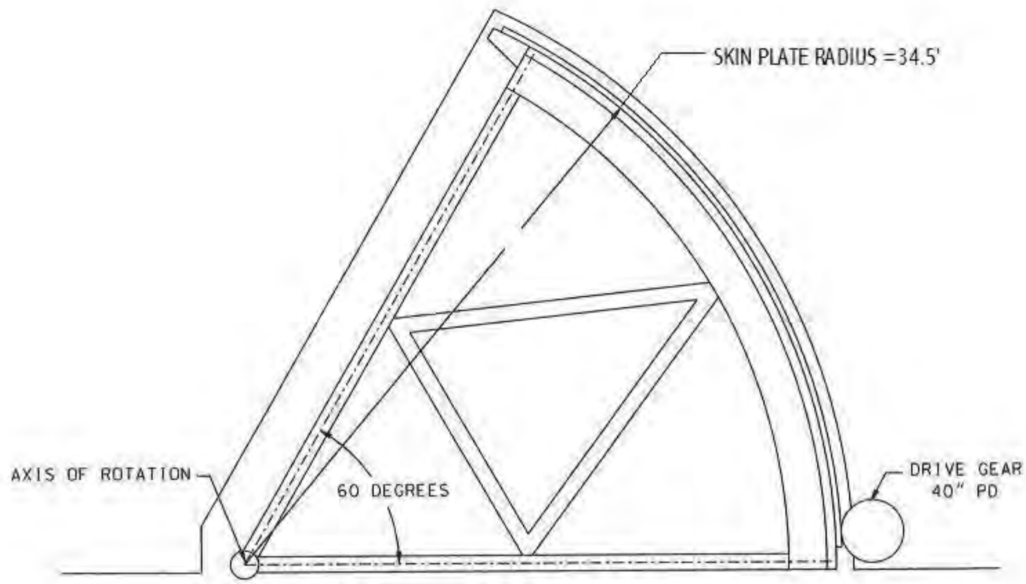
Gate Specifications and Design Data:

Top of Gate …………………….	EL 16.0
Bottom of Gate …………………	EL -10.3
Gate Weight (W_{gate}) …………..	110 Kips
C.G. …………………………….	24.1 from Hinge & Pintle
Skin Plate Radius (R_{spl}) ………..	34.5'
Hinge Elevation ………………….	14.5
Hinge Radius (R_{hinge}) ………..	Spherical 0.67'
Pintle Elevation …………………..	-8.8
Pintle Radius (R_{pintle}) ………..	Spherical 0.83'
Hinge & Pintle Coefficient of Friction ($f_{h/p}$) …….	0.25
Rack Gear Pitch Radius (R_{rack}) ..	34.9'
Pinion Pitch Radius (R_{pinion}) ……	1.6667'
Bottom Seal Type ……………….	J Type (See Below)
Bottom Seal Length (L_s) …..……	37'
Bottom Seal Radius (R_s) ………..	34.7'
Seal Coefficient of Friction (f_s) …	1.0
Preset Seal Load (F_{preset}) ………	10 Lbs/Ft

Bottom Seal Geometry:



BOTTOM SEAL CROSS SECTION



GATE GEOMETRY

Case 1: (4' Direct Head)

Loads considered are, seal friction (preset and hydrostatic) and hinge and pintle friction.

Seal Hydrostatic Friction:

Seal hydrostatic friction torque (T_{hyd}) is as follows:

$$T_{hyd} = (p_{water})(f_s)[(W_{seal})/2](h_d)(L_s)(R_s)$$

Where

p_{water} = weight of water = 62.4Lbs/Ft
f_s = seal friction = 1.0
W_{seal} = width of seal = 0.130'
h_d = differential head = 4'
L_s = seal length = 37'
R_s = seal radius = 34.7'

Substituting T_{hyd} = 62.4(1)(0.130/2)(4)(37)(34.7)/1000 = **20.8 Ft-Kips**

Seal Preset Friction:

Seal preset friction torque (T_{preset}) is as follows:

$$T_{preset} = F_{preset}(f_s)(L_s)(R_s)$$

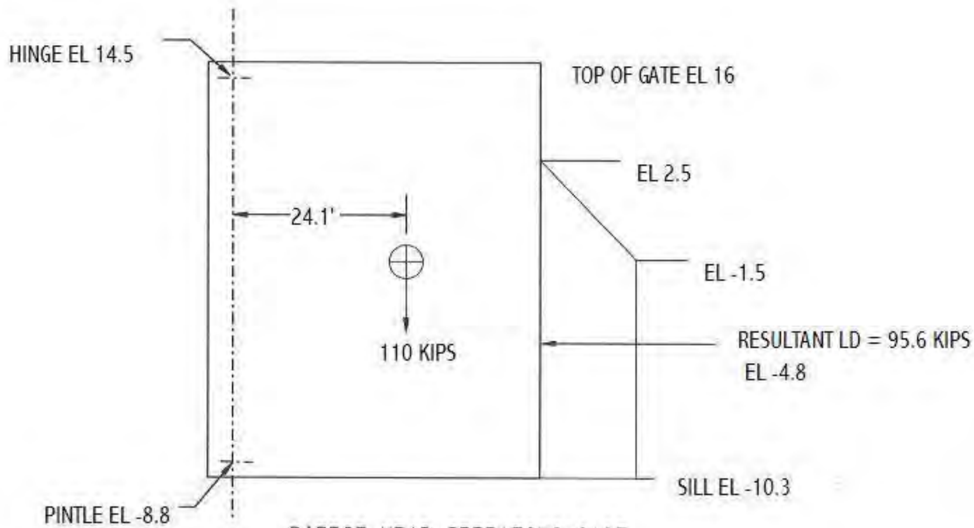
Where

F_{preset} = Seal preset load = 10Lbs/Ft
All other values are as above.

T_{preset} = 10(1)(37)(34.7)/1000 = **12.8 Ft-Kips**

Hinge and Pintle Friction:

Loads acting on the gate's hinge and pintle from the gate's weight and 4' direct head is as shown below. Hinge and pintle diameters are 18" and 22" respectively. Design frictional factor is 0.25. Hinge and pintle will be spherical, bronze on steel bearing surfaces.



DIRECT HEAD OPERATING CASE

Hinge horizontal (H_{hor}) load is as follows:

$$H_{hor} = [W_{gate}(C.G.)/(14.5+8.8)] - F_{hyd}[(y/(14.5+8.8)]$$

Where F_{hyd} = net hydrostatic load = 95.6 Kips
W_{gate} = gate weight = 110 Kips
C.G. = gate weight moment arm from hinge and pintle = 24.1'
y = vertical distance from center of pintle to application of hydrostatic load if the load was acting at one location = 3.95'

Substituting

$$H_{hor} = 110(24.1/23.3)-95.6(3.95)/23.3 = 97.6 \text{ Kips}$$

Pintle horizontal (P_{hor}) load is as follows:

$$P_{hor} = -[W_{gate}(C.G.)/(14.5+8.8] - F_{hyd}[(14.5+8.8-y)/(14.5+8.8)]$$

Substituting

$$P_{hor} = - 110(24.1/23.3) – 95.6(19.3/23.3) = 193 \text{ Kips}$$

The pintle carries the entire vertical load which is the gate's weight. The pintle resultant load $P_{resultant}$ is calculated as follows:

$$P(\text{resultant}) = [P^2_{vertical} + P^2_{hor}]^{1/2} = [110^2+193^2]^{1/2} = 222 \text{ Kips}$$

Frictional Torque (F_{torque}) is calculated as follows:

$$F_{torque} = (2/3\ R_{pintle}\ P_{resultant} + 2/3\ R_{hinge} H_{hor})(f_{h/p})$$

Where $f_{h/p}$ = hinge and pintle friction factor = 0.25

2/3[R] = effective radius of hinge and pintle
For the hinge R_{hinge} = 0.67'(.67) = 0.44'
For the pintle R_{pintle} = 0.83'(.67) = 0.56'

Substituting:

F_{torque} = [0.56(222) + 0.44(97.6)](0.25) = **41.8 Ft-Kips**

Case 1 torques are as follows:

Seal Hydrostatic Friction ……….	20.8 Ft-Kips
Seal Preset Friction ………………	12.8 Ft-Kips
Hinge and Pintle Friction ……….	41.8 Ft-Kips
Total for Case 1	**75.4 Ft-Kips**

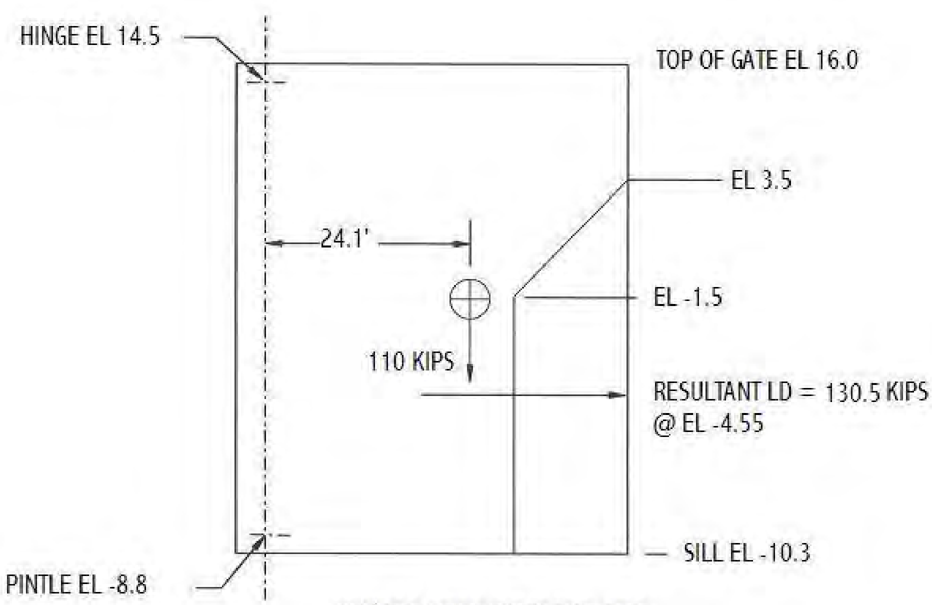
Case 2:

Loads considered for a 5.0' reverse head are hydrodynamic and hinge and pintle friction. There is no seal friction from a reverse head.

Hydrodynamic: Loads can be determined from using WES H-70-2 and applying correction factors and similarity. Alternatively, hydrodynamic loads can be estimated by applying 70% of the 5' differential head of water across the approximately 2' wide W24X68 I beam located on the miter end of the gate. This is based on a water elevation on the recess end of 3.5' and on the miter end of -1.5'. The distant from the centerline of the hinge/pintle to the centerline of the 2' wide end beam is 32.8'. Therefore the hydraulic torque =

0.7(5)(62.4)(5/2+8.8)(2)(32.8)/1000 = **162 Ft-Kips**

Hinge and Pintle Friction:
Loads acting on the gate's hinge and pintle from the gate's weight and 5' reverse head is as shown below. Hinge and pintle diameters are 16" and 20" respectively. Design frictional factor is 0.25. Hinge and pintle will be spherical, bronze on steel.



REVERSE HEAD OPERATING CASE

Hinge horizontal (H_{hor}) load is as follows:

$$H_{hor} = [W_{gate}(C.G.)/(8.8+14.5)] - F_{hyd}[(y/(8.8+14.5)]$$

Where F_{hyd} = net hydrostatic load = -130.5 Kips
W_{gate} = gate weight = 110 Kips
C.G. = gate weight moment arm from hinge and pintle = 24.1'
y = vertical distance from center of pintle to application of hydrostatic load if the load was acting at one location = 4.2'

Substituting

$$H_{hor} = 110(24.1/23.3)+130.5(4.2/23.3) = 137.3 \text{ Kips}$$

Pintle horizontal (P_{hor}) load is as follows:

$$P_{hor} = -[W_{gate}(C.G.)/(8.8+14.5)] - F_{hyd}[(14.5+8.8-4.2)/(8.8+14.5)]$$

Substituting

$$P_{hor} = -110(24.1/23.3) + 130.5(19.1/23.3) = -6.8 \text{ Kips}$$

The pintle carries the entire vertical load, which is the gate's weight of 110 Kips. The pintle resultant load $P_{resultant}$ is calculated as follows:

$$P(\text{resultant}) = [P^2_{vertical} + P^2_{hor}]^{1/2} = [110^2+6.8^2]^{1/2} = 110.2 \text{ Kips}$$

Frictional Torque (F_{torque}) is calculated as follows:

$$F_{torque} = (2/3\ R_{pintle}\ P_{resultant} + 2/3\ R_{hinge} H_{hor})(f_{h/p})$$

Where $f_{h/p}$ = hinge and pintle friction factor = 0.25
For the hinge R_{hinge} = 0.67'(.67) = 0.45'
For the hinge R_{pintle} = 0.83'(.67) = 0.56'

Substituting:

$$F_{torque} = [0.56(110.2) + 0.45(137.3)](0.25) = \mathbf{30.9\ Ft\text{-}Kips}$$

Cases 2 torques are as follows:

Hydrodynamic torque …………..	**162 Ft-Kips**
Hinge and Pintle Friction ……….	**30.9 Ft-Kips**
Total for Case 2	**192.9 Ft-Kips**

Machinery Loads:
Machinery loads are as shown in the following table. Formulas used to generate the values follows the table. The machinery selected to operate the gate is a gear/rack drive. Since the reverse head load case generates the largest load it will be used to size the machinery.

Case No.	Gate Operating Torque Ft-Kips	Gate Speed Deg/min (10^{-3}RPM)	Rack Force Kips[1]	Drive Pinion Torque Ft-LBS[2]	Pinion Speed RPM[3]	Motor[4] Pressure	Motor Flow GPM[5]	Pump HP[6]
1	75.4	6 (0.0167)	2.16	3676	0.35	669	2.8	2.6
2	192.9	6 (0.0167)	5.53	9405	0.35	1712	2.8	3.2

Sample calculations are for Case 2.

[1] Rack force (F_{rack}) = Gate Operating Torque/R_{rack} = 192.9/34.9= 5.53 Kips

[2] Drive pinion torque (T_{pinion}) = $F_{rack}(1/E_{gear})(R_{pinion})$ x 1000 = 9405 Ft-Lbs
Where: Gear efficiency (E_{gear}) = 0.98

[3] Pinion speed (S_{pinion}) = Gate Speed(R_{rack}/R_{pinion}) = 0.0167(34.9/1.6667) = 0.35 RPM (Starting value)

[4] Motor pressure (P_{motor}) = $(T_{pinion}/T_{motor})10^3$ = (9405/5492)(1000) = 1712 psi
Where: Motor output torque per 10^3 psi (T_{motor}) = 5492 Ft-Lbs
Based on a Hagglunds 44-06800 motor

[5] Motor flow req'd (Q_m) = $D_{theor}(S_{pinion})/231+ Q_i$
Where: D_{theor} is the theoretical displacement for the 44-06800 motor in in^3/revolution = 414(full)
Q_i is the volumetric loss for a given pressure. Use 2.5 gal/min for 2600 psi, 2.0 Gal/min for 2000 psi and 0.8 Gal/min for 1000 psi and less. Values are based on using a "D" type distributor.
Subsistuting: Q_m = (414/231)(.35) +2.0 = 2.6 GPM. Use 2.8 GPM to allow operating at 2600 psi. across the motor.

[6] Pump output horsepower (M_{hp}) = $(Q_m)(P_r)$/[(1714.3)] = 2.8(1712+400)/1714.3(.97) = 3.6HP
Where: P_r = Pump pressure = Motor pressure + 400 psi =1712 + 400 = 2112 psi
Relief valve setting is estimated at 3000 psi. A 3000 psi relief valve setting would allow for 50% overload (1714x1.5)+400 = 2971 psi.
Horsepower for 50% overload = 3000psi (2.8gpm) /1714.3= 4.9 hp

Rack Gear Design:
The gear will have a diametrical pitch =1. Tooth profile will be 20-degree full depth involute cut. Gear material will be structural steel or equivalent. Analysis will be based on the Lewis Beam Formula which analyzes the gear tooth as a cantilever beam. Assume only one tooth in contact.
Allowable stress is 60,000 psi/5 = 12,000 psi

$$s = F/bP_cy$$

Where: s = the tensile stress
P_c = the circular pitch = 3.1417"
c = tooth form factor = 0.124
b = width of gear teeth = 2"
F = tooth tangential load or transmitted load = 5530 lbs

Substituting: s = 5530/(2)(0.124)(3.1417) = 7098 psi
50% Overload = 1.5 x 7098psi =10,647 psi less than allowable of 12,000 psi.

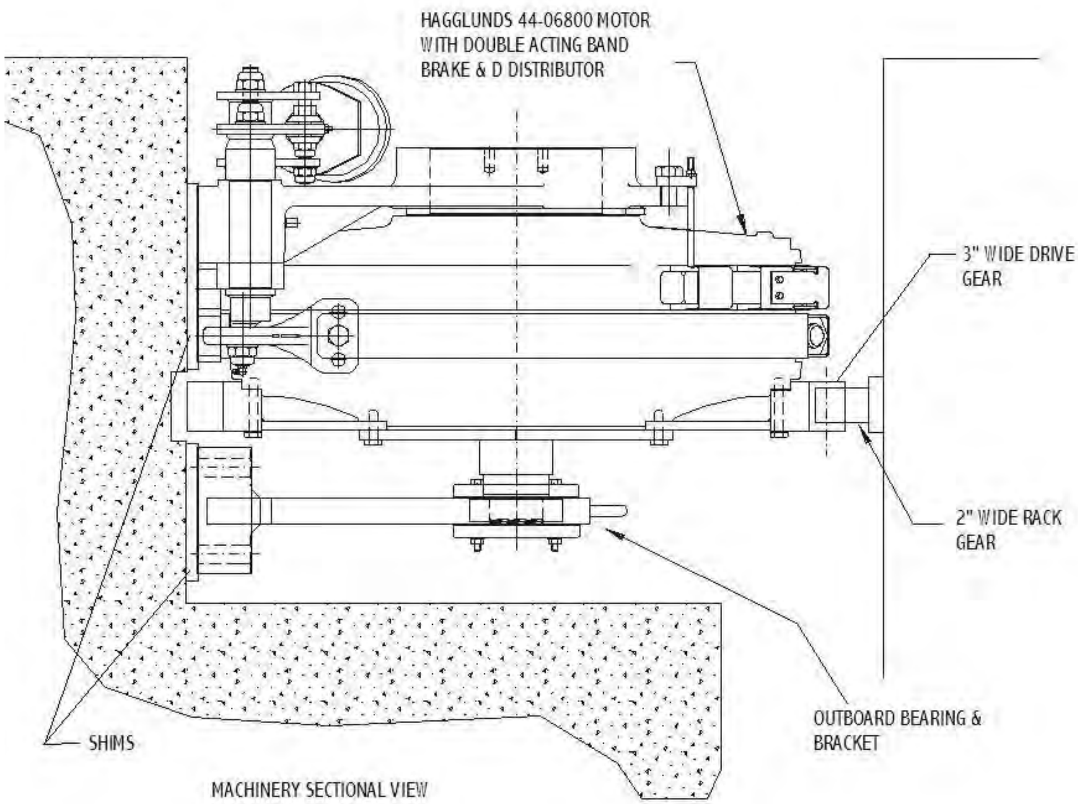
Drive gear will be 3" in width ASTM A148 Gr 80-40 or equivalent.
The rack would be plasma cut to reduce cost and because the stress and operating speeds are very low.

Pump motor size:
Assume 80% pump efficiency. Motor size = $M_{hp}/0.8 = 4.9/.8 = 6.1$ HP, therefore use 7.5 hp motor or 5 hp with 1.2 SF.

Reservoir:
Minimum usable volume is 2.8gpm x 3min. = 8.4 gallons
Recommend using a 20 gallon reservoir with usable capacity of 15 gallons.
Reservoir should be equipped with high temperature and low level shutdowns and a visible level indicator.
Pump should be flooded suction.

Note: Operating speed under low heads will be approximately 2 to 3 minutes and 10 minutes with reverse heads. If a 50% overload should occur based on the highest calculated reverse head operating time could be 20 minutes.

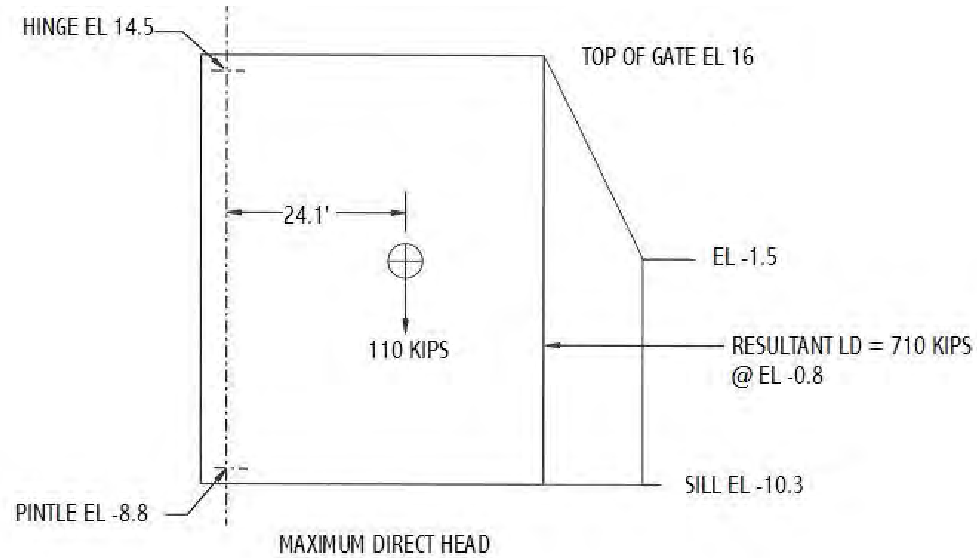
General machinery arrangement is as shown below.



MACHINERY SECTIONAL VIEW

Hinge and Pintle Requirements:

Maximum loading is from a direct head to the top of the gate, EL 16 and a protected side water EL -1.5. See diagram below.



Hinge horizontal (H_{hor}) load is as follows:

$$H_{hor} = [W_{gate}(C.G.)/(14.5+8.8)] - F_{hyd}[(y/(14.5+8.8)]$$

Where F_{hyd} = net hydrostatic load = 710 Kips
W_{gate} = gate weight = 110 Kips
C.G. = gate weight moment arm from hinge and pintle = 24.1'
y = vertical distance from center of pintle to application of hydrostatic load if the load was acting at one location = 8'

Substituting

$$H_{hor} = 110(24.1/23.3)-710(8)/23.3 = 130 \text{ Kips}$$

Add an additional load of 100 kips for boat impact. Total load =150 + 130 =280 Kips

Pintle horizontal (P_{hor}) load is as follows:

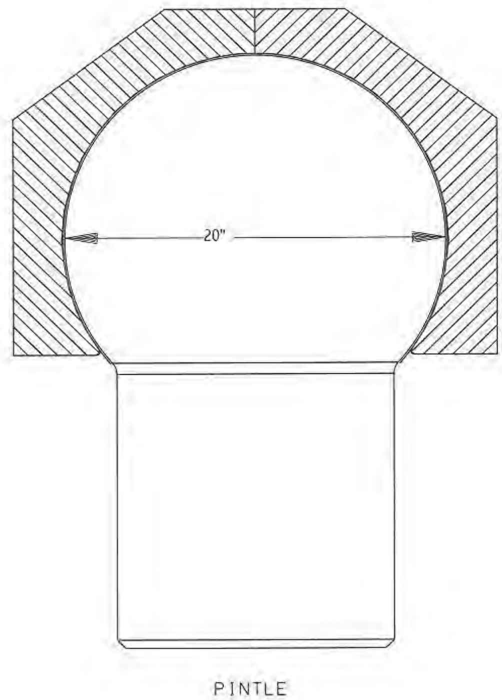
$$P_{hor} = -[W_{gate}(C.G.)/(14.5+8.8)] - F_{hyd}[(14.5+8.8-y)/(14.5+8.8)]$$

Substituting $P_{hor} = - 110(24.1/23.3) – 710(15.3/23.3) = -580$ Kips

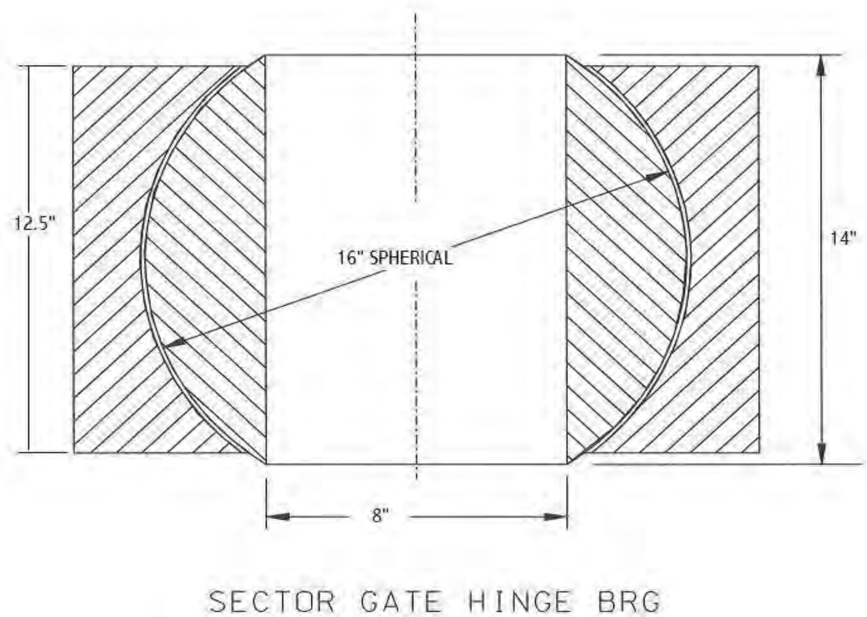
The pintle carries the entire vertical load which is the gate's weight. The pintle resultant load $P_{resultant}$ is calculated as follows:

$$P \text{ (resultant)} = [P^2_{vertical} + P^2_{hor}]^{1/2} = [110^2 + -580^2]^{1/2} = 590 \text{ Kips}$$

Pintle:
Use a 20” diameter spherical pintle.
Projected are $A_p = Pi(r^2) = 314 \text{ in}^2$
Limit maximum bearing pressure to 2000 psi. Use bronze CDA #932 on 17PH-4 stainless steel ball.
Limiting bearing pressure to 2ksi (314 in^2)= 628 Kips> 590 Kips



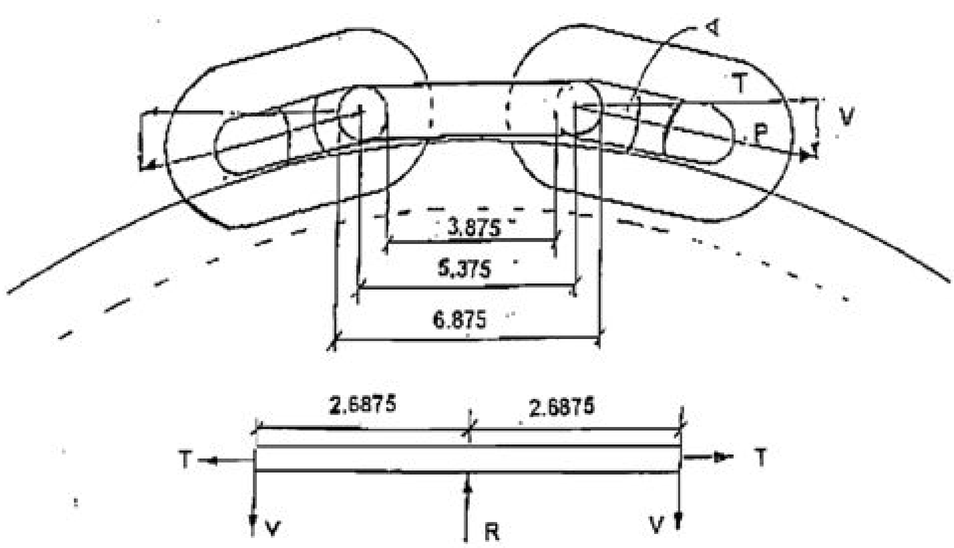
Hinge:
Use a 16" diameter spherical upper hinge.
Projected are A_p=Pi(r^2)-sections cut off by truncated area = 201.1-25.2= 176in^2
Limit maximum bearing pressure to 2000 psi. Use bronze CDA #932 on 17PH-4 stainless steel ball. Limiting bearing pressure to 2ksi (176 in^2)= 352 Kips> 280 Kips



3. Round Link Chain - Grooved Drum and Pocket Wheel

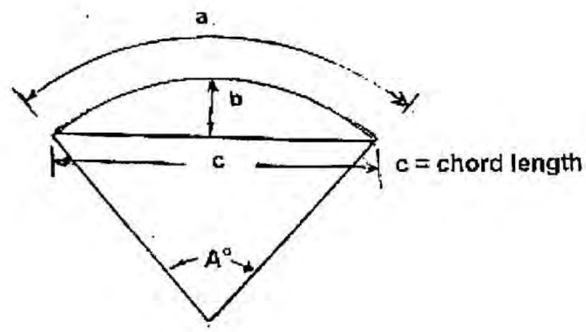
a. Chain Link Bending Stresses around Grooved Drum. Assume 1 ½ inch diameter round link with 3.875 inch pitch over a 41.69 inch diameter grooved drum.

Design load = 45,000 lb. x 275% overload
= 126,000 lb.



Material ASTM - A391, Fy = 160,000 psi, fy = 144,000 psi.

combined $\frac{fa}{Fa} + \frac{fb}{Fb} \leq 1$(AISC)

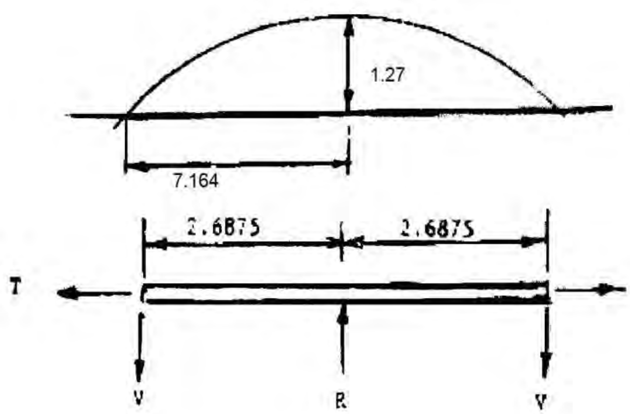


$a = 3.875 \times 3$ (links) $+ 1.5 \times 2 = 14.625$ in.

$$A^\circ = 57.29578 \times \frac{14.625}{20.845} = 40.2^\circ$$

$$c = 2 \times 20.845 \sin \frac{40.2}{2} = 14.327 \text{ in.}$$

$$b = \frac{14.327}{2} \tan \frac{40.2}{4} = 1.27 \text{ in}$$



$$\tan \sphericalangle = \frac{1.27}{7.1635} = 0.1773 \therefore \sphericalangle = 10.05^\circ$$

$T = P \cos 10.05^\circ = (126{,}000)(.9846) = 124{,}065$ lb.
$V = P \sin 10.05^\circ = (126{,}000)(.1745) = 22{,}000$ lb.

$A = 2\,(1.5)^2\,(\pi/4) = 3.53$ sq in.

$$S = \frac{(\pi)\,(1.5)^3\,(2)}{32} = 0.663 \text{ cu in.}$$

$$T = 124{,}065/3.53 = 35{,}145 \text{ lb/sq in.}$$

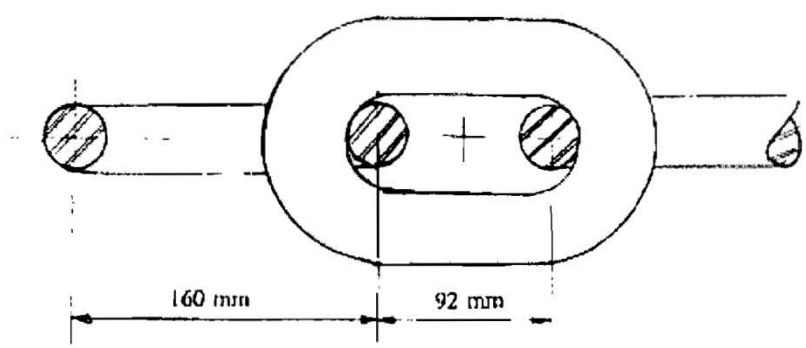
$$M = 2\,\frac{(22000)\;(5.375)}{4} = 59{,}125 \text{ lb in.}$$

$$v = \frac{M}{S} = \frac{59{,}125}{0.663} = 89{,}178 \text{ lb/sq in}$$

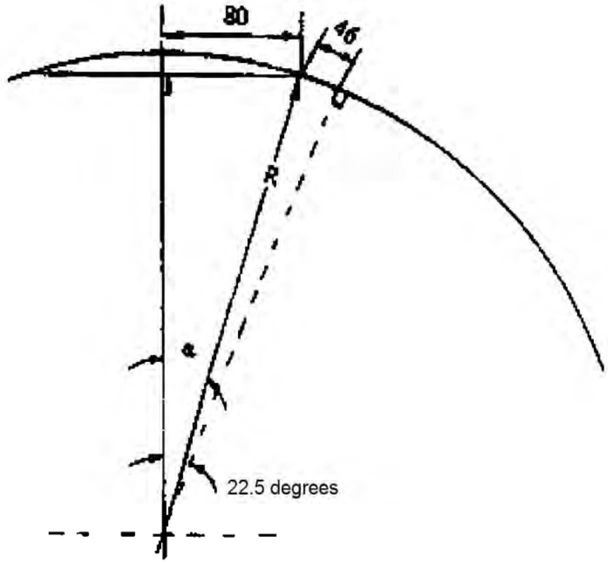
$$v \text{ combined} = \frac{35145}{144{,}000} + \frac{59{,}125}{144{,}000} = 0.65 < 1 \text{ (o.k.)}$$

b. Determine Pocket Wheel Pitch Diameter

(1) Based on Chain geometry
34 x 126 chain (DIN 22252)



Reference to shop drawing (Columbus McKinnon)

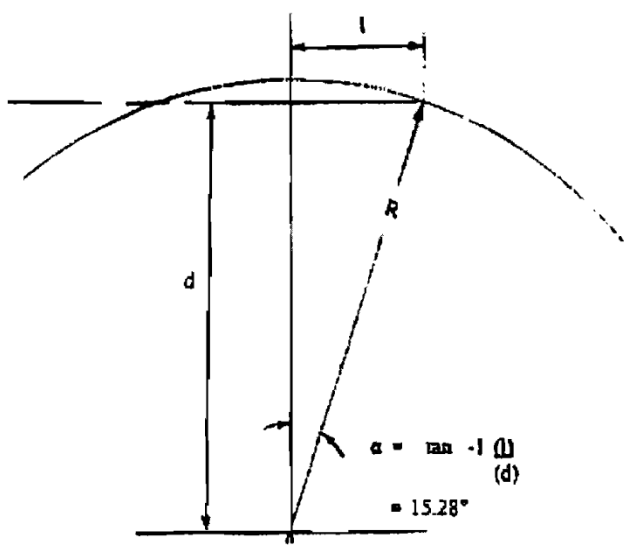


$$R = \frac{80}{\sin\alpha} = \frac{46}{\sin(22.5 - \alpha)}$$ solving for $\alpha = 14.32°$ and $R = 323.39$ mm

$$PD = 2R = 2 \times \frac{323.39}{25.4} = 25.46 \text{ in. (Pitch diameter based on chain geometry)}$$

(2) Based on Pocket Wheel Geometry

Reference to shop drawing 8-pocket sprocket (Columbus McKinnon)



$$d = \begin{pmatrix}\text{distance from Cutter face} \\ \text{to C.L. of Pocket Wheel bore}\end{pmatrix} + \begin{pmatrix}\text{radius of} \\ \text{chain stock}\end{pmatrix}$$

$$d = 11.54 + \frac{34\text{mm}}{2 \times 25.4} = 12.2093 \text{ in.}$$

$$l = \begin{pmatrix}\text{center of pocket} \\ \text{to center of cutter}\end{pmatrix} + \begin{pmatrix}\text{radius of} \\ \text{cutter}\end{pmatrix} - \begin{pmatrix}\text{radius of} \\ \text{chain stock}\end{pmatrix}$$

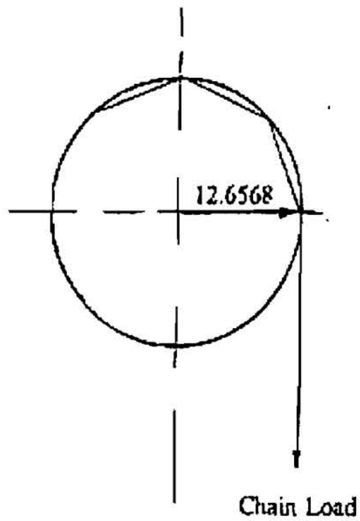
$$l = 1.845 + \frac{4.32}{2} - \frac{34 \text{ mm}}{2 \times 25.4} = 3.3357 \text{ in.}$$

$$R = \sqrt{(12.2093^2 + 3.3357^2)} = 12.6568 \text{ in.}$$

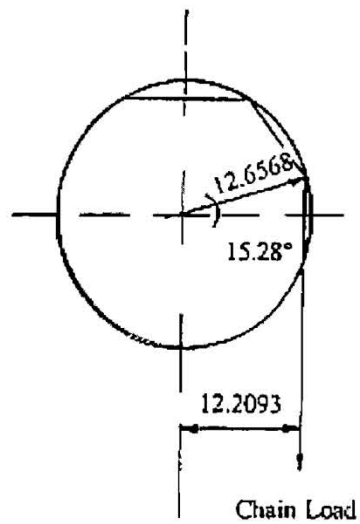
PD = 2R = 25.31 in. (Pitch diameter based on pocket wheel geometry)

(3) Pocket Wheel Pitch Diameter

The difference between the pitch diameter based on chain geometry and that based on the pocket wheel geometry is an indication of the slop desired to permit free engagement and disengagement of the chain links to the wheel pockets as the wheel rotates. The radius at which the load acts varies as the wheel rotates from a maximum of 12.6568 to a minimum of 12.2093, as determined by the computation based on pocket wheel geometry.



Position of pocket wheel
a max. radius



Position of pocket wheel
a min radius (15.28° rotation)

For purposes of determining speed of rotation and torque requirements for hoist equipment, a radius of 12.6568 will be used. This gives a pitch diameter of 25.3 in.

c. Presentation of Hoist Capacity

$$\text{Pocket Wheel r.p.m.} = \frac{\text{hoist speed x 12 in/ft.}}{\pi \text{ x pocket wheel dia-in}} \qquad \text{(Equation 1)}$$

$$\text{Gear train reduction} = \frac{\text{motor speed - rpm}}{\text{pocket wheel - rpm}} \qquad \text{(Equation 2)}$$

(1) Design Conditions

	MOVING		STALLED
	Normal	Peak	Max
Total load on chains:	120k (assumed)	136k	336k
Per side of gate:	60k(1)	68k	168k
Factor of Safety:	5(1)	3	-
Maximum unit stress:	-	-	75% yield
% Motor rated torque:	100%	≤ 115% Continuous	≤ 280%
Nominal hoist speed:	1.0 FPM	< 1.0 FPM	

Reference: EM 1110-2-2702

(2) Power Equation:

$$HP = \frac{vL}{33 \times \eta}$$

where: v = Chain speed (FPM)
L = Load (LBS)
η = efficiency of powertrain

η (powertrain) = η (triple box) x η (double box) x η (open gearing, 1 reduction)

Open gearing consists of the following:

- chain/pocket wheel efficiency 0.90 assumed
- spur gear set 0.97
- two sets of antifriction bearings @ 0.98 ea 0.98^2

Efficiencies quoted by gearbox manufacturers are typically high, therefore, use the following:

η (triple box) = 0.90
η (double box) = 0.95

η (powertrain) = 0.90 x 0.95 x 0.90 x 0.97 x 0.98^2 = 0.7169

Use 0.72

(3) Sample Calculations

$$\text{Solve for HP} = \frac{1.0 \times 120}{33 \times 0.72} = 5.05 \text{ say } \underline{5 \text{ HP}}$$

$$\text{and } v = \frac{5 \text{ HP} \times 33 \times 0.72}{120} = .99 \text{ say } \underline{1 \text{ FPM}}$$

d. Chain Locker Dimensions.

V_L required = Required volume of chain locker

where V_L required = $0.85\ d^2\ L$

and d = chain link diameter - in
L = chain length /6 - fathoms

Sample calculation:

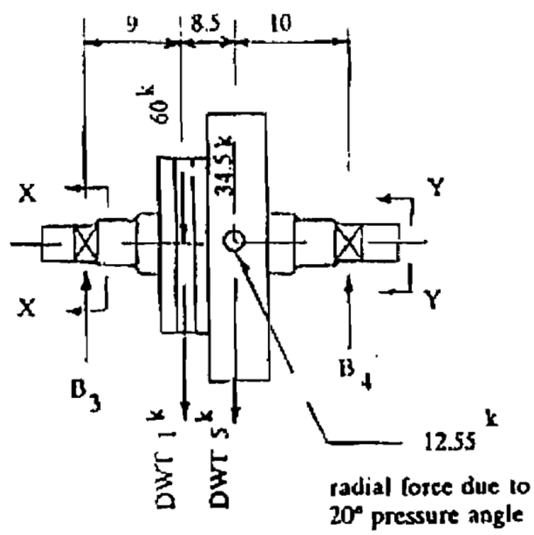
Assume 1½ in. diameter chain, 46 ft. in length

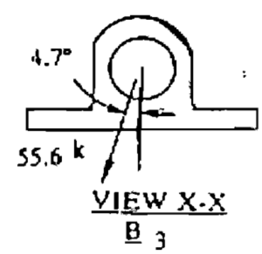
$$V_L \text{ required} = 0.85\ (1.5)^2 \left(\frac{46}{6}\right) = 14.7 \text{ cu. ft.}$$

The following tabulation showing chain locker depth for various selected diameters is helpful in determining the desired locker size, based on machinery locations and space limitations.

Dia-in.	Depth-in.
12	225
18	100
24	56
30	36

e. Bearing Selection - Pocket Wheel Shaft Load Computation



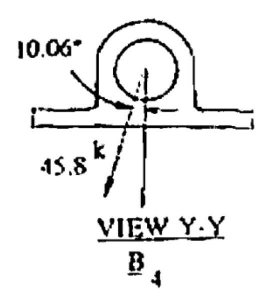


Check for normal load based on 60k chain load and FOS. = 5

$$\text{B3 vertical force} = \frac{(61)(18.5) + (39.5)(10)}{27.5} = 55.4\text{k}$$

$$\text{B3 horizontal force} = \frac{(12.55)(10)}{27.5} = 4.55\text{k}$$

$$\text{Resultant force} = 55.6\text{k} \ @ \ \tan^{-1}\frac{(4.55)}{(55.4)} = 4.7°$$



$$\text{B4 vertical force} = \frac{(39.5)(17.5) + (61)(9)}{27.5} = 45.1\text{k}$$

$$\text{B4 horizontal force} = \frac{(12.55)\,(17.5)}{27.5} = 8\text{k}$$

$$\text{Resultant force} = 45.8\text{k} \ @ \ \tan^{-1}\frac{(8)}{(45.1)} = 10.06°$$

(1) Similar computations should be made for chain loads based on a unit stress not in excess of 75% of the yield point of the material as called for in EM 1110-2-2702. Bearing sizing philosophy will be to specify for the highest resultant force as the minimum static load capacity required.

(2) The bearings should have a life expectancy requirement of 10,000 hours B-10 life with loads assumed equal to 75% of the maximum load.

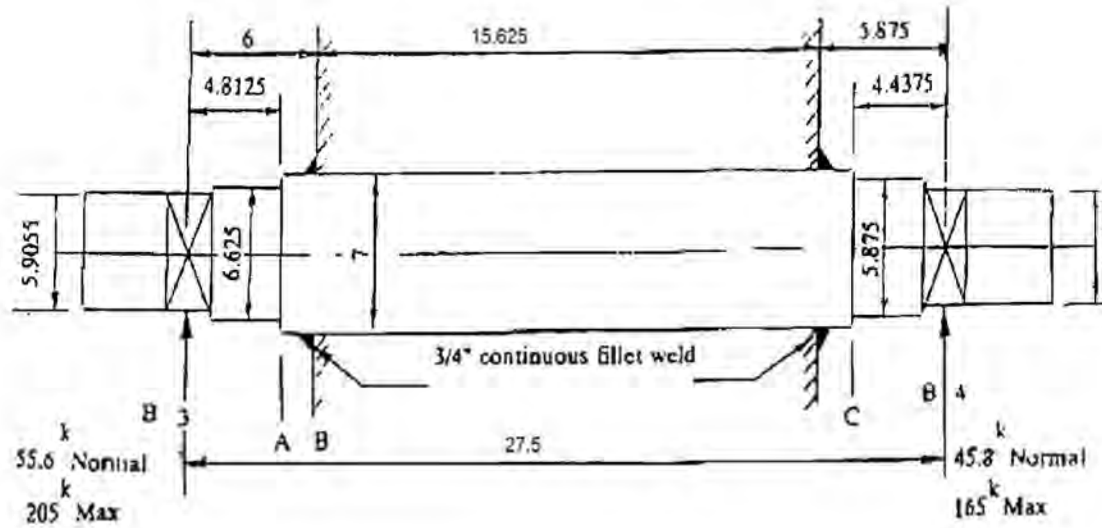
(3) End unit bearings may be subjected to a range of loads corresponding to the following:

a. Even-split of NORMAL rated load to each end unit.

b. Even-split of motor stall torque (½) (2.8 x normal rated load) to each end unit.

c. Maximum UNEVEN - split of motor stall torque to each end unit.

f. Pocket Wheel Shaft - Stresses and Material Selection.



(1) Compute Normal stresses at sections 'A', 'B' & 'C'.

	Section A	Section B	Section C
Area	34.5 in^2	38.5 in^2	27.1 in^2
Section Modulus ($\pi d^3/32$)	28.5 in^3	33.7 in^3	19.9 in^3
Normal Bending Moment	267.6 kp-in	333.6 kp-in	203.2 kp-in
Bending Stress (M/Z)	9.4 ksi	9.9 ksi	10.2 ksi
Shear Stress (V/A)	1.6 ksi	1.4 ksi	1.7 ksi

Combine stresses for 'C'

$$\tau_{max} = \sqrt{\left(\frac{10.2}{2}\right)^2 + (1.7)^2} = 5.38\ ksi$$

$$\sigma = \left(\frac{10.2}{2}\right) + 5.38 = 10.5\ ksi$$

(2) Compute Maximum stresses at sections 'A', 'B' & 'C'.

	Section A	Section B	Section C
Bending	$(9.4)\frac{(205)}{(55.6)} = 34.7$ ksi	$(9.9)\frac{(205)}{(55.6)} = 36.5$ ksi	$(11.4)\frac{(165)}{(45.8)} = 36.8$ ksi
Shear	$(1.6)\frac{(205)}{(55.6)} = 5.9$ ksi	$(1.4)\frac{(205)}{(55.6)} = 5.2$ ksi	$(1.7)\frac{(165)}{(45.8)} = 6.1$ ksi

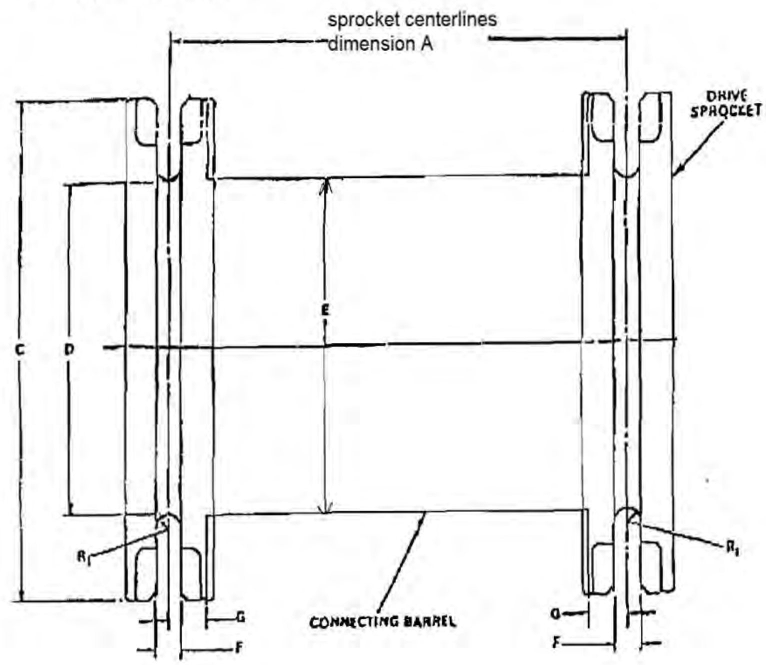
Combine stresses for 'C'

τ max = 19.4 ksi; σ = 37.8 ksi

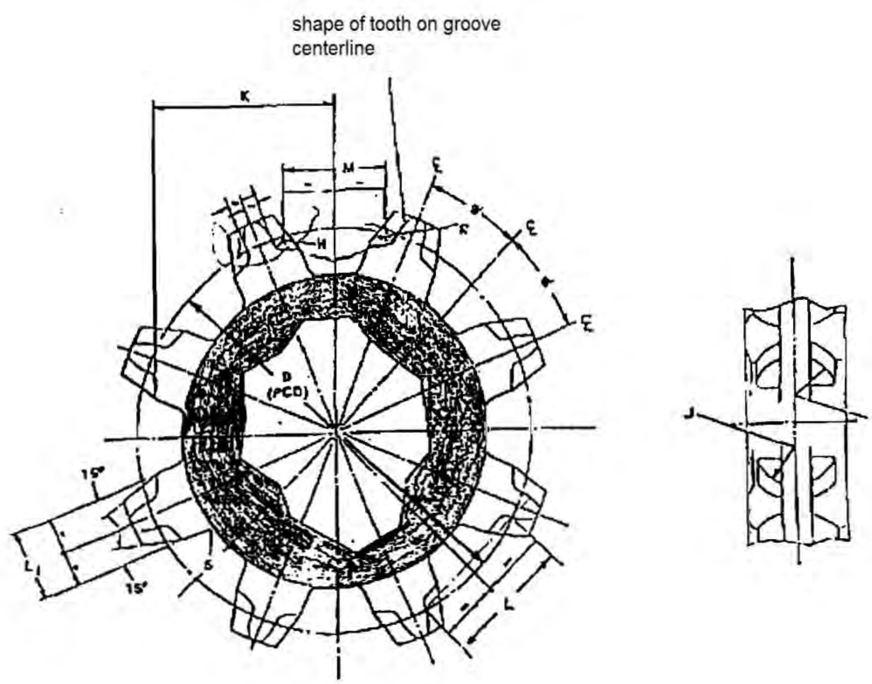
Use: ASTM A-668 Class E (UTS = 85 ksi, Fy = 44 ksi)
with supplementary requirement S4 (carbon content for welding)

F.O.S. = $\frac{85}{10.5} = 8.1$ and Fy = $\frac{37.8}{44} = 86\%$

g. Sprocket Assembly - Design Formula



Typical Sprocket Assembly



Sprocket Ring Profile

(1) Dimension B - Pitch circle diameter (theoretical)

$$B = \sqrt{\frac{P^2}{\sin^2\left(\frac{\theta}{2}\right)} + \frac{d^2}{\cos^2\left(\frac{\theta}{2}\right)}}$$

where:

d = Nominal diameter of chain link material.

P = Nominal pitch of chain link.

$\theta = \frac{360}{2N}$ degrees

N = Number of teeth in sprocket

The value for B obtained to be taken to nearest lower whole number.

(2) Dimension C - Overall diameter (reference)
C = B + 2d

NOTE - Actual diameter to be agreed between purchaser and manufacturer.

(3) Dimension D - Groove diameter
D = Diameter under vertical chain links minus a diametral clearance.

NOTE: Actual diameter to be agreed between purchaser and manufacturer.

(4) Dimension E - Barrel diameter
E = 2K + d – 2 (Bolt center line to bottom of scraper bar) – 5

(5) Dimension F - Sprocket groove width
F = 1.25d

(6) Dimension G - Groove center line to inside face of sprocket recess

$$G = b_t - (0.5e + 0.5V_u + 3.5)$$

where:

e = Diameter of nut across corners.

V_u = Clearance between bolt and hole of shackle connectors.

b_t = Chain center to hole center of shackle connector.

Dimension to be maintained in vicinity of nut and bolt only.

(7) Dimension H - Root radius
$H = 0.5d$

(8) Dimension J - Pocket plan radius (nominal)
J = Maximum outer radius of shackle connector and is measured on a line K + 0.5d from sprocket center line.

NOTE: If a working clearance is required it should be agreed to between purchaser and manufacturer.

(9) Dimension K - Height from sprocket center to bottom of the pocket

$$K = 0.5\left[\frac{P}{\tan\left(\frac{\theta}{2}\right)} - d\tan\left(\frac{\theta}{2}\right)\right] - 0.5d$$

The values for K obtained to be taken to nearest half millimeter.

(10) Dimension L - Length of pocket
$L = 1.075\ P + 2d$

(11) Dimension M - Pocket centers (reference)
$M = 1.075\ P + d$

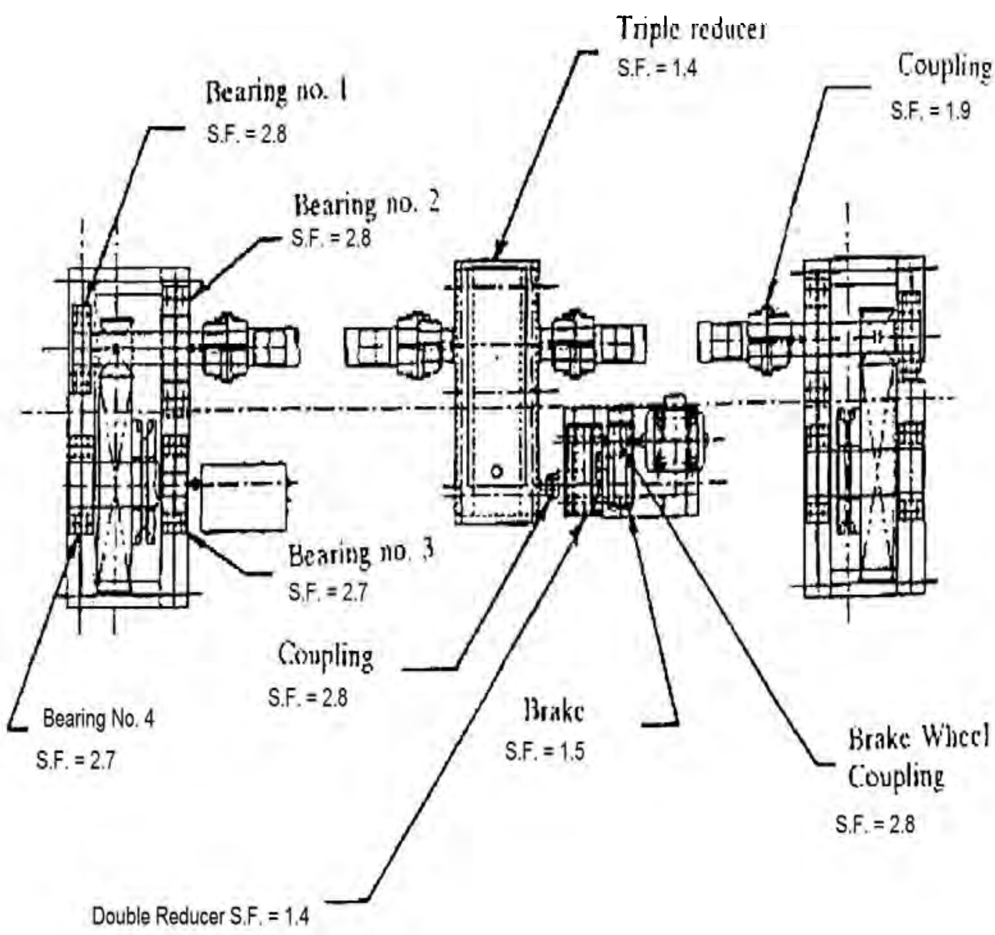
(12) Dimension R - Tooth flank radius (reference)
$R = P - 1.5d$

Radius to be struck from a line which is K + 0.5d from sprocket center line

(13) Dimension R1 - <u>Groove radius</u>
R1 = 0.5d

(14) Dimension S - <u>Radius at root of tooth stub</u>
S = 0.5d

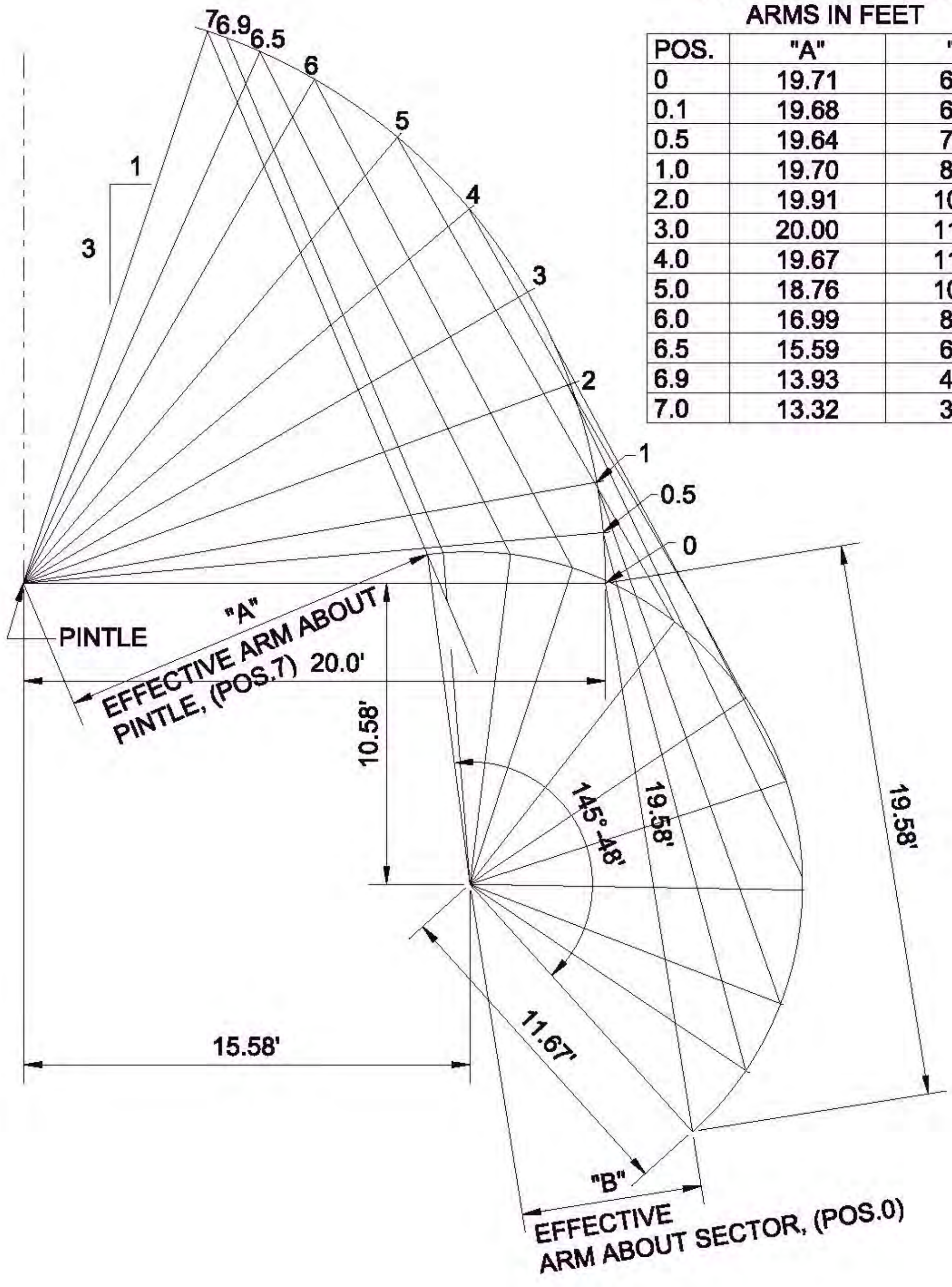
h. Typical Hoist Arrangement - Summary of Service Factors

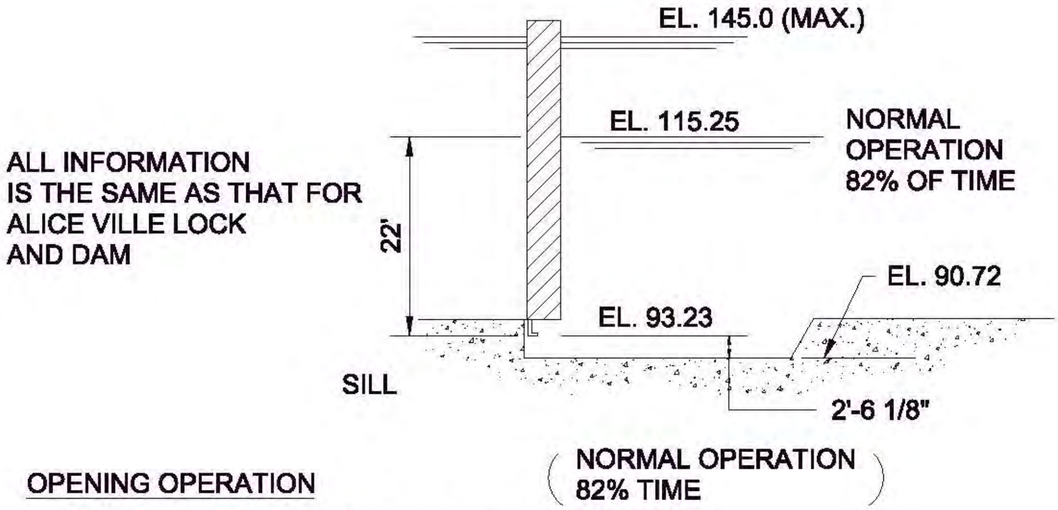


KINEMATICS
OHIO RIVER TYPE LINKAGE

EFFECTIVE MOMENT ARMS IN FEET

POS.	"A"	"B"
0	19.71	6.16
0.1	19.68	6.46
0.5	19.64	7.57
1.0	19.70	8.73
2.0	19.91	10.48
3.0	20.00	11.47
4.0	19.67	11.62
5.0	18.76	10.80
6.0	16.99	8.63
6.5	15.59	6.70
6.9	13.93	4.19
7.0	13.32	3.20





OPENING OPERATION

OPENING TIME 90 SEC (1 1/2 MIN)
UNDERGATE CLEARANCE = 2'-6 1/8"
SUBMG (ASSUMED) = 22 FT

CYLINDER DIAM = 12" I.D.
ROD DIAM. = 5.0'
STROKE = 12'-8 5/8"
OHIO TYPE LINKAGE

* TECHNICAL REPORT
NO. 2-651, JUNE' 64,

NOMENCLATURE		WES MODEL		PROPOSED LOCK ALICE VILLE		FACTOR
LENGTH GATE	-	3.0	L_1	60.0 FT	$\frac{L_1}{L}$	= 20
SUB'MG	S S_2	2.0 * (SEE PLATE B-64) 40.0	S_1	22.0	$\frac{S_1}{S_2}$	= 0.55
TIME	T T_2	20.1 SEC'S (SEE PLATE B-64) * 90.0 SEC	T_1	90.0		
ARC OF TRAVEL GATE	K	71.567	K_1	71.565	$\frac{T_2}{TA}$	$= \frac{90}{90}$

FACTOR $L_1 \div L = 60 \div 3 = 20$
S_1 = SUB. = 22 FT S. = S (L_1+L) = 2x20 = 40
T_1 = 90 SEC ADJ. $T_2 = T\sqrt{L_1/L} = 20.1\sqrt{20} = 20.1 \times 4.47 = 90$

ANGULAR TRAVEL $TA = T_1 \left(\frac{K}{K_1} \right) = 90 \left(\frac{71.565}{71.565} \right) = 90$

FACTOR $\frac{T_2}{T_A} = \frac{90}{90} = 1$

$$P_1 = P_0 \left(\frac{L_1}{L} \right)^4 \left(\frac{S_1}{S_2} \right)^{2.1} \left(\frac{T_2}{T_A} \right)^{1.0} \text{x BOTTOM EFFECT FACTOR}$$

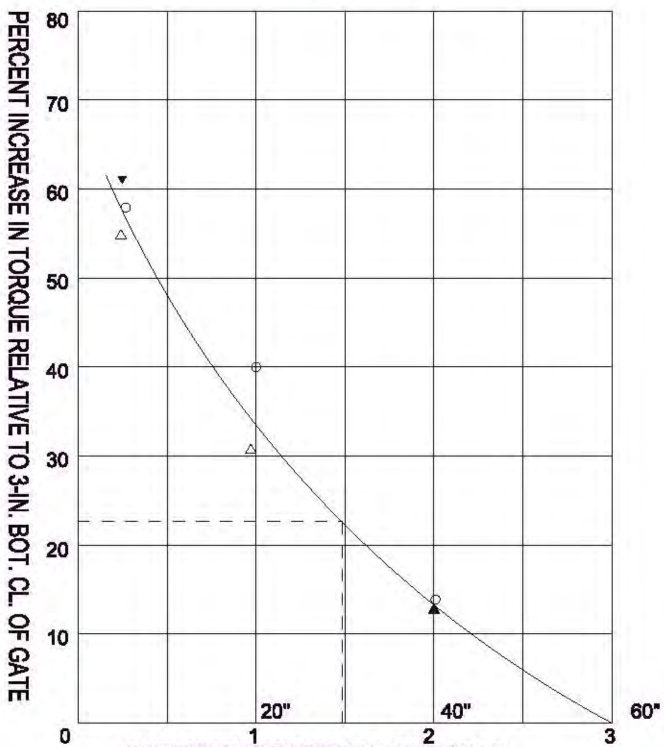
$$= P \left(20 \right)^4 \left(0.55 \right)^{2.1} \left(\frac{90}{90} \right)^{1.0} \text{x } 1.23 = \text{Px160,000x0.285x1x1.23}$$

$$= P \text{ x } 45{,}600 \text{ x } 1.23$$

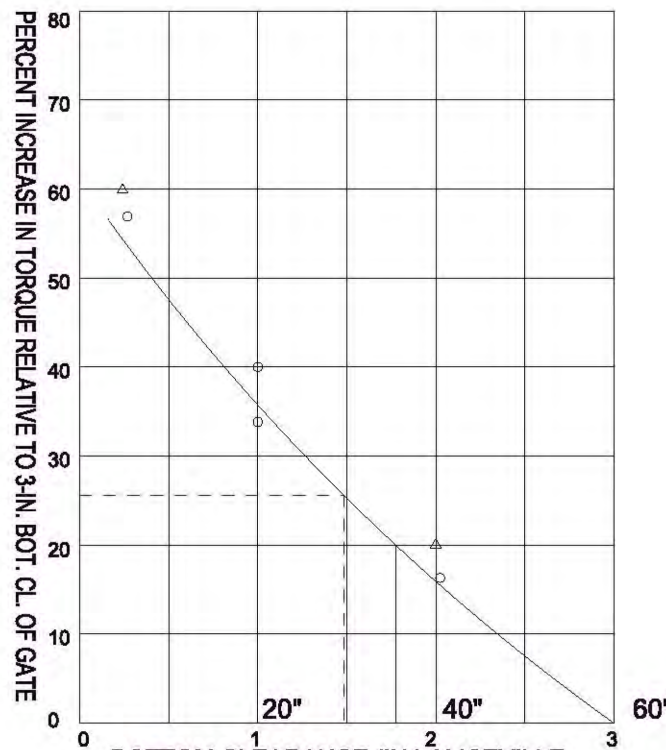
$$= P \text{ x } 56{,}100$$

EFFECT OF BOTTOM CLEARANCE SEE PLATE 52
OF REPORT 2-651 OPEN'G CYCLE - INCREASE TORQUE
23% FACTOR 1.23, CLOSING CYCLE - INCREASE TORQUE
27% FACTOR 1.27
AVAILABLE TORQUE BASED ON 900-PSI NET CYL. PRESS.

L = 3'-0" FOR MODEL STUDY



OPENING CYCLE



CLOSING CYCLE

$\frac{L_1}{L} = \frac{60}{3}$ = 20:1 ADJUST BOTTOM CLEARANCE IN RELATION TO SCALE RATIO

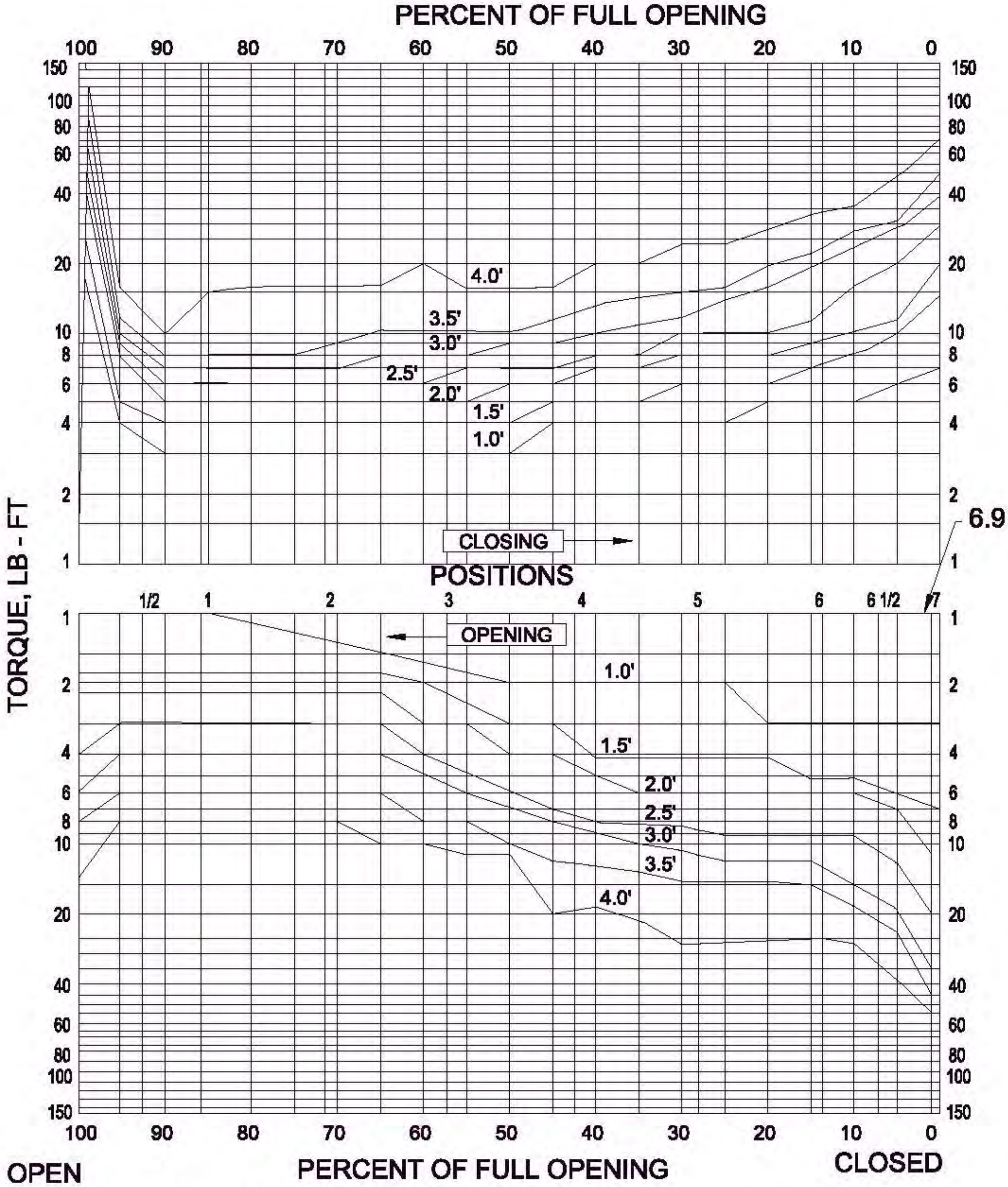
OPERATING TIME TORQUE LB-FT (3-IN BOT. CLEAR)

SYMBOL	SEC	OPENING CYCLE	CLOSING CYCLE
○	13.4	40	35
△	20.1	29	20
▽	26.8	18	12

CLEARANCE UNDER
ALICEVILLE GATE = 2'-6 1/8"
(30-1/8")

RELATIVE EFFECT OF
GATE BOTTOM CLEARANCE
ON TORQUE
4.0-FT SUBMERGENCE

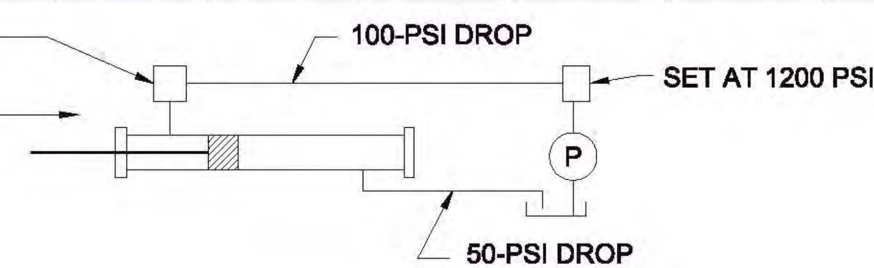
OHIO RIVER LINKAGE
EFFECT OF SUBMERGENCE ON INSTANTANEOUS TORQUE
OPERATING TIME, 20.1 SEC



SUBMERGENCE-AS SHOWN IN FEET
CHAMBER LENGTH
UPSTREAM - 25 FT
DOWNSTREAM - 25 FT
BOTTOM CLEARANCE 0.25 FT

OPENING OPERATION

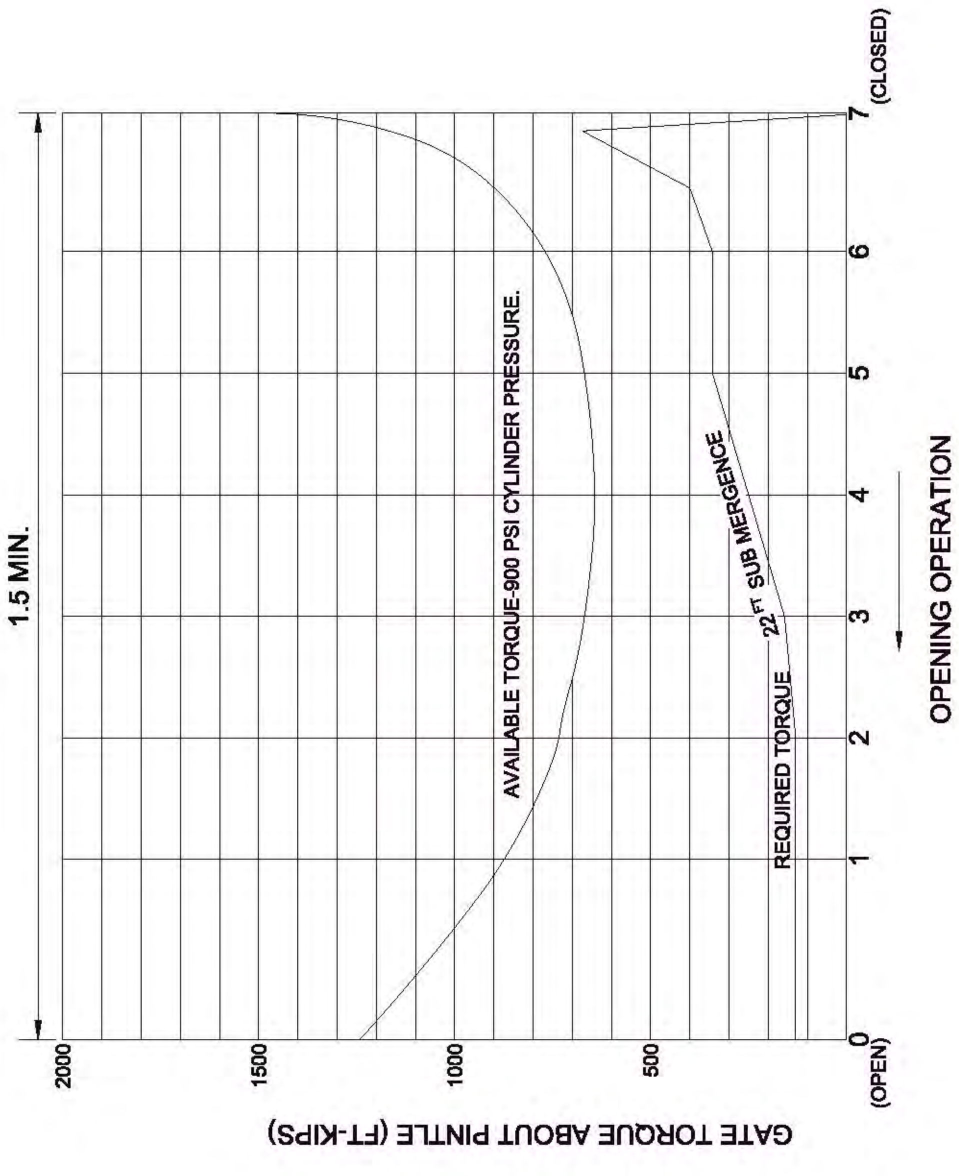
	REQUIRED LOADS										AVAILABLE TORQUE				
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
POSITION	WES TORQUE ABOUT PINTLE LB-FT	CORR. FACTOR	TORQUE ABOUT PINTLE (ALICE-VILLE) ③=①X② LB-FT	EFFECT. ARM ABOUT PINTLE FT	FORCE IN STRUT ⑤=③/(④x1000) KIPS	EFFECT. ARM ABOUT SECTOR FT	TORQUE ABOUT SECTOR (0.95 EFF.) ⑦=(⑤x⑥)/0.95 FT-KIPS	SECTOR GEAR RADIUS FT	FORCE ON PISTON (0.95 EFF.) ⑨=⑦/(⑧x.95) KIPS	CYL PRESS. (93.5□") ⑩=(⑨x1000.)/93.5 PSI +60 LOSS PSI	CYL PRESS. (93.5□") PSI	FORCE ON PISTON (0.95 EFF.) ⑫=⑪x93.5 x 0.95 KIPS	TORQUE ABOUT SECTOR (0.95 EFF.) ⑬=⑫x⑧ x 0.95 FT-KIPS	FORCE IN STRUT ⑭=⑬/⑥ KIPS	TORQUE ABOUT PINTLE ⑮=⑭x④ FT-KIPS
0	2.2	$56x10^3$	123,000	19.71	6.2	6.16	40.	5.0	8.4	150	900	79.9	379.5	61.6	1210
0.5	2.2		123,000	19.64	6.3	7.57	50.		10.5	170				50.1	980
1	2.2		123,000	19.70	6.2	8.73	57.		12.0	190				43.5	860
2	2.2		123,000	19.91	6.2	10.48	68.		14.3	210				36.2	720
3	3.0		168,000	20.00	8.4	11.47	101.		21.3	290				33.1	660
4	4.5		250,000	19.67	12.7	11.62	155.		32.6	410				32.7	640
5	6.0		340,000	18.76	18.1	10.80	206.		43.4	520				35.1	660
6	6.0		340,000	16.99	20.	8.63	182.		38.3	470				44.0	750
6.5	6.7		380,000	15.59	24.	6.70	169.		35.6	440				56.6	880
6.9	12.0		670,000	13.93	48.	4.19	212.		44.6	540				90.6	1260
7	0		0	13.32	0	3.20	0		0	0				119.0	1590



PRESSURE ON ROD END
DUE TO RETURN LINE FRICTION
= 50 PSIx113.1□"/93.5□" = 60 PSI
EFFECTIVE PRESSURE = 960-60 = 900 PSI

OHIO RIVER LINKAGE
MITER GATE MACHINERY
OPERATING DATA
OPENING

OHIO RIVER LINKAGE
GATE TORQUE ABOUT PINTLE
OPENING TIME
1.5 MIN



OPENING TIME = 1.5 MIN (ASSUMED)

$$\text{FLOW RATE} = \frac{(12^2 - 5^2)\ 0.7854 \times 12 \times 12.7234}{231 \times 1.5\ (\text{MIN})} = 41.2 \text{ GPM}$$

$$\text{CLOSING TIME} = \frac{12^2 \times 0.7854 \times 12 \times 12.7234}{41.2 \times 231} = 1.82 \text{ MIN.}$$

CLOSING OPERATION

NOMENCLATURE		WES MODEL		PROPOSED LOCK ALICE VILLE		FACTOR
TIME	T T_2	20.1 SEC'S 90 SEC'S	T_1	110		
ARC OF TRAVEL	K	71.567°	K_1	71.565°	$\frac{T_2}{T_A}$	$\frac{90}{110} = 0.82$

TIME T_1 = 110 SEC

ADJUST $T_2 = T\sqrt{\frac{L_1}{L}} = 20.1\sqrt{20} = 20.1 \times 4.47 = 90$

ADJ. ANGULAR TRAVEL $TA = T_1\left(\frac{K}{K_1}\right) = 110 \times 1 = 110$

$$\text{FACTOR} = \frac{T_2}{T_A} = \frac{90}{110} = 0.82$$

$$P_1 = P\left(\frac{L_1}{L}\right)^4\left(\frac{S_1}{S_2}\right)^{1.5}\left(\frac{T_2}{T_A}\right)^{1.0} \quad \text{(BOTTOM EFFECT)}$$

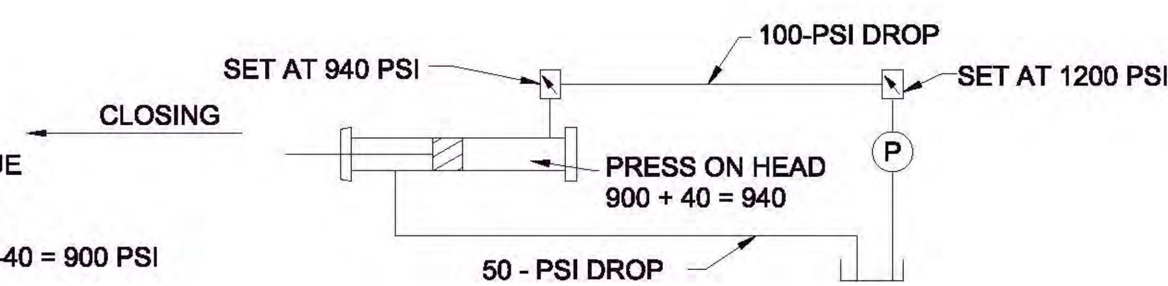
$$= P\,(20)^4\,(0.55)^{1.5}(0.82)^{1.0} \times \left(\begin{array}{c}\text{INCREASE DUE TO}\\ \text{BOTTOM EFFECT}\end{array}\right)$$

$$= P \times 160{,}000 \times 0.408 \times 0.82 \times 1.27$$

$$= P \times 68{,}000$$

CLOSING OPERATION

POSITION	REQUIRED LOADS										AVAILABLE TORQUE				
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
	WES TORQUE ABOUT PINTLE LB - FT	CORR. FACTOR	TORQUE ABOUT PINTLE (ALICE-VILLE) $③ = ① \times ②$ LB -FT	EFFECT. ARM ABOUT PINTLE FT	FORCE IN STRUT $⑤ = \frac{③}{④ \times 1000}$ KIPS	EFFECT. ARM ABOUT SECTOR FT	TORQUE ABOUT SECTOR (0.95 EFF.) $⑦ = \frac{⑤ \times ⑥}{0.95}$ FT - KIPS	SECTOR GEAR RADIUS FT	FORCE ON PISTON (0.95 EFF.) $⑨ = \frac{⑦}{⑧ \times .95}$ KIPS	CYL PRESS. (113.1□") $⑩ = \frac{⑨ \times 1000}{113.1}$ +40 LOSS PSI	CYL PRESS. (113.1□") PSI	FORCE ON PISTON (0.95 EFF.) $⑫ = ⑪ \times 113.1 \times 0.95 \times \frac{1}{1000}$ KIPS	TORQUE ABOUT SECTOR (0.95 EFF.) $⑬ = ⑫ \times ⑧ \times 0.95$ FT - KIPS	FORCE IN STRUT $⑭ = \frac{⑬}{⑥}$ KIPS	TORQUE ABOUT PINTLE $⑮ = ⑭ \times ④$ FT - KIPS
0	0	68×10^3		19.71		6.16		5.0		0	900	96.7	459.	74.5	1470
0.1	?0		2700×10^3	19.68	137.	6.46	930		195	1760				71.0	1400
0.5	6.3		430×10^3	19.64	22.	7.57	175		37.	370				60.6	1190
1.0	5.0		340×10^3	19.70	17.3	8.73	159		33.	330				52.6	1040.
2	5.0		340×10^3	19.91	17.1	10.48	189		40.	390				43.8	870
3	5.0		340×10^3	20.00	17.0	11.47	205		43.	420				40.0	800
4	6.4		440×10^3	19.67	22.	11.62	270		57.	540				39.5	780
5	8.0		540×10^3	18.76	29.	10.80	330		69.	650				42.5	800
6	9.0		610×10^3	16.99	36.	8.63	330		69.	650				53.2	900
7	20.0		1360×10^3	13.32	102	3.20	340		72.	680				143.4	1910.

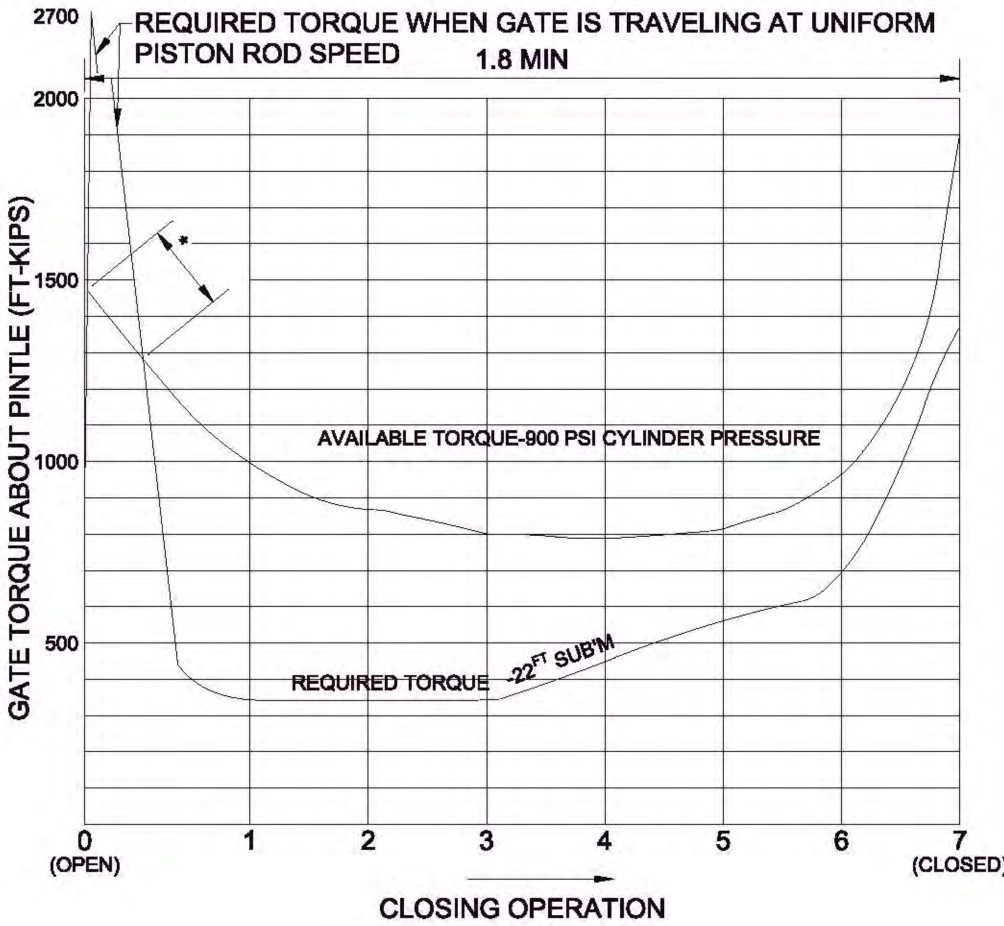


PRESSURE ON HEAD END DUE
TO RETURN LINE FRICTION =
50 X 93.5□" / 113.1□" = 40 PSI ±
EFFECTIVE PRESSURE = 940-40 = 900 PSI

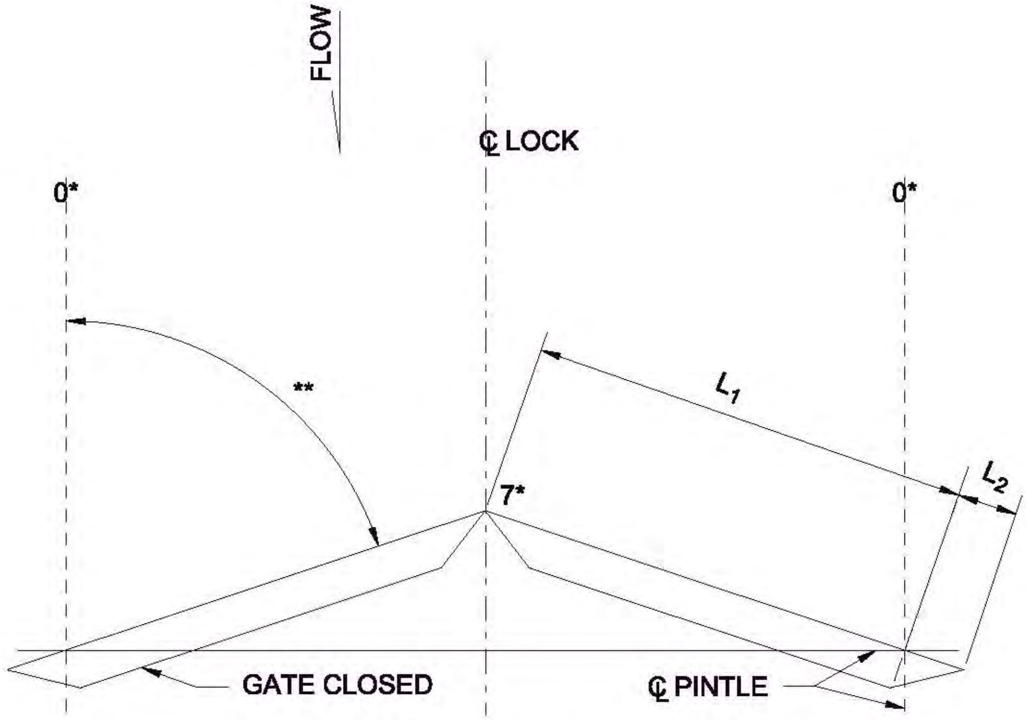
OHIO RIVER LINKAGE
MITER GATE MACHINERY
OPERATING DATA
CLOSING

OHIO RIVER LINKAGE
GATE TORQUE ABOUT PINTLE
CLOSING TIME
1.8 MIN

* WHEN REQUIRED GATE TORQUE EXCEEDS AVAILABLE TORQUE, SOME HYDRAULIC OIL WILL BE BYPASSED AT RELIEF VALVE AND THE GATE WILL AUTOMATICALLY TRAVEL AT A SPEED THAT WILL REDUCE THE REQUIRED GATE TORQUE TO THE GATE TORQUE AVAILABLE FROM 900 PSI EFFECTIVE CYLINDER PRESSURE.



TEMPORAL LOADING



** AREA OF TRAVEL WHERE NORMAL LOADS APPLY

* POSITION WHERE TEMPORAL LOADS APPLY

$$T_t = \frac{62.4}{4}\left(L_1^2 - L_2^2\right)\left(h_2^2 - h_1^2\right)$$

T_t = GATE TORQUE ABOUT PINTLE DUE TO TEMPORAL LOAD (FT-LB)

L_1 = GATE LENGTH PINTLE TO MITER (FT)

L_2 = GATE LENGTH PINTLE TO QUOIN (FT)

h_1 = GATE SUBMERGENCE (FT)

h_2 = GATE SUBMERGENCE PLUS TEMPORAL HEAD (FT)

TEMPORAL FORCES EXAMPLE COMPUTATIONS

$$T_t = \frac{62.4}{4}(L_1^2 - L_2^2)(h_2^2 - h_1^2)$$

FOR ALICEVILLE LOCKS

L_1 = 718.34" = 59.86'

L_2 = 21.65" = 1.804'

h_1 = 22'

h_2 = 23.25'

$$T_t = \frac{62.4}{4}\left[(59.86)^2-(1.804)^2\right]\left[(23.25)^2-(22)^2\right]$$

$$T_t = \frac{62.4}{4}\left[3579.96\right]\left[56.56\right]$$

T_t = 3,159,000 FT-LB

REFERENCE PLATE B-79, SHEET 9, POSITION 7

CALCULATE $\longrightarrow$ THE FORCE IN THE STRUT FOR A PINTLE TORQUE OF 3,159,000 FT-LB

$$\text{STRUT FORCE} = \frac{3{,}159{,}000}{13.32} = 237{,}000 \text{ LB}$$

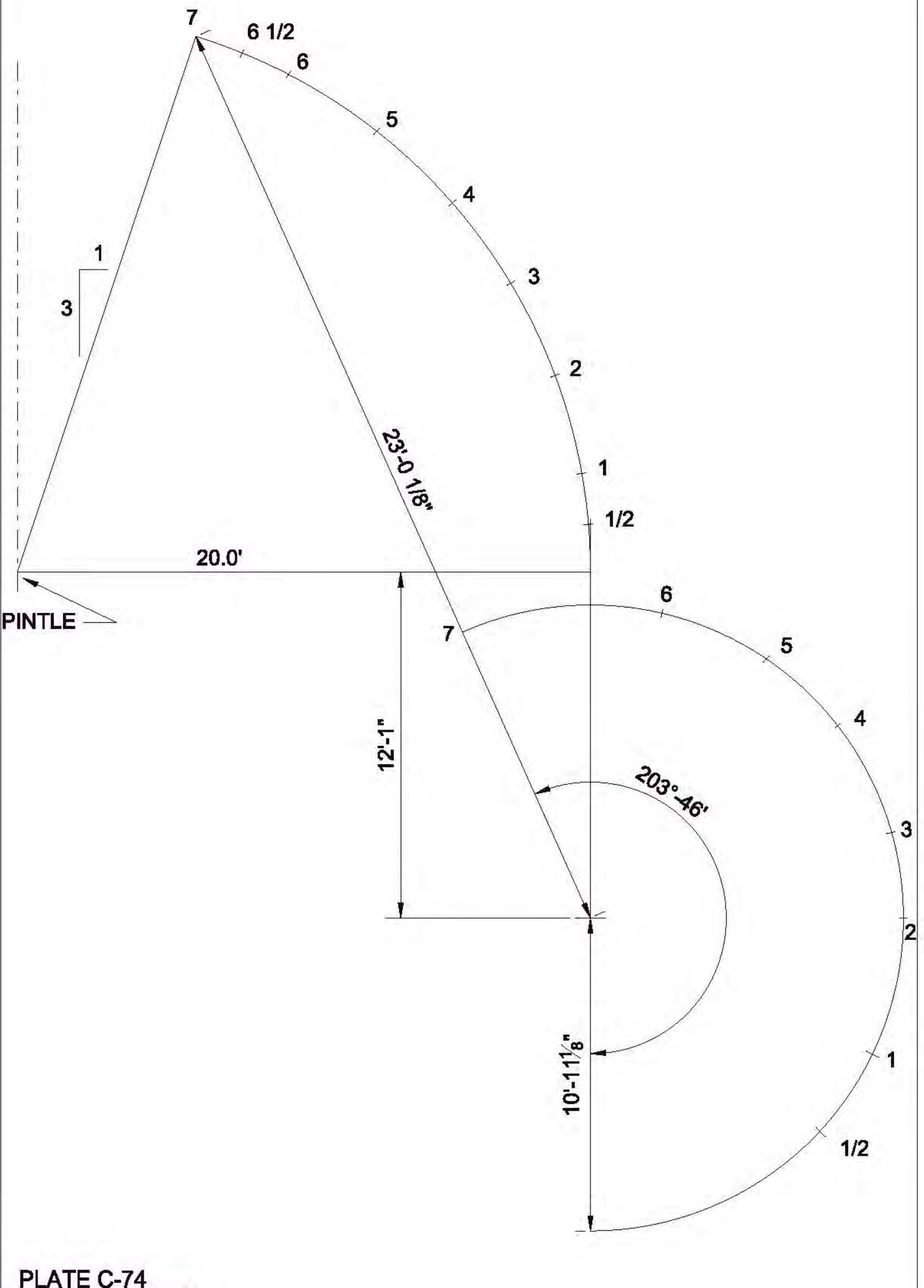
$$\text{TORQUE ABOUT SECTOR} = \frac{237{,}000}{0.95} = 798{,}000 \text{ FT-LB}$$

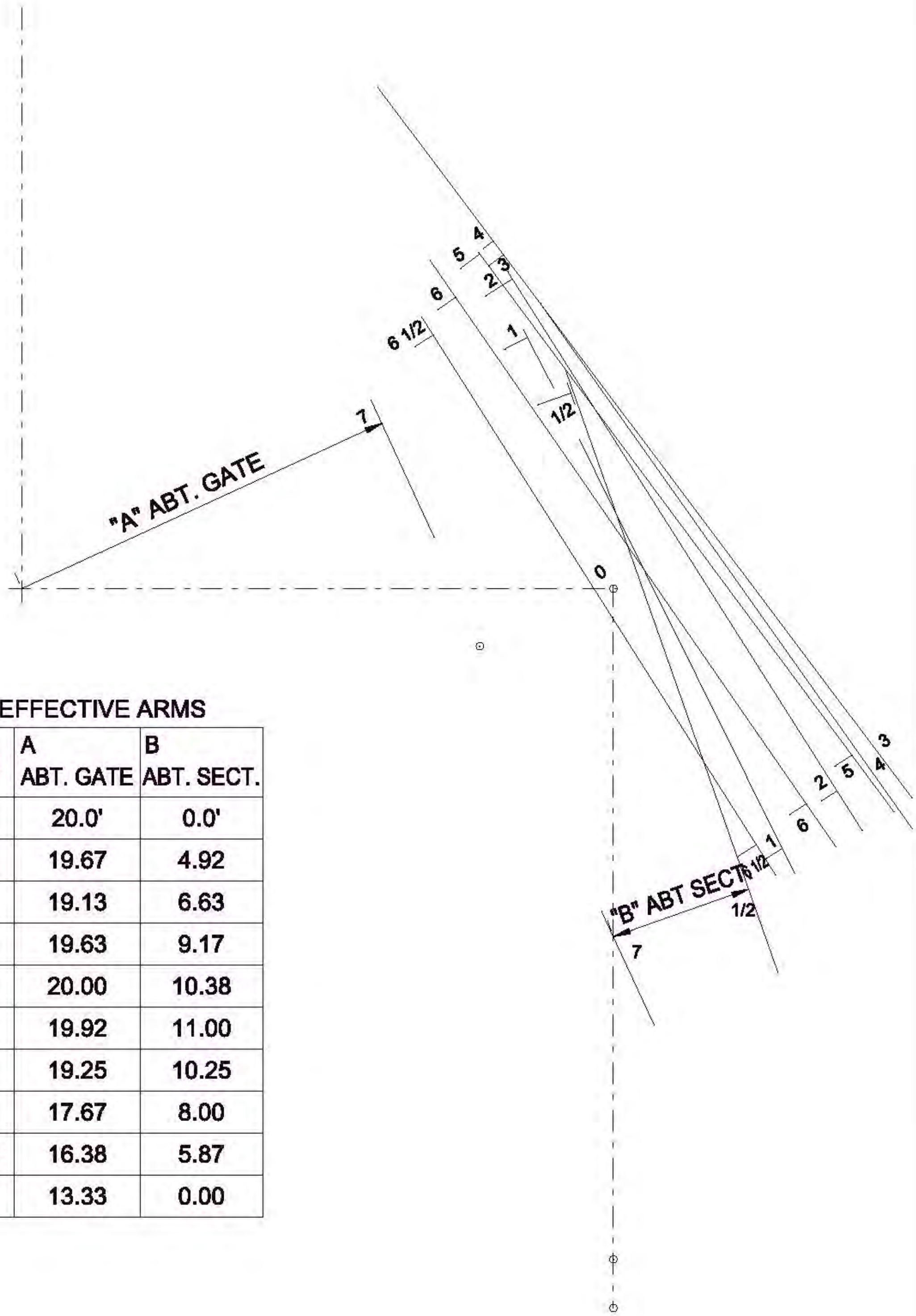
$$\text{FORCE ON PISTON} = \frac{798{,}000}{5.0 \times 0.95} = 168{,}000 \text{ LB}$$

$$\text{CYLINDER PRESSURE} = \frac{168{,}000}{113.1} + 40 \text{ PSI} = 1525 \text{ PSI}$$

A CYLINDER PRESSURE OF 1525 PSI IS REQUIRED TO MAINTAIN THE GATE IN THE MITER POSITION AGAINST A TEMPORAL HEAD OF 1.25 FT.

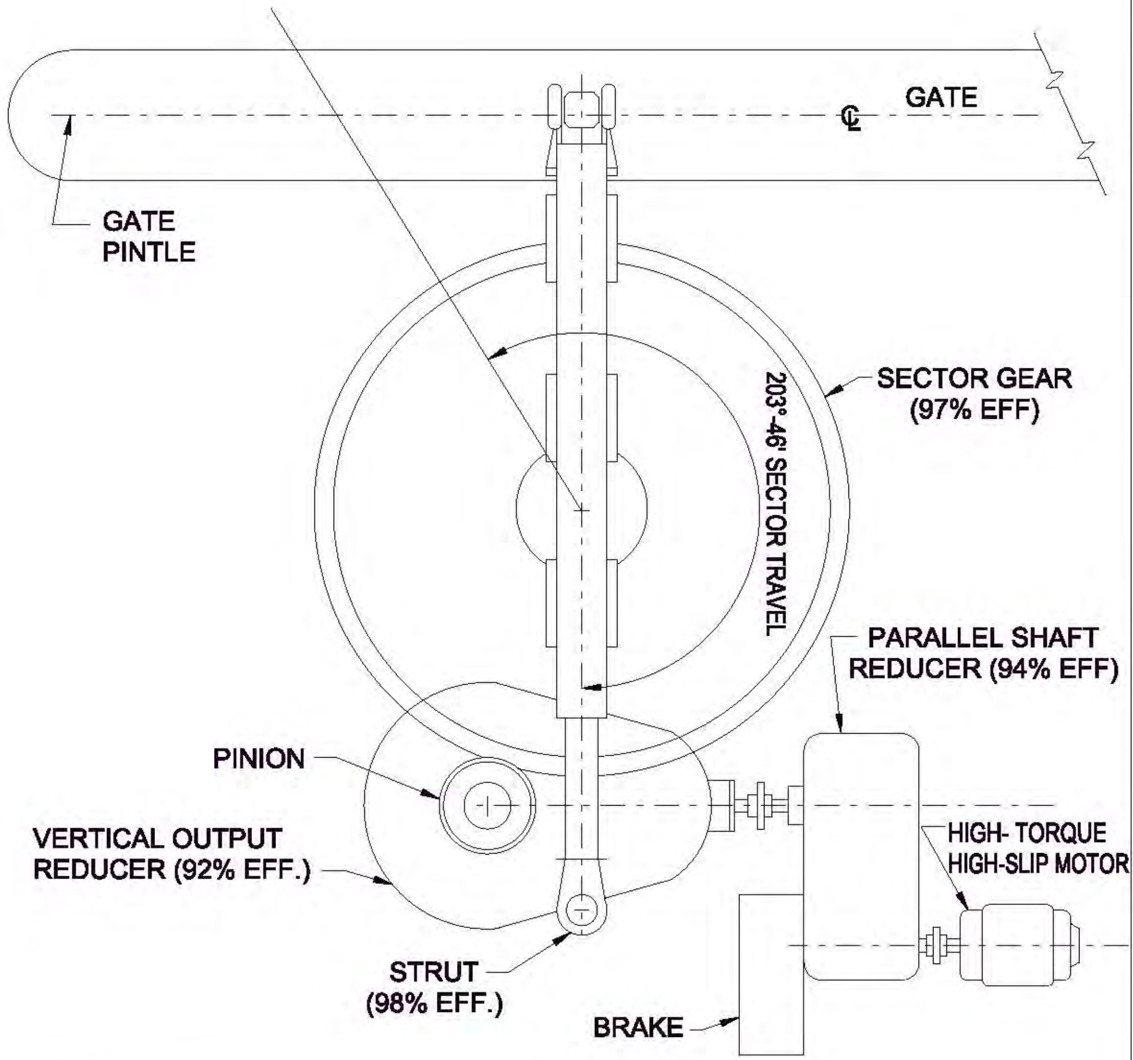
KINEMATICS
PANAMA CANAL LINKAGE
SCALE: 3/16" = 1'-0"





EFFECTIVE ARMS

POS.	A ABT. GATE	B ABT. SECT.
0	20.0'	0.0'
1/2	19.67	4.92
1	19.13	6.63
2	19.63	9.17
3	20.00	10.38
4	19.92	11.00
5	19.25	10.25
6	17.67	8.00
6 1/2	16.38	5.87
7	13.33	0.00



NL SPEED-MOTOR	1800
FL SPEED-MOTOR	1600

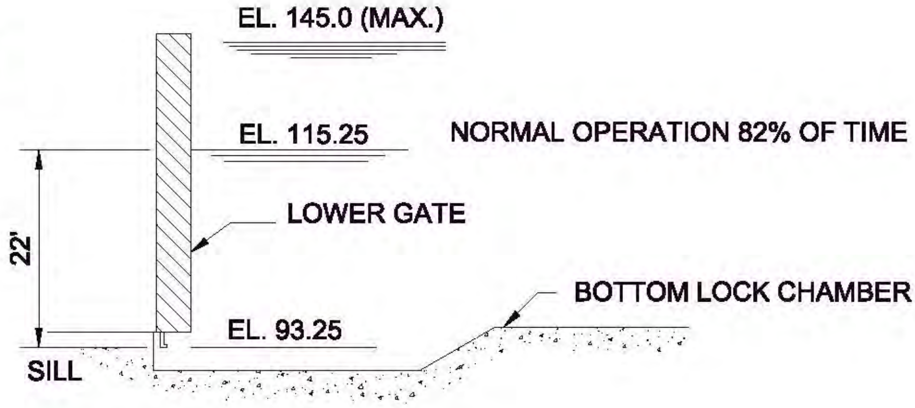
RPM OF SECTOR = $\dfrac{203.765}{1.65\text{x}60} = 2.07\ ^{\circ}\text{/SEC} \quad \therefore \quad \dfrac{2.07\text{x}60}{360} = 0.345$

AVERAGE OPERATING TIME

OVERALL RED. RATIO = 1600/0.345 = 4640 (MOTOR TO SECTOR)

OVERALL EFFICIENCY = 0.98x0.97x0.92x0.94 = 0.82

PANAMA LINKAGE



NORMAL OPERATION (ELECTRIC-MOTOR DRIVE)

ASSUME OPENING TIME = 99 SEC (SAME TOTAL TIME AS HYDR OPERATION)

$$\text{HYD OPER} = \left(\frac{1.5 + 1.8}{2}\right) 60 = 99 \text{ SEC}$$

SUBMERGENCE = 22^{FT}

NOMENCLATURE		PANAMA CANAL		PROPOSED LOCK (ALICE VILLE)		FACTOR
LENGTH GATE	L	84.304	L_1	60.0	L_1/L	= 0.712
SUBMERGENCE	S S_2	82.5 58.7	S_1	22.0	$\frac{S_1}{S_2}$	= 0.375
TIME	T T_2	120 SEC 101.3 SEC	T_1	99.0	$\frac{T_2}{T_1}$	= 1.023 SEC
ARC OF TRAVEL	K	63.436°	K_1 T_A	71.565 111.7	$\frac{T_2}{T_A}$	= 0.907

$$\text{FACTOR } \frac{L_1}{L} = \frac{60}{84.304} = 0.712$$

SUBMERGENCE = 22

$$\text{ADJ. SUBMERGENCE} = S = S_2 \left(\frac{L_1}{L}\right) = 82.5 \times 0.712 = 58.7$$

$$\text{FACTOR } \frac{S_1}{S} = \frac{22}{58.7} = 0.375$$

T_1 = TIME FOR PROPOSED LOCK = 99 SEC

T = TIME FOR PANAMA CANAL = 120 SEC

$T_2 = T \sqrt{\frac{L_1}{L}}$ = ADJ TIME = 101.3 MODEL

T_A = ANGULAR ADJ = $T_1 \left(\frac{K_1}{K}\right) = 111.7$

TIME T_1 = 99 SEC (PROPOSED GATE)

ADJ TIME = $T_2 = T \sqrt{\frac{L_1}{L}} = 120 \sqrt{0.712} = 120 \times 0.844 =$

= 101.3 MODEL

ALICEVILLE

ANGULAR TRAVEL $T_A = T_1 \left(\frac{K_1}{K}\right) = 99 \left(\frac{71.565}{63.436}\right) = 111.7$

PANAMA CANAL

FACTOR $\frac{T_2}{T_A} = \frac{101.3}{111.7} = 0.907$

EFFECT OF BOTTOM CLEARANCE-NO ADJUSTMENT
NECESSARY WHEN USING PANAMA TESTS

(NORMAL OPERATION 82% TIME)

$P_1 = P \left(\frac{L_1}{L}\right)^4 \left(\frac{S_1}{S}\right)^{1.7} \left(\frac{T_2}{T_A}\right)^{1.3}$ P_O(FROM PANAMA TEST CURVES)

$= P\ (0.712)^4 (0.375)^{1.7} (0.907)^{1.3}$

$= P\ (0.25)(0.188)(0.381) = P_O(0.0414)$

(MAX SUB'MG-LESS THAN 18% TIME)

H = 145.0-93.25 = 51.75-SAY 52'

ADJ FACTOR = $\left(\frac{52}{58.7}\right) = 0.8859$

$P_1 = P\ (0.714)^4 (0.8859)^{1.7} (0.907)^{1.3}$

$= P\ (0.25)(0.814)(0.881) = P_O(0.1793)$

CORRECTION FACTOR FOR 52' SUB'MG.)

$\left(\frac{P_{52}}{P_{22}}\right) = \frac{0.1793}{0.0414} = 4.33$

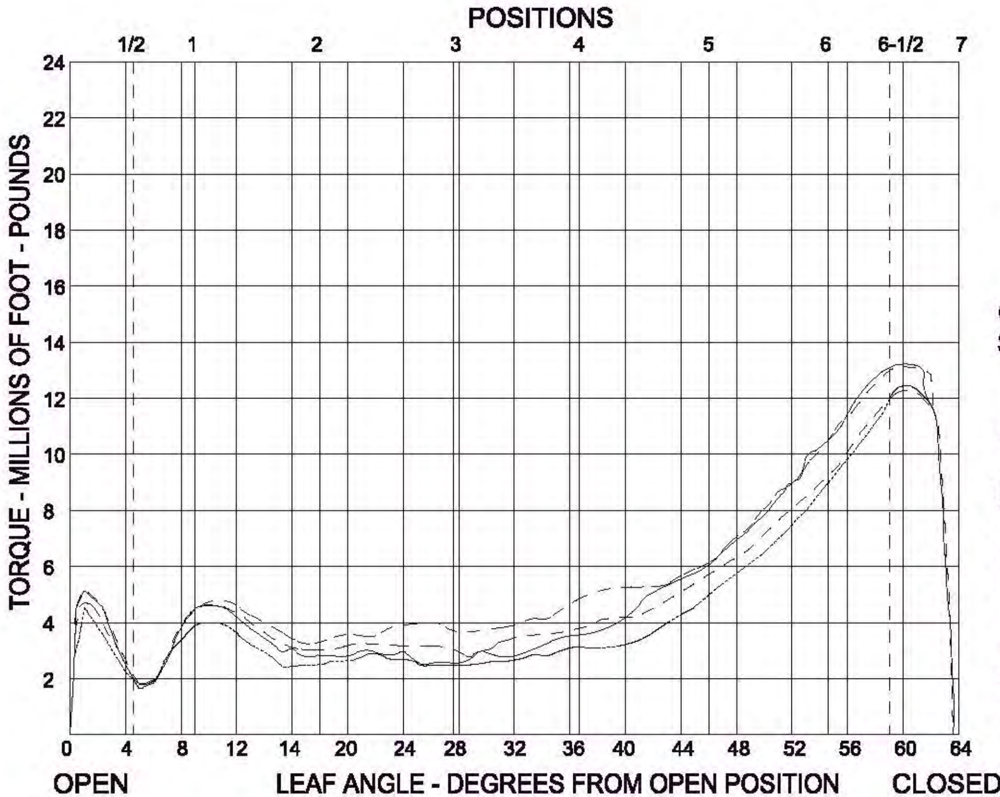
OPENING OPERATION

BASE CONDITIONS

OPERATING TIME : 2 MIN
SUBMERGENCE : 82.5 FT
CHAMBER LENGTHS :
UPSTREAM : 1,105 FT
DOWN STREAM : 200 FT

NOTE:
CURVES SHOW VARIATION OF RESULTS OBTAINED UNDER IDENTICAL OPERATING CONDITIONS.

THE PANAMA CANAL
SPECIAL ENGINEERING DIVISION
BALBOA HEIGHTS, CANAL ZONE
THE THIRD LOCKS PROJECT
HYDRAULIC MODEL Y
MITER GATE OPERATION
THIRD LOCKS TEST CURVES



CLOSING OPERATION

BASE CONDITIONS

OPERATING TIME : 2 MIN
SUBMERGENCE : 82.5 FT
CHAMBER LENGTHS :
UPSTREAM : 1,105 FT
DOWN STREAM : 200 FT

NOTE:
CURVES SHOW VARIATION OF RESULTS OBTAINED UNDER IDENTICAL OPERATING CONDITIONS.
FOR INFORMATION ONLY: MOTOR WOULD BE SELECTED ON THE OPENING CYCLE.

THE PANAMA CANAL
SPECIAL ENGINEERING DIVISION
BALBOA HEIGHTS, CANAL ZONE
THE THIRD LOCKS PROJECT
HYDRAULIC MODEL Y
MITER GATE OPERATION
THIRD LOCKS TEST CURVES

REQ'D POWER - OPENING OPERATION (22 FT SUB'M)

POS.	TORQUE ABT PINTLE PANAMA C. FT-LB	CORR. FACTOR	TORQUE ABT PINTLE ALICE VILLE FT-KIPS	EFFECT. ARM ABT PINTLE FT	FORCE IN STRUT (0.98 EFF)	EFFECT. ARM ABT SECTOR FT	TORQUE ABT SECTOR (0.97 EFF)	RE DUCT RATIO (0.865 EFF)	REQ'D MOTOR TORQUE FT-LBS	HP = $\frac{T \times RPM}{5250}$
0	0	0.0414	0	20.0	0	0	0	4640	0	0
0.5	4.0×10^6		165.6×10^3	19.67	8.59^K	4.92	43.6^K		10.9	3.32
1	3.8×10^6		157.3×10^3	19.13	8.39^K	6.63	57.4^K		14.3	4.4
2	3.0×10^6		124.2×10^3	19.63	6.46^K	9.17	61.0^K		15.2	4.6
3	5.9×10^6		244.3×10^3	20.00	12.46^K	10.38	133.4^K		33.2	10.1
4	7.9×10^6		327.0×10^3	19.92	16.75^K	11.00	190.0^K		47.3	14.4
5	14.0×10^6		579.6×10^3	19.25	30.70^K	10.25	324.7^K		80.9	24.7
6	18.4×10^6		761.8×10^3	17.67	43.99^K	8.00	362.8^K		90.4	27.5
6 1/2	15.4×10^6		637.6×10^3	16.38	39.72^K	5.87	240.4^K		59.9	18.3
7	0		0	13.33	0	0	0		0	0

FORCES AT MAX SUB'M. (52 FT)

REQUIRED POWER - OPENING OPERATION

POS.	TORQUE FROM P.C. CURVES	CORR FACTOR $= \frac{0.1793}{0.0414}$	FORCE IN STRUT 22-FT SUB'M.	FORCE IN STRUT 52-FT SUB'M.	TORQUE ABT PINTLE 22-FT SUB'M.	TORQUE ABT PINTLE 52-FT SUB'M.
0	0	4.33	0	0	0	0
0.5	4.0×10^{6}		8.59^{K}	37.2	165.6×10^{3}	717.1×10^{3}
1	3.8×10^{6}		8.39^{K}	36.3	157.3×10^{3}	681.1×10^{3}
2	3.0×10^{6}		6.46^{K}	28.0	124.2×10^{3}	537.8×10^{3}
3	5.9×10^{6}		12.46^{K}	53.9	244.3×10^{3}	1057.8×10^{3}
4	7.9×10^{6}		16.75^{K}	72.5	327.0×10^{3}	1415.9×10^{3}
5	14.0×10^{6}		30.70^{K}	132.9	579.6×10^{3}	2509.6×10^{3}
6	18.4×10^{6}		43.99^{K}	190.5	761.8×10^{3}	3298.6×10^{3}
6 1/2	15.4×10^{6}		39.72^{K}	172.0	637.6×10^{3}	2760.8×10^{3}
7	0		0	0	0	0

MOTOR SELECTION

22' SUB. MOTOR MUST PROVIDE REQ'D TORQUE AT ALL POSITIONS AND AT NORMAL SPEED.

52' SUB. MOTOR MUST PROVIDE PEAK TORQUE W/OUT EXCEEDING 150% OF F.L.T. GATE WILL BE SLOWED DUE TO MOTOR SPEED LOSS AT OVERLOAD. THEREFORE REQ'D TORQUE AS SHOWN IN PLATE B-80, SHEET II, WOULD REQUIRE CORRECTION.

PEAK OCCURS AT POS 6.
TORQUE ABOUT PINTLE = 3,298.6 $^{\text{FT-LB}}$ FOR NORMAL SPEED.

AT 150 % O.L.
MOTOR SPEED = (1,800) (0.77) = 1,386 RPM

1. GATE SPEED AT 1,386 MOTOR RPM

   $T_1 = (99)\left(\frac{1,386}{1,600}\right) = 114$ SEC

2. ADJ PINTLE TORQUE FOR CHANGED TIME
   $T_1 = 114$
   $T_2 = 101.3$

   $T_A = 114\left(\frac{71.565}{63.436}\right) = 128.6$

   FACTOR $T_2 / T_A = 101.3 / 128.6 = 0.788$

3. CALCULATE PINTLE TORQUE AT POS 6

$$P_1 = P_O\,(0.25)\,(0.814)\,(0.788)^{1.3}$$
$$= P_O\,(0.204 * 0.73) = P_O\,(0.149)$$

$$P_1 = 18.4 \times 10^6 * 0.149 = 2{,}742^{\text{FT-KIPS}}$$

4. MOTOR TORQUE AT 150% 0L.

$$\text{MOTOR TORQUE} = \frac{2742^{\text{FT-KIPS}} \times 8.00}{17.67 * 0.98 * 0.97 * 4640 * 0.865} = 322.^{\text{FT-LB}}$$

$$\text{NORMAL F.L. TORQUE} = \frac{322}{1.5} = 214.7^{\text{FT-LB}}$$

$$\text{MOTOR HP} = \frac{214.7 \times 1{,}600}{5250} = 65.4^{\text{HP}}$$

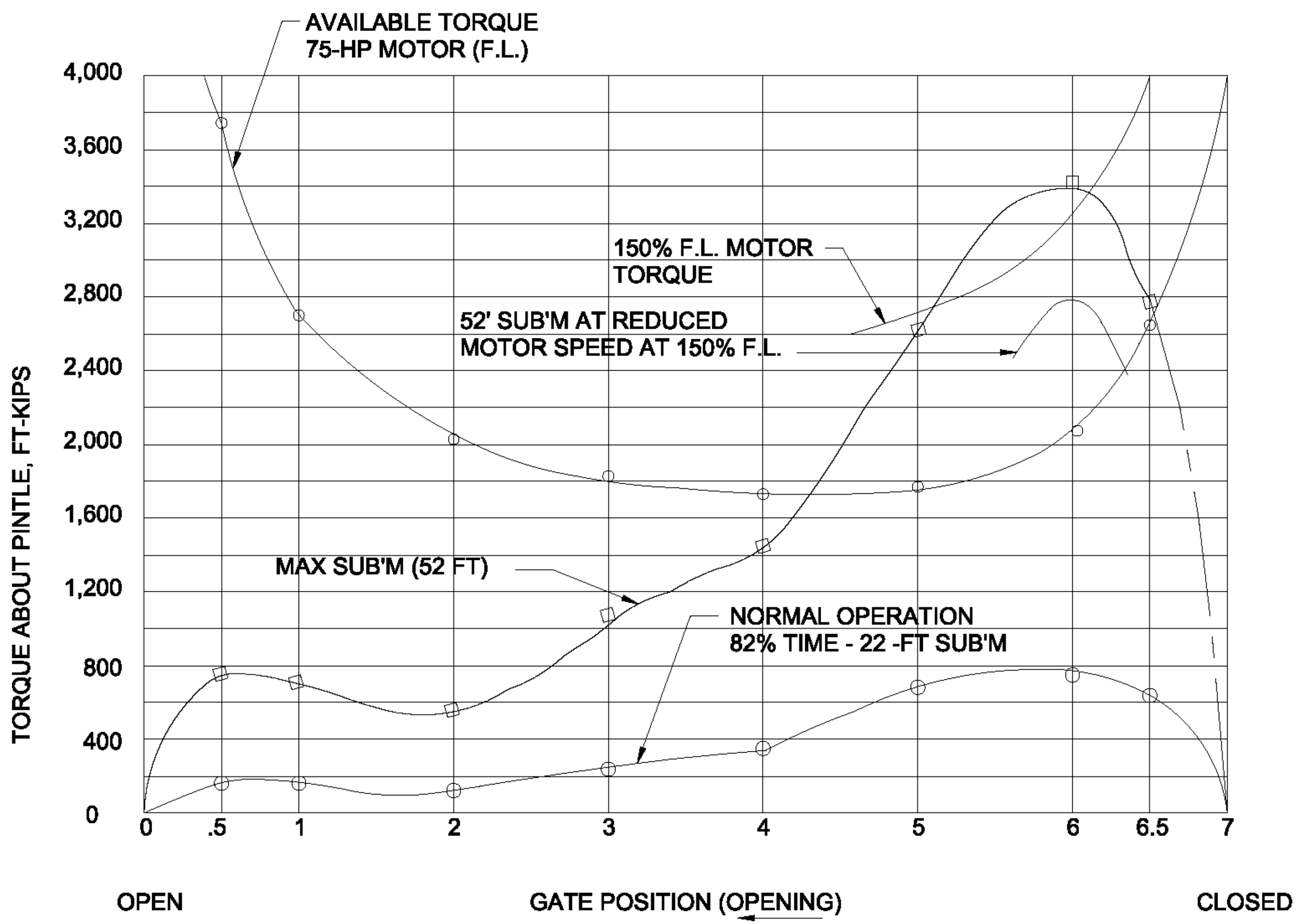
75 HP REQUIRED.

AVAILABLE POWER - OPENING OPERATION

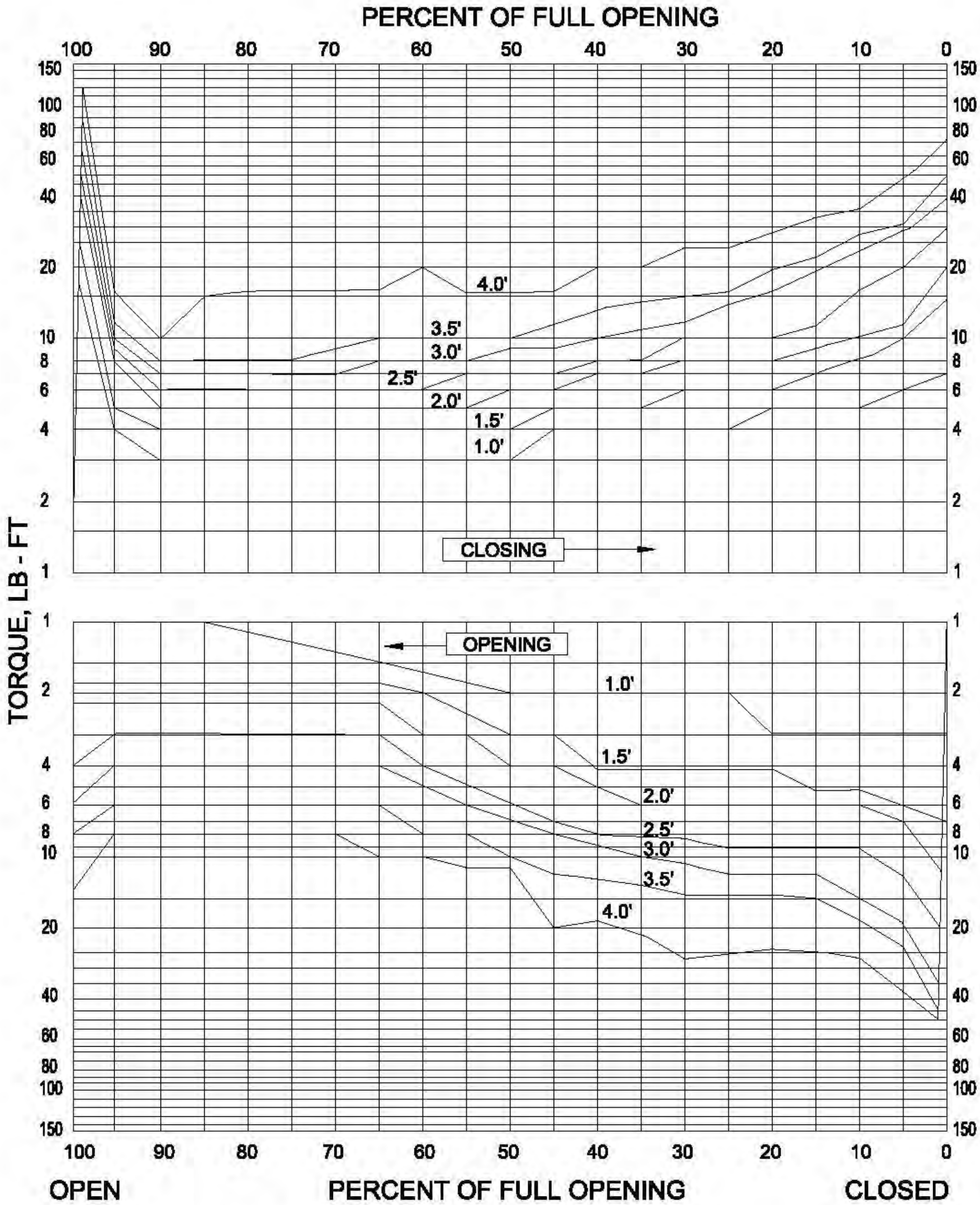
POS	F.L. * MOTOR TORQUE	RED. RATIO (0.865 EFF)	TORQUE ABT SECTOR	EFFECT. ARM ABT SECTOR (0.97)	FORCE IN STRUT	EFFECT. ARM ABT PINTLE	TORQUE ABT PINTLE (0.98 EFF.)
0	246 FT. LB.	4640	987.3 ^{1K}	0	0	20.00	0
0.5				4.92	194.6 ^{K}	19.67	3752 $^{FT/K}$
1				6.63	144.4	19.13	2708
2				9.17	104.4	19.63	2009
3				10.38	92.3	20.00	1808
4				11.00	87.1	19.92	1700
5				10.25	93.4	19.25	1762
6				8.00	119.7	17.67	2073
6 1/2				5.87	163.1	16.38	2619
7				0	0	13.33	0

* 75 HP AT 1,600 RPM F.L. SPEED

$$T = \frac{HP \times 5250}{RPM} = 246 \text{ FT-LB}$$

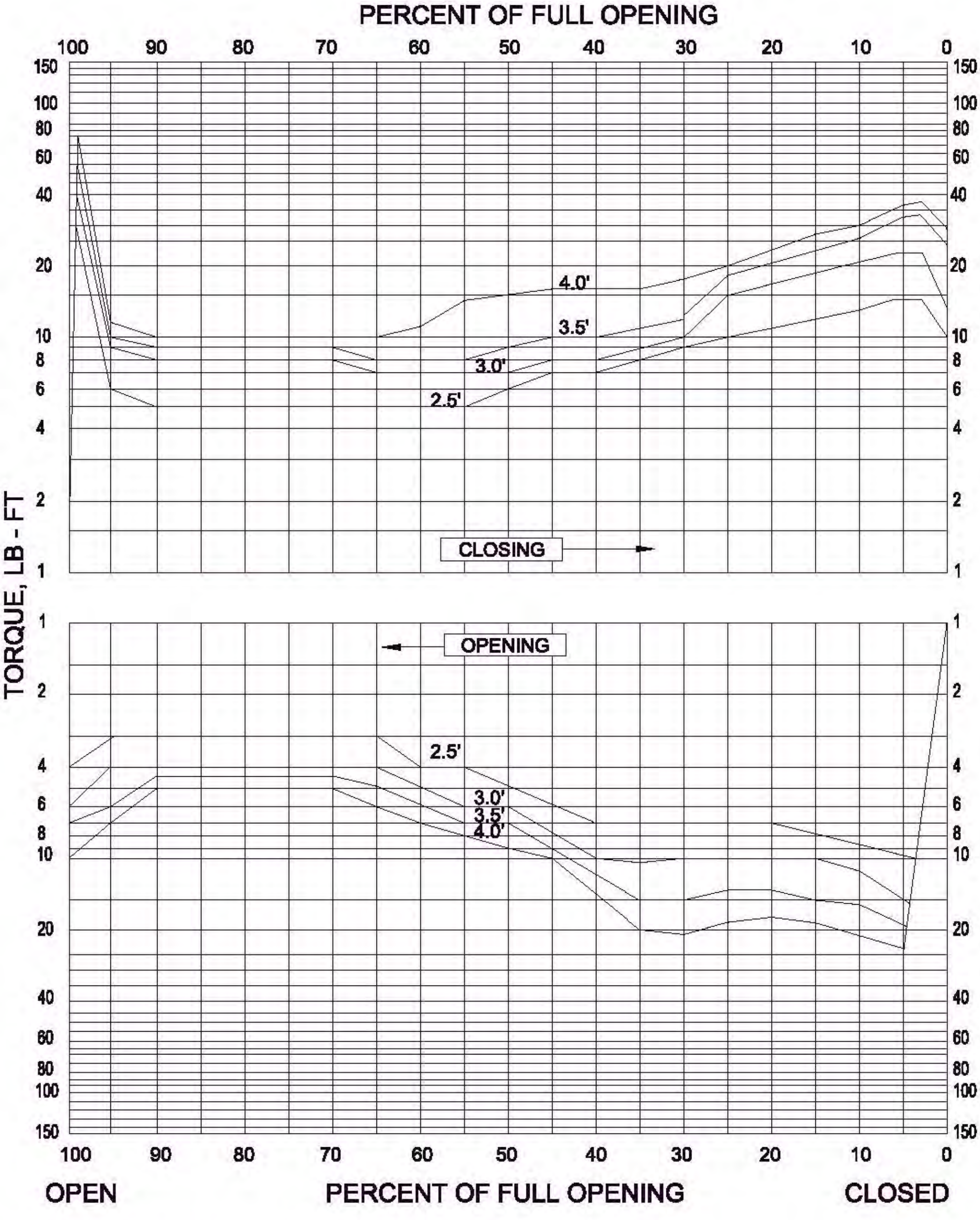

AVAILABLE TORQUE
75-HP MOTOR (F.L.)
150% F.L. MOTOR
TORQUE
52' SUB'M AT REDUCED
MOTOR SPEED AT 150% F.L.
MAX SUB'M (52 FT)
NORMAL OPERATION
82% TIME - 22 -FT SUB'M
TORQUE ABOUT PINTLE, FT-KIPS
4,000
3,600
3,200
2,800
2,400
2,000
1,600
1,200
800
400
0
0
.5
1
2
3
4
5
6
6.5
7
OPEN
GATE POSITION (OPENING)
CLOSED

MITER GATES
OHIO RIVER LINKAGE
EFFECT OF SUBMERGENCE ON INSTANTANEOUS TORQUE OPERATING TIME, 20.1 SEC

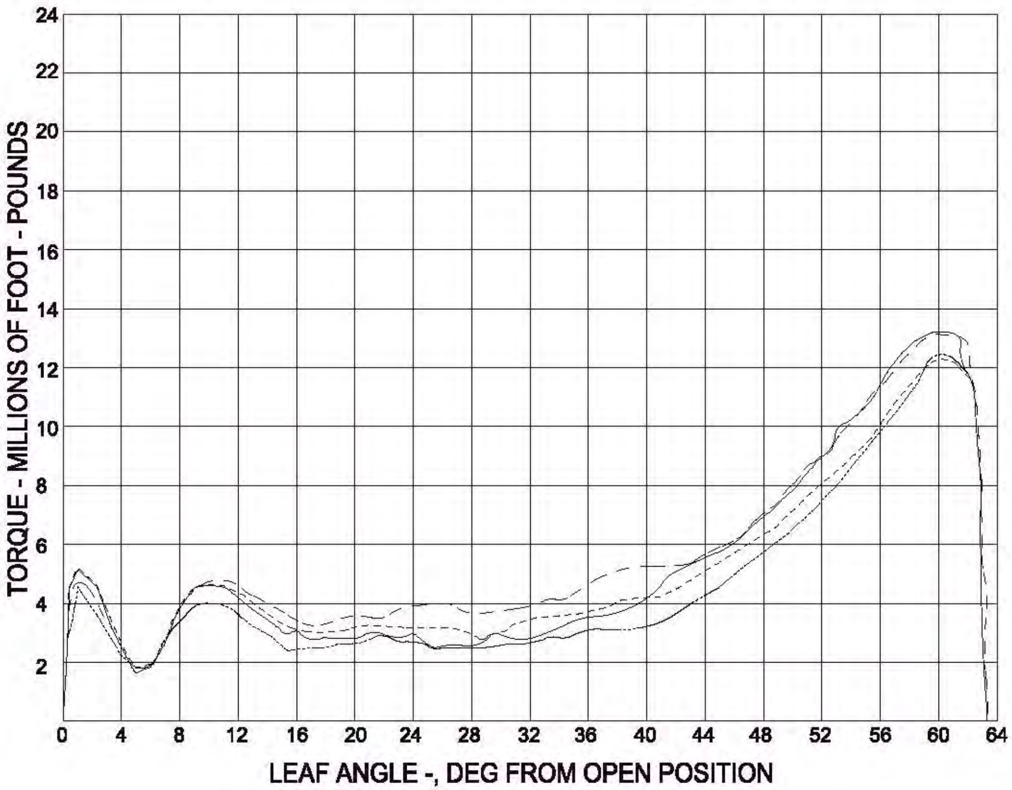


SUBMERGENCE-AS SHOWN IN FEET
CHAMBER LENGTH
UPSTREAM - 25FT
DOWNSTREAM - 25 FT
BOTTOM CLEARANCE 0.25 FT

MODIFIED OHIO RIVER LINKAGE
EFFECT OF SUBMERGENCE ON INSTANTANEOUS TORQUE
OPERATING TIME, 20.1 SEC
CHAMBER LENGTHS, 30 FT



SUBMERGENCE-AS SHOWN IN FEET
BOTTOM CLEARANCE 0.25 FT



CLOSING OPERATION

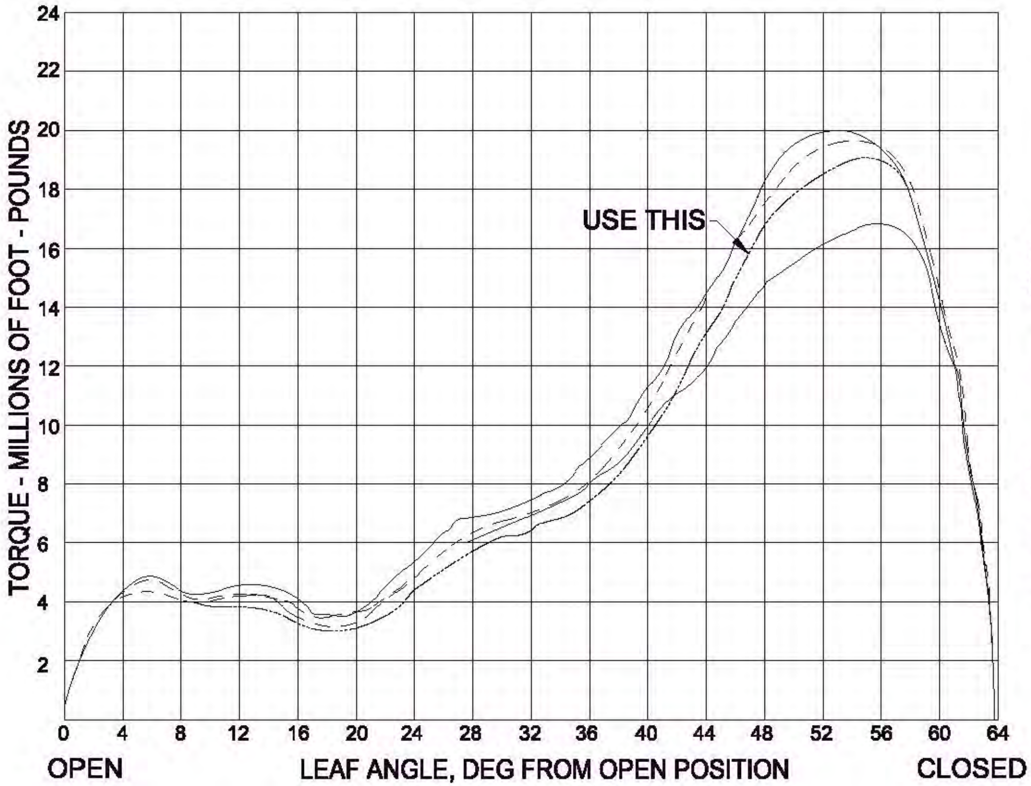
BASE CONDITIONS

OPERATING TIME : 2 MIN
SUBMERGENCE : 82.5 FT
CHAMBER LENGTHS :
UPSTREAM : 1,105 FT
DOWN STREAM : 200 FT

NOTE:
CURVES SHOW VARIATION OF RESULTS OBTAINED UNDER IDENTICAL OPERATING CONDITIONS.

MITER GATES
THE PANAMA CANAL

SPECIAL ENGINEERING DIVISION
BALBOA HEIGHTS, CANAL ZONE
THE THIRD LOCKS PROJECT
HYDRAULIC MODEL Y
MITER GATE OPERATION
THIRD LOCKS TEST CURVES



OPENING OPERATION

BASE CONDITIONS

OPERATING TIME : 2 MIN
SUBMERGENCE : 82.5 FT
CHAMBER LENGTHS :
UPSTREAM : 1,105 FT
DOWN STREAM : 200 FT

NOTE:
CURVES SHOW VARIATION OF RESULTS OBTAINED UNDER IDENTICAL OPERATING CONDITIONS.

MITER GATES
THE PANAMA CANAL
SPECIAL ENGINEERING DIVISION
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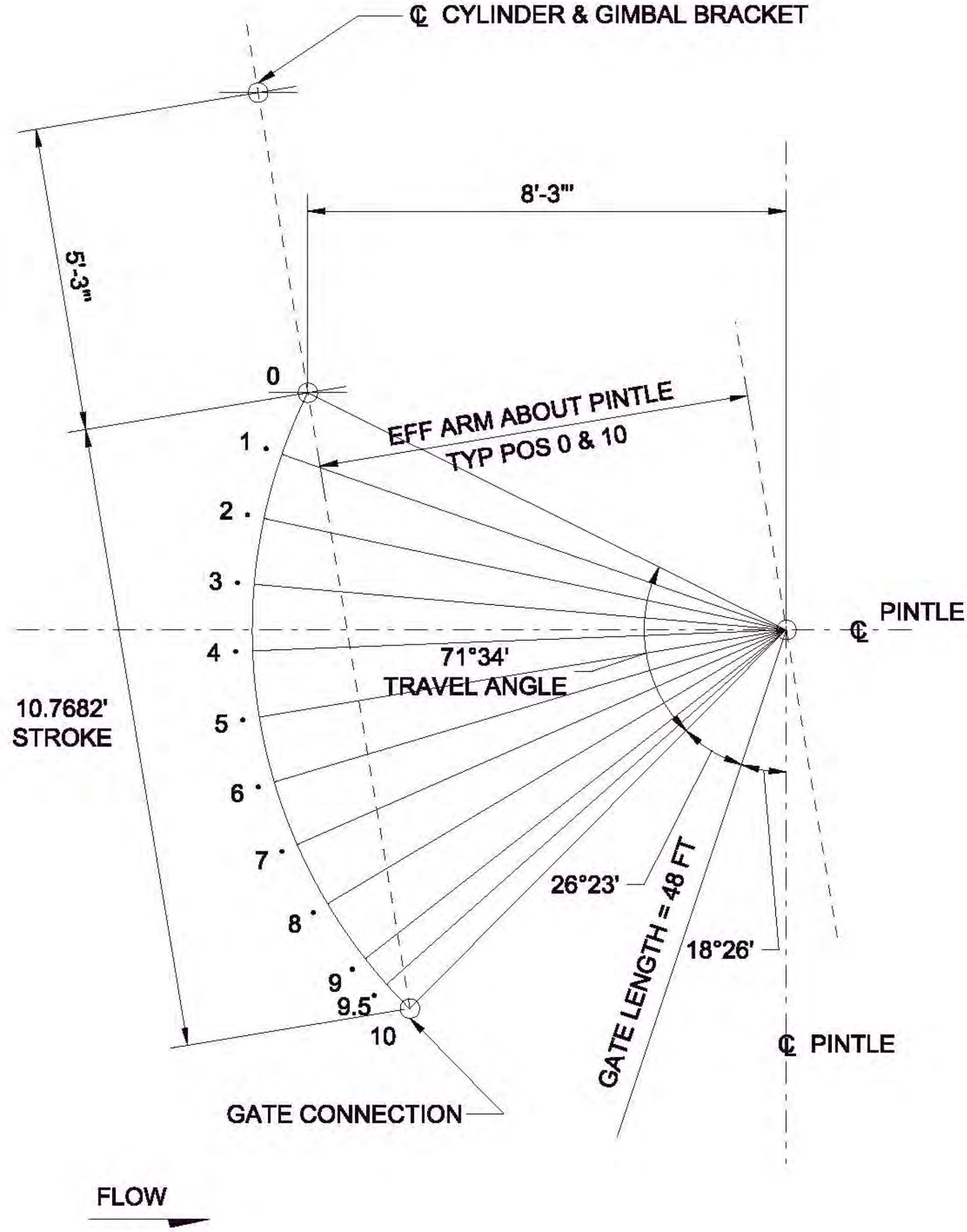
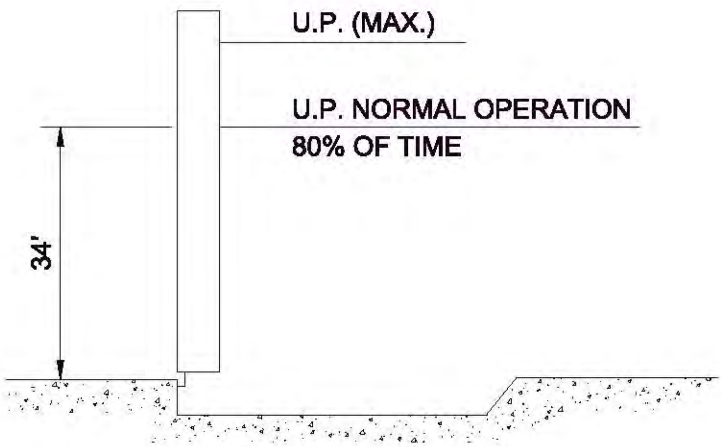
KINEMATICS
DIRECT CONNECTED CYLINDER
SCALE: 1" = 3'-0"





DESIGN CRITERIA (OPENING OPERATION) 80% TIME

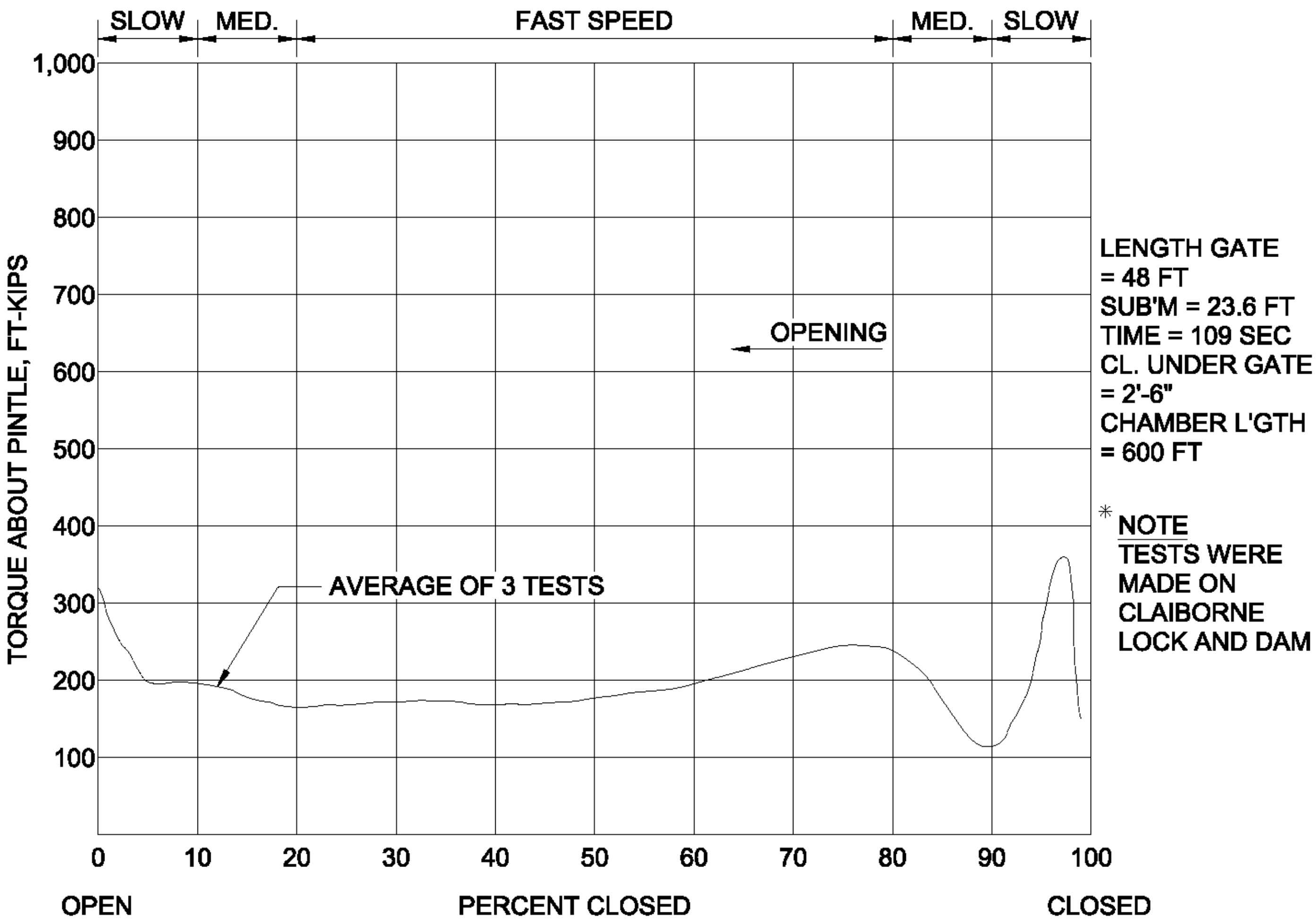
OPENING TIME: ASSUME 90 SEC (1 1/2 MIN) SEE PLATE B-87, SHEET 3

SUBMERGENCE: ASSUME 34 FT. (80% TIME OR LESS)

STROKE OF PISTON: 10.768 FT.

NOMENCLATURE		MODEL (CLAIBORNE L.)		PROPOSED LOCK		FACTOR
LENGTH GATE	L	48.0	L_1	48.0	$\frac{L_1}{L}$	1.0
SUBMERGENCE	S S_2	23.6' 23.6	S_1	34.0	$\frac{S_1}{S_2}$	1.441
TIME	T T_2	109 109	T_1	90	$\frac{T_2}{T_1}$	1.211
ARC OF TRAVEL	K	71.567°	K_1	71.567°	$\frac{K}{K_1}$	1.00

SINCE LENGTH OF GATE, ARC OF TRAVEL, AND CLEARANCE UNDER GATE ARE THE SAME ON MODEL AND PROPOSED LOCK, NO ADJUSTMENTS ARE REQUIRED, FOR THESE FACTORS.



TESTS 1A, 2A, AND 3A
VARIABLE SPEED OPERATION
TIME, 109 SEC

OPENING GATE

$$P_1 = P\left(\frac{L_1}{L}\right)^4\left(\frac{S_1}{S_2}\right)^{1.7}\left(\frac{T_2}{T_1}\right)^{1.3}$$

$$= P\,(1.0)^4\,(1.441)^{1.7}(1.211)^{1.3} = Px1.86x1.283$$

$$P_1 = 2.39xP$$

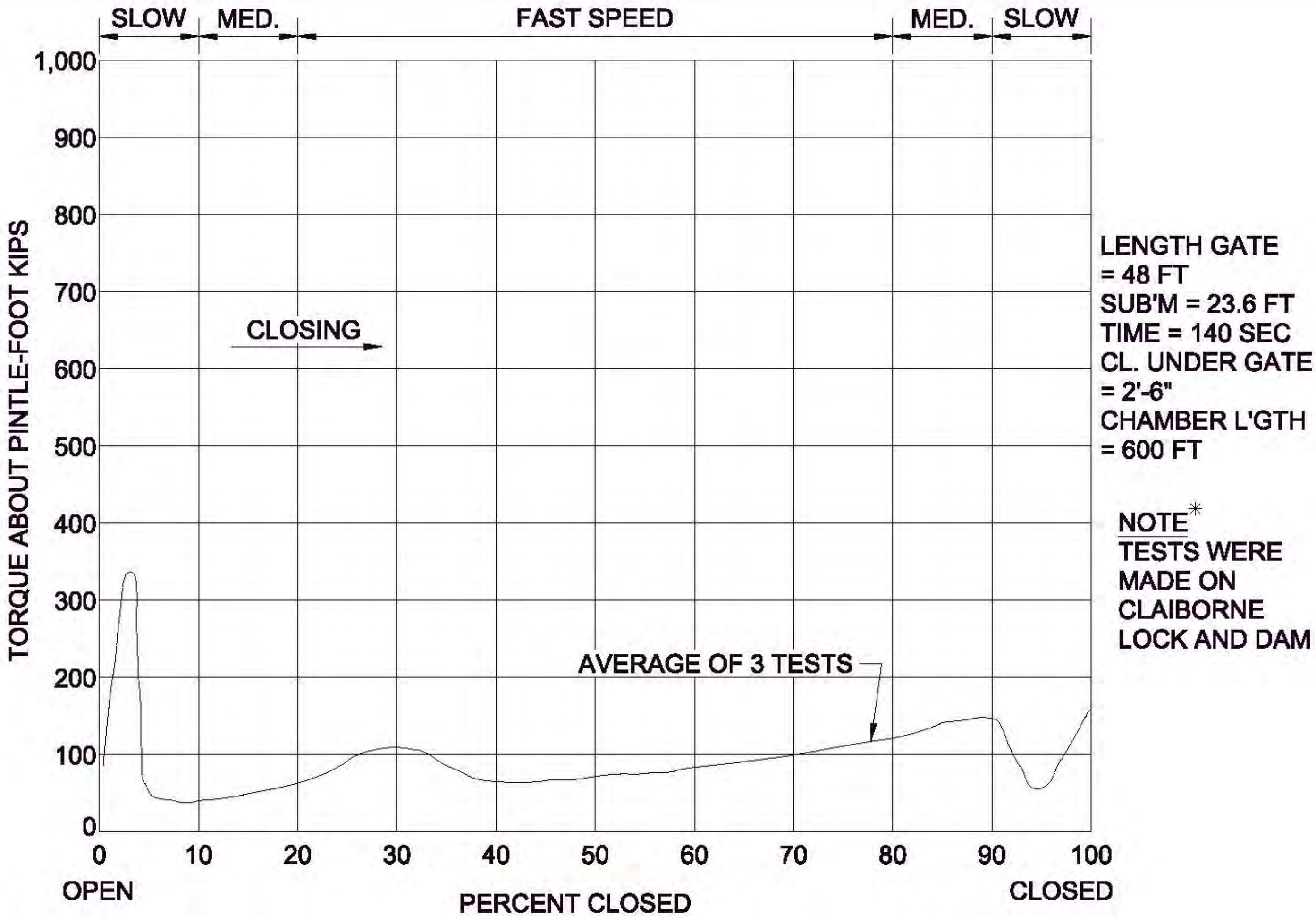
CLOSING GATE

NOMENCLATURE		MODEL (CLAIBORNE L.)		PROPOSED LOCK		FACTOR
LENGTH GATE	L	48.0 FT	L_1	48.0 FT	$\frac{L_1}{L}$	1.0
SUBMERGENCE	S S_2	23.6 23.6	S_1	34.0	$\frac{S_1}{S}$	1.441
TIME	T T_2	140 SEC 140 SEC	T_1	113	$\frac{T_2}{T_1}$	1.24
ARC OF TRAVEL	K	71.567	K_1	71.565	$\frac{K}{K_1}$	1.0

$$P_1 = P\left(\frac{L_1}{L}\right)^4\left(\frac{S_1}{S}\right)^{1.5}\left(\frac{T_2}{T_1}\right)^{1.1}$$

$$= P\,(1.0)^4\,(1.441)^{1.5}(1.24)^{1.1} = Px1.0x1.73x1.267$$

$$P_1 = 2.19P$$



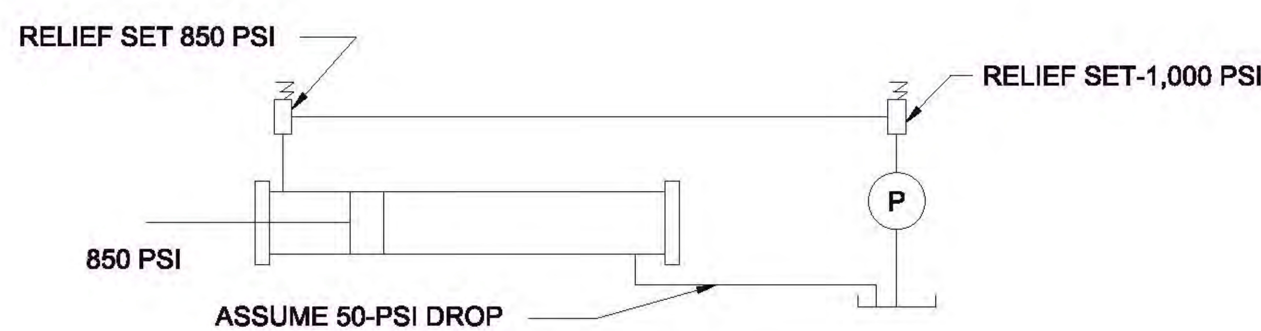
TESTS 1A, 2A, AND 3A
VARIABLE SPEED OPERATION
TIME, 140 SEC

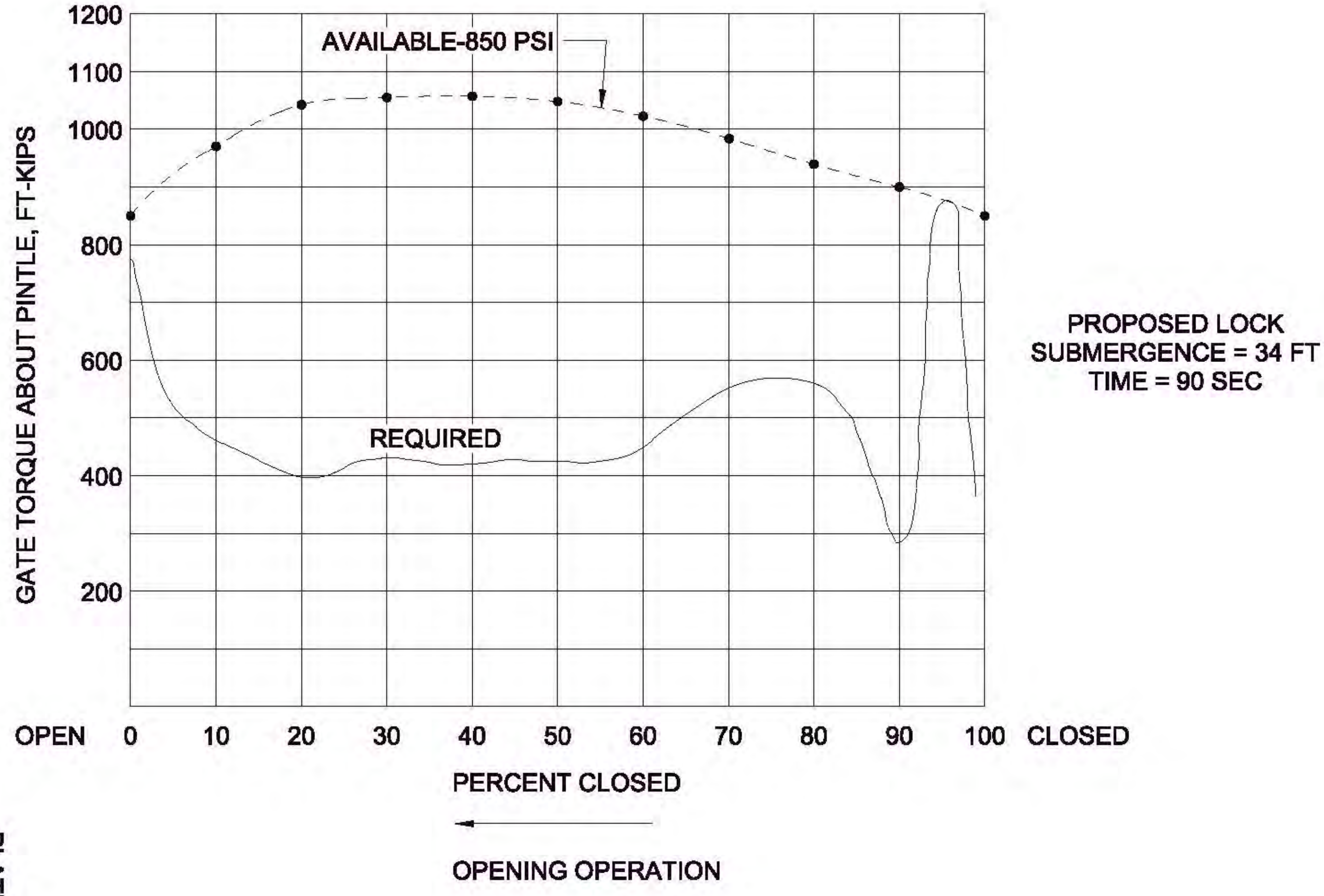
OPENING GATE

REQUIRED TORQUE										AVAILABLE TORQUE		
1	2	3	4	5	6	7	8	9	10	11	12	13
POS	MODEL TORQUE (CLAIBORNE) (FT-KIPS)	ADJ FACTOR	TORQUE ABT. PINTLE (PROPOSED L) FT-KIPS 2 X 3	EFFECT. ARM ABOUT PINTLE (FT)	FORCE ON PISTON ROD (ROD END) KIPS 4 ÷ 5 ÷ 0.97 *	NET AREA HEAD END (SQ IN)	RETARDING FORCE DUE TO 50-PSI FRICT. KIPS 7 X 50 PSI	REQ'D FORCE ROD END KIPS 6 + 8	REQ'D CYL PRESS. ROD END PSI 9 ÷150.83 **	AVAIL. CYL PRESS. PSI	AVAIL. PISTON FORCE KIPS 11 x150.83x 0.97÷1,000- 8	AVAIL. TORQUE ABOUT PINTLE FT-KIPS 12 X 5
CLOSED 10	0	2.39	0	7.4792	0	201.1	10.1	0	0	850	114.3	854.6
9 1/2	360		860.4	7.6790	112.0			122.1	810			877.4
9	115		274.8	7.8958	35.9			45.9	305			902.2
8	230		549.7	8.2917	68.3			78.4	520			947.4
7	225		537.8	8.6250	64.3			74.3	493			985.5
6	185		442.1	8.8958	51.2			61.2	407			1016.4
5	175		418.3	9.1042	47.4			57.5	381			1040.2
4	170		406.3	9.2083	45.5			55.6	369			1052.1
3	175		418.3	9.1875	46.9			57.0	378			1049.7
2	160		382.4	8.9792	43.9			54.0	358			1025.9
1	190		454.1	8.4688	55.3			65.4	433			967.6
0	315		752.8	7.4792	103.8			113.9	755			854.6

OPEN

* 0.97-EFFIC MACHINE
* * AREA ROD END = 150.83?"



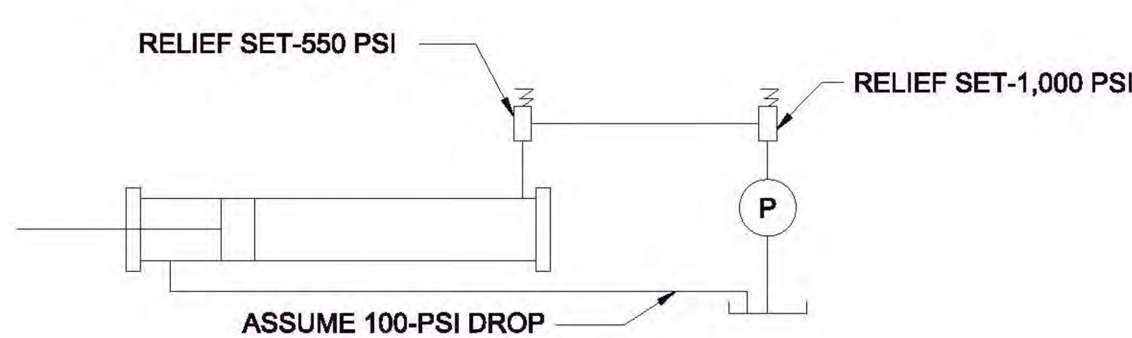

AVAILABLE-850 PSI
REQUIRED
GATE TORQUE ABOUT PINTLE, FT-KIPS
1200
1100
1000
800
600
400
200
OPEN
0
10
20
30
40
50
60
70
80
90
100
CLOSED
PERCENT CLOSED
OPENING OPERATION
PROPOSED LOCK
SUBMERGENCE = 34 FT
TIME = 90 SEC

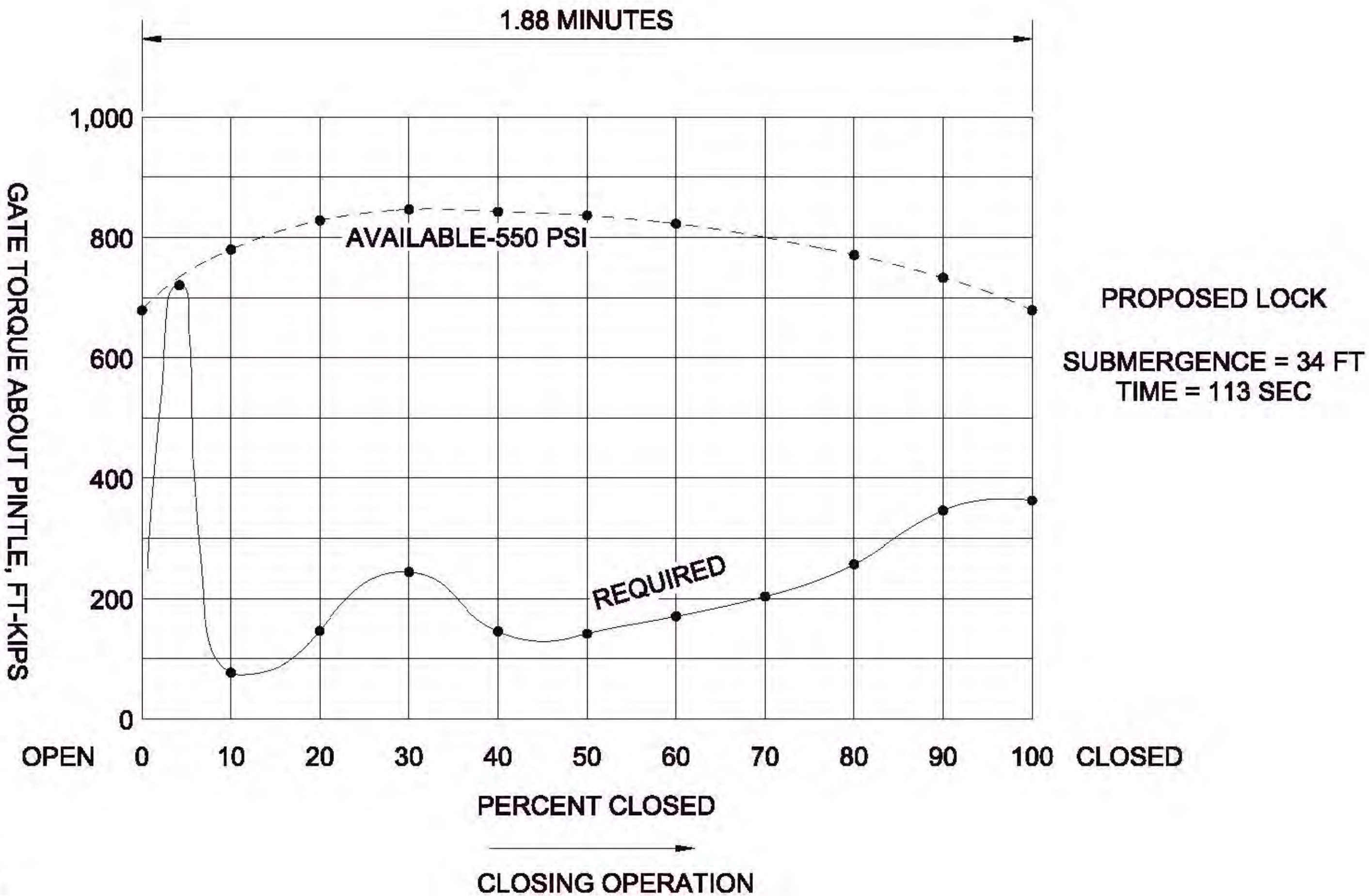
CLOSING GATE

REQUIRED TORQUE										AVAILABLE TORQUE		
1	2	3	4	5	6	7	8	9	10	11	12	13
POS	MODEL TORQUE (CLAIBORNE) (FT. KIPS)	ADJ FACTOR	TORQUE ABT. PINTLE (PROPOSED L.) FT-KIPS 2 X 3	EFFECT. ARM ABOUT PINTLE (FT)	FORCE ON PISTON ROD (HEAD END) KIPS 4 ÷ 5 ÷ 0.97 *	NET AREA ROD END (SQ IN)	RETARDING FORCE DUE TO 100-PSI FRICT. KIPS 7 X100 PSI	REQ'D FORCE HEAD END KIPS 6 + 8	REQ'D CYL PRESS. HEAD END PSI 9 ÷ 200.1 **	AVAIL. CYL PRESS. PSI	AVAIL. PISTON FORCE. KIPS 11 x200.1 ** x 0.97÷1000- 8	AVAIL. TORQUE ABOUT PINTLE FT-KIPS 12 X 5
OPEN 0	0	2.19	0	7.4792	0	150.83	0	0	0	550	91.7	686
1/2	360		722.7	8.0790	92.2		15.1	107.3	534			741
1	35		76.7	8.4688	9.3			24.4	121			777
2	65		142.5	8.9792	16.4			31.5	157			823
3	110		240.9	9.1875	27.0			42.1	209			842
4	70		153.3	9.2083	17.2			32.3	161			844
5	75		164.3	9.1042	18.6			33.7	168			835
6	85		186.2	8.8958	21.6			36.7	182			816
7	100		219.0	8.6250	26.1			41.3	206			791
8	125		273.8	8.2917	34.0			49.1	245			760
9	160		350.4	7.8958	45.8			60.9	304			724
10	165		361.4	7.4792	49.8			64.9	324			686

CLOSED

* 0.97-EFFIC MACHINE
** AREA HEAD END = 200.1




1.88 MINUTES
AVAILABLE-550 PSI
REQUIRED
PROPOSED LOCK
SUBMERGENCE = 34 FT
TIME = 113 SEC
GATE TORQUE ABOUT PINTLE, FT-KIPS
1,000
800
600
400
200
0
OPEN
0
10
20
30
40
50
60
70
80
90
100
CLOSED
PERCENT CLOSED
CLOSING OPERATION

PUMP CAPACITY:

USE 90 SEC OPERATING TIME-OPENING-SEE PLATE B-87 SHEET 2

NET AREA-ROD END = 150.83□"

$$GPM = \frac{150.83x12x0.1459x60}{231} = 68.6 \text{ FAST SPEED}$$

GPM = 68.6x0.8 = 54.9 MED SPEED

GPM = 68.6x0.3 = 20.6 SLOW SPEED

USE VARIABLE VOLUME PUMP WITH AT LEAST 4 SETTINGS, "NEUTRAL", "SLOW", "MEDIUM", AND "FAST", WITH PUMP DELIVERIES AS SHOWN ABOVE.

DETERMINATION OF LOADS REQUIRED TO RAISE UPSTREAM LEAF (MOVABLE SILL) OF A DOUBLE-LEAF VERTICAL-LIFT GATE:

A. WEIGHT OF UPSTREAM LEAF

B. WEIGHT OF SILT ON UPSTREAM LEAF

C. SLIDING FRICTION OF GATE ON RUBBING STRIPS

D. DOWNWARD HYDROSTATIC LOAD

E. WEIGHT OF RECESS PROTECTION AND TRASH SCREENS LESS WEIGHT OF WATER DISPLACED

F. SIDE SEAL FRICTION

A. WEIGHT OF UPSTREAM GATE

FROM PRELIMINARY CALCULATIONS, WEIGHT OF 14'x110'

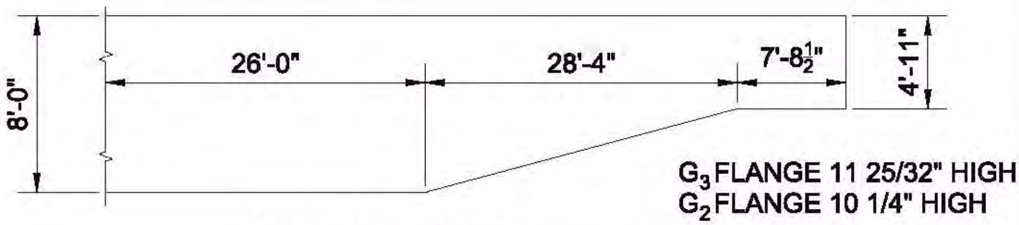
GATE WAS ESTIMATED TO BE 185,000 LB

$$\text{WT OF WATER DISPLACED} = \frac{185{,}000 \times 62.4}{489} = 23{,}600 \text{ LB}$$

TOTAL WEIGHT TO BE LIFTED = 185,000-23,600 = 161,400 LB

B. WEIGHT OF SILT (IN AIR)

WEIGHT OF SILT = 125 LB/FT^3



1/2 GIRDER- G2 & G3

WT OF SILT ON GIRDER G1 IS CONSIDERED NEGLIGIBLE

WT OF SILT ON GIRDER G2

$$W_{G2} = \left[2\,(125)\,\frac{10.25}{12}\right]\left[(26.0)\,(8.0) + \left(\frac{8.0+4.92}{2}\right)(28.33) + (7.71)\,(4.92)\right]$$

$$= \left[214\right]\left[(208)+(183)+(38)\right]$$

$$= (214)\,(429^{\square FT}) = \underline{91{,}800 \text{ LB}}$$

$$W_{G3} = \left[\frac{(214)\,(11.78)}{10.25"}\right] = (429) = (246)\,(429) = \underline{105{,}500 \text{ LB}}$$

$$W_{G1,\,G2,\,G3} = 91{,}800 + 105{,}500 = \underline{197{,}300 \text{ LB}}$$

C. SLIDING FRICTION OF GATE ON RUBBING STRIPS

SLIDING FRICTION $L_G = P_H A_V C_G H$

P_H = HYDROSTATIC PRESSURE PER FOOT OF HEAD

A_V = AREA OF UPSTREAM SKIN PLATE-SQ FT

C_G = COEF OF FRICTION FOR STEEL ON STEEL, ASSUMED TO BE 0.40

H = SWELL HEAD, ASSUMED TO BE 1.0 FT

$L_G = (62.4)(113.16x13)(0.40)(1.0) = 36{,}718$ LB

D. DOWNWARD HYDROSTATIC LOAD

DOWNWARD LOAD = $L_D = P_H A_T H$

P_H = HYDROSTATIC PRESSURE PER FOOT OF HEAD

A_T = AREA, TOP OF GATE-SQ FT

H = SWELL HEAD, ASSUMED TO BE 1.0 FT

$L_D = (62.4)\,[(54.71)(8.0)+(6.56)(1.44)+(4.4)(5.0)]\,[2]$
$= (62.4)[437+9.0+22.0]\,[2]$
$= 58{,}400$ LB

E. WT OF RECESS PROTECTION AND TRASH SCREEN MINUS WT OF WATER DISPLACED.

TRASH SCREENS AND RECESS PROTECTION = 18,200 LB*

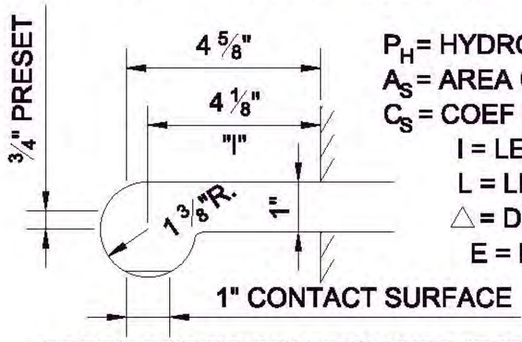
WT OF DISPLACEMENT = $\frac{18{,}200}{489}$ x62.4 = 2,320

NET WEIGHT = 15,900 LB

*ESTIMATE FROM PREVIOUS JOB

F. SIDE SEAL FRICTION

$L_S = 1/2\ P_H A_S C_S H + P_{PRESET} C_S$



P_H = HYDROSTATIC PRESSURE PER FOOT OF HEAD.
A_S = AREA OF SEAL SUBJECT TO DIFF HEAD
C_S = COEF OF FRICTION, RUBBER ON STEEL = 1.0
l = LENGTH OF CANTILEVER = 4-1/8"
L = LENGTH OF SEAL
$\triangle$ = DEFLECTION (PRESET = 3/4")
E = MOD ELASTICITY-RUBBER 60 DURO. = 2,000 PSI t = 1"

FORCE DUE TO SEALS ON SIDE (FOR 1 FT)

$$P_{PRESET} = L\left(\frac{3\triangle EI}{l^3}\right) = L\left(\frac{3(3/4)(2{,}000)(1/12)(12)\,(1)^3}{4.125^3}\right) = L\ (64.2)$$

L = (2 SIDES)(13'-7") = 2x13.58 = 27.16

P_{PRESET} = (27.16) (64.2) = <u>1747</u> LB

$L_S = 1/2\ P_H A_S C_S H + P_{PRESET} C_S \qquad A_S = \left(\frac{4.63}{12}\right)(27.16) = 10.5\ \square FT$

L_S = 1/2 (62.4)(10.5)(1.0)(1.0)+(1747)(1.0)

L_S = 328+1,747 = 2,075 LB (SIDE SEALS)

FORCE DUE TO SEALS IN RECESS (DISREGARD PRESET)

L = 4(7'-4 1/2")+2(5'-0") = 4(7.38)+2(5.00) = 39.5 FT

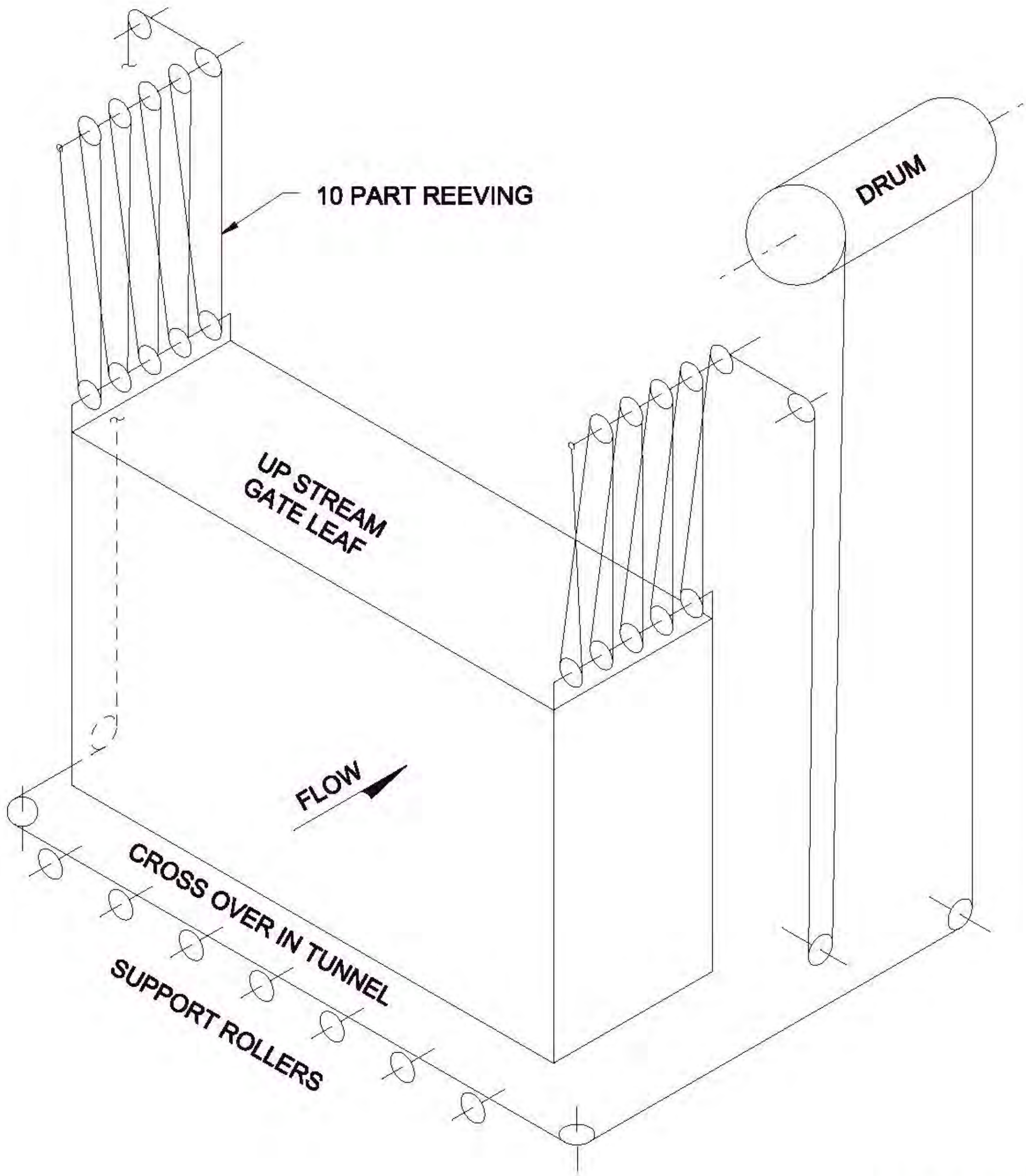
$A_S = \left(\frac{5.50}{12}\right)(39.5) = 18.1\ \square FT$

$L_S = 1/2\ P_H A_S C_S H$ = (1/2)(62.4)(18.1)(1.0)(1.0) = 565 LB

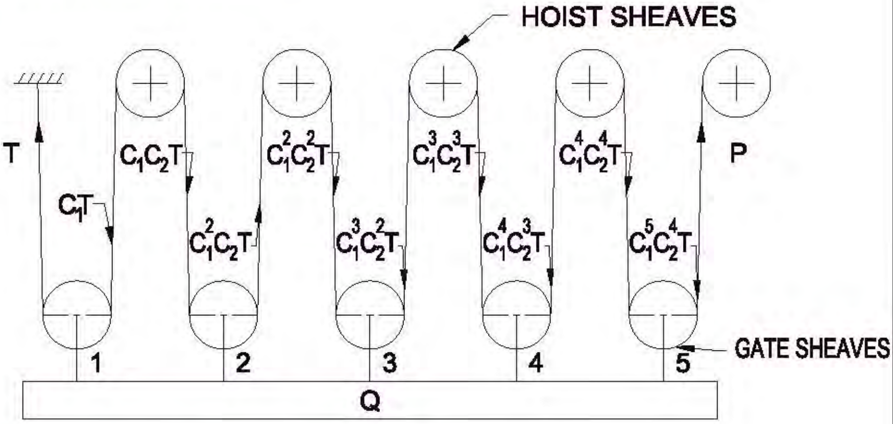
TOTAL SIDE-SEAL FRICTION

L_S = 2,075+565 = <u>2,640 LB</u>

HOIST REEVING FOR RAISING UPSTREAM EMERGENCY GATE.



LOAD IN ROPE THROUGH SHEAVES



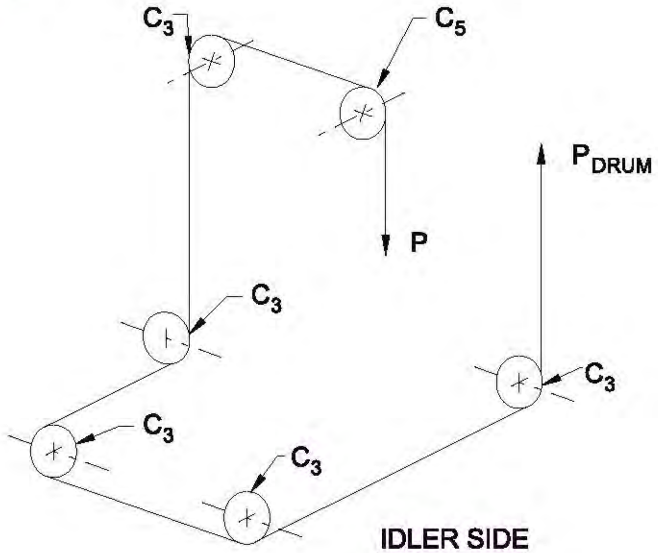
$C_1 = 1.087$ $C_2 = 1.012$ (SEE PLATE B-89, SHEET 12)

$$Q = T+C_1T+C_1C_2T+C_1^2C_2T+C_1^2C_2^2T+C_1^3C_2^2T+C_1^3C_2^3T+$$
$$C_1^4C_2^3T+C_1^4C_2^4T+C_1^5C_2^4T$$
$$= \Big[1+1.087+(1.087)(1.012)+(1.087)^2(1.012)+(1.087)^2(1.012)^2$$
$$+(1.087)^3(1.012)^2+(1.087)^3(1.012)^3+(1.087)^4(1.012)^3$$
$$+(1.087)^4(1.012)^4+(1.087)^5(1.012)^4\Big]T$$
$$= \Big[1.00+1.087+1.10+1.196+1.210+1.315+1.331+1.447$$
$$+ 1.464+1.592\Big] T = 12.742\ T$$

$$T = \frac{Q}{12.742}$$

$$P = C_1^5C_2^4T = (1.087)^5(1.012)^4 \frac{Q}{12.742}$$
$$= (1.518)(1.049) \frac{Q}{12.742} = Q/8.00$$

DRUM LOAD FROM IDLER SIDE



IDLER SIDE

$$P_{DRUM} = C_5 \overset{*}{C}_3^{5} P$$

$$= (1.008)(1.028)^{5} P$$

$$= \frac{(1.008)(1.148)\,Q}{8.00}$$

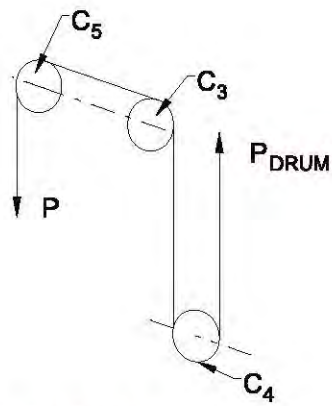
$$= \frac{2}{6.91}$$

REEVING EFF

$$= \frac{6.91 \times 100}{10 \text{ SHEAVES}}$$

$$= 69.1\%$$

DRUM LOAD FROM MACHINERY SIDE



MACHINERY SIDE

$$P_{DRUM} = C_5 \overset{*}{C}_3 \overset{*}{C}_4 P$$

$$= (1.008)(1.028)(1.033)\frac{Q}{8.00}$$

$$= \frac{1.070}{8.00}\,Q = \frac{Q}{7.47}$$

REEVING EFF

$$= \frac{7.47 \times 100}{10} = 74.7\%$$

* $C_3 C_1 C_4$ AND C_5 SEE PLATE B-89, SHEETS 12, 13, AND 15

TOTAL LOAD ON HOIST TO RAISE GATE

CONDITION I

DEAD WEIGHT OF GATE IN AIR	185.0 KIPS
SIDE-SEAL PRESET AND TRASH SCREEN WEIGHT	2.6
WEIGHT OF SILT	197.3
TOTAL LOAD	384.9 KIPS

CONDITION II

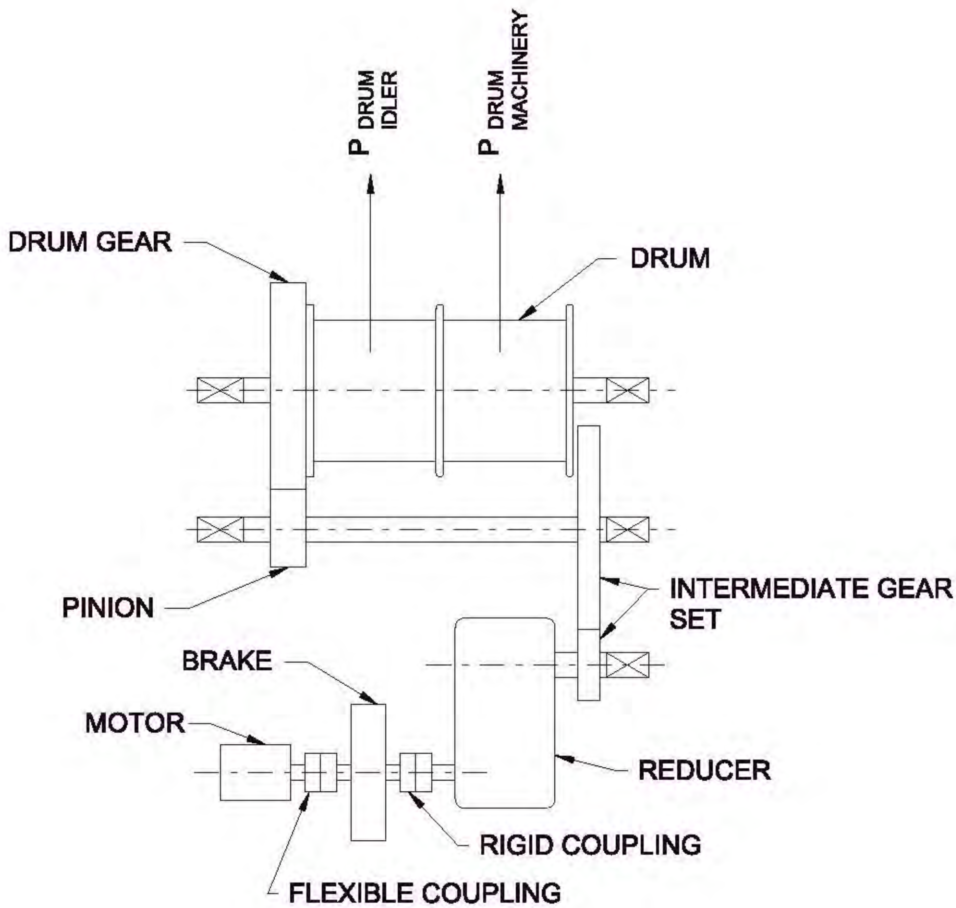
WEIGHT OF UPSTREAM LEAF IN WATER	161.4 KIPS
WEIGHT OF SILT (IN WATER) $197.3\left(\frac{125-62.4}{125}\right)$	98.8
SLIDING FRICTION	36.7
DOWNWARD HYDROSTATIC LOAD	58.4
WEIGHT OF RECESS PROTECTION AND TRASH SCREENS	15.9
SIDE SEAL FRICTION	2.6
TOTAL LOAD FOR DESIGN	373.8 KIPS
	SAY 385.0 (COND. I)

LOAD ON HOIST ROPES AT DRUM

$$P_{DRUM\ IDLER} = \frac{Q}{6.91} = \frac{\frac{385.0}{2}}{6.91} = 27.9\ KIPS$$

$$P_{DRUM\ MACHINERY} = \frac{Q}{7.47} = \frac{\frac{385.0}{2}}{7.47} = 25.8\ KIPS$$

$$P_{DRUM\ TOTAL} = 27.9+25.3 = \underline{53.7\ KIPS}\ (DESIGN\ LOAD)$$

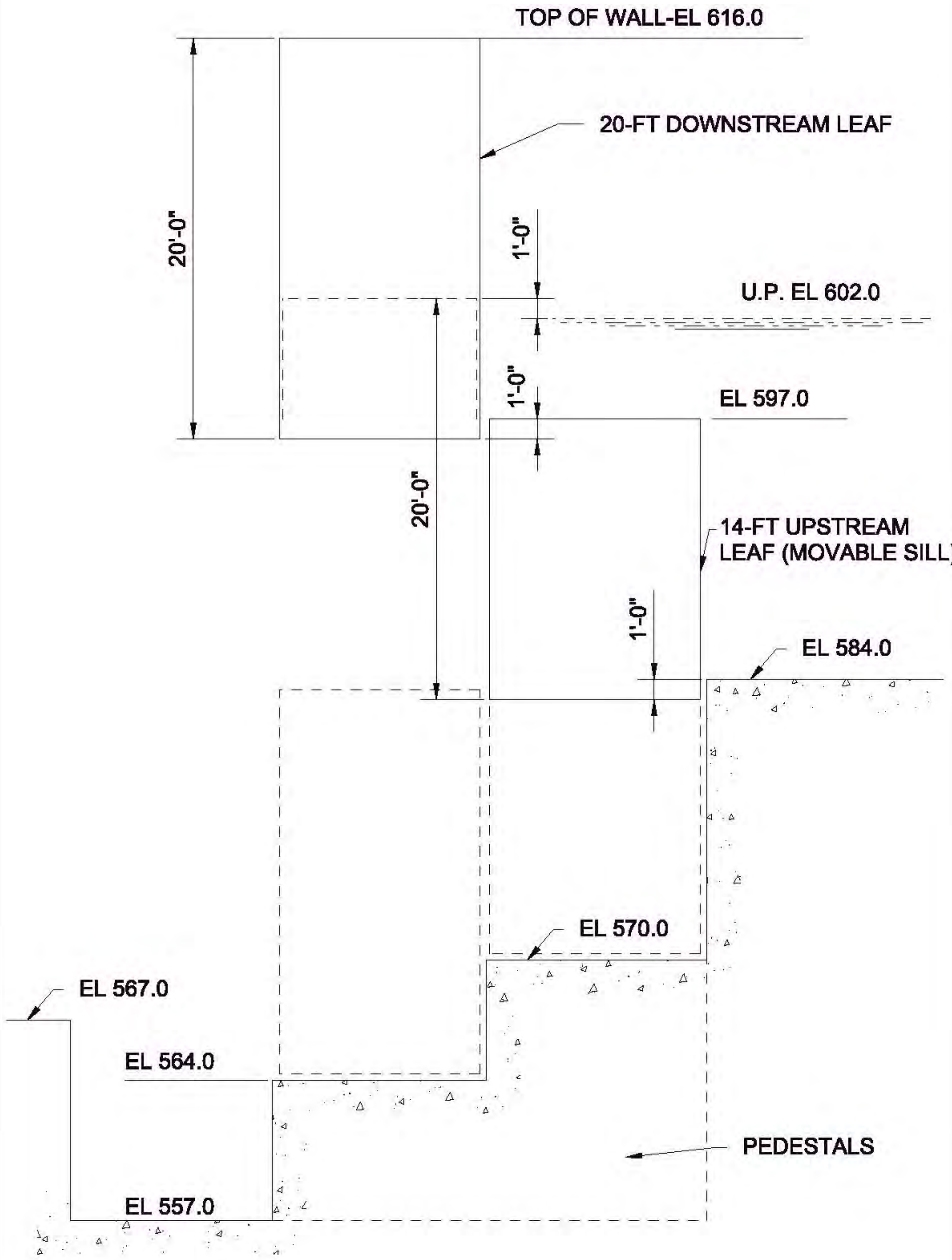


UPSTREAM GATE
HOIST

DETERMINATION OF LOADS REQUIRED TO RAISE DOWN STREAM LEAF OF A DOUBLE-LEAF VERTICAL LIFT GATE.

A. WEIGHT OF DOWNSTREAM LEAF.
B. WEIGHT OF SILT ON DOWNSTREAM LEAF.
C. HORIZONTAL FORCE ON GATE LEAF (HYDRAULIC)
D. VERTICAL FORCE ON GATE LEAF (HYDRAULIC)
E. GENERAL REEVING LAYOUT FOR DOWNSTREAM GATE
F. LOADS ON HOIST ROPE AT DRUM
G. GENERAL LAYOUT OF MACHINE.

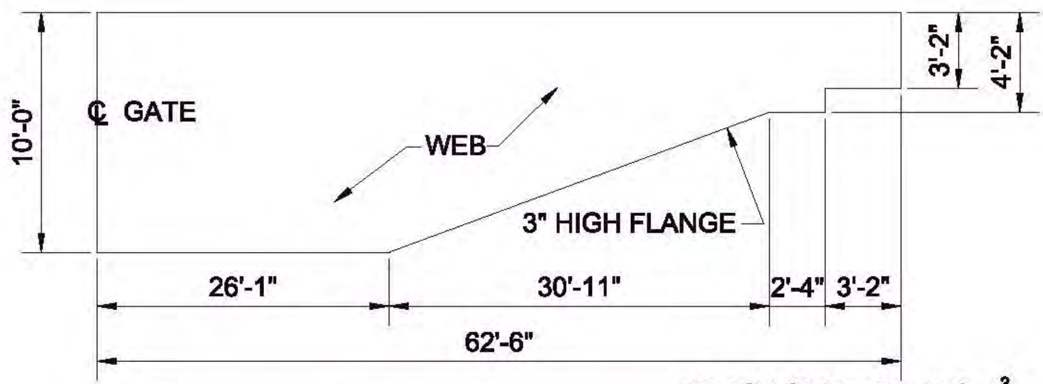
VERTICAL-LIFT GATES WITH UPSTREAM MOVABLE SILL



A. WEIGHT OF DOWNSTREAM LEAF

FROM PRELIMINARY CALCULATIONS, WEIGHT OF 20'x110' GATE WAS ESTIMATED TO BE 467,000 LB

B. WEIGHT OF SILT.



$\frac{1}{2}$ GIRDER G-1

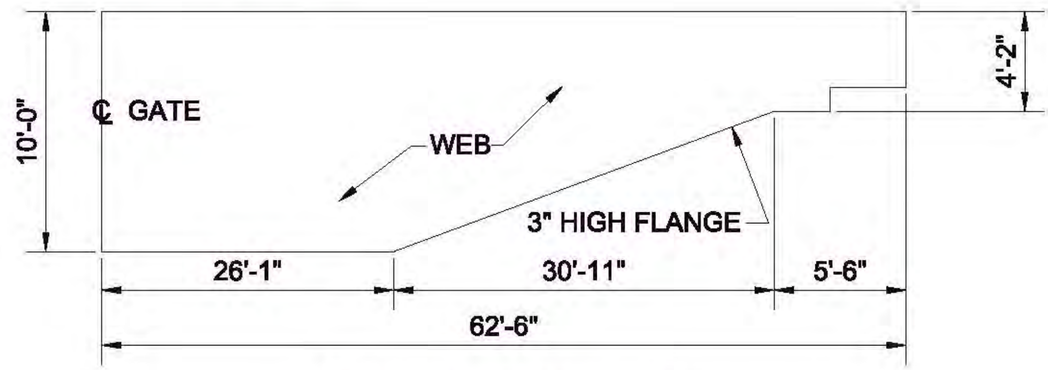
WT OF SILT ON G-1

$$W_{G\text{-}1} = 2(125)\left(\frac{3}{12}\right)\left[(26.08)(10.0)+\left(\frac{10.0+4.17}{2}\right)(30.92)+(4.17)(2.33)+(3.17)(3.17)\right]$$

$$= (62.5)[260.8+219.07+9.7+10.0]$$

$$= 62.5\text{x}499.57 = \underline{31{,}223\text{ LB}}$$

WT OF SILT (CONT'D)



$\frac{1}{2}$ GIRDER G-2, G-3, AND G-4

WT. OF SILT ON G-2, G-3, AND G-4

$$= (3)(2)(125)\left(\frac{3}{12}\right)\left[(26.08)(10.0)+\left(\frac{10.0+4.17}{2}\right)(30.92)+(5.50)(4.17)\right]$$

$$= (187.5)[260.8+219.1+22.9]$$

$$= 187.5\text{x}502.8 = \underline{94{,}275 \text{ LB}}$$

TOTAL WT OF SILT ON GATE = 31,223+94,275 = 125.5 KIPS

DEAD WT OF DOWNSTREAM LEAF (TOTAL)

WT = 467.0+125.5 = 593.0 KIPS

C. HORIZONTAL FORCE ON GATE LEAF (HYDRAULIC)

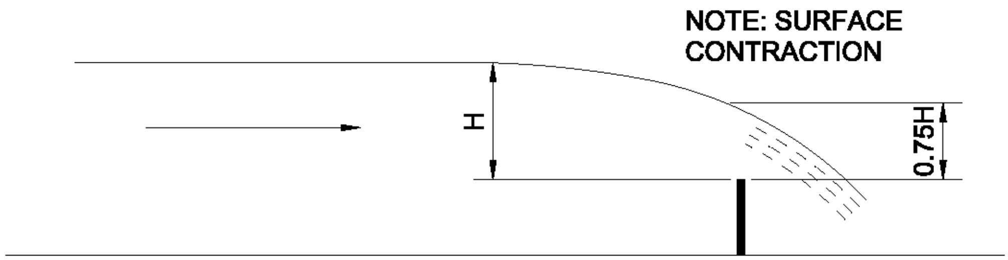
USING THE DATA SUPPLIED BY HYDRAULICS BRANCH IN GREEN UP NAVIGATION DAM-ANALYSIS OF HYDRODYNAMIC FORCES ACTING ON EMERGENCY GATE (DATED 7/15/54) AND SUPPLEMENTING THEIR TABLES WITH INTERMEDIATE VALUES, THE HORIZONTAL FORCES ON EMERGENCY GATE VS. FEET GATE RAISED WERE COMPUTED FOR THIS LOCK AND ARE AS FOLLOWS:

AS SOON AS THE EMERGENCY GATE IS LIFTED THE WEIR BECOMES SHARP CRESTED. USE C = 3.5

Q = FLOW IN CFS
L = WIDTH OF WEIR
H = HEIGTH OF WATER ABOVE WEIR

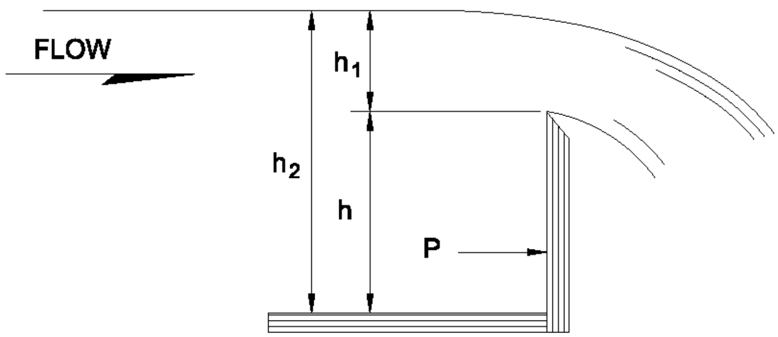
$Q = CL(H)^{\frac{3}{2}}$ (USE END CONTRACTION SUPPRESSED TO GIVE MAX. Q-HANDBOOK OF HYDRAULIC'S BY KING AND BRATER, PAGE 5-3

$Q = (3.5)(110)(H)^{\frac{3}{2}} = 385\,(H)^{\frac{3}{2}}$



1	2	3	4	5	6
EMERGENCY GATE RAISED (FT)	ELEVATION TOP OF GATE	H = 602- 2	$(H)^{\frac{3}{2}}$	3.5 L = 3.5x110	Q = $3.5L(H)^{\frac{3}{2}}$ 4 x 5
O	584	18	76.36	385	29,401
1	585	17	70.09		26,980
2	586	16	64.00		24,640
3	587	15	58.09		22,360
4	588	14	52.38		20,170
5	589	13	46.87		18,040
6	590	12	41.57		16,000
7	591	11	36.50		14,060
8	592	10	31.62		12,170
9	593	9	27.00		10,400
10	594	8	22.63		8,713
11	595	7	18.60		7,160
12	596	6	14.70		5,659
13	597	5	11.20		4,312
14	598	4	8.00		3,080
15	599	3	5.20		2,000
16	600	2	2.83		1,089
17	601	1	1.00		385
18	602	0	0		0

HORIZONTAL FORCE ON EMERGENCY GATE (CONT'D)



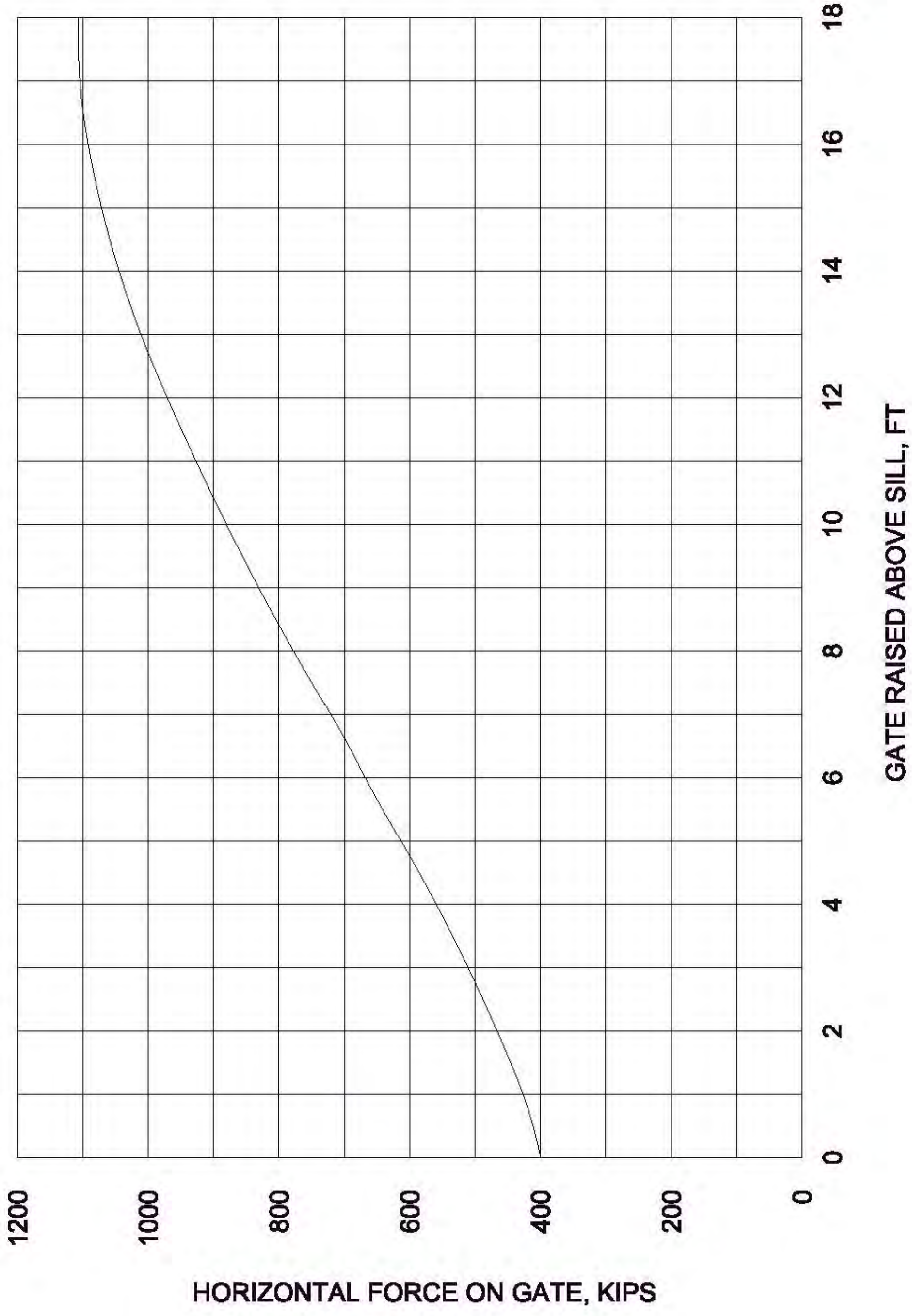
THE FORCE ON THE EMERGENCY GATE WAS COMPUTED USING THE FORMULAS FOR LOADS ON "SHARP-CRESTED WEIRS" FROM ENGINEERING FOR DAMS, VOL.2 (1945) BY HINDS, CREAGER, AND JUSTIN, PAGES 255-257.

$$P = 1/2\, h_2^2\, w_2 - q/g\, w_2 \left(k\sqrt{2gh_1} - q/h_2\right)$$

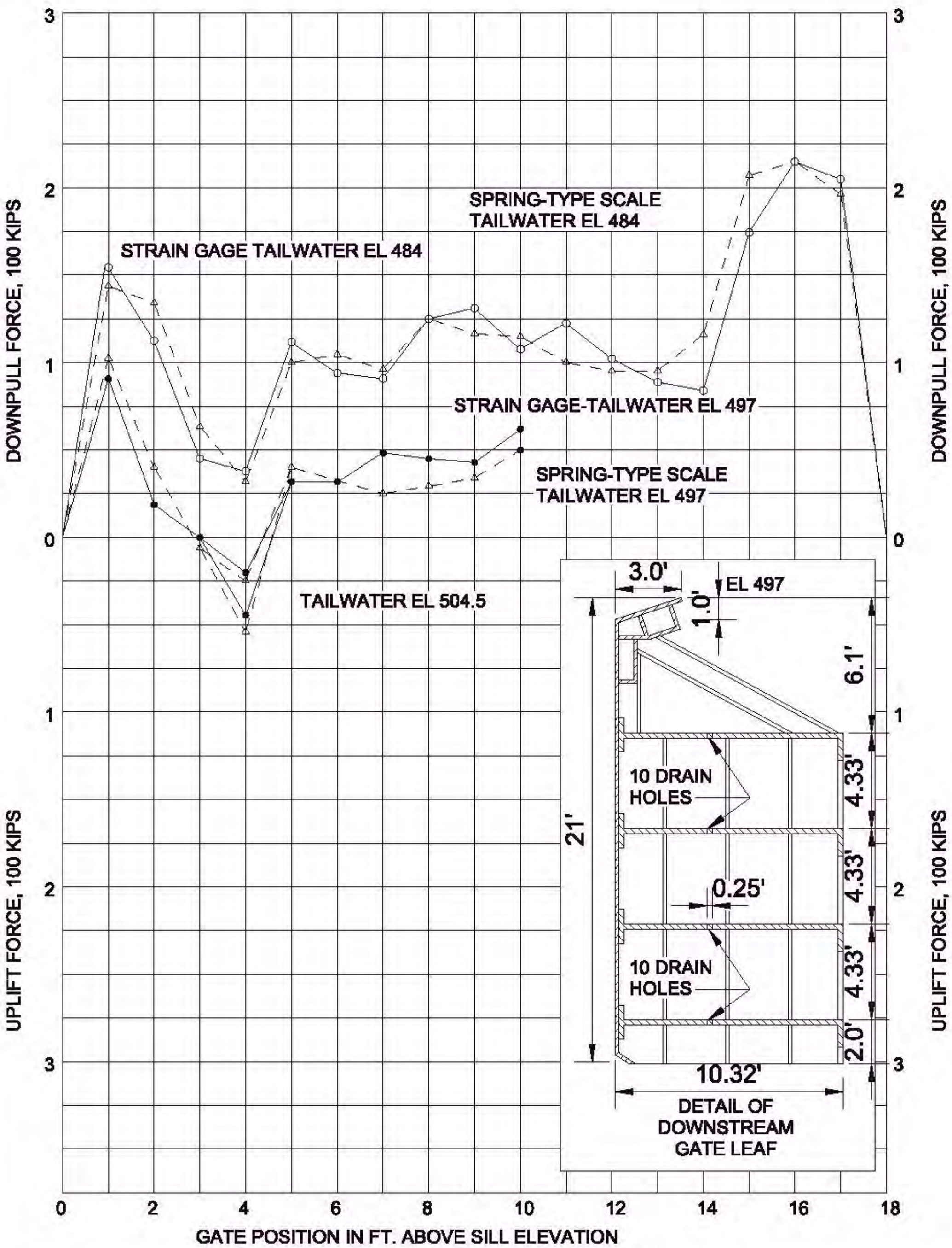
P' = TOTAL HORIZONTAL LOAD ON GATE
P = HORIZONTAL LOAD ON 1-FT WIDTH OF GATE
h_2 = UPSTREAM HEAD IN FEET
W_2 = WEIGHT OF WATER-62.4 LB/CU FT
q = FLOW IN 1-FT OF CHANNEL (SEE PRECEDING PAGE)
g = ACCELERATION DUE TO GRAVITY-32.2 FT/SEC/SEC
K = 0.81 (DETERMINED BY EXPERIMENT)
h_1 = UPSTREAM HEAD ABOVE WEIR (SEE H, COL 3 OF PRECEDING PAGE)

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
GATE RAISED (FT) h	Q (CFS)	q $=\frac{Q}{110}$	$\frac{W_2}{g}q$ =1.938 (3)	h_1 $=h_2-h$ =18-(1)	$\sqrt{h_1}$	$K\sqrt{2g}\sqrt{h_1}$ =6.49 (6)	h_2	$\frac{q}{h_2}$ $\frac{(3)}{(8)}$	$K\sqrt{2gh_1}$ $-\frac{q}{h_2}$ (7)-(9)	$\frac{q}{g}W_2(K\sqrt{2gh}$ $-q/h_2)$ (4) x (10)	$\frac{1}{2}h_2^2$ $\frac{(8)^2}{2}$	$\frac{1}{2}h_2^2 xw_2$ (12) x62.4	P(1FT) (13)-(11)	P^1 (TOTAL LOAD) (14)x110'
0	29,400	267	518	18	4.24	27.5	18	14.8	12.7	6,561	162	10,109	3,548	390KIPS
1	26,980	246	477	17	4.12	26.7		13.7	13.0	6,200			3,909	430
2	24,640	224	434	16	4.00	26.0		12.5	13.5	5,860			4,249	467
3	22,360	204	395	15	3.87	25.2		11.3	13.9	5,490			4,619	508
4	20,170	184	357	14	3.74	24.3		10.2	14.1	5,030			5,079	559
5	18,040	164	318	13	3.61	23.4		9.1	14.3	4,550			5,559	611
6	16,000	145	281	12	3.46	22.5		8.1	14.4	4,040			6,069	667
7	14,060	128	248	11	3.32	21.6		7.1	14.5	3,600			6,509	716
8	12,170	111	215	10	3.16	20.5		6.2	14.3	3,070			7,039	774
9	10,400	95	184	9	3.00	19.5		5.3	14.2	2,610			7,499	825
10	8,713	79	153	8	2.83	18.4		4.4	14.0	2,140			7,969	876
11	7,160	65	126	7	2.65	17.2		3.6	13.6	1,710			8,399	923
12	5,659	51	99	6	2.45	15.9		2.8	13.1	1,300			8,809	968
13	4,320	39	76	5	2.24	14.6		2.2	12.4	942			9,167	1,008
14	3,080	28	54	4	2.00	13.0		1.6	11.4	615			9,494	1,043
15	2,000	18	35	3	1.73	11.2		1.0	10.2	480			9,629	1,060
16	1,089	9.9	19	2	1.41	9.2		0.6	8.6	164			9,945	1,093
17	385	3.5	6.8	1	1.0	6.5		0.2	6.3	43			10,066	1,108
18	0	0	0	0	0	0	0	0	0	0			10,109	1,112

K = 0.8


GATE RAISED ABOVE SILL, FT
0
2
4
6
8
10
12
14
16
18
HORIZONTAL FORCE ON GATE, KIPS
1200
1000
800
600
400
200
0

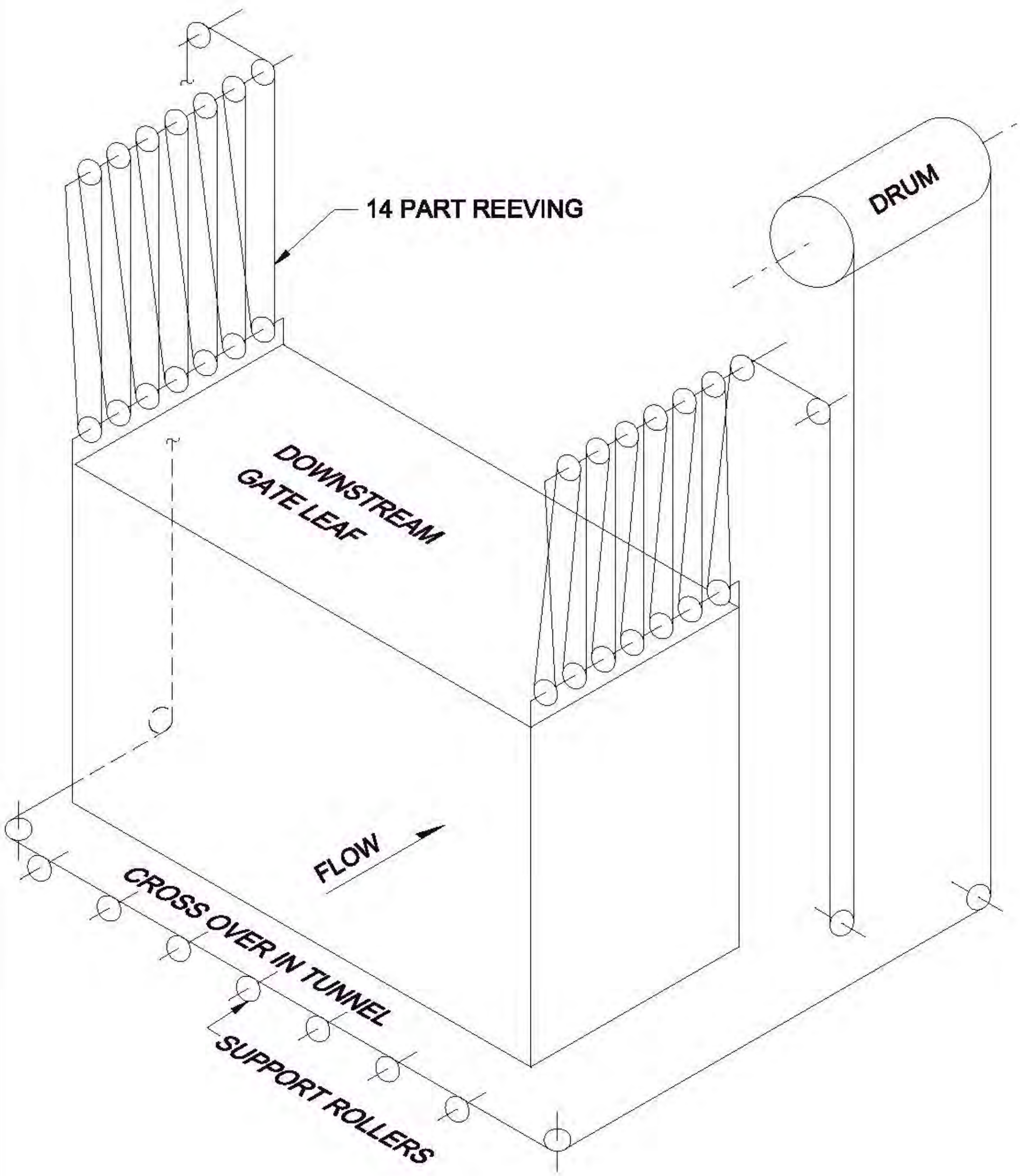
HYDRAULIC FORCES
TRIANGULAR-SHAPED
GATE CREST-TYPE 9 (FINAL) DESIGN



NOTE: OPEN SYMBOLS DESIGNATE LOW TAILWATER. CLOSED SYMBOLS DESIGNATE HIGH TAILWATER GATE SILL ELEVATION 497. DURING OPERATION OF GATE, UPPER POOL ELEVATION WAS MAINTAINED AT 515. AERATION TUBES OPEN DURING TESTS.

EMERGENCY GATE, GREEN UP LOCKS, OHIO RIVER, KENTUCKY. LOAD ON DOWNSTREAM LEAF (COM-P, FOR RACINE LOCKS)

E. HOIST REEVING FOR RAISING DOWNSTREAM EMERGENCY GATE.



TENSION RATIOS OF WIRE ROPE SHEAVES

(FROM MACHINE DESIGN BY BLACK AND ADAMS, THIRD EDITION, PAGES 293 THROUGH 299)

$$C = \frac{D+fd+2e}{D-fd-2e'}$$

WHERE
C = TENSION RATIO
D = PITCH DIAMETER OF SHEAVE
f = COEFFICIENT OF FRICTION
d = DIAMETER OF PIN
e AND e' = OFFSET OF ROPE DUE TO BENDING

NOTE: e AND e' ARE ASSUMED TO BE NEGLIGIBLE. DIMENSIONS AND ARE ESTIMATED AND MUST BE CHECKED.

<u>GATE SHEAVES</u> <u>TENSION RATIO C_1</u>

D = 36" d = 11.5"
f (BRONZE ON BRONZE) = 0.133
<u>REF.</u> KENT'S MECH. ENG'R HANDBOOK, 1961, PAGE 7-28

$$C_1 = \frac{36+(0.133)\,(11.5)}{36-(0.133)\,(11.5)} = \frac{36.0+1.53}{36.0-1.53} = \frac{37.53}{34.47} = \underline{1.087}$$

<u>HOIST SHEAVES</u> <u>TENSION RATIO C_2</u>

D = 36" $d \cong 14\frac{3}{4}$ <u>180° TURN</u>
f = 0.0131 (ROLLER BEARING)
<u>REF.</u> KENT'S MECH. ENG'R HANDBOOK, 1961, PAGE 7-31

$$C_2 = \frac{36+(0.0131)(14.75)}{36-(0.0131)(14.75)} = \frac{36.0+0.193}{36.0-0.193} = \frac{36.2}{35.8} = \underline{1.012}$$

<u>90° TURN</u>

$$C_5 = \frac{36+(0.0131)(14.75)(0.707)}{36-(0.0131)(14.75)(0.707)} = \frac{36+0.137}{36-0.137} = \frac{36.14}{35.86} = \underline{1.008}$$

TENSION RATIOS OF WIRE ROPE SHEAVES (CONT'D)

90° TURN

D = 36"   d = 4.75

f = 0.133 (BRONZE BUSHED)

$$C_3 = \frac{36.0+(0.133)(4.75)(0.707)}{36.0-(0.133)(4.75)(0.707)} = \frac{36.0+0.446}{36.0-0.446}$$

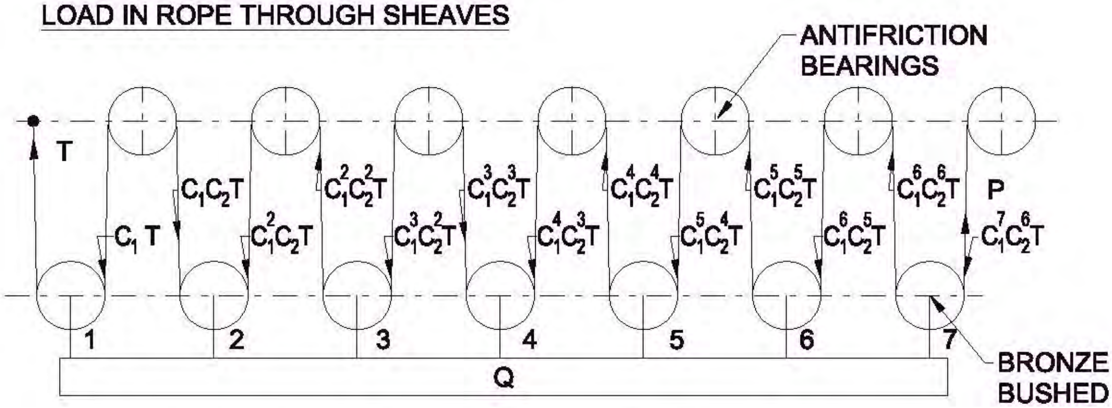
$$\frac{36.45}{35.55} = \underline{1.025}$$

180° TURN

$$C_4 = \frac{36.0+(0.133)(4.75)}{36.0-(0.133)(4.75)} = \frac{36.0+0.632}{36.0-0.632} = \frac{36.63}{35.37}$$

$$= \underline{1.036}$$

LOAD IN ROPE THROUGH SHEAVES



C_1 = LOWER SHEAVES TENSION RATIO COEF.

C_2 = UPPER SHEAVES TENSION RATIO COEF.

ASSUME GATE SHEAVE NO. 4 CARRIES 1/7 OF THE LOAD.

$$\frac{Q}{7} = C_1^3C_2^3T + C_1^4C_2^3T$$

$$\frac{Q}{7} = \left(C_1^3C_2^3 + C_1^4C_2^3\right)T$$

$$T = \frac{Q}{7\left(C_1^3C_2^3 + C_1^4C_2^3\right)}$$

$$= \frac{Q}{7\left[(1.087)^3(1.012)^3 + (1.087)^4(1.012)^3\right]}$$

$$= \frac{Q}{7\left[(1.285)(1.036) + (1.397)(1.036)\right]}$$

$$= \frac{Q}{7\left[(1.331) + (1.446)\right]} = \frac{Q}{7\ (2.777)}$$

$$= \frac{Q}{19.4}$$

LOAD IN ROPE THROUGH SHEAVES (CONT'D)

$$P = C_1^7 C_2^6 7T = (1.087)^7 (1.012)^6 \frac{Q}{19.4}$$

$$P = (1.793)(1.074)\frac{Q}{19.4} = \frac{Q}{10.1}$$

DRUM LOAD FROM IDLER SIDE

$$P_{DRUM} = C_5 C_3^5 P$$

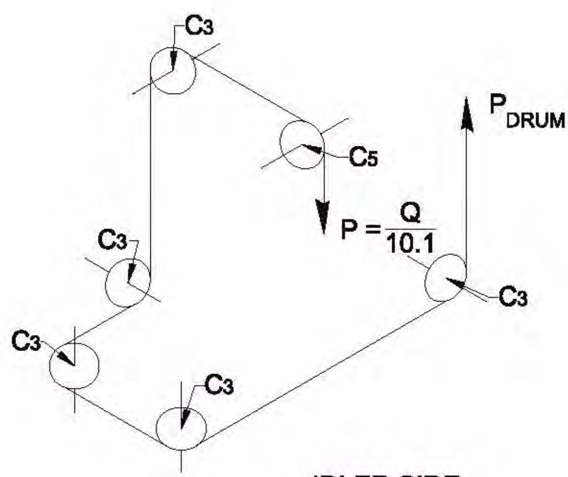
$$= (1.008)(1.028)^5 P$$

$$P_{DRUM} = \frac{(1.008)(1.148)\,Q}{10.1}$$

$$P_{DRUM} = \frac{Q}{8.72}$$

$$\text{REEVING EFF.} = \frac{8.72 \times 100}{14}$$

$$= 62.2\%$$



IDLER SIDE

DRUM LOAD FROM MACHINERY SIDE

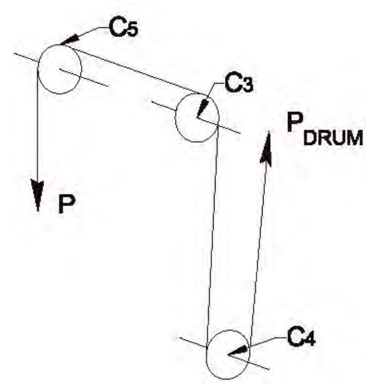
$$P_{DRUM} = C_2 C_3 C_4 P$$

$$= (1.008)(1.028)(1.033)\ \frac{Q}{10.1}$$

$$= \frac{1.070}{10.1} Q$$

$$P_{DRUM} = \frac{Q}{9.44}$$

$$\text{REEVING EFF.} = \frac{9.44 \times 100}{14.0} = 67.4\%$$

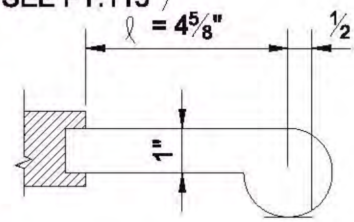


1	2	3	4	5	6	7	8	9	10
FT ABOVE SILL	$\frac{1}{2}$ $W_T/2$ DEAD W_T OF GATE & SILT KIPS	$\frac{1}{2}$ P/2 FORCE NORMAL TO GATE KIPS	LS FRICTION FORCE OF ONE SEAL KIPS	$\frac{1}{2}$ LR FRICTION FORCE OF ROLLERS KIPS	$\frac{1}{2}$ $F_{0/2}$ DOWN-WARD HYD. FORCE SEE PL. 109 KIPS	LOAD ON ONE GATE SHEAVE ASSY. 2 + 4 + 5 + 6 =Q KIPS	P_{DRUM} IDLER SIDE ROPE LOAD AT DRUM 7 ÷ 8.72 KIPS	P_{DRUM} MACH. SIDE ROPE LOAD AT DRUM 7 ÷ 9.44 KIPS	P_{DRUM} TOTAL 8 + 9 KIPS
0	297	197	0	9.90	0	307	35.2	32.5	67.7
1		215	1.65	10.75	78	387	44.4	41.0	85.4
2		234	1.74	11.70	57	367	42.1	38.9	81.0
3		254	1.81	12.70	23	335	38.4	35.5	73.9
4		280	1.91	14.01	19	332	38.1	35.2	73.3
5		306	2.01	15.31	57	371	42.5	39.3	81.8
6		334	2.12	16.70	47	363	41.6	38.5	80.1
7		358	2.21	17.90	46	363	41.6	38.5	80.1
8		387	2.32	19.36	63	382	43.8	40.5	84.3
9		413	2.43	20.70	67	387	44.4	41.0	85.4
10		438	2.53	21.90	56	377	43.2	39.9	83.1
11		462	2.62	23.10	62	385	44.2	40.8	85.0
12		484	2.70	24.20	52	376	43.1	39.8	82.9
13		504	2.78	25.20	44	369	42.3	39.1	81.4
14		522	2.85	26.10	39	365	41.9	38.7	80.6
15		530	2.88	26.50	87	413	47.4	43.8	91.2
16		547	2.94	27.40	109	436	50.0	46.2	96.2
17		554	2.97	27.70	104	432	49.5	45.8	95.3
18		556	2.98	27.80	14	342	39.2	36.2	75.4

EXAMPLE COMPUTATION USING GATE RAISED 1 (ONE) FT
COL 2-DEAD WEIGHT OF 1/2 OF GATE (INCLUDING SILT)
W/2 = 593.0/2 = 297 KIPS
COL 3-NORMAL FORCE ON 1/2 OF GATE
SEE GRAPH "HORIZONTAL FORCES ON EMERGENCY GATE VS GATE RAISED," PLATE B-89, SHEET 9.
P 1/2 = 430/2 = 215 KIPS

COL 4-FRICTION FORCE OF ONE SEAL (FOR LOCATION OF COL. NUMBERS SEE PT.113)

C_S= 1.0 (DESIGN CRITERIA)



FORCE DUE TO WATER PRESSURE (P_S)

$$P_S = \frac{(F_N)(A_S)(C_S)}{2\,(A_P)}$$

F_N= HORIZONTAL FORCE NORMAL TO GATE

A_S= AREA (SQ FT) SEAL SUBJECT TO DIFFERENTIAL HEAD

A_P= AREA (SQ FT) GATE RAISED ABOVE SILL

$$P_S = \frac{(430)\,(1.0)\,(5\tfrac{1}{8}/12)\,(1.0)}{(2)\,(110.5)\,(1.0)} = 0.830^K$$

FORCE DUE TO PRESET OF 3/4" (P_P)

$$P_P = \frac{3\triangle EI}{\ell 3}\,(L)\,(C_3)$$

L = LENGTH OF SEAL COMPRESSED

E = 2000-ELASTIC MOD.-DUROMETER 60-(SEE KENT'S 5-68)

t = 1"

$$= \frac{3\,(0.75)\,(2000)\,[\tfrac{1}{12}(12")\,(1.0")^3]\,(18')\,(1.0)}{4.625^3}$$

$$= 0.820^K$$

$L_S(TOTAL) = P_S+P_P = 0.830+0.820 = 1.65^K$

COL 5-ONE HALF FRICTION FORCE OF ROLLERS (L_R)

$$L_R = \frac{F_N}{2}\,(C_R) \text{ WHERE } C_R = 0.05 \text{ (DESIGN CRITERIA)}$$

$$= \frac{430}{2}\,(0.05) = 10.75^K$$

COL 6-HYDRAULIC DOWNWARD FORCE ON ONE-HALF GATE-(FD/2)

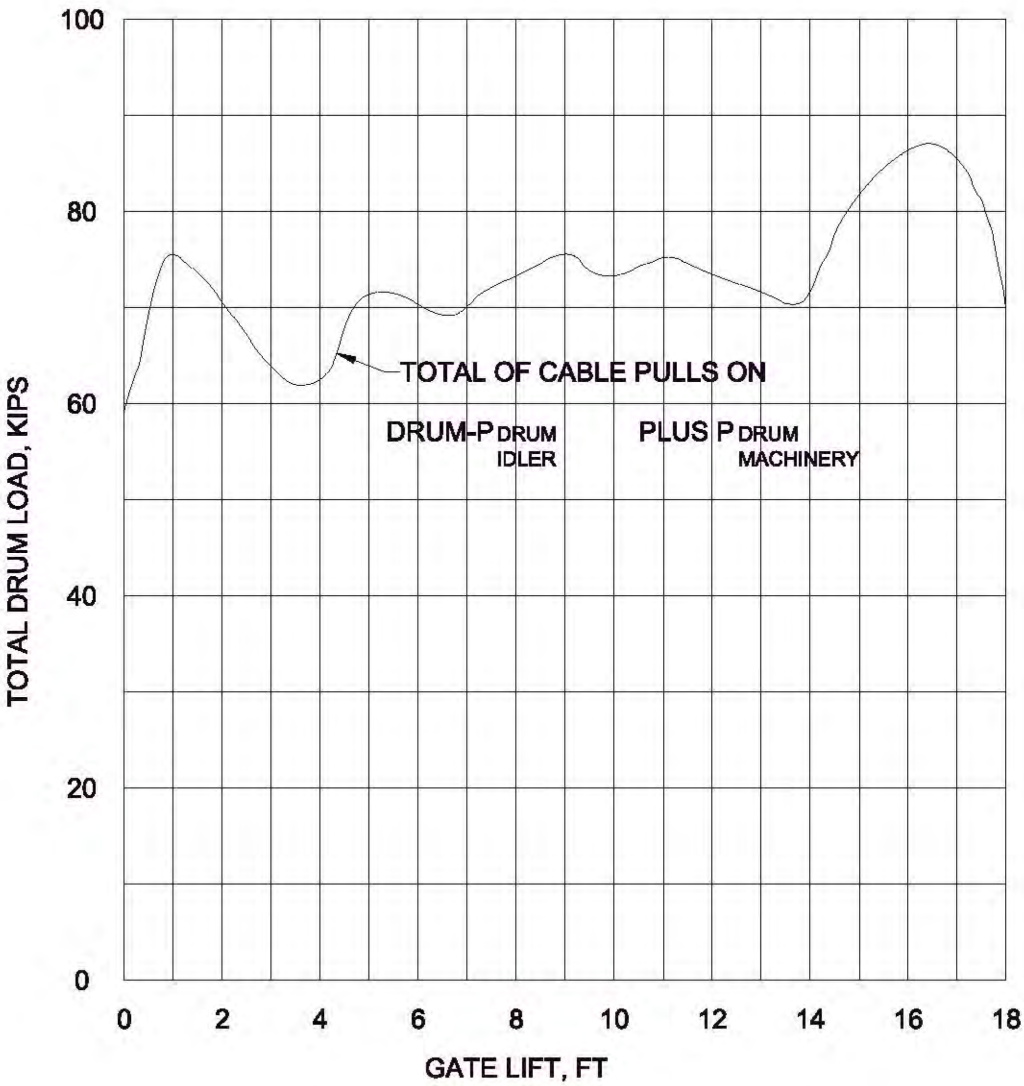
SEE GRAPH "FORCE UPLIFT OR DOWNPULL VS GATE POSITION ABOVE SILL" (PLATE 107)

$$\frac{F_D}{2} = \frac{156}{2} = 78^K$$

COL 7-TOTAL LOAD ON ONE GATE SHEAVE ASSEMBLY

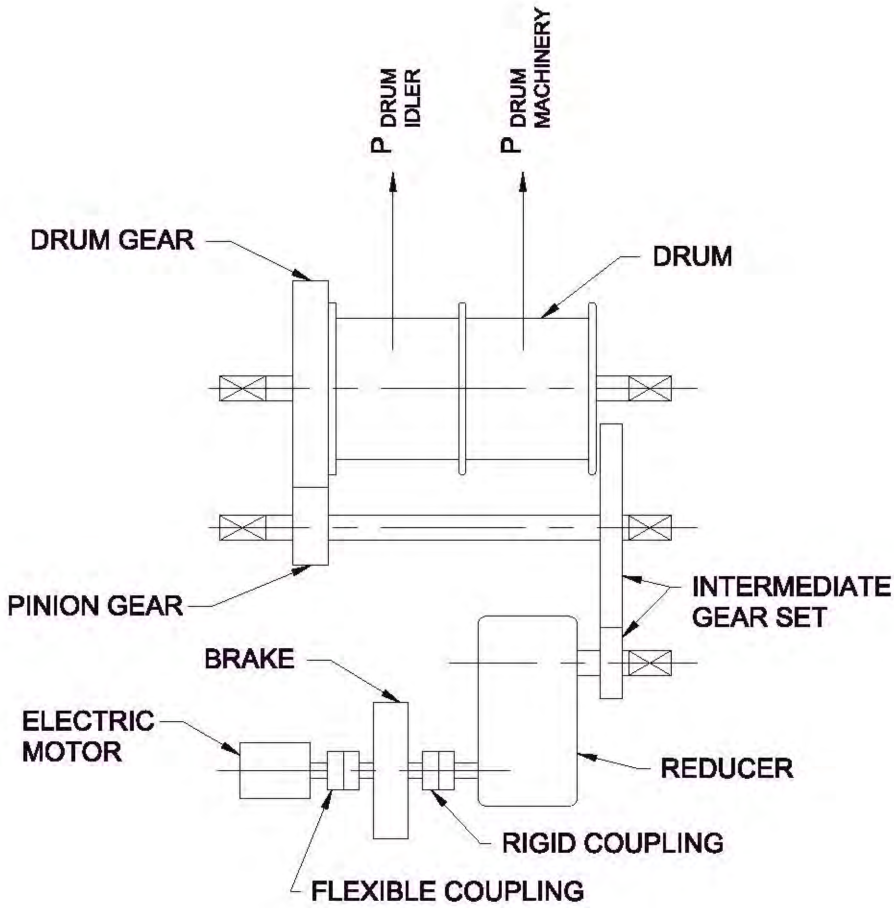
$$Q = \frac{W}{2}\; L_S + L_R + L_{HYD} = 387^K$$

DOWNSTREAM GATE
GATE LIFT VS. DRUM LOAD
DESIGN LOAD



DOWNSTREAM GATE HOIST

GENERAL LAYOUT OF HOIST MACHINE



VERTICAL LIFT GATES,
DESIGN OF WHEELS

GATE WHEELS (CROWNED TREAD)

LOAD ON WHEEL-MAXIMUM WHEEL LOAD FOR THE FOLLOWING EXAMPLE IS 337.5 KIPS (WHEEL LOADS ARE DETERMINED BY THE STRUCTURAL DESIGNER)

WHEEL-ASSUME THE DIAMETER OF TREAD TO BE 36"

WIDTH OF TREAD:

$$\text{WIDTH} = \frac{\text{LOAD}}{\text{800xDIAM (INCHES)}} = \frac{337.5\text{x}1000}{800\text{x}36}$$

= 11.75" USE 12"

SEE ROARK, "FORMULAS FOR STRESS AND STRAIN, 2ND. EDITION" PAGE 292, PARAGRAPH 2, WHERE MATERIAL IS EQUAL TO "TOOL STEEL," LOADING IS "SLOW MOTION," AND TYPE OF LOADING IS "CYLINDRICAL ROLLER ON FLAT PLATE".

COMPRESSIVE STRESS AT POINT OF CONTACT

ASSUME THE SURFACE OF THE WHEEL IS TO BE MADE OF WELD DEPOSIT OF E410 OR ER420 ELECTRODES. THESE ELECTRODES ARE 12% CHROMIUM AND GIVE GOOD RESISTANCE TO CORROSION, EROSION, AND ABRASION. SEE WELDING HANDBOOK, SECTION 5, SIXTH EDITION, PAGE 94-29, BY AMERICAN WELDING SOCIETY, 1973.

THE SURFACE OF THE ROLLER SHOULD HAVE A BRINELL HARDNESS OF APPROXIMATELY 360, WHICH IS NORMALLY OBTAINED BY AIR HARDENING.

THE DISTANCE FROM THE CENTER OF WHEEL, X, THAT THE REACTION ON THE TRACK WOULD OCCUR FOR MAXIMUM SLOPE AT VARIOUS CROWN RADII OF WHEEL TREAD.

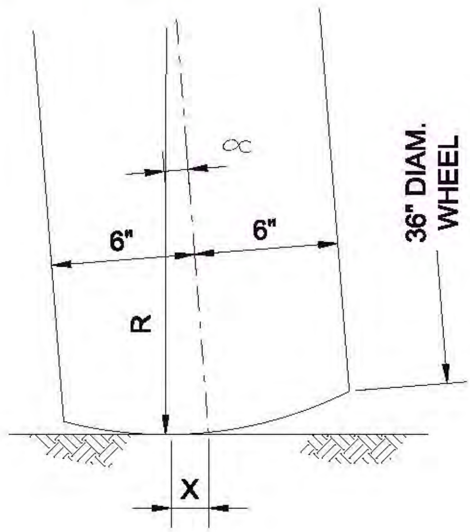
THE SLOPE OF THE GATE UNDER MAXIMUM LOADING CONDITIONS WAS CALCULATATED TO BE AS FOLLOWS:

α (SLOPE) = 0°-12' (AT END OF GATE)

Y MAX = 2.457" (TANα xGATE LENGTH+2)

$\triangle$ MAX = 1.17" (DEFLECTION-CENTER OF GATE)

R = CROWN RADIUS
WHEEL TREAD



CROWN RADIUS-R	SIN 0°-12'	DISTANCE X (IN.)
20'-0" = 240"	0.00349	0.838
30'-0" = 360"		1.257
40'-0" = 480"		1.675
50'-0" = 600"		2.090
60'-0" = 720"		2.520
70'-0" = 840"		2.930
80'-0" = 960"		3.350

DETERMINE COMPRESSIVE STRESS AT CENTER OF ELLIPTICAL CONTACT SURFACE OF GATE WHEEL RUNNING ON A FLAT PLATE TRACK.

A COMPUTER PROGRAM MAY BE USED TO RAPIDLY COMPUTE THE MAJOR AXIS OF ELLIPSE, MINOR AXIS OF ELLIPSE, MAXIMUM COMPRESSIVE STRESS, AND TOTAL DEFLECTION OF BOTH BODIES.

THE PROGRAM SHOULD BE WRITTEN USING FORMULA'S WHICH ARE TAKEN FROM ROARK, FORMULAS FOR STRESS AND STRAIN, 1965 EDITION, PAGE 321, CASE NO. 8, "GENERAL CASE OF TWO BODIES IN CONTACT."

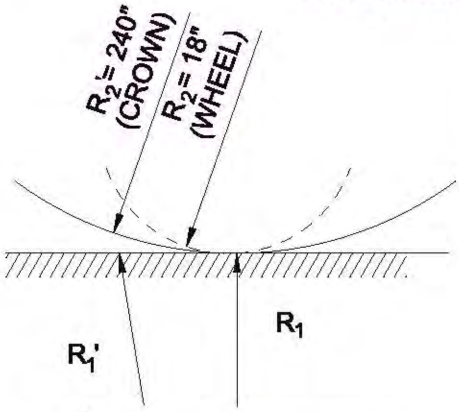
THE PROGRAM SHOULD ASSUME, FOR THIS PROBLEM, A MODULES OF ELASTICITY FOR BOTH WHEEL AND FLAT PLATE TRACK TO BE $30x10^6$ PSI AND POISSON'S RATIO FOR BOTH BODIES TO BE 0.26.

THE COMPUTER READOUT WOULD BE AS FOLLOWS:

WHEEL DIAM (IN.)	CROWN RADIUS FT	MAX. COMP STRESS KPSI *	MAJ AXIS ELLIPSE (IN.)	MIN AXIS ELLIPSE (IN.)	TOTAL DEFLECTION (IN.)
36.00	20.00	184	4.34	0.80	0.0142
	30.00	164	5.30	0.73	0.0131
	40.00	158	5.86	0.69	0.0122
	50.00	154	6.30	0.66	0.0116
	60.00	146	6.84	0.64	0.0111
	70.00	142	7.27	0.62	0.0107
	80.00	139	7.62	0.60	0.0104

* MAXIMUM WHEEL REACTION = 337.5 KIPS

SAMPLE COMPUTATION USING 36-IN. DIAMETER WHEEL, 20 FT CROWN RADIUS, AND 337.5 KIPS WHEEL LOAD (USING FORMULAS FROM ROARK)



(R_1' AND R_1 ARE INFINITE)

$$\delta = \frac{4}{\frac{1}{INF.} + \frac{1}{R_2} + \frac{1}{INF.} + \frac{1}{R_2'}} = \frac{4}{\frac{1}{18} + \frac{1}{240}}$$

$$= \frac{4}{0.055555+0.004167} = \frac{4}{0.0597216}$$

$$= 66.9774$$

$$K = \frac{8}{3}\,\frac{E_1E_2}{E_2(1-\nu^2)+E_1(1-\nu^2)}$$

$$E_1 = E_2 = 30\times10^6 \qquad \nu = 0.26$$

$$K = 2.66667 \quad \frac{(30\times10^6)\,(30\times10^6)}{[30\times10^6(1-0.26^2)]+[30\times10^6(1-0.26^2)]}$$

$$= 2.66667 \quad \frac{9\times10^{14}}{5.5944\times10^7} = 2.66667\,(1.60875\times10^7)$$

$$= 4.29\times10^7$$

$$\theta = \text{ARC COS } 1/4\,\delta\sqrt{\left(\frac{1}{R_1}-\frac{1}{R_1'}\right)^2+\left(\frac{1}{R_2}-\frac{1}{R_2'}\right)^2+2\left(\frac{1}{R_1}-\frac{1}{R_1'}\right)\left(\frac{1}{R_2}-\frac{1}{R_2'}\right)\text{COS}2\varnothing}$$

$$= \text{ARC COS } \frac{66.9774}{4}\sqrt{\left(\frac{1}{INF.}-\frac{1}{INF.}\right)^2+\left(\frac{1}{18}-\frac{1}{240}\right)^2+2\left(\frac{1}{INF.}-\frac{1}{INF.}\right)\left(\frac{1}{18}-\frac{1}{240}\right)\text{COS}2\varnothing}$$

$$= \text{ARC COS } 16.74435\sqrt{(0)^2-(0.055555-0.00416)^2+0}$$

$$= \text{ARC COS } 16.74435\,(0.0513884)$$

$$= \text{ARC COS } 0.8604653 = 30°\text{-}38'$$

FROM TABLE (ROARK PAGE 321) INTERPOLATE

θ	30°	30°-38'	35°
α	2.731	2.6876	2.397
β	0.493	0.4978	0.530
λ	1.453	1.4656	1.550

SAMPLE COMPUTATION (CON'TD)

$$C = \text{MAJOR AXIS} = \alpha \sqrt[3]{\frac{P\delta}{K}} = 2.6876 \sqrt[3]{\frac{(337{,}500.0)(66.9774)}{4.29\text{x}10^7}}$$

$$= 2.6876\text{x}0.8077 = 2.17\text{" (SEMI-AXIS)}$$

$$d = \text{MINOR AXIS} = \beta \sqrt[3]{\frac{P\delta}{K}} = 0.4978\text{x}0.8077 = 0.402\text{" (SEMI-AXIS)}$$

S_C = MAX COMPRESSIVE STRESS AT CENTER OF CONTACT

$$= \frac{1.5\ P}{\pi cd} = \frac{(1.5)\ (337.5)}{(3.1416)\ (2.17)\ (0.402)}$$

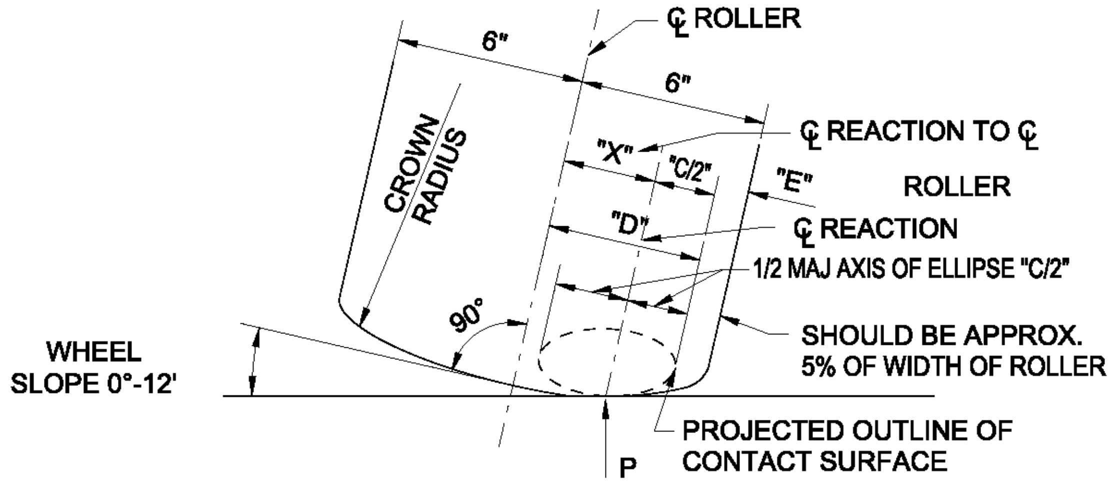
$$= \frac{5.0625\text{x}10^5}{2.74054}$$ = 184.7 K/SQ IN. (COMPARED TO 184 KSI FROM THE COMPUTER PROGRAM, SEE PLATE B-90, SHEET 3)

y = COMBINED DEFORMATION OF BOTH BODIES AT CONTACT.

$$= \lambda \sqrt[3]{\frac{P^2}{K^2\delta}} = 1.4656 \sqrt[3]{\frac{(337,\ 500.0)^2}{(4.29\text{x}10^7)^2(66.9774)}}$$

$$= 1.4656\ (0.009739)$$

y = 0.0142"



THE CROWN RADIUS OF TREAD SHOULD BE AS LARGE AS POSSIBLE YET SELECTED SO THAT THE EDGE OF THE COMPRESSION AREA SHALL FALL WITHIN THE FACE OF THE ROLLER.

"E" = DISTANCE BETWEEN OUTSIDE EDGE OR ROLLER AND OUTSIDE EDGE OF CONTACT SURFACE (ELLIPSE)

"D" = "X"+"C/2"   "E" = 1/2 WIDTH ROLLER MINUS "D"

CROWN RADIUS	"X" -℄ REACTION TO ℄ ROLLER (PL B-90 SHEET 2)	1/2 MAJOR AXIS (C/2) OF ELLIPSE (PL B-90 SHEET 2)	DISTANCE "D"	EDGE DIST. 'E"
20 FT	0.838 IN.	2.17 IN.	3.01 IN.	2.99 IN.
30	1.257	2.65	3.91	2.09
40	1.675	2.93	4.61	1.39
<u>50</u>	2.09	3.15	<u>5.24</u>	0.76
60	2.52	3.42	5.94	0.06
70	2.93	3.64	6.57	-0.57
80	3.35	3.81	7.16	-1.16

THE 50-FOOT CROWN RADIUS SHOULD BE USED SINCE IT IS THE LEAST RADIUS THAT THE ELLIPSE WITHIN A REASONABLE DISTANCE FROM EDGE OF ROLLER.

(SAY 12" WIDEx5% = 0.6")

WHEEL DESIGN (CON'TD)

THE WHEEL TREAD SHOULD BE DESIGNED WITH A FACTOR OF SAFETY OF 0.5 BASED ON THE YIELD POINT OF THE MATERIAL INVOLVED AND THE MAXIMUM COMPUTED WHEEL LOAD.

CROWN RADIUS	STRESS IN CONTACT AREA OF WHEEL	MINIMUM YIELD POINT OF REQUIRED MATERIAL
20 FT	184 KSI	92.0 KSI
30	164	82.0
40	158	79.0
50	154	77.0 = <89 KPS
60	146	73.0
70	142	71.0
80	139	69.5

THE ACTUAL YIELD POINT OF THE WELD DEPOSIT OF E 410 WELDING ELECTRODES CANNOT BE FOUND BUT CAN BE ESTIMATED FROM ITS ULTIMATE STRENGTH AND BRINELL HARDNESS NUMBER, AND THE KNOWN ULTIMATE STRENGTH, YIELD STRENGTH AT ANOTHER BRINELL NUMBER, AS FOLLOWS:

	BHN	YIELD	ULTIMATE
KNOWN	150 *	35,000*	70,000*
UNKNOWN	360**	X ⌖	178,000***

* PROPERTIES OF ELECTRODE USED

** HARDNESS OF DEPOSITED WELD METAL

*** PLATE B-90, SHEET 8.

$$\oplus \quad \frac{X}{178{,}000} = \frac{35{,}000}{70{,}000} \quad X = 89{,}000 \text{ PSI}$$

THE STRESS AT 50-FT CROWN RADIUS FALLS WITHIN THE ALLOWABLE STRESS, SO USE 50 FT

HARDNESS NUMBERS CONVERSION CHART