

RETURN TO ~~STEEL~~ PROJECT
SITE DEVELOPMENT
INFRASTRUCTURE DEPT

ACTUALLY THIS BELONGS TO HYDRO
KERRIGAN'S STYLE IT.

ASCE Manuals and Reports on Engineering Practice No. 79

Steel Penstocks

Prepared by the
ASCE Task Committee on
Manual of Practice for
Steel Penstocks
Energy Division
American Society of Civil Engineers
Richard D. Stutsman, Chairman



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ABSTRACT

Steel Penstocks, ASCE Manuals and Reports on Engineering Practice No. 79 covers the design, manufacture, installation, testing, startup, and maintenance of steel penstocks, including branches, wyes, associated appurtenances, and tunnel liners. The manual was prepared by a special ASCE Task Committee under the supervision of the Hydropower Committee of the ASCE Energy Division. The Task Committee's goal was to review all available penstock literature in order to bring the information together into one document. Since *Steel Penstocks* is intended for the engineer or designer who has very little experience with penstocks or pressure vessels, the manual provides a commentary on background as well as specific design, manufacture, installation, testing, startup, and maintenance requirements.

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- 79 Steel Penstocks

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PREFACE

This *Manual of Practice for Steel Penstocks* covers the design, manufacture, installation, testing, startup, and maintenance of steel penstocks, including branches, wyes, associated appurtenances, and tunnel liners. Standards presented in this manual are applicable only to steel penstocks. As used in the manual, a *penstock* is defined as a pressurized, closed water conduit located between the first free water surface and a hydroelectric power/pumping station. *Pressurized conduit* refers to pipe in which shell thickness is determined by external pressures rather than by other criteria. Standards presented in this manual are applicable only to steel penstocks.

The manual was prepared by a special ASCE Task Committee under the supervision of the Hydropower Committee of the ASCE Energy Division. The Hydropower Committee was formed to "develop and disseminate information on all phases of hydroelectric power to the hydro engineering community."

In 1985, the Hydropower Committee began work on the *Civil Engineering Guidelines for Planning and Designing Hydroelectric Developments*, commonly referred to as the *Guides*. Upon the work's completion, the committee decided to expand the section on penstocks into a nationally accepted standard or code.

The penstock design documents most commonly used by the penstock industry are the ASME Boiler and Pressure Vessel Code and the *Steel Plate Fabricator's Manual*.

The ASME Boiler and Pressure Vessel Code (ASME Code) has been in existence since 1914 and originally was based on an allowable stress of one-fifth the tensile strength of the steel. An ASME Code section specifically for pressure vessels came out in the mid-1920s. In the 1950s, the stress basis was changed to one-fourth of the tensile strength of the steel. The high-stress alternative rules, based on one-third of the tensile strength of the steel, came out in the 1960s, although the theory and stress basis had been in use for some time before that. Fabrication based on these rules has resulted in high-quality structures. The rules for both divisions of Section VIII of the ASME Code are accepted worldwide and have been applied to many types of structures other than pressure vessels, including penstocks. The alternative rules of Section VIII, Division 1 and 2 have been used extensively in preparing this manual.

The development of the *Steel Plate Fabricator's Manual* was based primarily on the 40-plus years of laboratory research and practical hydroelectric experience of the United States Bureau of Reclamation. The manual provides more specific penstock design information than the ASME Code but refers to the latter for fabrication and welding.

Although used extensively, neither of these two documents has been accepted as a comprehensive penstock code. Therefore, the goal of this committee was to review all the currently available penstock literature and bring the information together into one document as a standard for the design, manufacture, installation, testing, startup, and maintenance of steel penstocks.

This manual is intended for the engineer or designer who has very little experience with penstocks or pressure vessels. Therefore, the manual is presented in a format that provides a commentary on background as well as specific design, manufacture, installation, testing, startup, and maintenance requirements. However, the engineer or designer should be aware that there may be specific areas in this manual that require a higher level of experience or expertise.

The design procedures presented in this document are recommended by this committee. However, the committee recognizes that there are other methods and procedures that may be equally as valid. This document does not preclude their use, provided the engineer and designer are satisfied with their validity and applicability.

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Don Matchett and Arvids Zagars provided substantial contributions and support for the project. Special thanks goes to Tom Logan, who acted as a mentor and advisor, based on his spearheading the preparation of the *Guides*.

Finally, special gratitude is due to Tom Ahl. His dedicated support as co-chairman provided valuable assistance and perspective during the preparation of this manual.



Richard D. Stutsman, P.E.
Chairman, Manual of Practice for Steel Penstocks
ASCE Task Committee

MANUAL OF PRACTICE COMMITTEE MEMBERS

Richard Stutsman, Chairman, Pacific Gas and Electric Company, San Francisco, CA
 Thomas Ahl, Co-Chairman, Chicago Bridge and Iron, Oak Brook, IL
 Chuck Feild, Black & Veatch Engineering-Arch., Overland Park, KS
 John Bambei, Board of Water Commissioners, Denver, CO
 Don Beard, United States Bureau of Reclamation, Denver Federal Center, Denver, CO
 Clay Brogdon, Georgia Power Company, Atlanta, GA
 Jim Burton, Baker Coupling Co. Inc., Los Angeles, CA
 Frank Cortellessa, Ameron Steel Fabrication Division, Fontana, CA
 Theodore Critikos, Stone & Webster Engr. Corp., Denver, CO
 Jerry Dodd, Consulting Engineer and Geologist, Greenwood Vlg., CO
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 Paul Flanagan, State of California, Santa Nella, CA
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 Graeme Plant, Graeme D. Plant, PE, Napa, CA
 Dan Rogers, Duke Power Company, Charlotte, NC
 Malcolm Stephens, Civil Engineer, Carmichael, CA
 Ray Sullivan, EBASCO Services Inc., New York, NY
 Chris Sundberg, CH₂M Hill, Bellevue, WA
 Ray Toney, Ray Toney Associates, Redding, CA
 George Tupac, G.J. Tupac & Associates, Inc., Pittsburgh, PA
 Donald Wagner, Progressive Fabricators, St. Louis, MO

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 Duke Power Company, Civil Engineering Dept., Charlotte, NC

Preface

Malcolm Stephens, PE, Carmichael, CA
EBASCO Services Inc., New York, NY
CH₂M Hill, Bellevue, WA
Ray Toney Associates, Redding, CA
George Tupac, G.J. Tupac & Associates, Inc., Pittsburgh, PA
Progressive Fabricators, St. Louis, MO

MANUAL OF PRACTICE BLUE RIBBON COMMITTEE

William Allerton, FERC, Washington, D. C.
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Thomas Haag, Black & Veatch Engineers-Arch., Overland Park, KS
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Jimmy Throckmorton, Southern California Edison, Rosemead, CA
Alexic Vircol, ATC Engineering Consultants Inc., Englewood, CO
Arvids Zagars, Harza Engineering Company, Chicago, IL

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Yuba County Water Agency, Marysville, CA

ASCE ENERGY DIVISION EXECUTIVE COMMITTEE CONTACT MEMBERS

Arvids Zagars, P.E., Harza Engineering Company, Chicago, IL
Don Matchett, P.E., Stone & Webster Engineering Corp., Denver, CO

ASCE HYDROPOWER COMMITTEE CONTROL GROUP MEMBERS, 1988-1991

Edgar Moore, P.E., Chairman, ASCE Hydropower Development Committee,
Harza Engineering Company, Chicago, IL
Thomas Gebhard, Gebhard Engineers, Austin, TX
Toney Ferreira, Northeast Utilities, Holyoke, MA
Thomas Logan, P.E., Consultant, Denver, CO

TECHNICAL EDITOR

JLS Language Associates, Menlo Park, CA

CAD FIGURES

Frank Kanemitsu, Pacific Gas and Electric Company, San Francisco, CA

Zoe Henderson, Pacific Gas and Electric Company, San Francisco, CA

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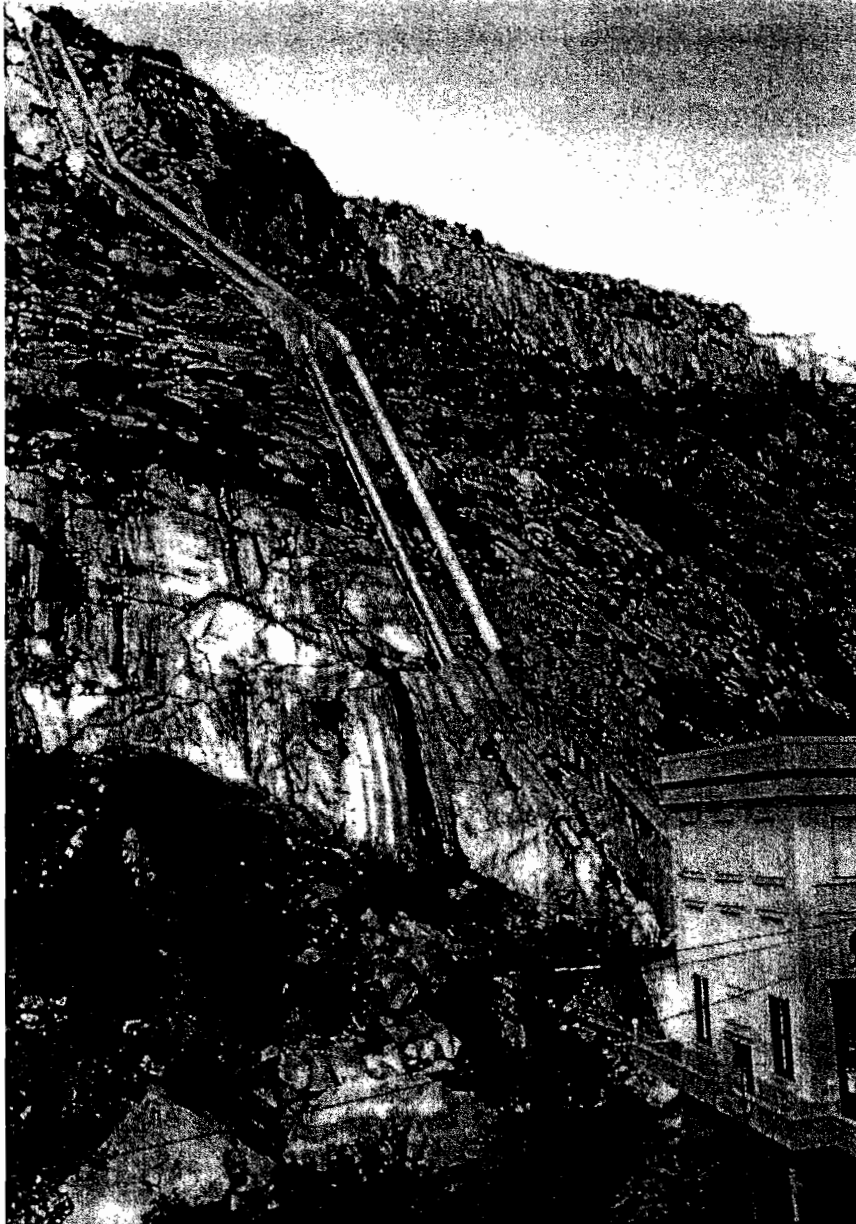
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This section presents an overview of general penstock design and maintenance issues. More detailed descriptions can be found in subsequent sections. The following pages address general design considerations; internal and external pressure design; economic diameter; shutoff systems; vibration; wind, snow, and earthquake loading; corrosion; maintenance; and geologic considerations.

1.1 Design Considerations

The design of a safe and cost-effective penstock system requires the consideration of technical, environmental, economic, and constructibility factors. Figure 1-1 shows the flow of considerations that impact the design of a penstock system. Detailed considerations, guidelines, criteria, and design methodology are presented in subsequent sections.

1.1.1 Required Information

It is important for the designer to gather as much information as possible about the installation. Essential information includes: (1) owner requirements, and (2) site-specific requirements.

1.1.1.1 Owner Requirements

The owner's installation preference and requirements must be clearly delineated, because these preferences affect the overall approach to the penstock system, including methodology, material selection, and design. Consideration must be given to:

- (1) Preferred material and design type
- (2) Plant operation—base load, voltage control peaking, or a combination of both; also important is the likely number of unit operating cycles, i.e., daily start-stop and load change requirement
- (3) Parameters and criteria for determining the annual cost of capital investment and the annual cost of power revenue loss
- (4) Inspection and maintenance philosophy
- (5) Applicable internal and governmental guidelines, criteria, and design requirements
- (6) Legal and political issues, including environmental, permit, and licensing issues.

1.1.1.2 Site-Specific Requirements

Equally important are site-specific requirements, which affect design by imposing environmental restraints, limitations on size and weight of penstock sections, geologic restraints, hydrologic considerations, and limitations (alignment and support) to the penstock physical layout. Consideration must be given to:

- (1) Land ownership, right-of-way limitations, mineral rights, and limitations relating to excavation/quarrying operations
- (2) Environmental restraints, including aesthetics, fish, game and wildlife preservation, archaeological excavations, disposal of material, clearing, and erosion

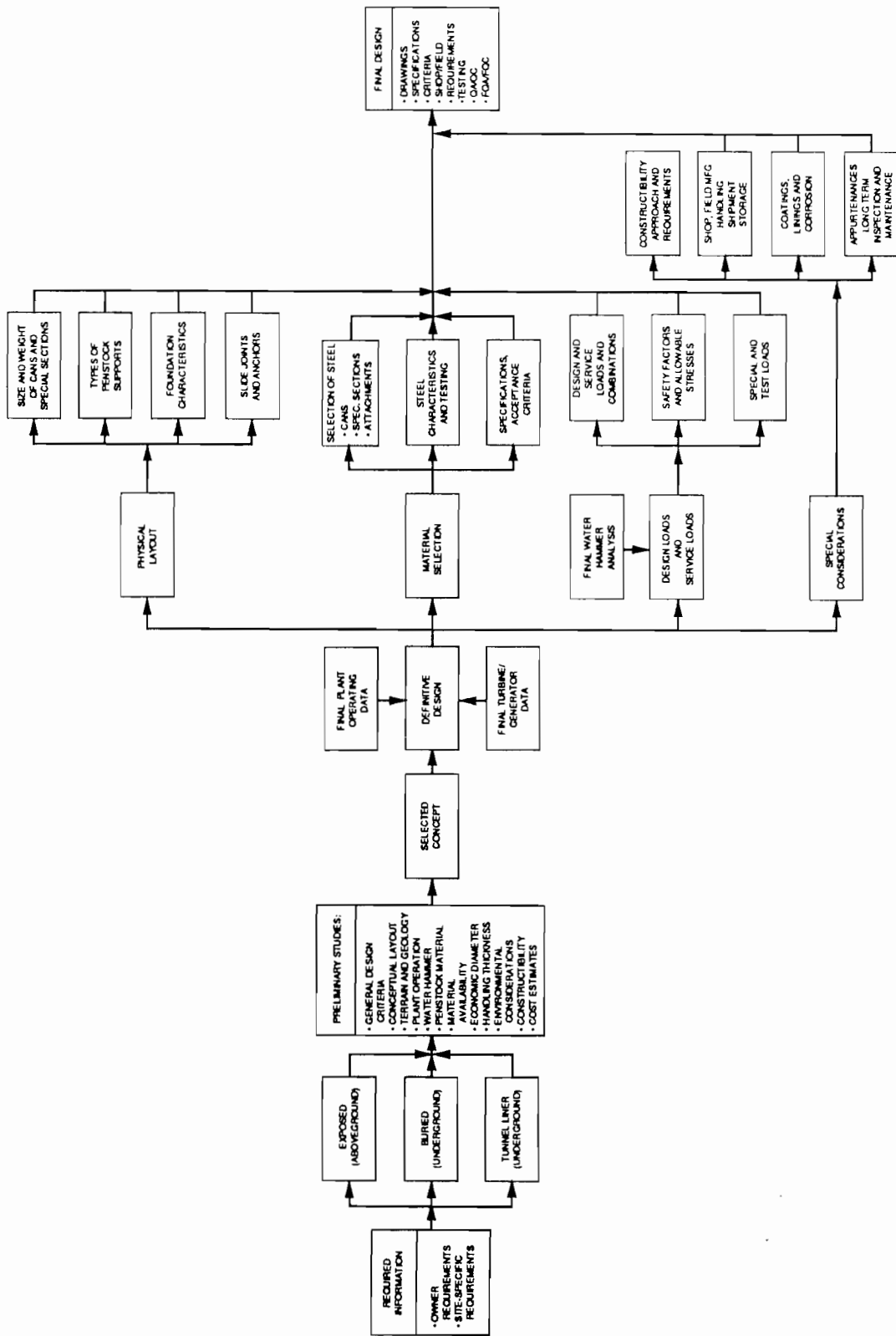


Figure 1-1 General Considerations for Penstock Design

- (3) Terrain configuration
- (4) Site geology, hydrology (groundwater conditions), and soils
- (5) Applicable codes and mandatory requirements
- (6) Other site-specific considerations.

1.1.2 Type of Installation

The type of installation selected should reflect the above considerations. A given penstock installation could include all of the following types:

- (1) Exposed penstock (aboveground)
- (2) Buried pipeline (underground)
- (3) Steel tunnel liner (underground).

Each penstock type has different associated design, material, and construction costs.

1.1.3 Preliminary Study

The preliminary study phase is an important phase of the general design effort and requires an experienced designer. The final penstock configuration, alignment, design, and other key requirements and parameters must be determined during this study phase.

The designer must investigate the site conditions and make several layouts of various alignments. Terrain, geologic characteristics, and foundation conditions play important roles during this study phase. Since the ultimate goal of this study phase is to determine the most economic and constructible alignment, it is not necessary to approach the study phase with great precision.

1.1.3.1 General Design Criteria

During the preliminary study phase, consideration must be given to criteria and parameters relating to design pressure and allowable hoop stress.

Shell thickness then must be determined by the classic formula:

$$t = \frac{Pr}{SE} \quad (\text{Equation 1-1})$$

Where:

- t = shell thickness, in.
- P = internal pressure considered for the study phase, psi

- r = internal radius of the penstock cylinder, in.
 S = allowable hoop stress considered for the study phase, psi
 E = joint reduction factor

Conservative values must be used for allowable hoop stress, depending on the type of steel shell material strength characteristics and the most likely factors of safety that will be considered for the installation type.

The internal design pressure must be at least equal to the maximum static head at the point of the penstock being investigated. Some refinement to this is possible by performing a hydraulic transient pressure analysis using preliminary hydropower equipment characteristics and plant operating data. As an alternate to the hydraulic transient analysis, a conservative pressure rise at the turbine of 10% to 20% can be assumed for Pelton type units; 30% to 40% for Francis type units without pressure-reducing valves, and 10% to 20% with pressure-reducing valves; and 30% to 40% for pump-turbine units. The pressure line then would vary linearly from the turbine to the first free water surface.

The designer must consider at least two types of suitable steel (low, medium, or high strength). High-strength steel may be required by design where the installation of heavy sections would be difficult (necessitating lighter can sections) or where shop and field welding of thick sections might present problems.

The designer should evaluate the potential for penstock corrosion and compare costs of the various methods available for corrosion protection. The cost of coatings and corrosion protection systems should be compared to the cost of added steel shell material.

Economic diameters for each of the penstock types must be calculated for the given energy costs and economic criteria, as well as the costs for the selected steel material and probable construction costs. Several formulas can be used to determine the economic diameter. Flow velocities and regulation characteristics of the system must be checked to determine that they are reasonable and conform to the accepted practice for similar installations. The required minimum thickness of steel pipe for handling must be compared to the thickness required to resist internal pressure and external loads. The greater thickness value should control design.

1.1.4 Selected Concept

The selected penstock configuration should incorporate material and designs that minimize life cycle costs, with prudent consideration given to technical, environmental, constructibility, and maintainability issues.

1.1.5 Definitive Design

The definitive design phase consists of compiling final design-related data, finalizing the conduit alignment and layout, and confirming the final plant operating characteristics. Turbine, generator, governor, and closure valve characteristics that will influence water-hammer analysis also must be determined.

Definitive guidelines and design criteria must be prepared for:

- (1) Physical layout, conduit alignment, supports, and anchor blocks
- (2) Material selection for the penstock, tunnel liners, and appurtenances
- (3) Design and service loads
- (4) Special considerations for manufacturing and field installation.

Within each of these definitive design groups are the following subgroups that form the basis for the design.

1.1.5.1 Physical Layout

The effort expended during the preliminary study phase for the selected conduit arrangement, layout, and alignment now must be refined to achieve a working concept. These refinements primarily apply to:

- (1) Can sections

The size and weight of can sections must be selected to be compatible with the needs and difficulty of the field installation. Traffic, shipping, and access limitations and restraints should be considered. Customarily, can sections up to 12 feet in diameter, 40 feet long, and weighing from 25 to 35 tons can be shipped and handled reasonably in the field for installation.

- (2) Specialty sections

Similar consideration must be given to specialty sections such as mitered bends, wyes, and tee connections. Special attention must be given to the detailing of these sections, particularly wyes with external support (girder beam and stiffener rings). These sections quickly become oversize and heavy, making shipment and handling difficult. The designer must be willing to accept some heavy field welding for these structures and should take this into account when preparing specifications and for QA/QC requirements.

- (3) Supports

Supports must be refined to reflect environmental considerations, can size, geology, and local foundation conditions. Reinforced concrete saddles or steel ring girders are the two most commonly used support types for exposed aboveground installations. Anchor and thrust blocks are special applications of localized support systems. Cable harnesses for suspended pipe that bridges large openings and steel or concrete bent supports are some specialty types of supports. The spacing of ring girders is governed by practical considerations. Shell diameter, thickness, and material type have a major influence on the spacing, which may range from 40 to 200 feet.

Saddles are used in conjunction with stiffened or unstiffened steel pipe. Factors to consider in the spacing of saddles include the profile slope, the maximum acceptable deflection of the pipe when acting as a beam, the shell stress conditions at the saddle support horns, and the foundation conditions for the saddle supports. Also, the designer must be aware that above each saddle support circumferential bending moments develop in the penstock shell and cause the upper portion of an unstiffened pipe to deform. This results in an upper shell portion that is ineffective as a beam, reducing the effective cross section of the member, i.e., moment of inertia.

For pipe buried in soil, key considerations are the flexibility of the pipe, the allowable vertical pipe deflection due to settlement, the ability of the surrounding soil to resist lateral loads, and the type of lining used. Accepted practice and methods should be applied with regard to trenching and backfilling for steel pipe based on soil mechanics, consistent with field experience for this type of pipe support.

Support requirements and considerations for a steel liner relate primarily to the interaction between the steel shell, concrete lining, and surrounding rock. Special supports, which may be required during field installation, are customarily of a temporary nature. However, the designer is cautioned about the potential for "hard spot" formation, resulting in potentially high, localized shell stresses. This is particularly applicable to penstock sections subjected to hydrostatic pressure testing and concrete encasement.

(4) Geology/foundation

Geology and foundation characteristics are important for all types of installations. For ring girder and saddle support designs, the foundation is required to resist primarily downward loads, some small loads parallel to the pipe due mainly to frictional loads, and some loads that result from temperature effects. In earthquake-prone areas, support foundations and sloping terrain also must safely withstand seismic loadings.

Penstock installations on sloping terrain are subject to potential slope stability problems. Slope instability must be addressed and investigated to determine geologic and soil properties of the terrain and the potential for slope movement and potential differential settlement. Long-term monitoring of slopes or the incorporation of structural improvements should be considered in the foundation analysis. Ice and snow as well as avalanche conditions also must be considered.

The cumulative loads acting in line with the pipe should be resisted by appropriate anchor blocks. It is recommended that the foundation conditions at each anchor block location be determined individually for use in the design of anchor blocks. If geologic and soil conditions permit, supports can be divided into groups with like foundations, and appropriate conservative foundation designs prepared for each grouping, resulting in an economy of design. See Section 1.11.2 for relevant geologic considerations.

(5) Mechanical joints

Mechanical joints are used in both exposed and buried penstocks to accommodate longitudinal movement caused by temperature changes and to facilitate construction. The joints also allow for movement where differential settlement or deflections are anticipated. Should the designer decide not to use mechanical joints, the penstock and anchor blocks must be adequately designed to handle loads due to temperature changes, settlement, or deflections.

Slide joints fall into two major categories: (a) expansion joints, and (b) bolted sleeve-type couplings.

Expansion joints permit only longitudinal movement. The joints are used primarily with aboveground installations and are located between girder or saddle supports at the point where the penstock deflections are of equal magnitude and direction. If used in a buried installation, the joints should be located in a vault to permit inspection and maintenance. Expansion joints also may be used where warranted to permit expansion and contraction.

Bolted sleeve-type couplings are used in both aboveground and underground installations. For underground installations, the penstock must be pressurized and the couplings checked for leakage before the penstock is buried. Sleeve-type couplings allow for longitudinal, angular, and some differential settlement movements.

For bolted sleeve-type couplings used on very steep slopes, consideration should be given to preventing the coupling from sliding downslope as the penstock sections move with temperature changes. This is accomplished by using a pipe stop, internal to the coupling, or by anchoring the coupling at some point to the outside of the penstock shell. In either case, consideration must be given to the need to slide the coupling beyond the joint for future maintenance.

Other types of joints and couplings are available, particularly for underground installations. These include O-ring joints and mechanical Victaulic couplings.

Slide (expansion or contraction) joints that keep resulting forces from reaching operating equipment (valves, pumps, etc.) also should be considered.

(6) Anchor blocks

Anchor blocks fix the pipe in place during installation and operation. They are designed to resist the loads resulting from gravitational, hydrodynamic, hydrostatic, and earthquake forces, as well as from temperature changes and changes in alignment.

For aboveground designs, anchor blocks are normally located at all points of significant change in slope or horizontal alignment and sometimes at intermediate points in long tangent runs. For installations using expansion joints, an anchor spacing of up to 500 feet may be used between anchors and expansion joints, depending on the design temperature range.

Welded, buried penstocks generally do not require anchors at points of minor vertical/horizontal alignment changes. This should be verified by analysis. Buried penstocks with expansion joints require anchors similar to those used for aboveground installations.

1.1.5.2 Material Selection

(1) Steel selection

The selection of the steel for the design and construction of penstocks, tunnel liners, water conduit, and appurtenances depends on plate thickness, availability, service temperature, particular use, economics, and the ease of shop and field fabrication. Plate thickness is important because it affects heat treatment requirements, mechanical testing, nondestructive testing examination needs, and the cost of material and fabrication. Steels should be manufactured and tested in accordance with appropriate ASTM specifications or their equivalent accepted standards (i.e., European, Canadian, or Japanese standards). Pressure-vessel-quality steels or the equivalent are normally used, unless the design loadings do not control the shell thickness. In these cases structural quality steels can be considered. In addition to yield and tensile strength characteristics, the following important steel properties require careful consideration by the designer: workability, ductility, fracture toughness, weldability, and cost. Depending on the application, spiral welded pipe can be used with special consideration given to preheat interruption, offset tolerances, skelp splices, coil feed speed, welding/welder quality requirements, testing, and use requirements.

The selection of the steel material, specified nondestructive examination (NDE) requirements, and hydrostatic pressure testing become more critical for special sections such as wyes, bends, thrust ring sections, and expansion/reducer sections. Special steels and more stringent NDE and QA/QC requirements may be justified for these sections.

Attachments such as ring girders, stiffener rings, and support systems may be designed for and fabricated from plate or structural shapes made from structural quality steel. Caution should be exercised, however, because the selection of some high-strength alloy steel materials for the penstock may preclude the use of attachments made from different materials.

Refer to Section 2.5 for steels commonly used in the manufacture of penstocks and associated structural supports. Again, the key criteria for selection are:

- (a) Yield and tensile strength
- (b) Ductility and workability
- (c) Weldability
- (d) Resistance to brittle fracture
- (e) Cost.

(2) Steel characteristics

The steel must be manufactured and tested in accordance with ASTM steel characteristic standards. ASTM supplemental requirements warranted for a particular installation also must be specified. If "foreign" steel is used for the penstock, review the applicable foreign standards (in English) and compare them with the applicable ASTM standards. Subtle differences may exist among standards and must be identified; "special" requirements must be called out in the specifications. Similarly, foreign testing methods may differ from U.S. practices. In some instances, foreign standards may require substantial, additional testing. The weldability of foreign steels also must be considered.

(3) Specification and acceptance criteria

Specification and acceptance criteria can take the form of individually prepared specifications for design, fabrication, and construction, or a single specification covering material, shop fabrication, and field installation. These documents should address all requirements in a clear and concise manner.

Material specifications, along with applicable drawings, should indicate clearly the material for component parts of the penstock. This applies to all can sections, appurtenances, and attachments. Fabrication requirements under applicable codes must be identified clearly. Requirements and responsibilities to be imposed upon the rolling mills for plates, forgings, and structural shapes must be spelled out clearly. Mechanical and chemical analyses and frequencies of testing must be specified. ASTM standards relating to mechanical, strength, chemical, and other requirements must be reviewed carefully by the designer. If other characteristics are desired, the ASTM requirements should be modified or supplemented, being careful not to alter beneficial characteristics.

Material inspection, fabrication shop welder qualifications, shop quality assurance, and installation processes must be clearly defined either through the use of applicable codes and standards or by calling out the requirements directly in the specification. It is recommended that critical inspection requirements and "hold points" be specified, along with a "requirement" for the inspector.

For nondestructive testing, the designer must be familiar with visual, ultrasonic, magnetic particle, radiographic, and dye-penetrant methods. Experience, safety, and economics must guide the designer in the selection of appropriate NDE inspection requirements. The specification must be sufficiently flexible to allow other forms of NDE inspection as alternatives.

Hydrostatic testing at 125% to 150% of the design pressure (see Section 13.4.2) is considered a "nondestructive" test that can be used in addition to the more conventional NDE. The necessity for hydrotesting depends on:

- (a) Site location, automatic shutoff valves, and head
- (b) Risk of loss of life and damage to property

- (c) Complexity of weldments and structures
- (d) Extent and type of NDE performed in the shop and field.

The construction specification for the installation of the penstock, appurtenances, and attachments must be consistent with the design of the penstock. Field welding requirements and welder qualifications must be consistent with those for the shop fabricating process.

During storage and installation, the penstock and its appurtenances must be handled and supported in a manner that prevents stress concentrations and distortions (hard spots) of the shell material. This is particularly applicable to previously stress-relieved parts. Alignment and positioning of parts prior to welding, weld preparation, the welding, and NDE processes must reflect the design intent and requirements. Plate offsets resulting from poor fabrication, partial penetration welds, and use of fillet welds where full-penetration welds are specified in the design will result in a poor product.

1.1.5.3 Design and Service Loads

Design and service loads vary, but the most important load imposed on a penstock system results from hydrostatic and hydrodynamic internal pressure. Other key load conditions include external pressure loads, gravity-related loads, loads resulting from temperature changes, and seismic, wind, and snow loads; also included are loads from vibrational and cyclical pressure fluctuation caused by plant operations and turbine equipment. The designer must identify and quantify these loads. Service levels and load combinations recommended in this manual may need to be augmented to include other load combinations more critical to a specific installation. Cyclical water-hammer pressure fluctuations must be considered in determining the potential for in-service cracking. The effects of temperature-related loads must not be underestimated, particularly for exposed penstock sections and restrained shell configurations, and in the development of secondary stress conditions and brittle fracture. Designs based on selected service levels can be checked for additional loads by the method of superposition or other acceptable method.

Factors of safety and resulting allowable stresses must be specified, taking into consideration both the yield and tensile strength attributes of the material. Other critical steel properties to be considered include ductility, resistance to brittle failure, and weldability. A wide range of pressure vessel steels are available for use in penstock design. These steels exhibit varying ranges between the yield strength and the tensile strength for the steel. For economy of design, localized stress conditions exceeding allowables should be considered for acceptance, given the probability of local yielding and the redistribution of stresses.

The designer must be aware of possible special loads that may be imposed on the structure, particularly during construction, filling, dewatering, and testing. Stresses developing from these special loads must be determined and adjustments made to accommodate unacceptable stress levels. As stated earlier, special consideration must be given to stress conditions that can develop within the structure during hydrostatic testing and to the configurations for supporting

the structure during testing. It is recommended that the designer develop the test procedure as well as the preferred method for supporting the structure during testing.

1.1.5.4 Special Considerations

The designer, in conjunction with personnel experienced in field installations, constantly must ask how the installation will be fabricated and built in the field. Constructibility is an important aspect that is often overlooked. A design may appear good on paper, but be extremely difficult to construct. The designer should think out the entire installation, item by item. Can it be handled in the shop and field? What are the size and weight limitations for shipment? How will each piece be stored? Are there special storage/cribbing requirements? What about internal supports and stiffeners? Will special welding and heat treatment procedures be needed in the shop or field? What are the specifications for coating and painting? Are provisions incorporated in the design to permit or accommodate long-term inspection and maintenance requirements?

Corrosion is also an important factor. Water analysis is necessary to determine the chemical content and hardness of the water. Corrosion from possible other sources, i.e., dissimilar metals, stray electrical currents, corrosive soils, high flow velocity, etc., needs to be evaluated. Also, special attention should be given to corrosion/erosion protection at bends, wyes, and other special components. Cathodic protection may be required for some penstock types. Special consideration must be given to accessories such as pressure relief valves, standpipes, air valves, penstock accessways, and penstock flow shutoff systems.

1.2 Internal Pressure Design

1.2.1 Internal Pressure Design for Free-Standing Penstocks

The minimum required plate thickness should be computed considering the maximum pressure rise due to full load rejection on all penstock units operating at full or partial gate discharge. Turbine wicket gates are assumed to close in the normal governor closure time during load rejection. The envelope of the maximum pressures that occur over time along the penstock is generally assumed to vary linearly from the maximum pressure head due to water hammer at the units to the maximum surge at the surge tank or, in the absence of a surge tank, it varies linearly to the reservoir surface at the intake. The pressure rise due to full closure of a turbine guard valve or a bypass pressure relief valve should also be investigated. Guard valve closure times are normally set to produce less pressure rise than a turbine wicket gate closure.

Load acceptance always must be investigated to determine that the hydraulic grade line does not drop below the penstock profile at any point. This can result in a separation of the water column. Internal pressure drops to the vapor pressure of water in the separation area, and a potential for collapse due to external air pressure then exists. In addition, the rejoining of the water column could cause excessive and undesirable water-hammer pressures.

1.2.2 Internal Pressure Design for Steel Tunnel Liners

The maximum internal pressure to which the steel tunnel liner can be subjected is discussed in Section 1.2.1. However, the minimum required plate thickness can generally be determined considering load sharing with the contiguous rock surrounding the tunnel in elastic interaction when adequate rock cover and rock conditions exist. A transition between load sharing and no load sharing is considered near an underground powerhouse cavern. A distressed zone of jointed rock can exist adjacent to the cavern walls for several reasons. These include the effect of the trajectory of very high in-situ horizontal rock stresses passing over and under the cavern, unfavorable rock conditions, gravitational effects, and loosening due to blasting during cavern excavation. The distressed rock zone will have a low modulus of deformation and as a result can share little of the internal pressure load with the steel tunnel liner.

1.3 External Pressure Design

1.3.1 External Pressure Design for Exposed Penstocks

Under certain extreme conditions, an exposed, free-standing penstock, if dewatered for inspection and maintenance, may be subjected to submergence and possible buoyancy forces at its lower end near the powerhouse. This might occur, for example, when the spillway is passing large floods creating extremely high tailwater. External ring stiffeners spaced at intervals along the penstock spans can be considered for providing the necessary resistance to buckling from external pressure. Flotation hold-down devices must also be considered to carry the component of buoyant force acting normal to the axis of the penstock and to prevent lateral displacement of the penstock.

1.3.2 External Pressure Design for Buried Penstocks

Buried penstocks must be analyzed for buckling due to external soil pressures and possible live loads on the backfill surface when the penstock is dewatered. The resistance of the pipe to buckling is affected by the density of the backfill material and the surrounding in-situ soil. Resistance to buckling can be improved by increasing the density of the surrounding soil. If the buried penstock can be periodically submerged, the analysis should consider buckling due to external water pressure and additionally check that uplift forces will not displace the dewatered penstock. If live loads on backfill dictate the thickness of the penstock shell, a reinforced concrete encasement should be considered for vehicle crossings.

1.3.3 External Pressure Design for Steel Tunnel Liners

External pressure design for unwatered steel tunnel liners must take into account the potential external pressure head that can develop on the steel liner from various sources.

One external pressure source is from groundwater. During construction the tunnel will act as a drainage sump and can appear drier over time, giving the impression that there is no established water table over the tunnel. Upon completion of the installation of the steel liner, however, the groundwater table will be reconstituted slowly. The phreatic level should be investigated for

seasonal effects in exploratory bore holes that extend down to tunnel level. The condition of the bore hole (degree of jointing, fracturing, iron staining, etc.) will indicate the potential for ground water at higher levels to communicate to lower depths within the rock mass. Groundwater pressure head may be higher than the vertical distance to the ground surface at a particular location due to water from adjacent, higher elevations migrating down along favorable joint systems. Also, water could migrate down from adjacent higher areas along the distressed and fractured zone at the periphery of an inclined shaft or tunnel that was not adequately closed by contact and consolidation grouting during construction.

Another source is from high-pressure water in the upstream power conduit migrating outward through cracks in the concrete tunnel lining and passing downstream through joints in the surrounding rock mass. If the grout curtain around the upstream end of the steel liner is not effective, high-pressure water eventually may penetrate through and around the grout curtain and upstream seepage cutoffs to act on the upstream portion of the steel liner. Long-term seepage through the curtain then may develop pressures along the upstream end of the liner equal to static operating pressure. These pressures may reduce very slowly when the tunnel is dewatered and still be effective when the steel liner is dewatered. Only a small accumulation of water is required to initiate buckling when the tunnel is dewatered for inspection and maintenance if the tunnel liner is not designed to resist the external pressure that develops.

Additionally, another source is from leakage out of the tunnel when pressurized through grout plugs that were not adequately sealed during construction.

Grouting pressures used during construction also must be considered in the analysis. Consolidation grouting pressures outside of the backfill concrete and contact grouting pressures between the steel liner and backfill concrete must be determined on the basis of the expected internal operating pressures. If consolidation grouting is used to improve the modulus of deformation of the contiguous rock, it must effectively fill all voids near the tunnel periphery, especially those above the springline. Consolidation grouting must be done at a pressure equal to or greater than one half of the maximum expected static internal operating pressure. Low grout pressures are very ineffective, especially when the grout must penetrate into and fill large overbreak areas in the tunnel roof that contain a massive amount of lagging as support to prevent further deterioration of the rock face.

Steel tunnel liners with a high D/t ratio should be investigated for their capacity to resist undesirable deformations during concreting.

When drains or pressure relief devices are used in an attempt to reduce the expected external pressure, considerable caution should be exercised. A network of piezometer cells should be installed behind the steel liner to monitor external pressures during dewatering. The rate at which the tunnel is dewatered then can be controlled to prevent an undesirable differential pressure acting on the outside of the steel liner. For long-time service life reliability, the system of installed pressure cells will require some redundancy in number.

Drain holes drilled into the rock around the steel liners from a drainage adit above the penstock must be on very close centers to be everywhere effective. Also, when drains are installed in or below rock strata with a content of calcium carbonates, they are usually rendered ineffective in a

short time due to an accumulating surface encrustation of calcite. The drains must be redrilled frequently to continue to be effective. Leachants from cement grout and concrete backfill can produce similar undesirable plugging of drains.

Pressure relief devices have been used successfully on some projects. However, pressure relief devices installed in the tunnel steel liner are subject to mineral encrustation from the sources previously mentioned every time they function to release water from a zone of high pressure to atmospheric pressure within the tunnel during dewatering. Before the tunnel can be watered up, the pressure relief devices must be examined. Every one that functioned must be removed, cleaned, lubricated, and replaced to ensure that it will function during a subsequent dewatering. Similarly, all nonfunctioning relief devices should be replaced. The number installed also requires some redundancy for long-time service life reliability of the system.

1.4 Economic Diameter

The economic diameter of a penstock is the diameter that minimizes the total cost of installation and the present worth of the power revenue loss due to hydraulic head loss in the penstock over the repayment period. The cost items directly affected by the penstock diameter are included in the analysis. Other large cost items such as valves and trashracks must be evaluated by separate value engineering.

There are several published economic diameter equations available to the designer. Examples may be found in AWWA Manual M11¹ and the ASCE/EPRI *Civil Engineering Guidelines*.²

This section develops equations that are practical and convenient to use. These equations apply to penstocks conveying water to conventional power plants. By moving the efficiency term (E) to the denominator, the equations are equally applicable to pumping plant discharge lines.

These equations are not recommended for sizing pump storage penstocks because major modifications are necessary to account for the changes in parameter values, such as the composite power values and hours per year of operation, during the pumping and power generation modes. Additional information for sizing penstocks for pump storage plants may be found in the proceedings of the American Society of Civil Engineers.³

Economic diameter equations are developed for the following two cases:

Case 1—Minimum penstock shell thickness for shipping and handling

Case 2—Internal pressure.

1.4.1 Installation Cost

(1) Case 1—Minimum penstock shell thickness for shipping and handling

The installation cost for a Case 1 penstock is calculated by the equation:

$$UI = \pi WCtD \quad (\text{Equation 1-2})$$

Where:

- UI = capital cost of penstock, Case 1, in dollars
- W = specific weight of steel, lb/ft³
- C = installation cost of penstock, \$/lb
- t = shell thickness, ft
- D = inside pipe diameter, ft

Equation 1-2 gives the cost of the penstock per unit length. The penstock length cancels out in developing the economic diameter equations. The Pacific Gas and Electric Company formula for determining the required minimum shell thickness in terms of diameter is:

$$t = \frac{D}{288} \quad (\text{Equation 1-3})$$

If $D/288$ is substituted for t in Equation 1-2, and the constant terms are combined, the installation cost equation may be rewritten as:

$$UI = 0.0109WCD^2 \quad (\text{Equation 1-4})$$

Additional information about minimum shell thickness may found in Section 4.

(2) Case 2—Internal pressure governs

For a Case 2 penstock, the shell thickness (t) is calculated by the hoop stress equation:

$$t = \frac{FHD}{2S} \quad (\text{Equation 1-5})$$

Where:

- F = conversion factor for feet of water to lb/in.², 0.4333 lb/ft/in.²
- H = design head (weighted average), ft
- D = inside diameter, ft
- S = allowable stress, lb/in.²

By substituting this expression for t into Equation 1-2 and simplifying, the installation cost (U_2) for a Case 2 penstock may be written as:

$$U_2 = \frac{0.6806WCHD^2}{S} \quad (\text{Equation 1-6})$$

1.4.2 Power Revenue Loss

The power revenue loss (PRL) of a penstock is calculated by using the following equation from the U. S. Bureau of Reclamation⁴ for the power-plant capacity (kW) and inserting the head loss term (H_L) for head.

$$kW = 0.0846QE(H_L) \quad (\text{Equation 1-7})$$

Where:

$$\begin{aligned} Q &= \text{design flow, ft}^3/\text{s} \\ E &= \text{overall turbine/generator efficiency} \end{aligned}$$

The Darcy-Weisbach equation is used to calculate the head loss (H_L) due to turbulent flow in the penstock. The equation is given as:

$$H_L = \frac{fLV^2}{D 2g} \quad (\text{Equation 1-8})$$

Where:

$$\begin{aligned} f &= \text{friction factor} \\ L &= \text{length, ft} \\ V &= \text{average velocity, ft/s} \\ D &= \text{inside diameter, ft} \\ g &= \text{acceleration due to gravity, 32.2 ft/s}^2 \end{aligned}$$

The equation is modified by replacing the velocity term (V) with:

$$\frac{4Q}{\pi D^2}$$

A unit length is used which results in an expression that computes head loss per unit length of penstock. This modified version of the Darcy-Weisbach equation is given as:

$$H_L = \frac{16fQ^2}{2g\pi^2 D^5} = \frac{0.02517fQ^2}{D^5} \quad (\text{Equation 1-9})$$

Substituting this head loss expression into Equation 1-7, the power-plant capacity equation may be rewritten as:

$$kW = \frac{0.002129fEQ^3}{D^5} \quad (\text{Equation 1-10})$$

To calculate the annual power revenue loss (PRL), Equation 1-10 is multiplied by the following parameters:

- (1) Composite value of power (M)
- (2) Hours of operation per year (h)

The annual power revenue loss (PRL) then may be written as:

$$PRL = \frac{0.002129fEQ^3hM}{D^5} \quad (\text{Equation 1-11})$$

It is important to note that the value of power is a single composite value obtained by adding the capacity and energy components. The composite value of power is expressed in units of $\$/kWh$. Often the capacity is expressed in $\$/kWyr$, in which case it is necessary to divide it by the number of hours per year of power generation.

The present worth of the annual power revenue loss (X) is obtained by multiplying Equation 1-11 by the present worth factor (pwf) for a given interest rate and repayment period, resulting in:

$$X = \frac{0.002129fEQ^3hM(pwf)}{D^5} \quad (\text{Equation 1-12})$$

1.4.3 Economic Diameter Equations

The economic diameter equations are developed by adding the Case 1 or Case 2 installation cost equation to Equation 1-12 to obtain an equation for the total cost. The total cost equation is differentiated with respect to diameter (D), set equal to zero, and then solved for D . The resulting economic diameter equations for Case 1 and Case 2 penstocks are as follows.

- (1) Case 1—Minimum thickness for shipping and handling

When the shell thickness (t) is determined by $D/288$, the economic diameter (D) is given as:

$$D = 0.9025 \left[\frac{fhMEQ^3(pwf)}{WC} \right]^{0.1429} \quad (\text{Equation 1-13})$$

When a specific value is used for t , the economic diameter is:

$$D = 0.3867 \left[\frac{f h M E Q^3 (p w f)}{W C t} \right]^{0.1667} \quad (\text{Equation 1-14})$$

(2) Case 2—Internal pressure governs

The economic diameter (D) for a Case 2 penstock is given as:

$$D = 0.05 \left[\frac{S M h f E Q^3 (p w f)}{W C H} \right]^{0.1429} \quad (\text{Equation 1-15})$$

It is very important to know how the power plant will be operated when determining the penstock diameter. Some of the parameter values, such as the composite value of power and the number of hours of power generation, will vary greatly for a base-load power plant compared to a peaking-load power plant. Selecting the design flow (Q) is most important because this term is cubed in the economic diameter equations. Variations in the design flow value have a significant impact on the resulting diameter calculations. For example, in Equation 1-15, if the value of Q is doubled and the other parameter values remain unchanged, the diameter increases by approximately 35%. By doubling any one of the other parameter values and leaving the flow term Q unchanged, the diameter increases by approximately 10%.

After the economic diameter has been determined, it can be shown that for a Case 1 penstock the installation cost is approximately 5 times the present worth of the annual power revenue loss. Similarly, for a Case 2 penstock, the installation cost is approximately 2.5 times the present worth of the power revenue loss.

1.4.4 Examples

The following examples illustrate the use of the equations for calculating the present worth factor and economic diameter for Case 1 and 2 penstocks.

(1) Example 1—Case 1 Penstock

Given data:

f	=	0.01, friction factor
h	=	6,500, hours per year of operation
M	=	0.05 \$/kWh, composite value of power
E	=	0.85, turbine/generator efficiency in decimal form
i	=	8.75%, interest rate
n	=	50 years, repayment period
Q	=	3,000 ft ³ /s, design flow
W	=	490 lbs/ft ³ , specific weight of steel
C	=	2.00 \$/lb, capital cost of penstock installed

The penstock shell thickness (t) is equal to $D/288$.

The present worth factor (pwf) for both Case 1 and Case 2 penstocks is calculated by the standard equation:

$$pwf = \frac{(i+1)^n + 1}{i(i+1)^n} \quad (\text{Equation 1-16})$$

Where:

- i = interest rate, %
- n = repayment period, years

Substituting in the values gives:

$$pwf = \frac{(1.0875)^{50} - 1}{0.0875(1.0875)^{50}} = 11.26$$

The economic diameter is calculated by using Equation 1-13:

$$D = 0.9025 \left[\frac{f h M E Q^3 (pwf)}{W C} \right]^{0.1429}$$

Substituting in the values gives:

$$D = 0.9025 \left[\frac{(0.01) (6,500) (0.05) (0.85) (3,000^3) (11.26)}{(490) (2)} \right]^{0.1429} = 17.06 \text{ ft}$$

(2) Example 2—Case 2 Penstock

Given data:

- f = 0.01 (dimensionless) friction factor
- h = 7,075 hours, annual hours of operation
- M = 0.05 \$/kWh, composite value of power
- E = 0.85, turbine/generator efficiency, in decimal form
- i = 8.75%, interest rate
- n = 50 years, repayment period
- Q = 2,900 ft³/s, design flow
- W = 490 lb/ft³, specific weight of steel
- C = 3.50, capital cost of penstock installed
- H = 442.4 ft, design head (weighted average)
- S = 20,000 lb/in.², allowable stress

The present worth factor (*pwf*) equals 11.26, the same as in Example 1. The economic diameter is calculated using Equation 1-15:

$$D = 0.5 \left[\frac{SMhfEQ^3(pwf)}{WCH} \right]^{0.1429}$$

Substituting in the values gives:

$$D = \left[\frac{(20,000) (0.05) (7,075) (0.01) (0.85) (2,900)^3 (11.26)}{(490) (3.50) (442.4)} \right]^{0.1429} = 15.00 \text{ ft}$$

1.5 Shutoff Systems

Flow shutoff control systems are installed in hydraulic conduit systems because of the potential for tunnel blowouts, penstock ruptures or major leaks, or blocked wicket gates. Shutoff systems can be installed at the upper end, an intermediate location, or the lower end of the hydraulic system. In the event of an excess flow signal, the shutoff system automatically closes under the transient flow conditions.

For a sudden penstock failure or a major penstock leak, the automatic shutoff system:

- (1) Prevents rapid dewatering of the tunnel
- (2) Minimizes the loss of water from the upstream reservoir
- (3) Minimizes the resulting damage or loss of life
- (4) Provides a means of regaining control of the system.

1.5.1 Intake Shutoff Gates

For hydraulic conveyance systems, such as steel pipelines, penstocks, or tunnels, intake shutoff gates (ISGs) must be installed at the intake structure to provide protection in the event of a system failure. When the intake is deeply submerged in the reservoir, gates are usually installed in a shaft or chamber downstream of the intake. Gate types include slide gates, fixed wheel gates, roller gates, radial gates, tainter gates, coaster gates, and tractor gates. Gates must close under their own weight rather than being forced down. An adequate air venting system must be designed and installed downstream of intake gates to prevent the development of negative pressures that could result in tunnel collapse.

1.5.2 Penstock Shutoff Valves

For hydraulic systems with tunnels and an aboveground penstock, penstock shutoff valves (PSVs) must be located downstream of the tunnel portal but just upstream of the penstock. There must be no unprotected steel liner between the tunnel and the PSV, unless an intake

shutoff gate is installed upstream. This type of system has the added benefit that the penstock can be dewatered without dewatering of the tunnel.

As with the intake gate system, an adequate air venting system must be designed and installed downstream of the valve(s) gates to prevent development of downstream negative pressures that could result in penstock collapse.

1.5.3 Automatic Shutoff Control System

Emergency shutoff valves and gates usually are unattended and are often located some distance from the powerhouse operator. Therefore, a local shutoff control system, which can sense an increase in flow and automatically close a gate or valve, must be installed. This automatic control system also must be controlled remotely from the powerhouse.

Examples of sensing systems used for automatic control systems are differential float-well systems, Rolex devices, and sonic flow measuring systems. These sensing systems usually activate the gate or valve operating system when the tunnel or penstock velocities exceed a certain percentage over the maximum normal transient velocity. Normal transient velocities are due to rapid loading or unloading of the unit. The valve or gate operating system usually is DC electrically powered with a stored energy (nitrogen) backup, or mechanically counterweighted.

For exposed penstocks in seismic zones, seismically triggered shutoff systems should be considered.

1.5.4 Turbine Shutoff Valve

A turbine shutoff valve (TSV) must be located at the lower end of long penstocks (just upstream of the turbine). The valve must be designed to close during excessive flow conditions, such as when turbine wicket gates become blocked and cannot be closed. Generally, the operating system for a TSV incorporates water-operated cylinders, mechanical counterweighted systems, or a combination of both.

A TSV also is used for maintenance of the turbine equipment (upstream clearance point). Use of the penstock shutoff valve at the upper end of the penstock for this purpose would require draining of the penstock each time maintenance is performed.

Numerous large projects with short, large-diameter penstocks, such as Grand Coulee, Robert Moses, Chief Joseph, Guri (Venezuela), and Itaipu (Brazil) operate successfully without turbine shutoff valves. For short and especially large-diameter penstocks, use of TSVs may be uneconomical. Spherical valve diameters are limited to approximately 15 feet, and butterfly valves usually are limited to 25 feet. Since these types of projects have short penstocks, uncontrolled turbine flow can be shut off by closing the intake shutoff gates.

1.6 Prevention of Vibration

Pressure pulsations generated within the hydraulic conduit system may cause vibration of a free-standing penstock if the frequency of the pressure pulsations within the system is very close to one of the natural frequencies of the penstock. If the forcing frequency of the pressure pulsation is exactly equal to a natural frequency in one or more of the penstock's modes of vibration, a state of resonance may develop and severe vibration may occur. When the amplitude of the vibrations is excessive, fatigue and sudden failure of the penstock shell is a possibility since an excessive number of oscillations may occur in a relatively short period. Failure is most likely near supports where high localized restraint is introduced.

1.6.1 Sources of Pressure Pulsations

Known sources of pressure pulsations during operation of the turbines include:

- (1) Draft tube pulsations associated with vorticity and vortex shedding in the draft tube
- (2) Turbine blade passing frequency
- (3) Rotational speed—dynamic imbalance
- (4) Water-hammer pressure waves within the penstock.

1.6.1.1 Draft Tube Pulsations

The forcing frequency (F) of draft tube pulsations associated with vorticity and vortex shedding (vortex rotating core phenomenon) has been reported to vary from about

$$\frac{RPM}{60} \times \frac{1}{3.6} \text{ to } \frac{RPM}{60} \times \frac{1}{2.8}$$

in the part load range, and from about

$$\frac{RPM}{60} \times \frac{1}{1.3} \text{ to } \frac{RPM}{60} \times 1.0$$

in the full load to overload operation range, where RPM = unit operating speed and F = frequency in Hz.

1.6.1.2 Turbine Blade Passing Frequency

The turbine blade passing frequency (F) can be determined by:

$$F = \frac{RPM}{60} \times B_1 \quad (\text{Equation 1-17})$$

or

$$F = \frac{RPM}{60} \times B_1 \times B_2 \quad (\text{Equation 1-18})$$

Where:

B_1 = number of turbine runner blades on Francis turbine

B_2 = number of wicket gates

1.6.1.3 Rotational Speed—Dynamic Imbalance

The frequency is determined from:

$$F = \frac{RPM}{60}$$

1.6.1.4 Water-Hammer Pressure Waves Within Penstock

Water-hammer pressure waves are generated within a penstock by changes in flow due to turbine wicket gate movements or valve movements. The intensity of the pressure wave produced is proportional to the speed of propagation of the pressure wave and the change in water velocity. These pressure waves normally damp out rapidly when the turbine wicket gate movements terminate. However, in the presence of governor hunting, for example, resonance could develop. The theoretical frequency of these pressure surges generated within the penstock is determined by:

$$F = \sum_{i=1}^{i=n} \frac{a_i}{4L_i} = \frac{a_1}{4L_1} + \frac{a_2}{4L_2} + \dots + \frac{a_n}{4L_n}$$

Where:

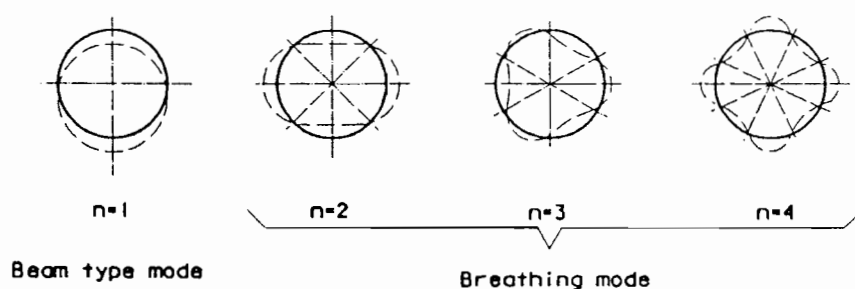
L_i = length of pipe sections of different diameter, ft

a_i = velocity of water-hammer wave in penstock section of length L_i , ft/s

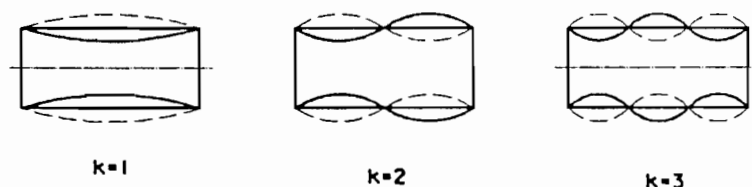
When a penstock has varying diameters, the apparent frequency will be slightly less than the theoretical frequency. The higher frequency harmonics of the penstock will differ slightly from being exact integer multiples of either the theoretical or the apparent frequency.

1.6.2 Vibration Frequency of Penstocks

Figure 1-2 shows the vibration mode shapes of simply supported, unstiffened cylindrical shells. In addition to the obvious factors of penstock diameter, plate thickness, and support span, other factors that affect the natural frequencies of penstocks include the internal water pressure, axial tension, and specific support geometry. It has been noted that penstock spans along steeply inclined sections can have different vibration frequencies due to differences in internal pressure.



(A) CIRCUMFERENTIAL MODE



(B) LONGITUDINAL MODE

Figure 1-2 Vibration Mode Shapes of Simply Supported, Unstiffened Cylindrical Shells

Circumferential mode flexing and vibration can be particularly troublesome for penstocks with low head and large discharge with their relatively large diameters and thin shell thicknesses (large D/t ratio).

The most appropriate method of determining the vibrating frequencies of penstocks is to perform a three-dimensional Finite Element static and dynamic analysis of the penstock that carefully models the support conditions and geometry. A three-dimensional Finite Element program that provides static and dynamic stresses and displacements can be used.

The selected final penstock design must ensure that the natural frequencies of the penstock are different by at least 20% from the expected forcing frequencies of pressure pulsations generated within the hydraulic system.

Circumferential ring stiffeners spaced at intervals along the span of a penstock are very useful in changing the frequency of the penstock in the circumferential, breathing modes of vibration.

1.7 Wind, Snow, and Ice Loading

Loading on the penstock due to a combination of wind and snow should be considered during the design of the penstock. As a conservative practice, it is recommended that the combination of both loads be taken into account and provided for in the penstock design. Both loads are considered to be equivalent static loads in the following sections.

1.7.1 Wind Loading

Wind loading and accompanying design criteria must comply with local codes and regulations. As a minimum, the loads and design criteria specified in Chapter 23, Part V of the Uniform Building Code⁵ (UBC) must be used. The design wind pressure (P) should be determined by using the following formula from Section 2311 of the UBC:

$$P = (C_e) (C_q) (q_s) (I) \quad (\text{Equation 1-19})$$

The importance factor (I) should be equal to 1.15. The C_q factor (pressure coefficient from Table No. 23-H of the UBC) should be 0.8. The C_e factor for the appropriate exposure should be used. The value of q_s and the minimum basic wind speed should be in accordance with Chapter 23 (Figure No. 1 and Table No. 23-F) of the UBC.

Wind-loading design must include the diameter of the penstock plus the ice layer. Local codes must be reviewed and incorporated as required.

Although there is little chance that wind vortex shedding will force an empty penstock into resonance, it is recommended that the penstock be checked for this condition, using the available methods for calculating resonant wind velocity.

1.7.2 Snow and Ice Loading

Snow-loading design must comply with local codes and, as a minimum, with requirements specified in the UBC Chapter 23, Section 2305. Values listed for flat or rise less than 4 inches per foot must be used, as a minimum. Lighter loads are not recommended even in areas without snow because of the live loads that must be resisted during construction and also in service. Ice loads in localized areas may be significant. Regions adjacent to portals and expansion joints or mechanical couplings could be subjected to ice buildup of up to 2 feet. In these areas, the weight of the ice must be included in the dead-load assessment.

1.8 Earthquake Loading

1.8.1 General Loading Criteria

The seismic design criteria for a penstock (including supports, anchor blocks, and the related structural system) are intended to: (1) protect the economic investment in the facility, (2) provide acceptably high functional reliability within a short time following an earthquake, and (3) ensure safety to life.

Where seismically induced damage or failure of the penstock poses no risk of uncontrolled release of water from the reservoir, the seismic criteria derived from the UBC are applicable. These criteria serve as guidelines for a minimum seismic design level. Special conditions, such as unusually high economic value or very high reliability requirements, could lead the owner to select a more stringent level of seismic design.

For penstocks in which seismically induced damage or failure could result in the uncontrolled release of stored water, with the possibility of significant flooding and loss of life or significant economic loss, seismic criteria that are equivalent to the probability levels of the criteria used for the design or safety review of the dam must be specified. Such seismic criteria are usually site-specific and are based on a maximum credible earthquake. This higher level of seismic design, discussed in Section 1.8.1.2, is intended to provide a level of penstock earthquake safety equivalent to that of the associated dam or water-retaining structure.

1.8.1.1 Design Basis Criteria

The Design Basis Criteria (DBC) specify the minimum seismic design standards for use in penstock design. The criteria are intended to enable the penstock to safely withstand the DBC-specified seismic loads, although some local repairable damage is acceptable for this loading. The existence of special conditions, noted in Section 1.8.1, may require more conservative design criteria. The DBC are determined by reference to the Uniform Building Code, using a zone factor (Z) appropriate to the location of the penstocks, an I factor value of 1.25 (not directly related to the importance factor I of the UBC), and an R_w factor equal to 3.0. The vertical criteria are equal to two-thirds of the horizontal criteria.

1.8.1.2 Dam Safety Criteria

When the penstock and its related structural system are an integral part of the structure of the associated dam or water-retaining structure, and the damage or failure of the penstock in an earthquake may jeopardize the safety of the dam, seismic criteria must be used that are appropriate for dam design or dam safety analysis at the site. Such criteria usually are based on an assessment of the maximum credible earthquake for the site. The maximum credible earthquake is defined as that earthquake which would cause the most severe vibratory ground motion at the site, considering the geologic evidence of past fault movement and the recorded seismic history of the area. For the Dam Safety Criteria (DSC), the penstock design is evaluated to ensure that it can withstand the seismic loading associated with a maximum credible earthquake without any sudden or uncontrolled release of the reservoir. Some

penstock damage may occur. The structure, systems, and components critical to retention or controlled release of the reservoir are required to function during and after a maximum credible earthquake.

Normally, in regions subject to frequent earthquakes of moderate to major intensity (UBC Seismic Zone No. 3 or 4), automatic control inlet gates are installed to prevent or mitigate the uncontrolled release of penstock water.

1.8.1.3 Special Considerations

Penstocks may be situated on noncohesive soils, soils subject to liquefaction (which alters soil conditions along the penstock alignment), or geologic discontinuities such as faults or other structural weaknesses. These types of geotechnical conditions may result in differential movement and settlement during moderate to major earthquake activity. The performance of a penstock supported by these types of foundation conditions is difficult to predict. The designer must utilize design details that can accommodate differential movement.

Wherever possible, the penstock alignment and foundation type should be adjusted to locate the penstock on suitable rock and soil that will minimize tendencies for differential movement and settlement.

1.8.2 Seismic Analysis (Simplified Method)

The method given in Section 2312(i) of Chapter 23 of the Uniform Building Code may be applied for the DBC discussed in Section 1.8.1.1. This method should be used to calculate the stresses in the two horizontal directions, one at a time, with the vertical direction acting simultaneously. The final stresses should be combined by the sum of the squares (SRSS) method.

For UBC Zone 3 and 4, further analysis, as directed below, may be required.

1.8.3 Seismic Analysis (Classical Method)

A qualified specialist must be retained to evaluate the equivalent DBC and DSC for the project, including consideration of all pertinent seismic zones. Also, the specialist should propose horizontal and vertical ground motions for the above-described earthquakes, with corresponding mean plus one standard deviation response spectra curves using 5% damping.

1.8.3.1 Pseudostatic Analysis

This method is applicable only if the penstock support system is very stiff and close to the ground. To check the assumption, the fundamental frequency (f) should be greater than 33 Hz. The equivalent static loading may be determined by multiplying the mass of the penstock and the contained water by the applicable zero period acceleration (ZPA). This method should be used to calculate stresses in the two horizontal directions, one at a time, with the vertical direction acting simultaneously. The final stresses should be combined by the sum of the squares (SRSS) method.

1.8.3.2 Pseudodynamic Analysis

A pseudodynamic analysis, or response spectra analysis, more accurately determines seismic effects. The response spectra derived by the seismic geologist are used in a modal-type finite element analysis. This analysis gives the stress maximums of the different modes of the penstock. Usually such analyses are limited to the first 10 modes and the results are combined by the SRSS method. However, if the frequency of successive modes is less than 10% apart, it may be more appropriate to combine them by the absolute sum method. Mass participation in the first 10 modes must be not less than 90%; if less, additional modes must be added until 90% mass participation is achieved.

A plot of the first 10 mode shapes will enable the engineer to predict the movement of the penstock during the earthquake and determine if certain deflections are within the acceptable range.

A response spectra analysis yields a combined maximum stress for several modes and is not really the true state of stress at any one time in the penstock. This method, while more accurate than the pseudostatic method, is appropriate for most penstock analysis and design.

1.8.3.3 Time History Analysis

To assess completely the effect of any seismic loading on a penstock, a time history analysis is necessary. However, because of economic considerations, this may not always be feasible, but is certainly worthwhile in areas of heavy seismic activity.

A time history analysis is simply a dynamic finite element analysis that yields stresses and deflections at specified time intervals for a corresponding ground motion. Because the volume of output is quite extensive, critical areas that are suspected to be greatly affected by the proposed ground motion parameters should be identified by either a pseudostatic analysis or a response spectra analysis. The output of the time history analysis should be summarized with post-processing computer programs and plotted to form a time history of stresses to allow the designer to determine the maximum instantaneous stress and the number of excursions beyond yield stress. This allows the designer to judge the severity of such excursions on the structural integrity of the penstock during and after the earthquake. A time history of deflections also may be useful in assessing the penstock. This analysis may have significant meaning for penstocks located on flexible substructures such as bridges or high piers.

1.9 Corrosion

Corrosion in penstocks can be a very complex problem. Generally, protection against corrosion is handled by coatings, linings, and cathodic protection (see Section 10). Water quality, environment, and high-velocity corrosion and erosion should be evaluated to determine the type of system needed.

1.9.1 Water Quality

All types of water can attack the penstock metal. Water quality ranges from balanced mountain water to aggressive water. Chemical tests should be performed to determine if the water is corrosive. Since normally very little can be done to change the quality of the water, special types of coatings and linings should be specified for the particular operating environment.

1.9.1.1 Pit Depth

Much can be learned about the corrosiveness of water by examining the depth of pits in existing pipes or penstocks. As a rule of thumb, for internal corrosion only, pit depth increases with the cube root of time. Thus, if a pit depth has reached 1/4 inch in 1 year, it will be about 1/2 inch in 8 years.

1.9.1.2 Dissolved Oxygen

Corrosion occurs only in the presence of oxygen. Water from lakes or streams usually contains dissolved oxygen. Loss of dissolved oxygen may indicate oxidizable organic matter or gross corrosion.

1.9.1.3 Corrosive Index

The Langelier Index is a common index used to indicate calcium carbonate saturation. Calcium carbonate coatings protect metal from corrosion.

1.9.1.4 pH

pH is an expression of the intensity of the basic or acid condition of a liquid. The pH may range from 0 to 14, where 0 is the most acidic, 14 is the most basic, and 7 is neutral. Natural waters usually have a pH between 6.5 and 8.5. The hydrogen ion is extremely active (corrosive) at pH values below 6; this is an acidic condition in which metal will be corroded.

1.9.2 Environment

In various parts of the country, where salt is used for deicing roads in the winter, severe exterior corrosion can occur unless the penstock is properly protected.

Penstocks supported on saddles experience their highest degrees of corrosion at the saddle horns and at the contact area between the shell and the support. At the horns, large concentrated bending stresses aggravate corrosion. At the shell/saddle contacts, rain or condensation runs down between them and becomes trapped. These areas are virtually impossible to maintain after their installation.

In colder regions, light-colored penstocks can develop internal ice build-up in the winter. Conversely, in warmer regions, dark-colored penstocks may absorb more heat and result in larger thermal gradients (uniform and nonuniform).

1.9.3 High-Velocity Corrosion and Erosion

The effect of water velocity on the corrosion rate depends on the characteristics of the material and the environment.⁶ High water velocities in uncoated and improperly coated steel pipelines can cause corrosion.

Velocity plays an important role in corrosion because it strongly influences the mechanisms of the corrosion reaction. Figure 1-3 shows typical effects of velocity on the corrosion rate. Curve A is typical of a metal that is easily passivated, such as stainless steel. There is an active-to-passive transition after an initial increase in the corrosion rate with increasing passivation.

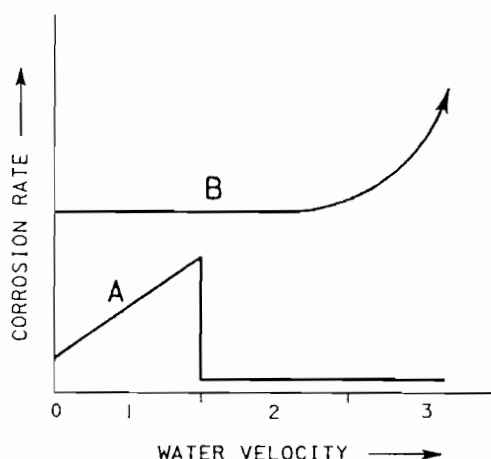


Figure 1-3 Effect of Water Velocity on Corrosion Rate

Some metals acquire their corrosion resistance from bulk protective surface films, such as oxide layers. These films are different from passivating films (such as on stainless steel) because they are visible and loosely adhered. At higher velocities, damage or removal of these films occurs. This results in accelerated attack as shown by curve B.

For certain alloy/environmental combinations (as in curve B), there is no effect up to a critical velocity, but at that point rapid attack occurs. For carbon steel the corrosion rates are minor when the flow is increased from 1 to 4 feet per second, but destructive attack occurs at approximately 27 feet per second (about 60 mils per year). Table 1-1 shows corrosion rates for several materials in water at different velocities.

Table 1-1 Corrosion by Water Moving at Different Velocities

MATERIAL	TYPICAL CORROSION RATE, mdd*		
	1 FT/S	4 FT/S	27 FT/S
CARBON STEEL	34	72	254
CAST IRON	45	—	270
SILICON BRONZE	1	2	343
ADMIRALTY BRASS	2	20	170
HYDRAULIC BRONZE	4	1	339
G BRONZE	7	2	280
Al BRONZE (10% Al)	5	—	236
ALUMINUM BRASS	2	—	105
90-10 Cu Ni (0.8% Fe)	5	—	99
70-30 Cu Ni (0.05% Fe)	2	—	199
70-30 Cu Ni (0.5% Fe)	<1	<1	39
MONEL	<1	<1	4
STAINLESS STEEL TYPE 316	1	0	<1
HASTELLOY C	<1	—	3
TITANIUM	0	—	0

* mdd (milligrams/square decimeter/day) = .183 mpy (mils/year)

The actual corrosion mechanism is thought to be due either to an increase in oxygen (or other gas) in contact with the surface or the velocity causing an increase in the diffusion or transfer of ions when the film thickness is reduced.

The potential effects of high-velocity corrosion/erosion should be considered in developing the penstock diameter (and therefore velocities) and in the selection of protective coatings and linings.

The coating application quality assurance program is extremely important in penstocks with high water velocities. The coating material must be holiday free; otherwise, the effects of corrosion/erosion may be more severe than for uncoated penstocks.

1.10 Maintenance Provisions

1.10.1 Inspection/Maintenance Program

The owner must establish, document, and maintain a comprehensive, routine inspection/maintenance program for the installed penstock system. The program must describe the inspection items included, the type of inspection/maintenance required for each item, acceptance tolerances and criteria, the frequency of inspection/maintenance, descriptions of possible breakdowns and other problems, and records of dates and actions taken of any inspection, maintenance, or repair. Any maintenance provisions specified by the material supplier or the fabricator must be included in the program for warranty purposes. The inspection/maintenance instructions must include diagrams of the installed component or system.

The owner's inspection/maintenance program must include provisions for:

- (1) Unusual movement
- (2) Excessive vibration

- (3) Leakage
- (4) Penstock age and material specification
- (5) Penstock shell condition (inside and outside)
- (6) Welds
- (7) Bolts/rivets
- (8) Expansion joints/bolted sleeve-type couplings
- (9) Air valves
- (10) Valves or other water control systems
- (11) Manholes and other penetrations
- (12) Anchor blocks and supports
- (13) Coating/linings (inside and outside)
- (14) Instrumentation (as appropriate).

1.10.2 Spare Parts Data

Each material supplier must furnish spare parts data to the owner for each different item of material and equipment specified, prior to final acceptance by the owner. The data must include a complete list of parts, special tools, supplies (including shelf life if applicable), and installation procedures, with current unit prices and supply sources. The spare parts data should be included in the inspection/maintenance procedure for each applicable component or system.

1.10.3 Personnel Training

If required by the owner in the contract documents, the penstock fabricator/installer must train plant operators and the maintenance staff to familiarize them with the maintenance provisions and action requirements of each pertinent system, component, and piece of equipment. Included in the training should be the specifics of what to look for, and why it is important, as well as possible breakdowns, causes, and corrective action. It is important that the plant staff have a thorough working knowledge of the entire penstock system to ensure adequate maintenance of the system after it is in operation. The training should be conducted after the penstock system is functionally completed but before final acceptance tests.

1.11 Geologic Considerations

This section briefly describes the geologic aspects of a penstock installation and is not intended to serve as a complete reference on the subject. Suggestions for further reading include the ASCE/EPRI *Civil Engineering Guidelines*,² Vol. 1, Division II, Chapter 6; *Handbook of Geology in Civil Engineering*;⁷ and *Geotechnical Engineering Investigation*.⁸

The terms *rock* and *soil* are used here in an engineering context. Namely, *rock* is a hardened, coherent, solid mass of earth material, and *soil* is a loose to consistent particulate earth material.

1.11.1 Investigations and Related Designs

One convenient method for establishing the scope of geotechnical investigations and designs is to identify all the geologic considerations pertinent to a given penstock installation. A geologic consideration involves a relationship between engineering and the geologic domain that must be considered when planning, designing, constructing, and maintaining a project. Each geologic consideration becomes an individual objective within the overall geotechnical program. Section 1.11.2 identifies relevant geologic considerations for a penstock installation. Identification of geologic considerations does not require proficiency in geology and is accomplished under the supervision of the engineer responsible for the penstock.

A principal challenge for each geologic consideration is the prediction of ground behavior, which does not necessarily require a complex mathematical analysis. Ground behavior may be analyzed by inference using existing knowledge, past experience and practice, or by observation of physical models, prototypes, and nature. One analysis may combine two or three of these methods. Selection of the proper method or methods depends upon various factors, such as the existence of suitable numerical expression, available knowledge, regulatory requirements, risks, and cost. Timely evaluation and selection of the appropriate means of behavioral analysis is crucial because the type of analysis determines the nature and quantity of data necessary to perform the evaluation. Some data will have to be collected during the exploration phase of the geotechnical program.

The geology of the site obviously dictates the ground behavior. Specific geologic concerns mark geologic conditions that are particularly favorable or adverse to the resolution of a geologic consideration. For each geologic consideration, background and site-specific information is collected to characterize the geologic materials and structure and their behavioral properties. Also, the collected information must support the prediction of ground behavior at an acceptable level of confidence. Geologic data-gathering operations incorporate surface and subsurface geologic probings, exploration and borehole geophysics, and tests for material properties and energy states of the ground, e.g., stress.

If the site geology does not provide suitable foundations and conditions for a penstock installation, engineering works can be used to enhance the strength and deformation characteristics of the natural ground and to control or cut off ground water.

The methods and results of these explorations, tests, analyses, and designs must be fully documented. If the consequences of a faulty prediction of ground behavior are severe or

unacceptable, a validation program using visual and measuring techniques must be instituted to confirm the geologic interpretations and to monitor geologic behavior during construction and operation.

Because they can be compiled early, geologic considerations (along with their individual geologic concerns and likely means of analysis) are very useful for defining the scope of investigations to be performed by a geotechnical group or consulting firm. Also, the geologic concerns established for each geologic consideration highlight important geologic information that may be needed in the construction specification.

1.11.2 Relevant Geologic Considerations

Following are general geologic considerations relevant to the design and construction of steel penstocks and tunnel liners. Geologic considerations for tunneling are not included. The applicability of some geologic considerations depends upon the penstock's profile and geologic setting.

- (1) Foundation for anchor block, pier, and penstock bedding
 - (a) Bearing capacity, including liquefaction potential
 - (b) Sliding resistance
 - (c) Settlement and swelling-ground displacements
 - (d) Rock-anchor bar and tendon pullout capacity
 - (e) Damage to foundation by construction
 - (f) Erosion (from penstock leakage or natural watercourse)
- (2) Piles and drilled piers
 - (a) Depth to "sound" ground
 - (b) Ground bearing capacity at pile point or pier end
 - (c) Ground to shaft (skin) shear resistance
- (3) Permanent excavations
 - (a) Cut-slope stability (includes falling rocks that may hit penstock)
 - (b) Bottom heave

- (4) Buried steel penstock—external pressures and loads
 - (a) Groundwater (from any source) pressures
 - (b) Earth material loads
- (5) Steel tunnel liner, including concrete backfill
 - (a) External pressures and loads, such as groundwater (from any source) pressures, and squeezing, swelling, and other unstable ground loads
 - (b) Load-carrying characteristics of the ground, e.g., modulus of deformation used in composite design method for tunnels and susceptibility to hydraulic fracturing
- (6) Availability and quality of earth construction materials
- (7) Effects of geology on construction operations
 - (a) Excavating and excavation surfaces
 - (b) Pile driving
 - (c) Grout injection
 - (d) Dewatering
 - (e) Traffic access bearing capacity and abrasiveness
 - (f) Processing of earth materials for aggregates, etc.
 - (g) Stability and maintenance of temporary cut-slopes, trench walls, staging and stockpile areas
- (8) The alteration of shallow geologic materials (e.g., weathering) and the deterioration of engineering materials (e.g., subsurface corrosion and erosion, alkali-aggregate reaction) subjected to geologic agents and ground currents
- (9) Natural geologic hazards
 - (a) Earthquakes and fault movements
 - (b) Volcanic lava, ash flows and lahars (debris flows)
 - (c) Landslides and hillside stability (includes falling rocks that may hit the penstock)

- (d) Natural (and man-caused) ground subsidence and collapse
- (e) Erosion and sedimentation

The geologic considerations for any needed ground-improvement works, such as the example of piles and drilled piers mentioned in (2) above, are identified after the decision is made on the kind of works to be used.

1.11.3 Adverse Geologic Conditions

Common geologic features that may adversely affect the behavior of the ground include:

- (1) Low-strength ground
- (2) Cohesive and noncohesive particulate beds and seams
- (3) Weak beds or seams relative to the surrounding ground
- (4) Soluble and partially soluble ground
- (5) Volcanic sequences
- (6) Frost and permafrost
- (7) Fractures and fracture zones
- (8) Joints
- (9) Shears, including bedding/foliation shears
- (10) Faults
- (11) Distorted zones in the ground adjacent to shears and faults
- (12) Shear fractures
- (13) Irregular weathering fronts caused by decomposition, disintegration, and solutioning that create highly variable geologic profiles and material properties.

1.11.4 Engineering Works to Improve Ground Behavior

Categories of common works for improving the ground behavior include:

- (1) Removal of any undesirable, unstable, or burdensome ground (excavation, washing, or jetting, and moat or open annulus around structural support for isolation from unstable ground)
- (2) Filling of larger re-entrants and cavities in the ground (backfill concrete and select earth material)
- (3) Replacement of unsuitable ground with suitable materials (soil/bentonite-, cement/bentonite-, or concrete-filled slurry trenches, sheet piling, grouting-induced and filled fractures, and concrete supplantations)
- (4) Protection of the ground surface from excessive or concentrated mechanical, thermal, or chemical loads and agents (bearing pads, surface coatings, vegetation covers, and surface drainage)
- (5) Solidification or cementation of the ground (void grouting, slush grouting, jet grouting, and mixed-in-place columns and walls)
- (6) Adjunct reinforcement of the ground for compression, tension, and shear resistance (steel, concrete, and wood piles, caissons, sand and stone columns, root piles and soil nails, anchor bars, dowels, bolts and tendons, shear keys, and graveling of finer surface soils)
- (7) Drainage of the ground (drainage holes and tunnels, wicks, sand drains, electro-osmosis, and consolidation)
- (8) Densification of the ground (blasting, vibrocompaction, heavy tamping, compaction grouting, compaction piles, and precompression)
- (9) Chemical treatment of clays (liming, salting, and electro-chemical injection)
- (10) Thermal treatment of the ground—usually temporary (freezing, drying, and heating)
- (11) External stabilization of the ground (retaining walls, earth berms, shot-concrete skins, and tunnel set supports).

References*

1. *Steel Pipe—A Guide for Design and Installation*. AWWA Manual M11. American Water Works Association, Denver, CO (1989).
2. *ASCE/EPRI Civil Engineering Guidelines for Planning and Designing Hydroelectric Developments*. American Society of Civil Engineers, New York, NY (1989).
3. Proceedings of the American Society of Civil Engineers. *Journal of the Power Division* (July 1971).
4. Selection of Hydraulic Reaction Turbines. Engineering Monograph No. 20, U.S. Bureau of Reclamation (1976).
5. Uniform Building Code. International Conference of Building Officials, Whittier, CA.
6. Fontana, M.G. and Greene, N.D. *Corrosion Engineering*. McGraw-Hill Book Co., New York, NY (1967).
7. Legget, R.F. and Karrow, P.F. *Handbook of Geology in Civil Engineering*. McGraw-Hill Book Co., New York, NY (1983).
8. Hunt, R.E. *Geotechnical Engineering Investigation*. McGraw-Hill Book Co., New York, NY (1984).

The following references are not cited in the text.

Bhave, S. K. et al. "Vibrations in Penstocks." *Water Power & Dam Construction* (Nov. 1987).

Experiences of the Bureau of Reclamation with Flow-Induced Vibrations. Hydraulics Branch, Laboratory Report No. Hyd.-538 (1964).

Fazalare, R. W. "Trends in Selecting and Procuring Hydro Turbines." *Water Power & Dam Construction* (Dec. 1986).

Jaeger, C. "Theory of Resonance in Hydro-Power Systems." *Water Power* (1963).

Kito, F. "The Vibration of Penstocks." *Water Power*. (Oct. 1959).

Shiraki, K., Nagata, O., and Honma, T. "The Penstock Vibration Characteristics—On the Vibrations of Simply Supported and Ring-Stiffened Cylindrical Shells Filled with Pressurized Water." ASME Publication (June 1975).

* The most current version of a standard, code, or specification should be used for reference.



2.1 General

All steel materials used in the fabrication of penstocks, including pressure-carrying components and non-pressure-carrying attachments such as flanges, ring girders, stiffener rings, thrust rings, lugs, support systems, and other appurtenances must be manufactured and tested in strict accordance with appropriate ASTM specifications and as specified in this manual.

The properties of steels are governed by their chemical composition, by the processes used to form the base metal, and by their heat treatment. The effects of these parameters on the properties of steels are discussed in the following sections.

2.1.1 Chemical Composition

Steels are a mixture of iron and carbon with varying amounts of other elements, primarily manganese, phosphorus, sulfur, and silicon. These and other elements are either unavoidably present or intentionally added in various combinations to achieve specific characteristics and properties of the finished steel products. The properties and uses of the commonly used chemical elements on the properties of hot-rolled and heat-treated carbon and alloy steels are presented in Table 2-1. The effects of carbon, manganese, sulfur, silicon, and aluminum are of primary interest.

Table 2-1 Properties and Uses of Alloying Elements

ELEMENT	PROPERTY/USE
CARBON (C)	PRINCIPAL HARDENING ELEMENT IN STEEL
	INCREASES STRENGTH AND HARDNESS
	DECREASES DUCTILITY, TOUGHNESS, AND WELDABILITY
	MODERATE TENDENCY TO SEGREGATE
MANGANESE (Mn)	INCREASES STRENGTH
	CONTROLS HARMFUL EFFECTS OF SULFUR
PHOSPHORUS (P)	INCREASES STRENGTH AND HARDNESS
	DECREASES DUCTILITY AND TOUGHNESS
	CONSIDERED AN IMPURITY, BUT SOMETIMES ADDED FOR ATMOSPHERIC CORROSION RESISTANCE
	STRONG TENDENCY TO SEGREGATE
SULFUR (S)	CONSIDERED UNDESIRABLE EXCEPT FOR MACHINABILITY
	DECREASES DUCTILITY, TOUGHNESS, AND WELDABILITY
	ADVERSELY AFFECTS SURFACE QUALITY
	STRONG TENDENCY TO SEGREGATE
SILICON (Si)	USED TO DEOXIDIZE OR "KILL" MOLTEN STEEL
ALUMINUM (Al)	USED TO DEOXIDIZE OR "KILL" MOLTEN STEEL
	REFINES GRAIN SIZE, THUS INCREASING STRENGTH AND TOUGHNESS
VANADIUM (V) AND COLUMBIUM (Nb)	SMALL ADDITIONS INCREASE STRENGTH
NICKEL (Ni)	INCREASES STRENGTH AND TOUGHNESS
CHROMIUM (Cr)	INCREASES STRENGTH
	INCREASES ATMOSPHERIC CORROSION RESISTANCE
COPPER (Cu)	PRIMARY CONTRIBUTOR TO ATMOSPHERIC CORROSION RESISTANCE
NITROGEN (N)	INCREASES STRENGTH AND HARDNESS
	DECREASES DUCTILITY AND TOUGHNESS
BORON (B)	SMALL AMOUNTS (0.0005%) INCREASE HARDENABILITY IN QUENCHED AND TEMPERED STEELS
	USED ONLY IN ALUMINUM-KILLED STEELS
	MOST EFFECTIVE AT LOW CARBON LEVELS

Carbon is the principal hardening element in steel; the addition of carbon increases the steel hardness and tensile strength. However, carbon has a moderate tendency to segregate, and increased amounts of carbon cause a decrease in ductility, toughness, and weldability.

Manganese increases the hardness and strength of steels but to a lesser degree than carbon. Manganese combines with sulfur to form manganese sulfides, thus decreasing the harmful effects of sulfur.

Sulfur is generally considered an undesirable element except where machinability is an important consideration. Sulfur adversely affects surface quality, has a strong tendency to segregate, and decreases ductility, toughness, and weldability.

Silicon and aluminum are the principal deoxidizers used in the manufacture of carbon and alloy steels. Aluminum is used to control and refine grain size.

2.1.2 Casting

In the traditional steelmaking process, molten steel is poured (teemed) into a series of molds to form castings known as ingots. The ingots are removed from the molds, reheated, and then rolled into products with square or rectangular cross sections. This hot-rolling operation elongates the ingot and produces semi-finished products known as blooms, slabs, or billets.

All ingots exhibit some degree of nonuniformity of chemical composition known as segregation, which is an inherent characteristic of the cooling and solidification of the molten steel in the mold.

The first liquid steel to contact the relatively cold walls and bottom of the mold solidifies very rapidly. However, as the rate of solidification decreases away from the mold sides, crystals of relatively pure iron solidify first. Thus, the first crystals to form contain less carbon, manganese, phosphorus, sulfur, and other elements than the liquid steel from which they were formed. The remaining liquid is enriched by these elements, which are continually being rejected by the advancing crystals. Consequently, the last liquid to solidify, which is located around the axis in the top half of the ingot, contains high levels of the rejected elements and has a lower melting point than the poured liquid steel. This segregation of the chemical elements is frequently expressed as a local departure from the average chemical composition. In general, the content of an element that has a tendency to segregate is greater than average at the center of the top half of an ingot and less than average at the bottom half of an ingot.

Certain elements tend to segregate more readily than others. Sulfur segregates to the greatest extent. The following elements also segregate, but to a lesser degree, and in descending order: phosphorus, carbon, silicon, and manganese. The degree of segregation is influenced by the composition of the liquid steel, the liquid temperature, and ingot size. The most severely segregated areas of the ingot are removed by cropping, which is the cutting and discarding of sufficient material during rolling.

Continuous casting is the direct casting of steel from the ladle into slabs. This steelmaking development bypasses the operations between molten steel and the semi-finished product which are inherent in making steel products from ingots. In continuous casting, molten steel is poured at a regulated rate into the top of an oscillating water-cooled mold with a cross-sectional size corresponding to the desired slab. As the molten metal begins to solidify along the mold walls, it forms a shell that permits the gradual withdrawal of the strand product from the bottom of the mold into a water-spray chamber where solidification is completed. The solidified strand is cut to length and then reheated and rolled into finished products as in the conventional ingot process. The smaller size and higher cooling rates for the strand result in less segregation and greater uniformity in composition and properties for steel products made by the continuous casting process than for ingot products.

2.1.3 Killed and Semi-Killed Steels

The primary reaction involved in most steelmaking processes is the combination of carbon and oxygen to form carbon monoxide gas. The solubility of this and other gases dissolved in the steel decreases as the molten metal cools to the solidification temperature range. Thus, excess gases are expelled from the metal and, unless controlled, continue to evolve during solidification. The oxygen available for the reaction can be eliminated and the gaseous evolution inhibited by deoxidizing the molten steel using additions of silicon or aluminum or both. Steels that are strongly deoxidized do not evolve any gases and are called killed steels because they lie quietly in the mold. Increasing amounts of gas evolution results in semi-killed, capped, or rimmed steels.

In general, killed steel ingots are less segregated and contain negligible porosity when compared to semi-killed steel ingots. Consequently, killed steel products usually exhibit a higher degree of uniformity in composition and properties than semi-killed steel products.

2.1.4 Heat Treatments for Steels

Steels respond to a variety of heat treatments, which can be used to obtain certain desirable characteristics. These heat treatments can be divided into slow cooling treatments and rapid cooling treatments. The slow cooling treatments, such as annealing, normalizing, and stress relieving, decrease hardness and promote uniformity of structure. Rapid cooling treatments, such as quenching and tempering, increase strength, hardness, and toughness. Heat treatments of base metals are generally mill options or ASTM requirements. Following are some of the more common heat treatments for steels.

(1) Annealing

Annealing consists of heating the steel to a given temperature followed by slow cooling. The temperature, the rate of heating and cooling, and the time the metal is held at temperature depends on the composition, shape, and size of the steel product being treated and the desired properties. Usually steels are annealed to remove stresses, induce softness, increase ductility and toughness, produce a given microstructure, increase uniformity of microstructure, improve machinability, and facilitate cold forming.

(2) Normalizing

Normalizing consists of heating the steel to between 1,650°F and 1,700°F followed by slow cooling in air. This heat treatment is commonly used to refine the grain size, improve uniformity of microstructure, and improve ductility and fracture toughness.

(3) Stress relieving

Stress relieving of carbon steels consists of heating and holding the steel to between 1,000°F to 1,200°F to equalize the temperature throughout the piece, followed by slow cooling. The stress-relieving temperature for quenched and tempered steels must be maintained below the tempering temperature for the product. The process relieves internal stresses induced

by welding, normalizing, cold working, cutting, quenching, and machining. It is not intended to alter significantly the microstructure of the mechanical properties.

(4) Quenching and tempering

Quenching and tempering consists of heating and holding the steel at the proper austenitizing temperature (about 1,650°F) for a significant time to produce a desired change in microstructure, then quenching by immersion in a suitable medium (such as water for bridge steels). After quenching, the steel is tempered by reheating to a temperature between approximately 800°F and 1,200°F, holding for a specified time at temperature, and then cooling under suitable conditions to obtain the desired properties. Quenching and tempering increase the strength and improve the toughness of the steel, but may decrease ductility.

(5) Controlled rolling

Controlled rolling is a thermo-mechanical treatment at the rolling mill that tailors the time-temperature-deformation process by controlling the rolling parameters. The parameters of primary importance are: (1) the temperature at the start of controlled rolling in the finishing stand, (2) the percentage reduction from the start of controlled-rolling to the final plate thickness, and (3) the plate finishing temperature.

Hot-rolled plates are deformed as quickly as possible at temperatures above 1,800°F to take advantage of the hot workability of the steel at high temperatures. In contrast, controlled rolling incorporates a hold or delay time to allow the partially rolled slab to reach a desired temperature before the start of final rolling. Controlled rolling involves deformations at temperatures between 1,500°F and 1,800°F. Because rolling deformation at these low temperatures increases the mill loads significantly, controlled rolling usually is restricted to less than 2-inch thick plates. Controlled rolling increases strength, refines the grain size, improves toughness, and may eliminate the need for normalizing.

(6) Controlled finishing-temperature rolling

Controlled finishing-temperature rolling is a less severe practice than controlled rolling and is aimed primarily at improving notch toughness of plates up to 2½ inches in thickness. The finishing temperatures (approximately 1,600°F) in this practice are higher than required for controlled rolling. However, because heavier plates are involved than with controlled rolling, mill delays still are required to reach the desired finishing temperatures. Fine grain size and improved notch toughness can be obtained by controlling the finishing temperature.

2.1.5 Mechanical Properties

The mechanical properties of a material are those properties that characterize its elastic and inelastic (plastic) behavior under stress or strain. These properties include parameters related to the material's strength, ductility, and hardness. The strength and ductility parameters under tensile loading can be defined and explained best by analyzing the tensile stress-strain curve for the material.

2.1.5.1 Stress-Strain Curves

Figure 2-1 shows an idealized tensile stress-strain curve for constructional steels. The curve is obtained by the tensile loading to failure of specimens that have rectangular or circular cross sections. A detailed discussion of definitions and details for tension test specimens and test methods is available in ASTM A370.¹

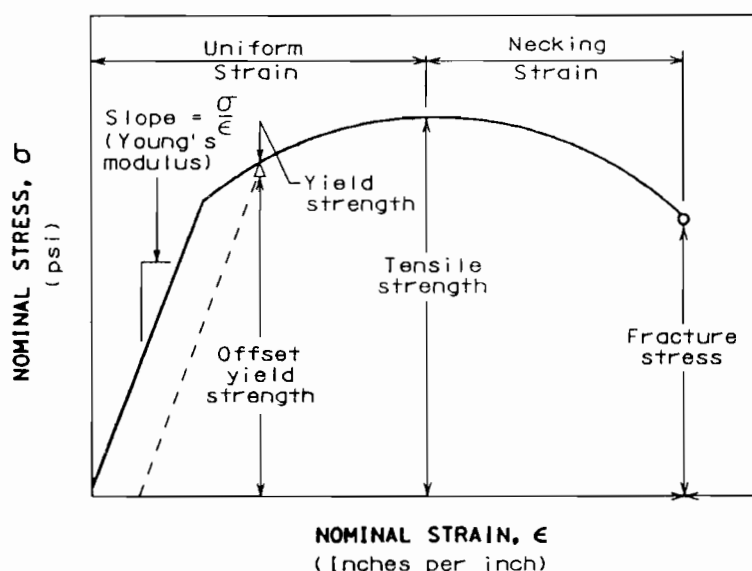


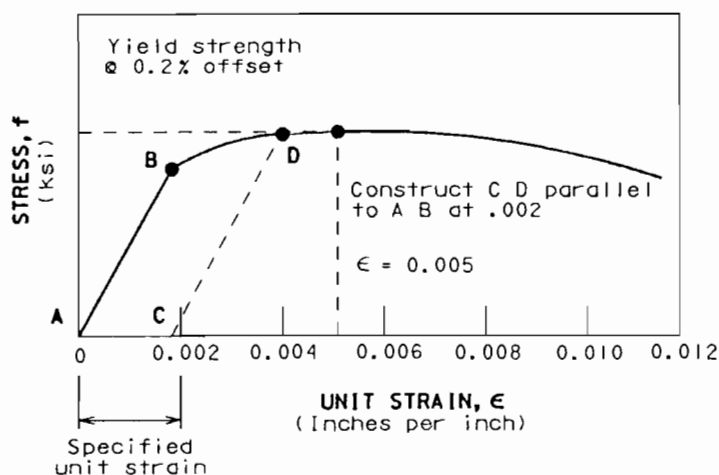
Figure 2-1 Idealized Tensile Stress-Strain Curve

The curve shown in Figure 2-1 is an engineering tensile stress-strain curve, as opposed to a true tensile stress-strain curve, because the plotted stresses are calculated by dividing the instantaneous load on the specimen by its original, rather than reduced, cross-sectional area. Also, the strains are calculated by dividing the instantaneous elongation of a gage length of the specimen by the original gage length.

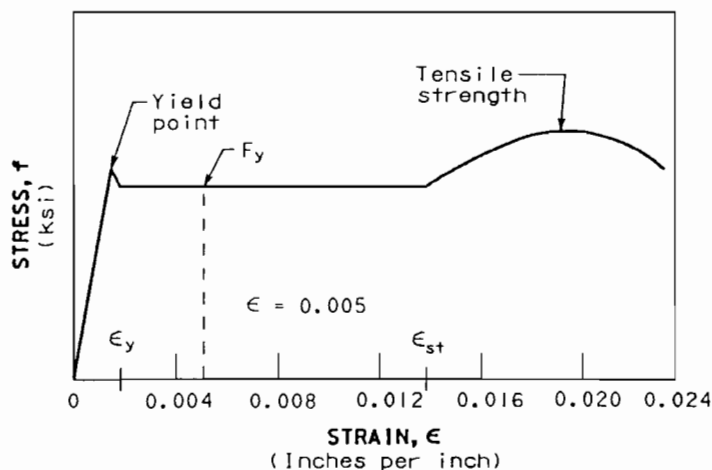
The initial straight line segment of the stress-strain curve, where stress is linearly related to strain, represents the elastic behavior of the specimen. In this region the strain is fully recoverable and the specimen returns to its original length when the load is removed. The slope of the line, which is the ratio of stress to strain in the elastic region, is the modulus of elasticity, or Young's modulus, and is equal approximately to 29×10^6 psi for steels. As the load increases, the stress and strain relationship becomes nonlinear, and the specimen experiences permanent plastic deformation. The stress corresponding to the initial deviation from linearity represents the yield strength of the material and the beginning of the plastic region. Usually, the stress required to produce additional plastic strain increases with increasing strain, and thus the steel strain hardens. The rate at which stress increases with plastic strain is the strain-hardening modulus.

Figure 2-2 shows tensile stress-strain curves for constructional steels that exhibit two types of behavior in the plastic region. The curve in Figure 2-2(A) exhibits a smooth deviation from linearity, and the stress continuously increases to a maximum value and then decreases until the specimen fractures. On the other hand, the stress for the curve in Figure 2-2(B) reaches a peak

immediately after the stress-strain curve deviates from linearity, dips slightly, and then remains at a constant value for a considerable amount of additional strain. Thereafter, the steel strain hardens and the stress increases with strain to a maximum and then decreases until the specimen fractures. The stress corresponding to the peak value represents the *yield strength* and is the stress at which the material exhibits a specific limiting deviation from linearity of stress and strain. The deviation may be expressed as a 0.2% offset or a 0.005 inch/inch total extension under load (see Figure 2-2(A)).



(A)



(B)

Figure 2-2 Schematic Stress-Strain Curves for Steels

The maximum stress exhibited by the engineering stress-strain curve corresponds to the *tensile strength* of the steel.

2.1.5.2 Ductility and Toughness

The tensile stress-strain curve can be divided into a uniform strain region and a nonuniform strain region which combine to give the total strain to fracture (see Figure 2-1). In the uniform strain region, the cross-sectional area along the entire gage length of the specimen decreases uniformly as the specimen elongates under load. Initially, the strain hardening compensates for the decrease in cross-sectional area and the engineering stress continues to increase with increasing strain until the specimen reaches its ultimate tensile strength. Beyond this point the plastic strain becomes localized in a small region of the gage length and the specimen begins to neck locally with a corresponding decrease in total stress until the specimen fractures.

The total percent elongation and the total percent reduction of area at fracture are two measures of ductility that are obtained from the tension test. The percent elongation is calculated from the difference between the initial gage length and the gage length after fracture. Similarly, the percent reduction of area is calculated from the difference between the initial and the final cross-sectional area after fracture. Both elongation and reduction of area are influenced by gage length and specimen geometry. The conversion between elongation of an 8-inch gage length strap specimen and a 2-inch gage length round specimen can be found in ASTM standard A370.¹

Ductility is an important material property because it allows the redistribution of high local stresses. Such stresses occur in welded connections and at regions of stress concentration such as holes and changes in geometry.

Toughness is the ability of a material to absorb energy prior to fracture and is related to the area under the stress-strain curve. The larger the area under the curve, the tougher the material.

2.1.5.3 Tensile Properties

The procedures and definitions for the tension test methods are presented in ASTM standard A370.¹ The general specifications and tolerances for acceptability of structural steel plates are presented in ASTM standard A6.²

Steel producers are aware of the chemical segregation and the variability in the properties of steel products. Consequently, based on their experience, they establish aim chemical compositions that ensure that the requirements of the material specifications will be met. Because of the recognized variations, the aim chemistries and processing practices usually are selected to result in properties that will exceed the minimum properties required by the material specifications.

2.1.6 Fracture Toughness

Most steels fracture either in a ductile or a brittle manner. The mode of fracture is governed by the temperature at fracture, the rate at which the loads are applied, and the magnitude of the constraints that would prevent plastic deformation. The effects of these parameters on the mode of fracture are reflected in the fracture-toughness behavior of the material. In general, the fracture toughness increases with increasing temperature, decreasing load rate, and decreasing

constraint. Furthermore, there is no single unique fracture-toughness value for a given steel even at a fixed temperature and loading rate.

Traditionally, the fracture toughness for low- and intermediate-strength steels has been characterized primarily by testing Charpy V-notch (CVN) specimens at different temperatures. However, the fracture toughness for materials can be established best by using fracture-mechanics test methods.

The Charpy V-notch impact specimen has been the most widely used specimen for characterizing the fracture-toughness behavior of steels. These specimens may be tested at different temperatures and the impact fracture toughness at each test temperature may be determined from the energy absorbed during fracture, the percent shear (fibrous) fracture on the fracture surface, or the change in the width of the specimen (lateral expansion).

At low temperatures, constructional steels exhibit a low value of absorbed energy (about 5 ft-lb), and zero fibrous fracture and lateral expansion. The values of these fracture-toughness parameters increase as the test temperature increases until the specimens exhibit 100% fibrous fracture and reach a constant value of absorbed energy and lateral expansion. This transition from brittle-to-ductile fracture behavior usually occurs at different temperatures for different steels and even for a given steel composition. Consequently, like other fracture-toughness tests, there is no single unique CVN value for a given steel, even at a fixed temperature and loading rate. Therefore, when fracture toughness is an important parameter, the design engineer must establish and specify the necessary level of fracture toughness for the material to be used in the particular structure at a given temperature or in a critical component within the structure.

2.2 Types of Materials

2.2.1 Steel Plate

Steel plate is manufactured to two standards—ASTM A6² for structural use and ASTM A20³ for pressure vessels. The physical and chemical properties of steel produced to these two standards can be very similar, particularly when Supplemental Requirements are specified. The standards differ, however, in the amount of testing required to ensure uniform quality.

The individual ASTM standards for plates give the design engineer a wide range of properties from which to select the economical material for a particular application. Structural steel (ASTM A6) is suitable for ring girders, stiffener rings, thrust rings, lugs, support systems, and for the pipe shell in many of the less critical penstocks where design stress does not exceed 21,000 psi.

Pressure vessel quality plates (ASTM A20) normally are used in the fabrication of the pipe shell for penstocks. They may also be used for crotch plates, sickle beams, ring girders, or other structural parts if desired.

2.2.2 Coils

Coils produced on modern rolling mills from continuous cast slab have proven to have the consistent chemical and mechanical properties throughout the coil required by ASTM standard A20.³ Currently, coil thickness is limited to 3/4 inch, and heat treatment of the coil (normalized or quenched and tempered) is not available.

When spiral weld pipe is produced directly from coils, all applicable portions of standards ASTM A6 or ASTM A20 apply, except that the testing provisions for chemical and physical properties are revised as follows.

The mill producing the coil must furnish certified chemical analysis of each heat. All required physical tests may be taken by the pipe manufacturer from the coil or from the completed pipe. The spiral weld pipe manufacturer must furnish certified reports of the physical tests. To qualify under ASTM A6, two sets of the required physical tests must be taken for each heat or each 50 tons of each heat. To qualify under ASTM A20, one set of the required physical tests must be taken from each coil used.

A set of physical tests must consist of three tests on a coil—one from the outside wrap of the coil, one from the middle third of the coil, and one from the inner wrap adjacent to the portion of the coil that is used. The set from the middle portion of the coil may be taken from the finished pipe. For test orientation, the longitudinal axis of the test specimens must be transverse to the final rolling direction of the coil.

Steel coil, used in the manufacture of spiral welded steel pipe, may be used if it qualifies as either PVQ (ASTM A20) or STR (ASTM A6) steel.

2.3 Temperature Considerations

The toughness of steel at low temperatures must be considered in penstock design. Toughness usually is measured by Charpy impact tests.

The lowest service temperature (LST) of a buried penstock is approximately 30°F. Also, the lowest service temperature of an aboveground penstock is close to 30°F for the pipe shell. Structural support systems, ring girders, and empty pipe may be subjected to lower temperatures.

Fine-grain, killed steel, with a 55 ksi minimum yield or less and a thickness of 5/8 inch or less, has adequate toughness at a service temperature as low as 20°F. Normalized, quenched, and tempered steels have improved toughness at even lower temperatures. See Section 2.6.3 for testing requirements for other steels and thicknesses.

2.4 Attachment Materials and Criteria

2.4.1 Nozzles

All openings in the penstock shell must be reinforced in accordance with the ASME Boiler and Pressure Vessel Code,⁴ Section VIII, Division 1, Paragraphs UG-36 and 37.

Suitable materials include:

- (1) Forged steel weld couplers, weldolets, nipplets, sweepolets, or other fittings conforming to ANSI B16.11 and of material conforming to ASTM A105,⁵ Grade 2
- (2) Steel pipe manufactured from material listed in Section 2.5 or fabricated in accordance with Section 11.

2.4.2 Bends, Tee and Reducers, and Caps

Forged steel weld fittings must be in accordance with ANSI B16.9, conforming to ASTM A234⁶ for the schedule and grade required.

2.4.3 Dished Heads

Heads may be of ellipsoidal, torispherical, hemispherical, conical, or toriconical design in accordance with the ASME Boiler and Pressure Vessel Code,⁴ Section VIII, Division 1, Paragraph UG-32, using materials listed in Section 2.5.

2.4.4 Attachments

Non-pressure carrying attachments such as ring girders, stiffener rings, thrust rings, lugs, support systems, and other appurtenances may be fabricated from any of the materials listed in Section 2.5.

2.5 Material Specifications

Table 2-2 lists some of the steels commonly used in the manufacture of steel penstocks and associated structural supports. These materials have been specified with and without supplemental requirements.

2 Materials

Table 2-2 Steels Commonly Used in the Manufacture of Penstocks and Associated Structural Supports

SPECIFICATION NO.	GRADE	MINIMUM YIELD STRENGTH (ksi)	TENSILE STRENGTH (ksi)	REMARKS
ASTM A36	—	36	58-80	STRUCTURAL
ASTM A53	B	35	60	TYPE E & S PIPE SPECIFICATION
ASTM A139	B	35	60	PIPE SPECIFICATION; SPECIFY .25 MAXIMUM CARBON, CON-CAST
	C	42	60	
	D	46	60	
	E	52	66	
ASTM A516	55	30	55-75	PVQ OVER 1-1/2 IN. IS NORMALIZED
	60	32	60-80	
	65	35	65-85	
	70	38	70-90	
ASTM A283	A	24	45-60	STRUCTURAL
	B	27	50-65	
	C	30	55-70	
	D	33	60-75	
ASTM A285	A	24	45-65	PVQ
	B	27	50-70	
	C	30	55-75	
ASTM A537 ALL PVQ CL-1 IS NORMALIZED CL-2 IS QUENCHED AND TEMPERED.	CL-1	50	70	2-1/2 IN. AND UNDER
	CL-1	45	65	OVER 2-1/2 IN. TO 4 IN.
	CL-2	60	80	2-1/2 IN. AND UNDER
	CL-2	55	75	OVER 2-1/2 IN. TO 4 IN.
	CL-2	46	70	OVER 4 IN. TO 6 IN.
ASTM A517 QUENCHED AND TEMPERED AND ALL PVQ-H.S.	—	100	115-135	2-1/2 IN. AND UNDER
	—	90	105-135	OVER 2-1/2 IN. TO 6 IN.
ASTM A572 ALL S91 KILLED FINE GRAIN	42	42	60	STRUCTURAL
	50	50	65	OVER 3/4 IN.
	50 (S81)	50	70	3/4 IN. AND UNDER
	60	60	75	1-1/4 IN. MAXIMUM
API-5L	B	35	60	PIPE SPECIFICATION INCLUDES 100% NDE
	X42	42	60	
	X46	46	63	
	X52	52	66	
	X56	56	71	
	X60	60	75	

Steels with physical characteristics within the same ranges as above, but with different labels or denominations, are used in the manufacture of penstocks. The designer should consult a metallurgist/fabricator when selecting steels.

2.6 Material Testing Requirements

Following are testing requirements for materials used for pressure retaining parts of exposed steel penstocks, tunnel linings, and attachments welded directly to these pressure retaining parts.

2.6.1 Chemical Analysis

Chemical analysis tests must be performed in accordance with ASME SA Specification⁷ or ASTM A Specification requirements.

2.6.2 Tensile and Bend Tests

Tensile and bend tests must be performed in accordance with ASME SA Specification⁷ or ASTM A Specification requirements.

2.6.3 Impact Tests

Materials other than those excluded in Section 2.3 must be impact tested in accordance with ASTM Standard SA370 or ASTM Standard A370¹ requirements and must meet the criteria specified in Table 2-3. All impact tests must be conducted at the lowest service metal temperature (LST).

Table 2-3 Required Cv Energy Values for Pressure Retaining Material

NOMINAL WALL THICKNESS (IN.)	ENERGY FOR BASE MATERIALS* OR SPECIFIED MINIMUM YIELD STRENGTH (FT-LB)					
	55 ksi or less		56 ksi to 75 ksi		76 ksi to 105 ksi	
	AVERAGE OF 3	LOWEST 1 OF 3	AVERAGE OF 3	LOWEST 1 OF 3	AVERAGE OF 3	LOWEST 1 OF 3
5/8 OR LESS**	-	-	-	-	-	-
OVER 5/8 TO 1	20	15	25	20	30	25
OVER 1 TO 1-1/2	25	20	30	25	35	30
OVER 1-1/2 TO 2-1/2	35	30	40	35	45	40
OVER 2-1/2	45	40	50	45	55	50

*: Where two base materials with different required energy values are joined, the weld metal impact energy requirements of the procedure qualification tests must conform to those of the base material with the higher value.

** : No test required.

2.6.4 Special Carbon Equivalent Criteria Requirements

For materials with specified yield point stresses over certain values, the following special requirements apply to the carbon equivalent (CE) criteria.

- (1) For specified yield point stress over 55 ksi, CE . 45 (ASTM A20, S20.2)
- (2) For specified yield point stress over 75 ksi, CE . 53 (ASTM A20, S20.2)

2.6.5 Additional Tests

Additional testing of material may be required, such as short bead, SR brittleness, deep notch, and double tension tests. Also, welding procedure tests of full-size models may be necessary. Consult the manufacturer for additional testing requirements.

References*

1. Standard Test Methods and Definitions for Mechanical Testing of Steel Products. ASTM A370. ASTM, Philadelphia, PA.
2. Standard Specification for General Requirements for Rolled Steel Plates, Shapes, Sheet Piling and Bars for Structural Use. ASTM A6. ASTM, Philadelphia, PA.
3. Standard Specification for General Requirements for Steel Plates and Pressure Vessels. ASTM A20. ASTM, Philadelphia, PA.
4. Rules for Construction of Pressure Vessels. ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. ASME, New York, NY.
5. Standard Specifications for Forgings, Carbon Steel, for Piping Components. ASTM A105. ASTM, Philadelphia, PA.
6. Standard Specifications for Piping Fittings of Wrought Carbon Steel and Alloy Steel. ASTM A234. ASTM, Philadelphia, PA.
7. ASME SA Specification. ASME Boiler and Pressure Vessel Code, Section 11. ASME, New York, NY.

The following references are not cited in the text.

Lankford, W.T., Jr., Samways, N.L., Craven, R.F., and McGannon, H.E. *The Making, Shaping and Treating of Steel*. Association of Iron and Steel Engineers, Pittsburgh, PA (10th Ed. 1985).

"Carbon and Alloy Steels." *Metals Handbook*. American Society for Metals, Metals Park, OH (1985).

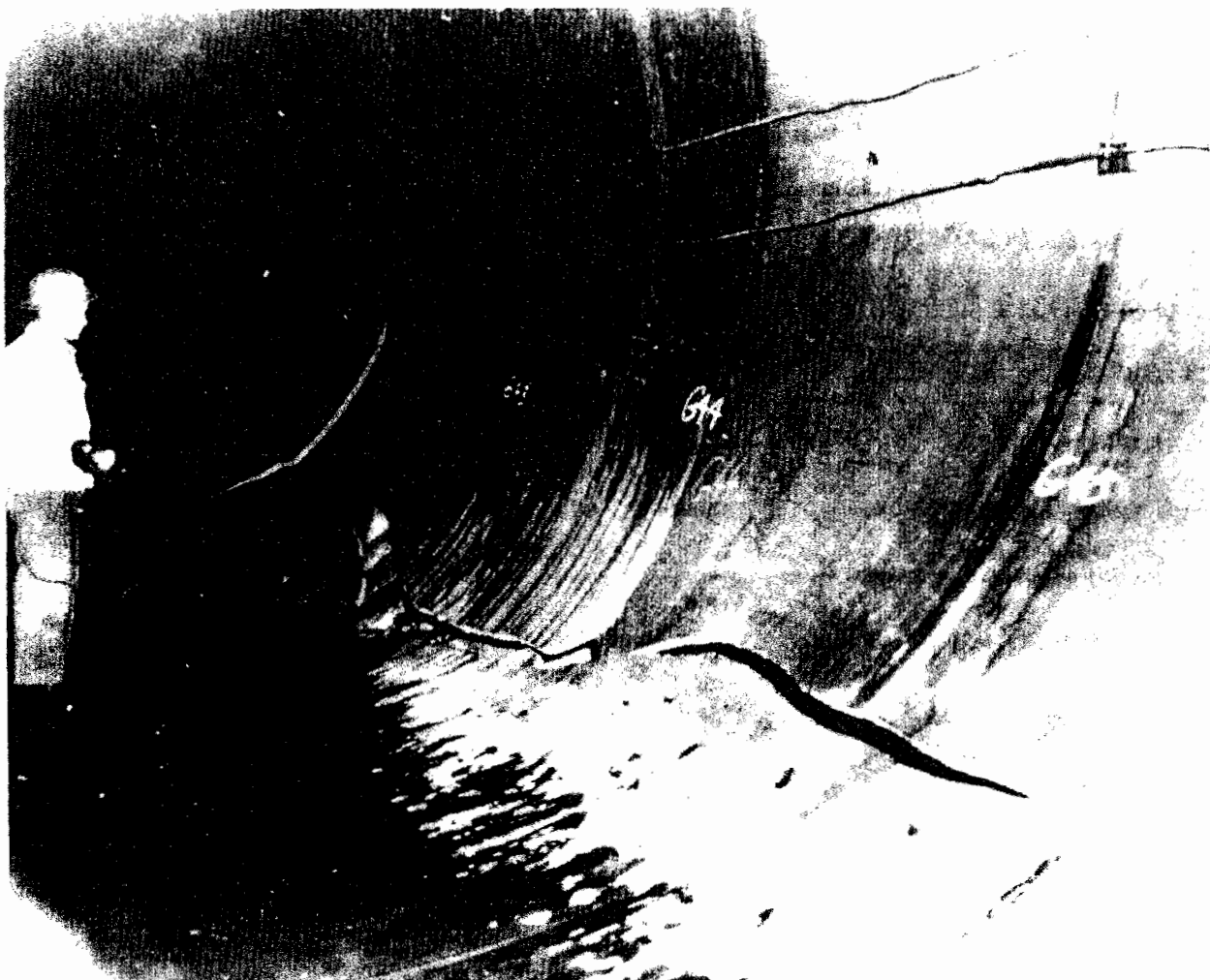
Brockenbrough, R.L. and Johnston, B.G. *USS Steel Design Manual*. U. S. Steel Corporation, Pittsburgh, PA (1981).

Barsom, J.M. and Rolfe, S.T. *Fracture and Fatigue Control in Structures—Applications of Fracture Mechanics*. Prentice-Hall, Inc., Englewood Cliffs, NJ (2nd Ed. 1987).

The Variation of Charpy V-notch Impact Properties in Plates. American Iron and Steel Institute, Washington, DC (1989).

Barsom, J.M. "Effect of Temperature and Rate of Loading on the Fracture Behavior of Various Steels." *Dynamic Fracture Toughness*. The Welding Institute, Abington, Cambridge (1976).

* The most current version of a standard, code, or specification should be used for reference.



3.1 General Considerations

3.1.1 Certified Design Criteria

Penstocks must be designed in accordance with the requirements of the Certified Design Criteria, which must be prepared by a qualified, experienced, and registered professional engineer who is knowledgeable in penstock design. The Certified Design Criteria gives the penstock designer the following design input information:

- (1) All internal and external pressure profiles
- (2) All dead loads

3.2.1 Construction Loads

P_{C1} Loads during shipping, handling, storing, and erecting

P_{C2} Uplift and external pressure from wet concrete

P_{C3} Grouting pressure.

3.2.2 Live Loads

LL1 Wind

LL2 Snow and/or ice

LL3 Vehicle loads

EQ1 Design Basis Criteria (DBC) earthquake

EQ2 Dam Safety Criteria (DSC) earthquake.

3.2.3 Dead Loads

DL1 Weight of structure and permanent accessories

DL2 Weight of water when full or partially full

DL3 Weight of soil or rock fill or soil surcharge

DL4 External hydrostatic pressure including flotation

DL5 Rock loads in tunnels.

3.2.4 Intermittent Loads

DL2 Filling and draining of penstock.

3.2.5 Service Loads—Internal and External Pressures

In addition to the above-defined loads, the penstock and tunnel liner must be designed for the following internal and external pressure service loads when applicable.

3.2.5.1 Internal Pressure Loads

P_{N1} : The maximum static head without surge or waterhammer based on water at the highest reservoir level.

3 Design Criteria

P_{N2} : The maximum static head minus the head loss plus the waterhammer and surge for a plant operation load rejection (TSV closure, wicket gate, PRV needle valve, etc.) when all units are operating with normal governor closure time.

P_{N3} : The minimum static head minus the waterhammer and downsurge pressures that would occur when all units operate from speed no load (SNL) to full load acceptance for all units.

P_{I1} : The penstock full of water but at zero surcharge pressure.

P_{I2} : The penstock half full of water.

P_{EM1} : The waterhammer and surge, calculated for final part gate closure to zero gate position at the maximum governor rate in 2L/a seconds. This simulates the condition of the governor cushioning stroke being inoperative.

P_{EX1} : The internal pressure value, which includes full gate closure with malfunctioning control equipment in the most adverse manner.

P_{T1} : The hydrostatic test condition internal pressure.

3.2.5.2 External Pressure Loads

P_{I3} : The penstock dewatered and subjected to maximum ground water seepage pressure, or for exposed penstocks, the maximum vacuum that can be generated by dewatering.

3.2.5.3 Construction External Pressure Loads

P_{C1} : The equivalent external pressure due to wet concrete consisting of the depth of the pour times the equivalent density of the concrete.

P_{C2} : The equivalent external pressure due to the grouting of the space between the concrete and the steel tunnel liner.

3.2.5.4 Temperature Loads

TL1: The loads due to the expansion and contraction from daily and seasonal temperature variations including construction and hydrotest cases.

TL2: The temperature gradient across the penstock diameter.

3.2.5.5 Sliding and Friction Loads

EJL: The longitudinal loads due to the friction at sliding expansion joints/mechanical couplings.

SFL: The longitudinal loads due to friction or sliding at the supports.

3.2.5.6 Differential Settlement of Adjacent Supports

DS1: The loads generated by the settlement of a single support.

3.2.6 Analysis of Loadings

The above-described loadings are to be applied to the penstock steel shell and support system by treating it as a continuous beam supported by ring girders, saddles, or soil. Analysis should incorporate consideration of the discontinuity of the system at the expansion joints/mechanical coupling. The effects of these loadings must be reflected in the beam analysis as follows.

3.2.6.1 Beam and Support Ring Loadings

The commonly used beam equations must be used for calculating longitudinal stresses due to loads for penstock span and support configuration.

3.2.6.2 Bending Stress Imposed by Ring Girder and Concrete Saddle Supports

When ring girder and concrete saddle supports are used, they impart a local bending stress into the penstock shell, which must be added to the longitudinal and/or circumferential loading.

3.2.6.3 Support Reactions

These loads vary according to the type of support normally encountered in conventional practice. For penstocks with welded field joints the continuous beam method may be used in the analysis. Consideration must be given to the adjusted support reactions for the case in which a single support experiences a differential settlement.

3.3 Service Conditions and Load Combinations

Penstocks must be designed for the load combinations included under each of the conditions in Table 3-1.

3.3.1 Normal Condition

The normal condition loads and load combinations consist of the three normal pressures, dead loads, live loads, thermal loads, and friction loads listed for the three cases shown in Table 3-1. The allowable stresses are given in Section 3.5.3.

3.3.2 Intermittent Condition

The intermittent condition loads and load combinations consist of the three normal pressures, three intermittent pressures, dead loads, live loads, earthquake loads, thermal loads, and friction loads listed for the eleven cases shown in Table 3-1. The allowable stresses for this condition are given in Section 3.5.3.

3 Design Criteria

Table 3-1 Load Combinations for Service Conditions

LOAD	NORMAL			INTERMITTENT											EMERGENCY			EXCEPTIONAL		CONSTRUCTION	HYDROTEST
	1	2	3	1	2	3	4	5	6	7	8	9	10	11	1	2	3	1	2	1	1
P _{N1}	•			•															•		
P _{N2}		•			•	•															
P _{N3}			•					•													
P _{I1}									•	•											
P _{I2}											•	•									
P _{I3}													•	•							
P _{EM1}															•	•	•				
P _{EX1}																		•			
P _{T1}																					•
DL1	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•
DL2	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•		•
DL3	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•
DL4													•	•						•	
DL5	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•
LL1					•		•		•		•		•			•				•	
LL2	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•		
LL3	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•		
EQ1				•		•		•		•		•								•	
EQ2																			•		
DS1															•						
TL1	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•
TL2													•	•							
EJL	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•		•
SFL	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•	•		•
P _{C1}																				•	
P _{C2}																				•	
P _{C3}																				•	

3.3.3 Emergency Condition

The emergency condition loads and load combinations consist of the emergency pressure, dead loads, live loads, differential settlement loads, thermal loads, and friction loads listed for the three cases shown in Table 3-1. The allowable stresses are given in Section 3.5.3.

3.3.4 Exceptional Condition

The loads and load combinations for the exceptional condition consist of one normal pressure, one exceptional pressure, dead loads, live loads, earthquake loads and friction loads as listed for the two cases shown in Table 3-1. The allowable stresses are given in Section 3.5.3.

3.3.5 Construction Conditions

Construction conditions include the dead loads, live loads and handling loads described in Section 3.2. This includes penstock fabrication, storage, shipping, handling, and installation. The major load to consider for this condition is the weight of the penstock itself. Precautions must be taken to prevent overstressing the shell due to the penstock supporting its own weight.

In addition, steps must be taken to prevent excessive deflection of the penstock, particularly after the lining and coating have been applied. Internal bracing may be used to prevent shell overstressing and excessive deflection. Padding supports and tie-down straps may be necessary to prevent damage to the lining and coating.

Internal bracing (pipe stulling) may be necessary for buried penstocks under backfill conditions.

Penstocks or tunnel liners to be encased in concrete must be designed with consideration given to the buoyant and concreting loads due to concrete placement.

Loads and load combinations for the construction conditions are shown in Table 3-1. The allowable stresses are given in Section 3.5.3.

3.3.6 Hydrostatic Test Condition

The loads and load combinations for the hydrostatic test condition consist of the one case shown in Table 3-1. The allowable stresses for this condition are given in Section 3.5.3.

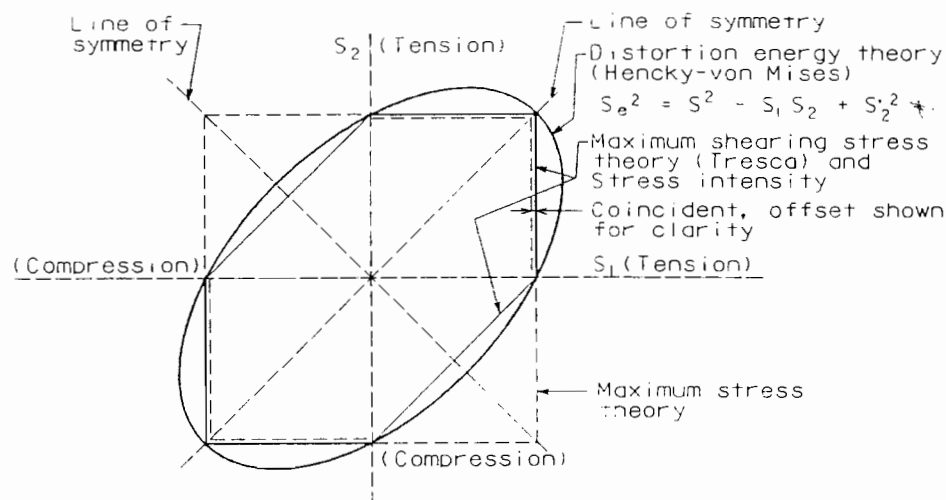
3.4 Stress Categories

3.4.1 Theories of Failure

The stress state at any point in a structure may be completely defined by giving the magnitudes and directions of the three principal stresses. When two or three of these stresses differ from zero, the proximity to yielding may be determined by means of a strength theory. The theories most commonly used are the maximum stress theory, the maximum shear stress theory (also known as the Tresca criterion), and the distortion energy theory (also known as the octahedral shear stress theory or the Huber-Hencky-von Mises criterion). Both the maximum shear stress theory and the distortion energy theory are much better than the maximum stress theory in predicting both yielding and fatigue failure in ductile materials in a biaxial or triaxial stress state. Failure is assumed to occur when the state of stress in the structure equals the same state of stress in the tension test at yield.

This manual recommends use of the maximum shear stress theory, because it is easier to apply and offers advantages in some applications of the fatigue analysis. This manual also recognizes as acceptable the Huber-Hencky-von Mises theory based on the two-dimensional state of stress equation. The various strength theories are compared in Figure 3-1 for a two-dimensional (biaxial) stress state for a constant stress intensity (S_I) or equivalent stress (S_e). S_1 and S_2 represent the principal stresses. Note that in the figure all theories give identical results for certain stress states. Also see Sections 3.4.13, 3.4.14, and referenced work by P. J. Bier.¹

3 Design Criteria



Note:

S_1 and S_2 = Principal stresses; each theory depicts a constant equivalent stress.

Figure 3-1 Comparison of Theories of Failure for Biaxial State of Stress

The maximum shear stress at a point is equal to one-half of the algebraic difference between the largest and the smallest of the three principal orthogonal stresses. Thus, if the principal stresses are S_1 , S_2 , and S_3 and $S_1 > S_2 > S_3$ (algebraically), the maximum shear stress is $\frac{1}{2}(S_1 - S_3)$. The maximum shear stress theory of failure states that yielding in a component occurs when the maximum shear stress reaches a value equal to the maximum shear stress at the yield point in a tensile test. In the tensile test, at yield, $S_1 = S_y$, $S_2 = 0$, and $S_3 = 0$; therefore, the maximum shear stress is $S_y/2$. The theory states that yielding in the component occurs when:

$$\frac{1}{2}(S_1 - S_3) = \frac{1}{2} S_y \quad (\text{Equation 3-1})$$

To avoid the unfamiliar and unnecessary operation of dividing both the calculated and the allowable stresses by two before comparing them, a term called *equivalent intensity of combined stress* or, more briefly, *stress intensity (SI)* has been formulated. Stress intensity is defined as twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal orthogonal stresses.

For a comparison of the Hencky-von Mises theory with the maximum shear stress theory in the two-dimensional state of stress consider the following cases:

- (1) Case 1—Principal stresses of $S_1 = 10$ ksi, $S_2 = 6$ ksi, $S_3 = 0$

Equivalent stress, $S_e = 8.72$ ksi

Stress intensity, $SI = 10$ ksi

- (2) Case 2—Principal stresses of $S_1 = 5$ ksi, $S_2 = -5$ ksi, $S_3 = 0$

Equivalent stress, $S_e = 8.66$ ksi

Stress intensity, $SI = 10$ ksi

- (3) Case 3—Principal stresses of $S_1 = 10$ ksi, $S_2 = 10$ ksi, $S_3 = 0$

Equivalent stress, $S_e = 10$ ksi

Stress intensity, $SI = 10$ ksi

Note that in Cases 1 through 3, the maximum shear stress theory gives results equal to or more conservative than the Hencky-von Mises theory. (However, the maximum stress theory is grossly unconservative for Case 2.)

3.4.2 Stress Intensity Limits

The allowable stress limits are expressed as limits on stress intensity. Stress intensity limits are applicable to the shell of the penstock and to biaxially stressed shell or flat plate elements. They are also applicable to linear elements stressed principally in one direction, along the axis of the element. Examples of linear elements are columns, tension members, beams and rings stressed by loads and moments lying in the plane of the ring. A linear element stressed along only its axis has the characteristic that the uniaxial stress is equal to the stress intensity.

3.4.3 Membrane and Bending Stress Categories

Stresses or stress intensities are categorized as primary, secondary, and peak. Peak stresses are of concern only in relation to fatigue and will not be given any further consideration in this section. However, if a fatigue analysis is considered necessary, refer to the ASME Code,² Section VIII, Division 2, Appendix 5.

The primary category of stresses or stress intensities are general membrane (P_m), local membrane (P_L), and bending (P_b).

The secondary category (Q) includes stresses or stress intensities caused by restraint of the penstock itself and which are generally characterized as being self limiting. The total range of stress intensities, secondary stresses combined with primary stresses and peak stresses, is considered for fatigue. Secondary stress intensities combined with primary stress intensities are limited to twice the yield strength to ensure shakedown to elastic action as discussed in Section 3.4.11.

3.4.4 Shear Stress Category

Shear stress (expressed as an average) in the web of a beam, plate girder, web of a stiffener ring, and shear keys is limited to $0.6S$ where S is the allowable stress intensity. Peak shear stress, such as from torsion acting on a round bar, is limited to $0.8S$.

3.4.5 Weld Joint Reduction Factor (E)

The weld joint reduction factor (E) is imposed on the calculated tension stress. The value of E is 1.0 for joints in compression. Its value depends on the degree of examination the joint receives and the type of joint. See Section 3.5.1 for permissible E values and the example in Section 3.4.14.

3.4.6 Temperature Stress Intensities

Temperature stress intensities in penstocks are due to several sources, including:

- (1) Solar sources, where the structure is restrained with respect to axial growth and/or lateral displacement and temperature gradients develop due to solar effects
- (2) Friction forces that develop from temperature changes at sliding and rolling supports, at sleeve type expansion joints, and from the backfill or encasement of buried penstocks or liners
- (3) Forces that are proportional to axial growth or angular rotation, such as the action in bellows-type expansion joints
- (4) Changes in ambient air or water temperatures.

Temperature stress intensities should be considered in the design of penstocks, supports, liners, and appurtenances when they are significant (stresses greater than 10% of the basic allowable SI).

Temperature stress intensities are generally considered secondary in nature and placed in the Q stress category.

3.4.7 Plasticity Shape Factor (X)

The plasticity shape factor (X) is accounted for in the classification of SI . For solid rectangular sections, such as in bending across the thickness of a flat plate or shell, a factor of 1.5 is permitted. For sections with a shape factor less than 1.5 (such as thin-walled pipe), a shape factor of 1.0 should be used. For sections with a shape factor greater than 1.5, the shape factor used in design should not exceed 1.5.

- (2) Case 2—Principal stresses of $S_1 = 5$ ksi, $S_2 = -5$ ksi, $S_3 = 0$

Equivalent stress, $S_e = 8.66$ ksi

Stress intensity, $SI = 10$ ksi

- (3) Case 3—Principal stresses of $S_1 = 10$ ksi, $S_2 = 10$ ksi, $S_3 = 0$

Equivalent stress, $S_e = 10$ ksi

Stress intensity, $SI = 10$ ksi

Note that in Cases 1 through 3, the maximum shear stress theory gives results equal to or more conservative than the Hencky-von Mises theory. (However, the maximum stress theory is grossly unconservative for Case 2.)

3.4.2 Stress Intensity Limits

The allowable stress limits are expressed as limits on stress intensity. Stress intensity limits are applicable to the shell of the penstock and to biaxially stressed shell or flat plate elements. They are also applicable to linear elements stressed principally in one direction, along the axis of the element. Examples of linear elements are columns, tension members, beams and rings stressed by loads and moments lying in the plane of the ring. A linear element stressed along only its axis has the characteristic that the uniaxial stress is equal to the stress intensity.

3.4.3 Membrane and Bending Stress Categories

Stresses or stress intensities are categorized as primary, secondary, and peak. Peak stresses are of concern only in relation to fatigue and will not be given any further consideration in this section. However, if a fatigue analysis is considered necessary, refer to the ASME Code,² Section VIII, Division 2, Appendix 5.

The primary category of stresses or stress intensities are general membrane (P_m), local membrane (P_L), and bending (P_b).

The secondary category (Q) includes stresses or stress intensities caused by restraint of the penstock itself and which are generally characterized as being self limiting. The total range of stress intensities, secondary stresses combined with primary stresses and peak stresses, is considered for fatigue. Secondary stress intensities combined with primary stress intensities are limited to twice the yield strength to ensure shakedown to elastic action as discussed in Section 3.4.11.

3.4.8 Stress Increase Factor (K)

Allowable stress increase factors (K) are given in Section 3.5.2. The K factor represents the proportional increase in S allowed for loading cases calculated over the basic allowable S .

3.4.9 Compression Stability

Shells, flat plates, and linear structural elements loaded to produce uniform compression stress across the thickness or section must be checked for buckling or instability. Appropriate changes must be made to the design to ensure an adequate factor of safety against buckling. Buckling and instability requirements in the ASME Code,² Section VIII, Division 1 for shells, and the AISC steel building specifications³ for plate and linear structural elements must be met. For stiffener rings on penstocks, refer to Section 4.4. Refer to Section 6 for stiffened or unstiffened liners.

3.4.10 Basic Allowable Stress Intensity (S)

The basic allowable stress intensity (S) should be the lesser of one-third of the minimum specified tensile strength or two-thirds of the minimum specified yield strength.

3.4.11 Stress Categories—Definitions and Limits

3.4.11.1 General Primary Membrane Stress (P_m)

General primary membrane stress (P_m) is the stress (averaged across the thickness) necessary to satisfy the simple laws of equilibrium. The value of P_m is limited to KS , where K is the stress increase factor and S is the allowable stress intensity. In penstock shells, an example is hoop stress from internal pressure.

3.4.11.2 Local Primary Membrane Stress (P_L)

Local primary membrane stress (P_L) is discussed at 4-112(i), Section VIII, Division 2 of the ASME Code,² (the ASME Code uses S_m for KS):

Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a primary and/or a discontinuity effect would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure. Conservatism requires that such a stress be classified as a local primary membrane stress even though it has some characteristics of a secondary stress. A stressed region may be considered as local if the distance over which the stress intensity exceeds $1.1 S_m$ does not extend in the meridional direction more than $1.0 \sqrt{Rt}$, where R is the midsurface radius of curvature measured normal to the surface from the axis of rotation and t is the minimum thickness in the region considered. Regions of local primary membrane stress which exceed $1.1 S_m$ shall not be closer in the meridional direction than $2.5 \sqrt{Rt}$, where R is defined as $(R_1 + R_2)/2$, and t is defined as $(t_1 + t_2)/2$, where t_1 and t_2 are the minimum thicknesses at each of the regions considered, and R_1 and R_2 are the midsurface radii of curvature measured normal to the surface from the axis of rotation at these regions where

the membrane stress exceeds $1.1 S_m$. Discrete regions of local primary membrane stress, such as those resulting from concentrated loads acting on brackets, where the membrane stress exceeds $1.1 S_m$ shall be spaced so that there is no overlapping of the areas in which the membrane stress exceeds $1.1 S_m$.

An example of a local primary membrane stress is the membrane stress in a shell produced by external loads acting on a permanent support or a nozzle connection. Another example is the membrane stress intensity at the junction of a cylindrical to conical section.

P_L must not exceed $1.5 KS$.

3.4.11.3 Primary Bending Stress (P_b) in Shells and Plates

The primary bending stress (P_b) as used in this manual applies to shells and plates in which the bending is across the thickness of the shell or plate, and is primary in the sense that no redistribution of load occurs as a result of yielding. An example is bending SI in the central portion of a flat head or cover due to pressure loading. P_b must not exceed $1.5 KS$. The sum of P_L and P_b likewise must not exceed $1.5 KS$.

3.4.11.4 Secondary Stress (Q)

A secondary stress (Q) is a normal stress, either bending or axial, or a shear stress developed by the constraint of adjacent parts or by self-constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions which caused the stress to occur. Failure from one application of the stress is not to be expected.

Examples of secondary stress include: (1) thermal stresses, generally, (2) bending stress intensities across the thickness of a shell or plate at a gross structural discontinuity, and (3) rim bending stresses at ring girder supports.

There is no need to limit Q type stress intensities acting alone. The combination of Q with primary stress intensities (P_L , P_m , P_b), however, should be limited to $3S$. The $3S$ limit ensures shakedown to elastic action after a few repetitions of the loads producing them and is a limit on the SI range. K factors (other than 1.0) should not be applied to the $3S$ limit.

3.4.12 Representative SI Classifications

Table 3-2 lists the SI classifications for parts typically found in penstocks and tunnel liners.

Table 3-2 SI Classifications for Typical Parts Found in Penstocks and Tunnel Liners

CASE	SI CATEGORY
RING GIRDER SUPPORTS AND STIFFENER RINGS	
RIM BENDING	Q
LOCAL MEMBRANE HOOP STRESS IN THE PENSTOCK IN THE CIRCUMFERENTIAL DIRECTION	P_L
BENDING STRESSES IN RING GIRDER AND/OR STIFFENERS ACTING ACROSS THE DEPTH OF THE GIRDER	P_m
BENDING + AXIAL SI IN RING	P_m
RIM BENDING IN THE PENSTOCK SHELL PLUS DIRECT/AXIAL LOAD	$(Q + P_m)$
SADDLES (PENSTOCK SHELL IMMEDIATELY ABOVE HORN OF SADDLE)	
HORN BENDING	Q
HORN HOOP MEMBRANE LOAD	P_L
SHEAR IN SHELL ADJACENT TO HORN	0.6S
BEARING OR COMPRESSION FROM BEARING LOADS	P_m
BENDING IN BASE PLATES AND SADDLE PLATES ACROSS THICKNESS OF ELEMENT	P_b
LOADED ATTACHMENTS AND NOZZLES	
MEMBRANE SI IN SHELL OF PENSTOCK DUE TO THE LOADS	P_L
BENDING SI IN SHELL OF PENSTOCK DUE TO THE LOADS	Q
NOZZLE NECKS	
NOZZLE NECKS OUTSIDE LIMITS OF REINFORCEMENT FROM EXTERNAL LOADS AND MOMENTS, AND PRESSURE	P_L
NOZZLE NECKS WITHIN LIMITS OF REINFORCEMENT FROM EXTERNAL LOADS AND MOMENTS, AND PRESSURE	P_m
BIFURCATIONS	
MEMBRANE STRESS INTENSITY AWAY FROM REINFORCING GIRDERS AND AWAY FROM CHANGES IN DIRECTION	P_m
MEMBRANE STRESS INTENSITY ADJACENT TO REINFORCING GIRDERS AND CHANGES IN DIRECTION	P_L
BENDING STRESS ACROSS THICKNESS OF SHELL	Q
MAXIMUM STRESS INTENSITY IN REINFORCING GIRDERS	$P_b + P_L$
MISCELLANEOUS	
MEMBRANE HOOP STRESS FROM INTERNAL PRESSURE IN CYLINDRICAL AND CONICAL PENSTOCK SECTIONS AND FOR FORMED SHELLS AWAY FROM DISCONTINUITIES	P_m
MEMBRANE SI IN THE VICINITY OF ATTACHMENTS UNDER LOAD, HORN AREA OF SADDLE SUPPORTS, CONICAL-TO-CYLINDRICAL INTERSECTIONS, AND PENSTOCK SHELLS ADJACENT TO NOZZLES	P_L
BENDING SI IN FLAT HEADS AND COVERS	P_b
TEMPERATURE SI	Q

3.4.13 Equivalent Stresses

For a three-dimensional state of stress in which the three orthogonal principal stresses are S_1 , S_2 , and S_3 , the equivalent stress (S_e) based on the distortion energy theory (the octahedral shearing stress theory) is given by:

$$S_e = \left\{ \frac{1}{2} [(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_3 - S_1)^2] \right\}^{1/2} \quad (\text{Equation 3-2})$$

For a two-dimensional state of stress, the equation reduces to:

$$S_e = \left\{ S_1^2 - S_1 S_2 + S_2^2 \right\}^{1/2} \quad (\text{Equation 3-3})$$

The equivalent stress (S_e) based on the distortion energy theory is equivalent to stress intensity (SI) based on the maximum shear stress theory.

In applying the distortion energy theory it is suggested that the two-dimensional equation be applied using the two principal stresses that result in the greatest algebraic difference.

3.4.14 Equivalent Stress Calculations (Example)

Assume the case of a 15-foot-diameter exposed penstock supported by ring girders spaced at 60 feet in which it is necessary to determine stress intensity and equivalent stress at the top of the penstock at mid-span. It is assumed that expansion joints in the line keep longitudinal pressure stresses in the penstock at negligible values.

The weld joint reduction factor in tension is 0.85 (see Section 3.5.1), and in compression is 1.0.

The following stresses are given:

Pressure: 100 psi

Circumferential membrane stress: 15 ksi (tension)

Longitudinal membrane stress: 10 ksi (compression).

Note that internal pressure produces compression stress in the penstock shell in the radial direction. This compression ranges from -100 psi at the inner surface to zero at the outside and is assumed to vary linearly for points between the inner and outer penstock surfaces.

The weld joint reduction factors are applied to the calculated stress.

The principal stresses are:

$$S_1 = 15 \text{ ksi} / 0.85 = 17.65 \text{ ksi}$$

$$S_2 = -10 \text{ ksi} / 1.0 = -10.0 \text{ ksi}$$

$$S_3 = -0.5 \text{ ksi}$$

The stress intensity (SI) is calculated as:

$$SI = 17.65 \text{ ksi} - (-10.0 \text{ ksi}) = 27.65 \text{ ksi}$$

The equivalent stress (S_e) is calculated using the two-dimensional equation (Equation 3-3) as:

$$S_e = \left[17.65^2 - 17.65(-10) + (-10)^2 \right]^{1/2} = 24.25 \text{ ksi}$$

The equivalent stress (S_e) is calculated using the three-dimensional equation (Equation 3-2):

$$S_e = \left\{ \frac{1}{2} [(17.65 - (-10))^2 + ((-10.0) - (-.5))^2 + (-.5 - 17.65)^2] \right\}^{1/2} = 24.33 \text{ ksi}$$

Note that the stress intensity method is the simplest and most conservative of the three methods.

3.5 Allowable Stress Intensities

The allowable stress intensities must be adjusted by the following weld joint reduction factors and the allowable stress increase factors.

3.5.1 Weld Joint Reduction Factor (E)

The weld joint reduction factors given in Table 3-3 must be used for butt joints.

Table 3-3 Weld Joint Reduction Factors
Exposed Penstocks and Tunnel Liners

CATEGORY	TYPE	100% RT or UT	SPOT RT ¹	NO RT OR UT ²
LONGITUDINAL BUTT WELDS	DOUBLE-WELDED BUTT JOINTS	1.00	.85	.70
CIRCUMFERENTIAL BUTT WELDS	SINGLE-WELDED BUTT JOINTS WITH BACKING STRIPS	.90	.80	.65
SPIRAL (HELICAL) BUTT WELDS	SINGLE-WELDED BUTT JOINT WITHOUT BACKING STRIPS (CIRCUMFERENTIAL ONLY, <5/8 in. THICK AND <24 in. O.D. ⁴)	NA ³	NA	.60
DOUBLE FULL-FILLET LAP	LONGITUDINAL $t_{max} = 3/8$ in.	NA	NA	.55
	CIRCUMFERENTIAL $t_{max} = 5/8$ in.	NA	NA	.55
SINGLE FULL-FILLET LAP JOINTS WITH PLUG WELDS	CIRCUMFERENTIAL ONLY, FOR <24 in. O.D. $t_{max} = 1/2$ in.	NA	NA	.50

Buried Penstocks

CATEGORY	TYPE	100% RT OR UT	SPOT RT ¹	NO RT OR UT ²
LONGITUDINAL BUTT WELDS	DOUBLE-WELDED BUTT JOINTS	1.00	.85	.70
CIRCUMFERENTIAL BUTT WELDS	SINGLE-WELDED BUTT JOINTS WITH BACKING STRIPS	.90	.80	.65
SPIRAL (HELICAL) BUTT WELDS	SINGLE-WELDED BUTT JOINTS WITHOUT BACKING STRIPS (CIRCUMFERENTIAL ONLY, <5/8 in. THICK AND <24 in. O.D.)	NA	NA	.60
DOUBLE FULL-FILLET LAP	LONGITUDINAL $t_{max} = 3/8$ in.	NA	NA	.55
	CIRCUMFERENTIAL $t_{max} = 5/8$ in.	NA	NA	.55
SINGLE FULL-FILLET LAP JOINTS WITH PLUG WELDS	CIRCUMFERENTIAL ONLY, FOR < 24 in. O.D. $t_{max} = 5/8$ in.	NA	NA	.50
BELL-AND-SPIGOT JOINTS	SINGLE FILLET LIMITED TO $t_{max} = 5/8$ in.	NA	NA	.45
	DOUBLE FILLET $t_{max} = 5/8$ in.	NA	NA	.55

See 4.6, page 100

Bifurcations

TYPE	100% RT OR UT	SPOT RT ¹	NO RT OR UT ²
LONGITUDINAL AND CIRCUMFERENTIAL BUTT JOINTS IN SHELL PLATES (DOUBLE-WELDED BUTT JOINTS)	1.00	.85	.70
SICKLE PLATE SPLICES ⁵ (DOUBLE-WELDED BUTT JOINTS)	1.00	.85	.70
SHELL PLATE TO SICKLE TEE WELDS	1.00	NA	NA

1: Spot RT is defined in the ASME Code, Section VIII, Division 1, Paragraph UW-52.

2: A minimum of 20% MT is required for these joints.

3: NA = Not Allowed

4: O.D. = Outside Diameter

5: Includes butt welds in webs and flanges of C-clamps

3.5.2 Allowable Stress Increase Factor (K)

Table 3-4 specifies the allowable stress increase factors.

Table 3-4 Allowable Stress Increase Factors

LOADING CASE	K FACTOR
NORMAL OPERATING CONDITIONS	1.00
INTERMITTENT CONDITIONS	1.33
EMERGENCY CONDITIONS	1.50
EXCEPTIONAL CONDITIONS	2.50
CONSTRUCTION AND HYDROTEST CONDITIONS	1.33

3.5.3 Allowable Stress Intensities

The basic allowable stress intensity (S) is the smaller of 1/3 tensile or 2/3 yield strength as specified in Section 3.4. Table 3-5 gives allowable stress intensities.

Table 3-5 Allowable Stress Intensities

K FACTOR OR SI CATEGORY	NORMAL CONDITIONS	INTERMITTENT CONDITIONS	EMERGENCY CONDITIONS	EXCEPTIONAL CONDITIONS	CONSTRUCTION AND HYDROSTATIC TEST CONDITIONS
INCREASE FACTOR, K	1.0	1.33	1.5	2.5	1.33
GENERAL PRIMARY MEMBRANE, P_m	1.0S	1.33S	1.5S	2.5S (SEE NOTE 2)	1.33S (SEE NOTE 3)
LOCAL PRIMARY MEMBRANE, P_L	1.5S	2.0S	2.25S	2.5S	2.0S
PRIMARY BENDING, P_b	1.5S	2.0S	2.25S	2.5S	2.0S
LOCAL PRIMARY MEMBRANE PLUS PRIMARY BENDING ($P_L + P_b$)	1.5S	2.0S	2.25S	2.5S	2.0S
SECONDARY ($P_L + P_b + Q$)	3.0S	3.0S	(SEE NOTE 1)	(SEE NOTE 1)	(SEE NOTE 1)

NOTES:

- (1) Secondary stress intensities need not be calculated except for emergency conditions when fatigue evaluation is required.
- (2) Not to exceed 0.9 tensile strength.
- (3) Not to exceed 0.5 tensile strength.

3.5.4 Allowable Shear Stresses (All Loading Conditions)

For the design of fillet welds and partial penetration welds, the following formula gives the allowable shear stress for the weld metal:

$$0.3 K \times (\text{tensile strength of welding material}) \quad (\text{Equation 3-4})$$

Where

K = stress increase factor (Table 3-4)

The allowable shear stress for the base metal is given by:

$$0.6 SK \quad (\text{Equation 3-5})$$

The allowable peak shear stress for the base metal is given by:

$$0.8 SK \quad (\text{Equation 3-6})$$

3.5.5 Buckling Factors of Safety

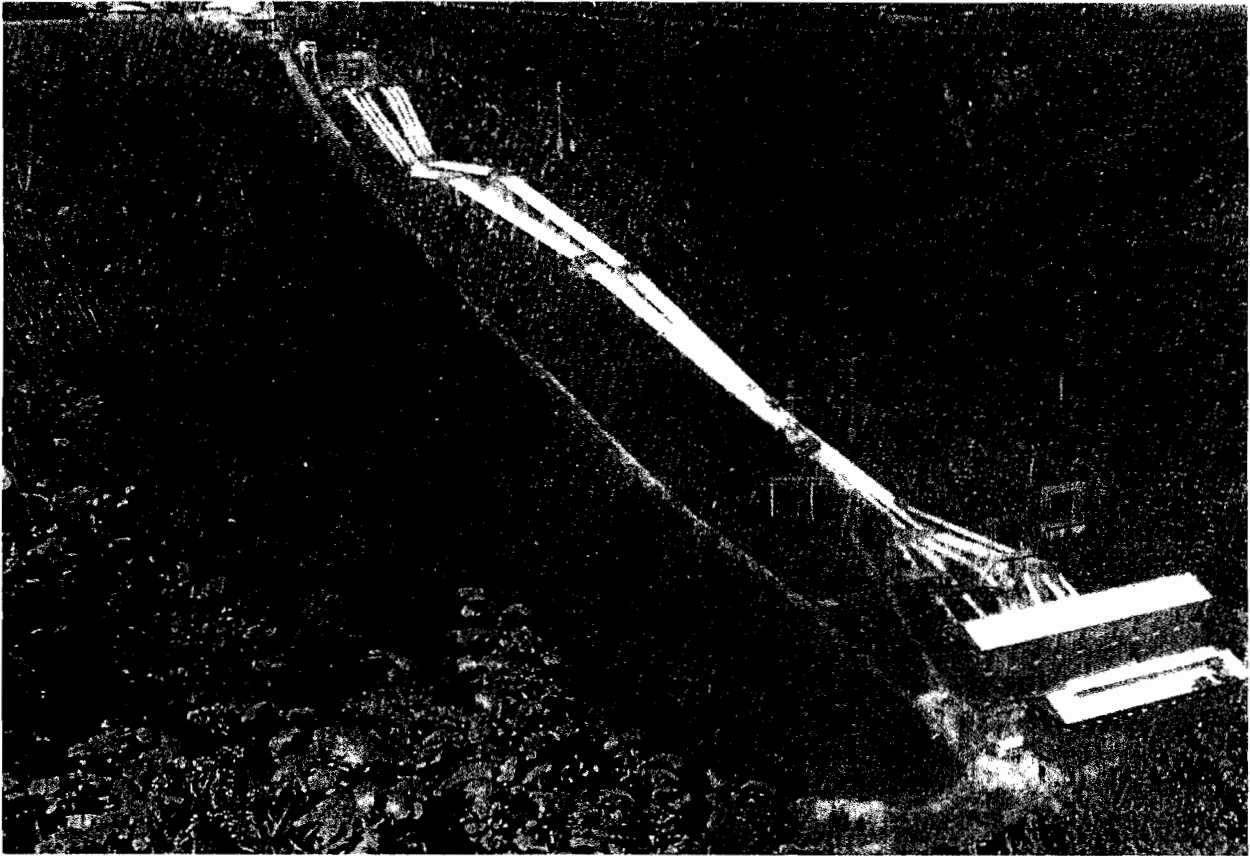
The buckling factor of safety for a dewatered and empty tunnel liner is 1.5.

The buckling factor of safety for an exposed penstock is 2.4 (based on a theoretical factor of safety of 3 and a knockdown factor of 0.8).

References*

1. Bier, P.J. "Welded Steel Penstocks—Design and Construction." Engineering Monograph No. 3. U.S. Bureau of Reclamation, Denver, CO (1966).
2. Rules for Construction of Pressure Vessels, Sec. VIII. ASME Boiler and Pressure Vessel Code. American Society of Mechanical Engineers, New York, NY.
3. Specification for Structural Steel Buildings—Allowable Stress Design and Plastic Design, June 1, 1989. American Institute of Steel Construction Inc., Chicago, IL.

* The most current version of a standard, code, or specification should be used for reference.



This section discusses the parameters to be considered in the design and analysis of exposed penstocks.

4.1 Penstock Shell Design and Analysis

The principal factors that govern the required shell thickness are:

- (1) Thickness required for shipment and handling
- (2) Thickness required to resist the imposed loads, considering the appropriate allowable stresses.

Additional factors in determining shell thickness include:

- (1) Acceptance criteria for mill and fabrication tolerances
- (2) Criteria for corrosion allowance, if elected in lieu of coating and lining.

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4.1.1 Minimum Shell Thickness

The minimum thickness of the penstock shell should be the larger of the following thickness values:

- (1) Thickness required to ensure adequate shipment and handling, as required by this manual.
- (2) Thickness required to resist the imposed loads, given certain allowable stress conditions and considering that the shell thickness is to be specified within recommended mill and fabrication tolerances. The possible allowance for corrosion must be considered as an addition to the required thickness for design.
- (3) In addition to minimum shell thickness for handling, the shell thickness, support types, and support spacing must be selected so that the maximum deflection of the pipe filled with water, acting as a beam between the supports, does not exceed 1/360 of the span.
- (4) Similarly, to avoid pipe buckling due to full internal vacuum, the D/t ratio should be not less than 158. If the ratio is less than 158, stiffeners may be required.

4.1.2 Determining Shell Thickness and Stresses

Although the predominant stress-causing load on a penstock section (unrestrained and free from internal or external appurtenances) is the stress from internal pressure, it is important to recognize that other stress conditions can exist and should be considered in determining the required shell thickness. These additional stresses can result from beam action, differential temperatures, and longitudinal stress due to end closures free to move in a longitudinal direction. The stress (not hoop stress) caused directly by the internal pressure may be considered negligible. As such, and in the majority of cases, the stress analyses for determining the shell thickness ends as a biaxial state of stress and is resolved by application of either the Hencky-von Mises theory or the stress intensity approach discussed in Section 3.4.

The following formulas can be used to determine shell thickness and stresses imposed by certain loading conditions.

4.1.2.1 Minimum Thickness for Shipping and Handling

The minimum thickness (t_{min}) of the penstock shell for shipping and handling can be calculated using the formulas of Pacific Gas and Electric Company (PG&E) or the Bureau of Reclamation. This manual recommends use of the larger of the minimum thickness values calculated by the formulas.

- (1) PG&E formula

$$t_{min} = \frac{D}{288} \quad \text{(Equation 4-1)}$$

(2) Bureau of Reclamation formula

$$t_{\min} = \frac{D + 20}{400} \quad (\text{Equation 4-2})$$

Where:

- $t_{\min}$ = minimum thickness, in.
- D = nominal penstock diameter, in.

The larger of the minimum thickness values calculated by the above formulas must be checked to determine if the shell can adequately support itself at a point load, as if it were resting on a flat surface and loaded by its own weight. The maximum stress ($S_{\max}$) under this loading condition is given by:

$$S_{\max} = \frac{9R^2W}{t} \quad (\text{Equation 4-3})$$

Where:

- t = shell thickness, in.
- W = unit weight of the shell material, lb/in.³
- R = radius of the middle surface of the pipe shell, in.

A smaller minimum thickness value, more in line with design thickness, is acceptable provided that bracing, special supports, or external stiffeners are used to accommodate handling, construction, or other conditions.

If corrosion allowance is considered in the design, the minimum thickness for shipping and handling must include the corrosion allowance.

4.1.2.2 Hoop Stress Due to Internal Pressure

To determine hoop stress due to internal pressure, either of the following formulas may be used:

$$t = \frac{Pr}{SE} \quad (\text{Equation 4-4})$$

$$S_H = \frac{Pr}{tE} \quad (\text{Equation 4-5})$$

Where:

- P = internal pressure (at centerline of pipe), psi
- t = penstock shell thickness required to resist the design pressure P , in.
- S = basic allowable stress intensity for design load condition resulting in pressure P , psi

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- S_H = hoop stress due to internal pressure, psi
 r = inside penstock radius, in.
 E = weld joint reduction factor, in decimal percentage value

Circumferential bending moments in a penstock shell occur whenever the penstock is partly filled. Shell design for a partly filled penstock has been treated in several articles, none of which gives a complete analysis. Good treatments of this complex analysis are given by R. J. Roark¹ and H. Schorer.² Calculations using the formulas presented by Roark must be done with great precision because the expression for the bending moment represents the algebraic sum of large numbers of nearly equal terms. Circumferential bending moments for a completely filled penstock (zero pressure) exist only for a penstock supported on saddles. Depending on the saddle-to-shell configuration, formulas developed by L. P. Zick³ can be used.

The shearing stresses developed in a transverse penstock section are due to the external loads, including weight of the shell and water. When the shell is held to a cylindrical shape by a stiffener ring, for example, the developed shear is tangent to the shell at all points and varies from zero at the top to zero at the bottom, with a maximum at mid-depth twice the average value over the entire section. When the shell is free to deform, as with a saddle support, the tangential shear stresses act on a reduced effective cross section, and the maximum stress occurs at the horn of the saddle. There is further increase in this shear stress because a portion of the shell above the saddle is noneffective, tending to increase the shear in the effective portion.

4.1.2.3 Longitudinal Stresses

Longitudinal stresses are imposed on a penstock shell from several loading conditions. These stresses are generally categorized by the following action conditions: (1) beam action, (2) stiffener ring restraint at rim, (3) buckling due to axial compression, (4) radial strain (Poisson's effect), and (5) temperature-related effects.

(1) Beam action

When a penstock rests on its supports, it acts as a beam. The beam load consists of the weight of the pipe, the contained water, and any external live loads such as ice, snow, wind, or earthquake. If the penstock is to function as a beam, deformation of the shell at the supports must be limited by the use of properly designed stiffener rings, ring girder supports, or saddle supports. On the assumption that large shell deformations can be prevented, the beam stresses can be computed using the theory of flexure. For a cylinder section, the resulting longitudinal stress intensity (S_L) is given by:

$$S_L = \frac{M_B}{\pi r^2 t} \quad (\text{Equation 4-6})$$

Where:

- S_L = resulting longitudinal stress intensity, psi
 M_B = bending moment due to beam action, in.-lb

- r = inside penstock radius, in.
 t = nominal shell thickness, in.

(2) Stiffener ring restraint

Because of restraint imposed on the shell by a rigid ring girder or stiffener ring, secondary longitudinal bending stresses are developed in the pipe shell adjacent to the ring girder or stiffener ring. Although this is a local stress, which decreases rapidly with an increase in distance away from the ring, it should be considered in designing the plate for longitudinal stresses. These secondary stresses are flexural stresses caused by the bending deformation of the shell near the stiffener because the shell at the stiffener cannot widen radically in the same manner as the more distant shell portion. The maximum longitudinal bending stress (S_{LR}) is given by:

$$S_{LR} = (1.82) \left(\frac{A_r - Ct}{A_r + 1.56 t \sqrt{rt}} \right) \left(\frac{Pr}{t} \right) \quad (\text{Equation 4-7})$$

Where:

- S_{LR} = resulting longitudinal bending stress intensity due to ring girder action, psi
 A_r = area of girder ring(s), plus shell area under and between rings, plus shell area to a distance on either side of the ring equal to $0.78 \sqrt{rt}$, in a plane along the axis of the shell, in.²
 C = length of penstock shell measured between the out-to-out of girder rings, in.
 t = nominal shell thickness, in.
 r = inside penstock radius, in.
 P = value of internal pressure at centerline of shell, psi

To show the effect of the weight of water in the penstock, the weight of the penstock shell, and the joint reduction factor on the rim bending stresses, the factor Pr/t in the above equation should be replaced by the hoop stress calculated for the shell. The recomputed S_{LR} should be used in determining shell thickness.

The longitudinal bending stress (S_{LRX}) in an axial direction, x distance from the ring girder edge, can be found by:

$$S_{LRX} = S_{LR} \left(e^{-x/z} \right) \left(-\sqrt{2} \right) \cos \left(\frac{x}{2} + \frac{\pi}{4} \right) \quad (\text{Equation 4-8})$$

Where:

- S_{LR} = resulting longitudinal bending stress intensity due to ring girder action, psi
 x = distance from the outside face of the girder ring at which point the ring restraining bending stress is being calculated, in.
 z is a constant for the cylindrical shell determined by:

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$$z = \frac{\sqrt{rt}}{\sqrt[4]{3(1-\mu^2)}}$$

(for steel with $\mu = 0.303$, $z = 0.78 \sqrt{rt}$)

In current practice, if the maximum longitudinal restraining stresses are excessive, the shell thickness is increased on each side of the stiffener ring for a minimum length of $L_R = 2.33 \sqrt{rt}$.

These secondary, longitudinal restraining stresses in the shell should be added algebraically to the longitudinal flexural stresses, if any, and the resultant used in shell thickness computations.

The above formula for stresses due to ring girder action are for a penstock under pressure. A good treatment for calculating stress conditions with a half-full penstock is given by H. Schorer.² Because it may control the design, this loading condition should be reviewed in combination with other applicable loads.

Secondary, longitudinal restraining stresses for an unstiffened penstock supported on saddles are assumed to not exist. This assumption is based on the fact that the shell is not restrained in any way and can rotate in a vertical direction. An excellent treatment of shell stresses for penstocks supported on saddles is given by L. P. Zick.³ Secondary stresses in a penstock shell near edges or corners of concrete anchors are not readily calculated. Some indication of the magnitude of these stresses may be realized by the use of finite element analyses. However, these stresses can be reduced by covering the penstock at these points, prior to concrete placement, with a tapering plastic material such as tar paper or cork sheeting.

(3) Buckling due to axial compression

Stresses caused by direct buckling or wrinkling failure of a thin shell occur whenever the shell is subjected to axial compression. Axial compression may be caused by bending action, temperature expansion in a longitudinally restrained penstock, by forces developed due to resistance against sliding, and by the compressive force developed by the weight of an inclined penstock with bottom anchorage. A treatment for determining the allowable, average buckling stress in a thin shell, considering the effects of imperfections due to fabrication, is given by L. H. Donnell and C. C. Wan.⁴ Their analysis shows that the safe compressive stress that can be imposed on a steel cylinder shell without failure by wrinkling is one-twelfth the theoretical critical stress. Similarly, experimental tests show that the safe compressive stress that can be carried without buckling failure by wrinkling is given by:

$$S_{allow} = 1.5(10^6) \left(\frac{t}{r} \right) \leq \frac{1}{3} \text{ yield point (for general buckling)} \quad (\text{Equation 4-9})$$

$$S_{allow} = 1.8(10^6) \left(\frac{t}{r} \right) \leq \frac{1}{2} \text{ yield point (for local buckling)} \quad (\text{Equation 4-10})$$

Where:

- S_{allow} = allowable compressive stress, psi
 t = nominal shell thickness, in.
 r = inside penstock radius, in.

A more refined treatment of penstock buckling under pressure is given by E.H. Baker, L. Kovalevsky, and F.L. Rish.⁵

(4) Longitudinal strain/stress

Radial expansion due to internal pressure on an axially restrained shell will cause longitudinal contraction (Poisson's effect) with a corresponding longitudinal tensile stress equal to:

$$S_{LP} = \mu S_H \quad (\text{Equation 4-11})$$

Where:

- S_{LP} = longitudinal stress from Poisson's effect, psi
 μ = Poisson's ratio (0.303 for steel)
 S_H = hoop stress due to internal pressure, psi

This stress occurs only if the pipe is axially restrained. This stress should be combined algebraically with other longitudinal stresses calculated for the condition causing the Poisson strain/stress effect.

(5) Temperature stresses

The conditions under which thermal stresses occur can be distinguished in two ways:

- (a) The temperature and shell conditions are such that there would be no stresses due to temperature except for the constraint from external forces and/or restraints. In this case, the stresses are calculated by determining the shape and dimensions the shell body would take if unrestrained and then finding the forces required to bring it back to its restrained shape and dimension. Having determined these "restoring" forces, the stresses in the shell are calculated using applicable formulas.
- (b) The form of the body and thermal conditions are such that stresses are produced in the absence of external constraints, solely because of the incompatibility of the natural expansions and contractions of the different parts of the body.

Where the position of ring girder footings, valve flanges, and other external attachments have to be fixed, thermal stresses and resulting displacements are important when the penstock is empty and when setting the penstock in place. Temperature differentials should be determined for the installation during both its construction and its operation. In addition, the position of the sun should be considered when determining thermal stresses

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and displacements, and the resultant stresses and displacements should be resolved into components for use in design.

If a thin-walled shell of a given length with both ends fixed is subjected to an outside temperature (T) on one side and an outside temperature ($T + \Delta T$) on the opposite side, and the temperature gradient between the two sides is assumed to be linear, then the fixed-end moments (M_T) that develop at the end of the shell are given by:

$$M_T = \frac{E_Y I \Delta T \alpha}{D} \quad (\text{Equation 4-12})$$

and the maximum resulting bending stress (S_{TB}) is given by:

$$S_{TB} = \frac{MC}{I} = \left(\frac{E_Y I \Delta T \alpha}{D} \right) \frac{D}{2I} = \frac{E_Y \Delta T \alpha}{2} \quad (\text{Equation 4-13})$$

Where:

- M_T = fixed-end moments
- S_{TB} = maximum resulting bending stress
- E_Y = Young's modulus for steel, lb/in.²
- ΔT = differential temperature or change in temperature, °F
- D = nominal penstock diameter, in.
- α = temperature coefficient of expansion, in./in./°F
- M = any moment
- C = distance to the most extreme fiber of shell subject to bending, in.
- I = moment of inertia of a section, in.⁴

If the thermal conditions described above are applied to a shell with both ends free, the shell normally would curve in the plane of the sun's rays at an arc of a circle having an inside radius (R_I) of:

$$R_I = \frac{D}{\Delta T \alpha} \quad (\text{Equation 4-14})$$

Where:

- R_I = radius of curvature of shell at its centerline, in.
- D = nominal penstock diameter, in.
- ΔT = differential temperature or change in temperature, °F
- α = temperature coefficient of expansion, in./in./°F

The deflection (Y) at any point on the shell parallel to the plane of the sun's rays, as determined by a line drawn perpendicular to the original axis of the shell from the point to the longitudinal tangent drawn from the fixed end of the shell, is given by:

$$Y = \frac{\Delta T \alpha L^2}{2D} \quad (\text{Equation 4-15})$$

Where:

- Y = centerline deflection at any point of the shell, with one end fixed, due to differential temperature, in.
- ΔT = differential temperature or change in temperature, °F
- L = length, in.
- D = nominal shell diameter, in.

If the supports are permitted to rotate in a horizontal plane but about a vertical axis, then the angle of rotation (ϕ) in radians is:

$$\phi = \frac{\Delta T \alpha L}{2D} \quad (\text{Equation 4-16})$$

Additional thermal stresses develop in a long, thin shell because of differential temperature conditions between the inside and outside surfaces of the shell. Assuming that the temperature gradient across the shell thickness is linear and that the outside temperature is higher, then the maximum circumferential stress at points remote from the ends of the shell is:

$$S_{CT} = \frac{\Delta T \alpha E_Y}{2(1-\mu)} \quad (\text{Equation 4-17})$$

Where:

- S_{CT} = circumferential shell stress intensity due to differential temperature between inside and outside of shell, psi
- μ = Poisson's ratio (0.303 for steel)

Under the same conditions, the maximum longitudinal stress (in psi) is:

$$S_{LT} = \frac{\Delta T \alpha E_Y}{2} \quad (\text{Equation 4-18})$$

Both of the above stresses are compression on the outside and tension on the inside of the shell. At the ends of the shell, if free, the maximum tensile stress is about 25% greater than the value given by the above equations.

If no restraint is imposed at the ends of a penstock, the load imposed due to temperature will be the resistance to sliding between shell and support and the resistance to sliding between shell and connecting joint (expansion or coupling). This latter resistance may be taken as 500 pounds per diameter inch.

4.1.3 Loading Combinations

Given the above potentially active stress conditions, the designer should take into consideration probable combinations of loadings that may result in higher principal stresses. The stresses considered under normal conditions are:

- (1) Between supports
 - (a) Longitudinal stresses due to beam bending
 - (b) Longitudinal stresses due to longitudinal movement under temperature changes and internal pressure
 - (c) Circumferential (hoop) stress due to internal pressure
 - (d) Equivalent stress based on the Hencky-von Mises theory of failure or the stress intensities approach.
- (2) At supports
 - (a) Circumferential stresses in the supporting ring girder or saddle due to bending, and direct stresses and tensile stress due to internal pressure
 - (b) Longitudinal stress in the shell at the supports due to beam action and longitudinal movement from temperature changes and internal pressure
 - (c) Bending stresses in the shell imposed by the support (ring girder or saddle)
 - (d) Equivalent stress based on the Hencky-von Mises theory of failure or the stress intensities approach.

Also, it may be important to consider the shear stress, as this may occur at or near supports and throughout the shell structure. It is important to keep in mind the secondary tension and compression stresses that can occur at an element of the shell and how the resulting Hencky-von Mises stress or stress intensities may govern design when these secondary stresses are combined with the primary hoop tension and beam bending longitudinal stresses.

4.1.4 Tolerances

Penstock material plate or prefabricated pipe should be ordered with a thickness equal to or greater than the minimum thickness for handling or the thickness calculated for design. No additional adjustment needs to be made to the shell thickness if the specified mill tolerance will

provide a plate thickness not less than the smaller of 0.01 in. or 6% of the *nominal* thickness. If greater tolerances are allowed, the design plate thickness should be increased to account for the undertolerance. Tolerances for steel plates and/or shapes are given by ASTM A20 and ASTM A6.

Also, for an acceptable design thickness of the shell plate without any upward adjustment, it is important to apply minimum/maximum acceptance tolerances for shop and field weldment alignments at weld joints. The requirements of the ASME Code, Section VIII, Division 1 should be specified.

4.2 Ring Girders

Ring girders normally are used to support long-span exposed steel penstocks. The purpose of the ring girder is to support the exposed penstock, its contents, and all live and dead loads as defined in Section 3.2. Also, ring girders stiffen the penstock shell and maintain the pipe section's roundness, thus allowing the penstock to be self-supporting, acting as either a simple or continuous beam when the penstock is supported by more than two supports. Ring girders allow the penstock to span relatively long distances (100 feet or more) compared to saddle-type supports (40 feet or less). Figure 4-1 shows a typical ring girder supporting a large-diameter penstock. See Section 3.4 for further definitions of types of stresses to be considered for ring girders.

4.2.1 Analysis

Detailed ring girder stress analysis includes combining circumferential and longitudinal stresses in the penstock shell at the ring girder junction in accordance with Section 3.4.

Added to the longitudinal beam stresses are longitudinal stress due to pressure on the exposed pipe end at the expansion joint, longitudinal stress due to frictional force at the supports, longitudinal stress due to frictional force at expansion joints or sleeve-type couplings, longitudinal force due to gravity (if the penstock is sloping), and localized bending stress in the shell due to ring restraint.

The penstock shell away from the supports is also designed for combined longitudinal and circumferential stresses using the same procedure; however, bending stress from localized ring restraint is neglected. For exposed penstocks, it is common to thicken the shell in the vicinity of the ring girder. A detailed analysis method for ring girders has been published by the U.S. Bureau of Reclamation.⁶ Also, an abbreviated version has been published.⁷

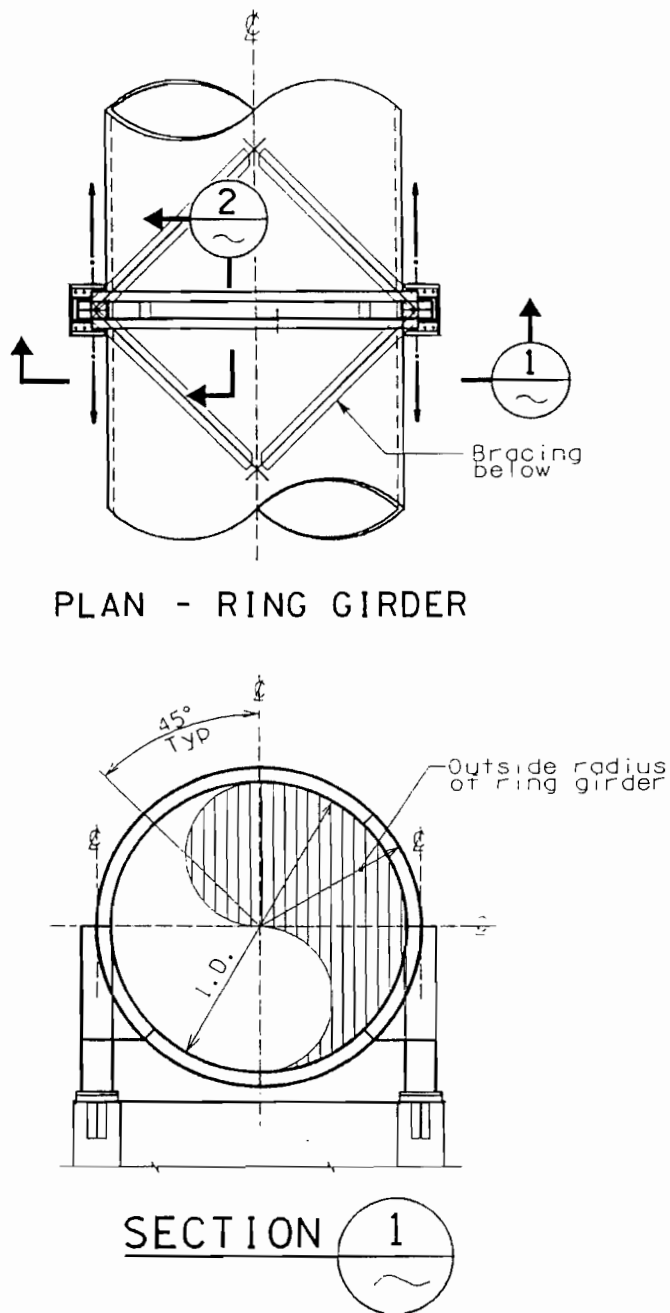


Figure 4-1a Typical Ring Girder Supporting a Large-Diameter Penstock



Figure 4-1b Typical Ring Girder Supporting a Large-Diameter Penstock

4.2.1.1 Vertical Loads

The basic procedure for ring girder design is to first locate the supports and then determine the reaction at the supports, assuming that the penstock acts as a continuous beam. A trial ring geometry is selected and the centerline of the support legs located such that the column centerline is approximately collinear with the centroid of the ring plus shell section. To minimize the ring bending moment, the centerline of the support legs should be located approximately $0.04r$ inches outside the centroid of the ring plus shell section. The shell length (L_l) that is assumed to be effective on either side of the ring is given by:

$$L_f = 0.78\sqrt{rt} \quad (\text{Equation 4-19})$$

Where:

- r = inside radius of the shell, in.
 t = shell thickness, in.

For ring girders that use more than one ring, the maximum shell length (L_2) that is assumed to be effective between the rings must not exceed $1.56\sqrt{rt}$.

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Next, section properties for the ring and the portion of the penstock shell that will participate in composite action with the ring are determined. Then the maximum ring and shell stresses for the vertical loads are calculated by combining the direct stress, bending stress, and pressure stress. Both the inside shell stress and outside ring tip stress then are compared with allowable stresses as defined in Section 3.4. The ring geometry is revised and the above analysis is repeated until stresses are acceptable.

4.2.1.2 Lateral Loads

Ring girders also must support lateral forces due to wind or seismic conditions. Lateral forces should be evaluated according to Sections 1.7 and 1.8. The equivalent static lateral force must be not less than 15% of the vertical load.

Horizontal seismic loading of ring girders produces a maximum bending at the horizontal springline of the penstock (where the support legs are attached to the ring). Vertical load stresses are combined with the seismic load stresses, and the area near the support leg attachments should be investigated to determine the stress magnitude and maximum stress location. The stresses then are compared with the allowable stresses defined in Section 4.2.4 to determine if the ring girder is adequate. If not, the ring geometry is revised and the above analysis is repeated until stresses are acceptable.

Also, stresses must be checked that occur in ring girders when the conduit is half full. These stresses are compared with the material and allowable stresses indicated in Section 3.4. The ring geometry must be revised until the stresses are acceptable.

4.2.2 Rocker Bearings

Rocker bearings provide low resistance to longitudinal forces acting on the penstock support as the result of temperature changes, pressure, and gravity loads on inclined penstocks. Figure 4-2 shows a typical rocker bearing detail.

The pin is designed as a beam with loadings introduced through the rocker and the side brackets. The following AISC formula (J8-2)⁸ establishes the basic rocker dimensions:

$$F_p = \left(\frac{F_y - 13}{20} \right) 0.66d \quad (\text{Equation 4-20})$$

Where:

- F_p = allowable bearing, kips/in.
- d = diameter of rocker, in.
- F_y = specified minimum yield stress of the steel, psi

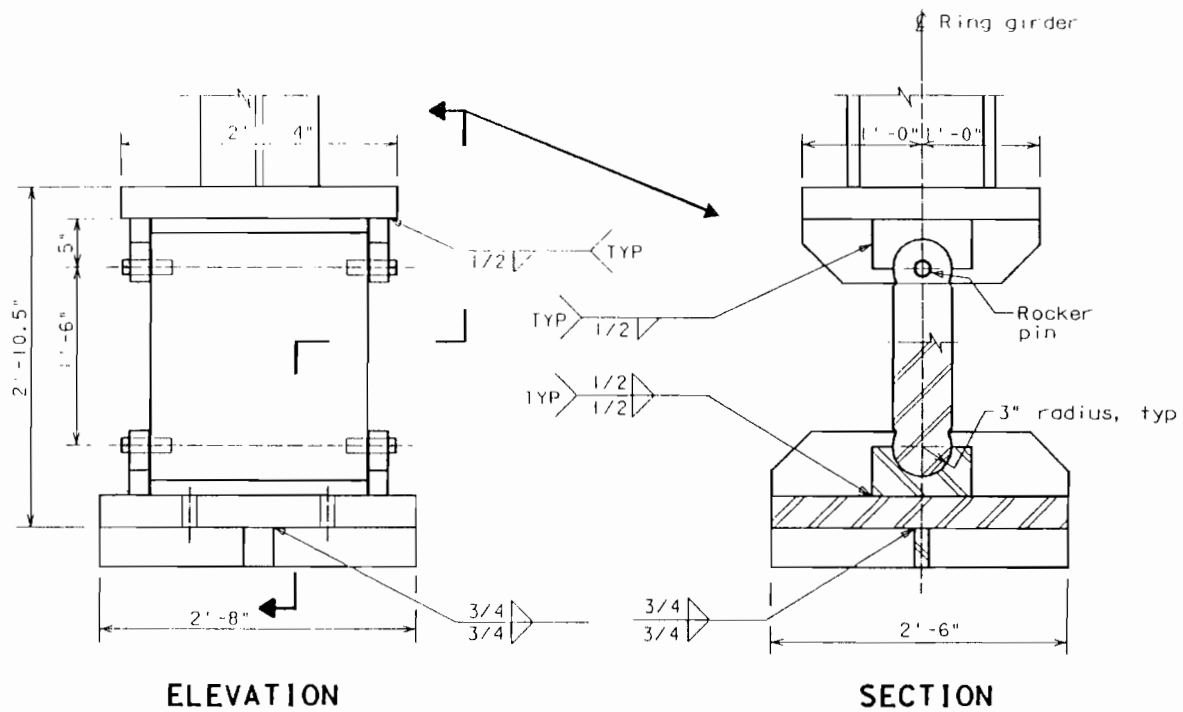


Figure 4-2 Rocker Bearing Detail

Low resistance to friction can also be obtained by using slide bearings made of virgin Teflon[®], high-density polyethylene (HDPE), or other products manufactured for the purpose of supporting heavy loads. These materials can reduce the coefficient of friction to 0.05 or less. Figure 4-3 shows a typical slide bearing detail that uses Teflon[®] bearing material. The upper surface of the bearing should be slightly larger than the lower surface to prevent debris from contaminating the contact surface of the bearing.

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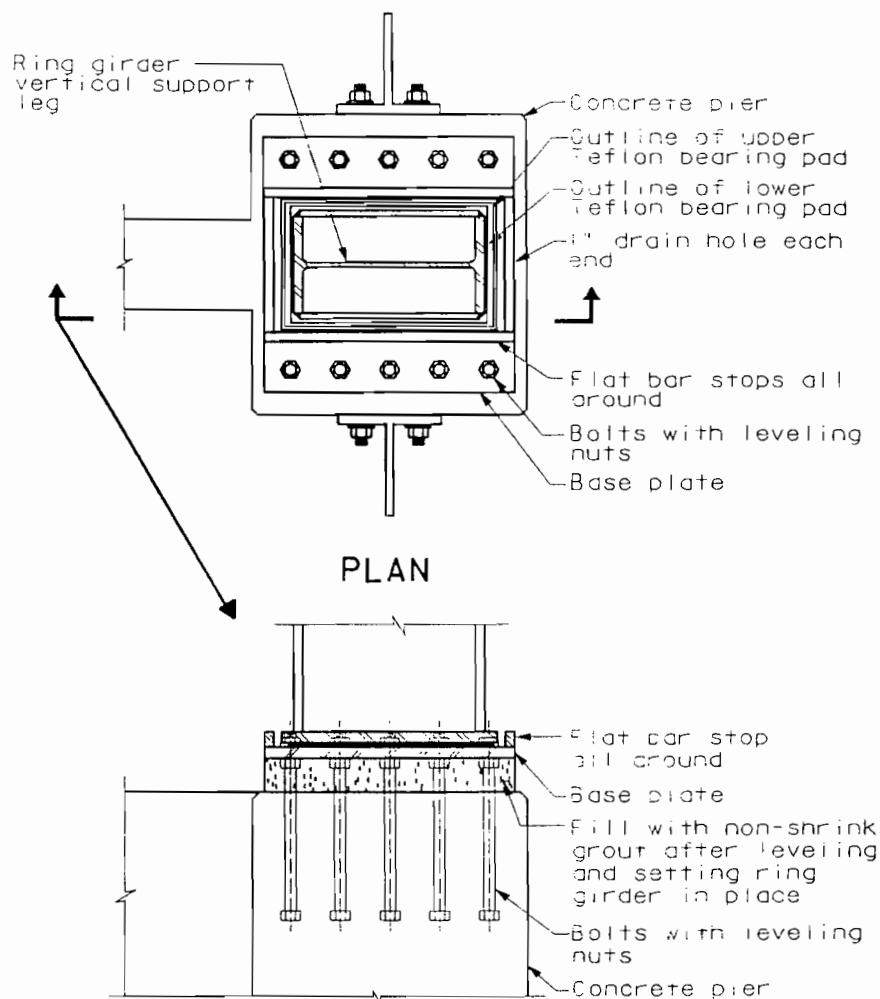


Figure 4-3 Ring Girder Slide Bearing

4.3 Saddles

Saddles are a type of support for exposed penstocks. The support engages less than the full perimeter of the penstock, generally between 90 and 180 degrees of arc, and typically 120 degrees. Saddles are simpler to construct than full-perimeter ring girder supports, but generally are spaced closer together than ring girders. The closer spacing is necessary because saddles do not stiffen the penstock shell against radial deformations to the same extent as ring girder supports.

Saddles, serving the same functions as ring girders, act as supports to carry water and penstock metal loads or as construction supports. They may be of steel or reinforced concrete. Figure 4-4 illustrates several typical saddle configurations.

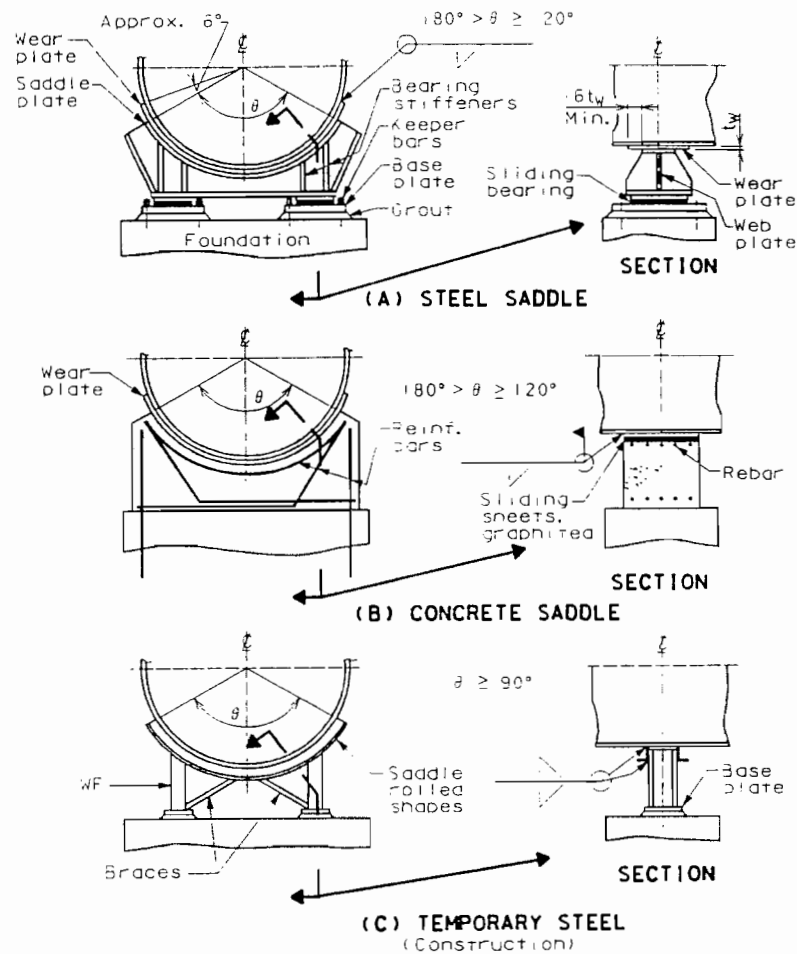


Figure 4-4 Saddle Configurations

4.3.1 Wear Plates

Wear plates sometimes are used between the saddle and penstock shell to stiffen the shell and limit local stresses in the shell immediately around the saddle plate. Wear plates can serve as the interface between the penstock and saddle that allows differential expansion to take place. To be effective in reducing shell stresses, wear plates must extend beyond the saddle plate they bear upon by an amount in the longitudinal direction of about 16 wear plate thicknesses plus the differential growth, and in the circumferential direction by about 6 degrees of arc. Wear plates are attached to the penstock shell by a continuous fillet weld. Corners of the wear plate are cut with a radius to reduce stress concentrations.

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4.3.2 Stiffener Rings

Full-circumference stiffener rings can be used on either side of the saddle to stiffen the penstock shell or can be placed in the same cross section as the saddle itself and made integral with the saddle. Stiffener rings make possible span lengths approaching those permitted by ring girder supports.

4.3.3 Steel Saddles

Steel saddles generally are fabricated from structural grades of steel (A36 is the most common) and are of welded construction. Wear plates and stiffener rings, which stiffen the penstock shell directly over the saddle, generally are welded to the penstock; therefore, these items must be of the same material as the penstock shell or of a compatible material with the same nominal chemical and mechanical strength as the penstock shell material. Also, the items must be heat treated to similar notch toughness as the shell, and have good weldability to the penstock. See Section 2.4 for specific material requirements for saddles, wear plates, and stiffener rings.

4.3.4 Expansion Provisions

Saddles may be designed to act as anchors to resist pressure forces at bends in the penstock and also loads directed along the length of the penstock. The loads acting along the length of the penstock are from friction forces generated at supports as a penstock expands or contracts due to temperature changes or pressure surges, and from the axial component of gravity loads on inclined penstocks.

If not fixed, saddles generally are designed to permit sliding, relative to the penstock, either at the saddle-to-penstock interface or at the base of the saddle. If sliding is at the base of the saddle, keeper bars are required to prevent the penstock from moving in the transverse direction.

4.3.5 Stability

Saddle design requires a stability check to ensure that the saddle will not overturn when acted upon by lateral forces in directions along the penstock axis or transverse to the axis.

Generally, no uplift or point of zero bearing is permitted anywhere on bearing surfaces with a factor of safety against overturning of at least 1.5 for Design Basic Criteria (DBC) earthquake and 1.0 for the Dam Safety Criteria (DSC) earthquake.

Stability considerations require that the minimum bearing on the right support shown in Figure 4-5 satisfies the expression:

$$\frac{W}{e} - \frac{Hh}{d} - L \left(h - R + \frac{b}{2} \right) \left(\frac{1}{c} \right) \geq 0.0 \quad (\text{Equation 4-21})$$

Where:

- W = total reaction normal to centerline, kips
- H = total transverse load, kips

L = total longitudinal load, kips

e = 3 for DBC and 2 for DSC

(The distances R , b , h , c , and d given in Figure 4-5 must be in consistent units of length.)

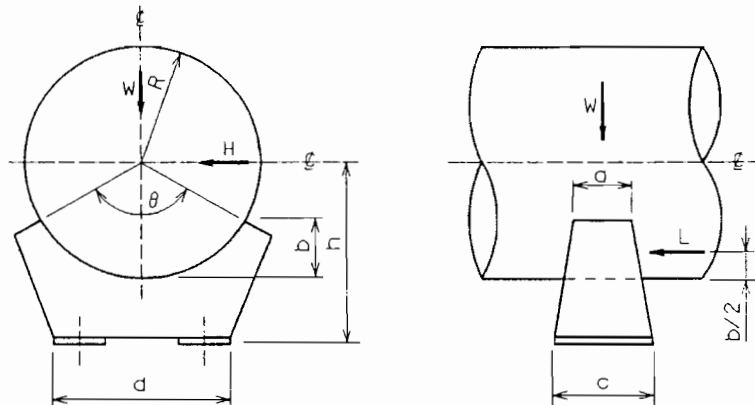


Figure 4-5 Saddle Stability Diagram

Longitudinal load (L) and transverse load (H) often are due to frictional forces caused by penstock expansion, in which case they become proportional to reaction (W), where the proportionality factor is the friction factor. Friction factors for design are given in Reference 7.

For saddles that are not welded to the penstock, stability considerations require that the following expressions be satisfied:

$$\tan^{-1} (H/W) \leq 0.125(\theta) \text{ for DBC} \quad (\text{Equation 4-22})$$

$$\tan^{-1} (H/W) \leq 0.25(\theta) \text{ for DSC} \quad (\text{Equation 4-23})$$

Where:

H = total transverse load, kips

W = total reaction normal to centerline, kips

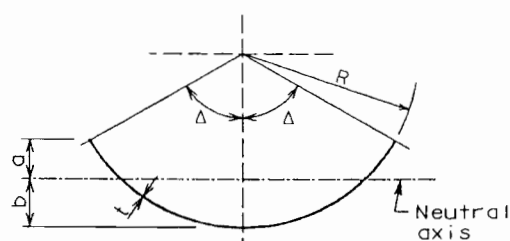
θ = total contact angle between saddle and penstock, degrees

These expressions ensure contact between the penstock and saddle over at least 3/4 of the arc subtended by the full saddle angle (θ) for the Design Basis Criteria (DBC) earthquake, and 1/2 the saddle angle for the Dam Safety Criteria (DSC) earthquake. The expressions assume that longitudinal force (L) is uniformly distributed along the saddle arc and does not affect transverse stability.

4.3.6 Saddle Design

4.3.6.1 Shell Stresses

Figures 4-6 and 4-7 show the shell stresses that must be considered in saddle design.



$$I = \text{Moment of inertia} = \frac{R^3 t (\Delta + \sin \Delta \cos \Delta - 2 \frac{\sin^2 \Delta}{\Delta})}{12}$$

$$Z_a = \text{Section modulus, top} = \frac{I}{a}$$

$$Z_b = \text{Section modulus, bottom} = \frac{I}{b}$$

θ = Saddle contact angle

$$\Delta = \frac{5}{12} \theta + 30^\circ$$

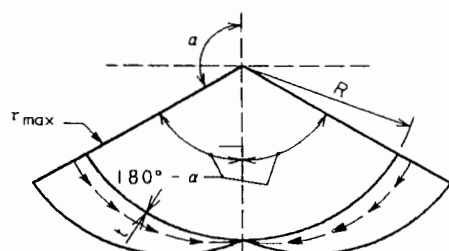
R = Mid-surface radius

t = Thickness

$$a = R \left(\frac{\sin \Delta}{\Delta} - \cos \Delta \right)$$

$$b = R \left(1 - \frac{\sin \Delta}{\Delta} \right)$$

(A) SECTION MODULUS OVER SADDLE (Stiffener rings not used)



θ = Saddle contact angle

$$\alpha = 171^\circ - \frac{19}{40} \theta$$

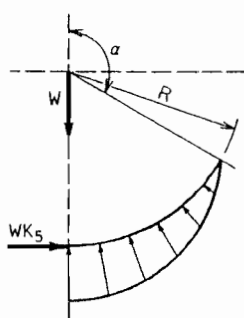
V = Shear at support

$$r_{\max} = \frac{VK_2}{Rt}$$

$$K_2 = \frac{\sin \alpha}{(\pi - \alpha) + \sin \alpha \cos \alpha}$$

(B) TANGENTIAL SHEAR STRESSES AT SADDLE

(Stiffener rings not used)



W = Total reaction, kips

WK_5 = Compression force, kips

$$K_5 = \frac{1 + \cos \alpha}{(\pi - \alpha) + \sin \alpha \cos \alpha}$$

Longitudinal length of shell resisting compression force = $1.5 \sqrt{Rt} + b$ (without wear plate) or = $1.56 \sqrt{Rt} + g$ (with wear plate of width g)

$$\text{Compressive stress} = \frac{WK_5}{1.56 \sqrt{Rt} + b)t}$$

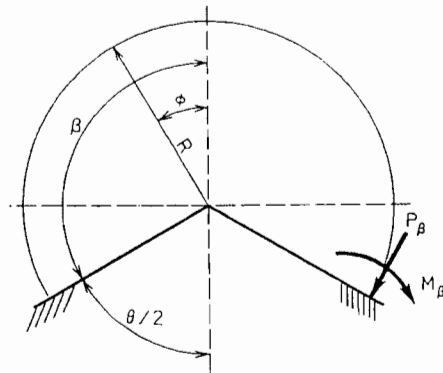
$$\text{or} = \frac{WK_5}{(1.56 \sqrt{Rt} + g) + gt_2}$$

t_2 = thickness of wear plate

(C) CIRCUMFERENTIAL COMPRESSION IN SHELL OVER INVERT OF SADDLE

(Stiffener rings not used)

Figure 4-6 Saddle Design Shell Stresses



FIXED-ENDED ARCH

M_ϕ = Bending moment at ϕ (in.-lbs)
 θ = Saddle contact angle (radians)
 W = Total load/reaction
 M_ϕ maximum at $\phi = \beta$

$$M_\phi = \frac{WR}{\pi} \left[\cos \phi + \frac{\phi}{2} \sin \phi - \frac{3}{2} \left(\frac{\sin \beta}{\beta} \right) \frac{\cos \beta}{2} - \frac{1}{4} (\cos \phi - \frac{\sin \beta}{\beta}) \left[g - \frac{4 - 6 \left(\frac{\sin \beta^2}{\beta} + 2 \cos^2 \beta \right)}{\frac{\sin \beta}{\beta} \cos \beta + 1 - 2 \left(\frac{\sin \beta^2}{\beta} \right)} \right] \right]$$

Thrust load, P_β , at $\phi = \beta$

$$P_\beta = \frac{W}{\pi} \left[\frac{\beta \sin \beta}{2(1 - \cos \beta)} - \cos \beta \right] + \frac{\cos \beta}{R(1 - \cos \beta)} (M_\beta - M_t)$$

Where $M_t = M_\phi$ when $\phi = 0^\circ$

Width of shell resisting P_β is $1.56\sqrt{Rt} + b$
 b = Width of saddle or wear plate, in.
 t = Penstock thickness, in.
 R = Penstock radius, in.

Width of shell resisting M_β equals the lesser of $8R$ or center-to-center span length of saddle supports.

Example:

If span length $l < 8R$, then horn bending = $\frac{6M_\beta}{lt^2}$

and total thrust plus bending = $\frac{6M_\beta}{lt^2} + \frac{P_\beta}{(1.56\sqrt{Rt} + b)t}$

For a wear plate of width g ($g > b$) and thickness t_2 ,

$$\text{total stress} = \frac{P_\beta}{(1.56\sqrt{Rt} + g)t + gt_2} + \frac{6M_\beta}{l(t^2 + t_2^2)}$$

Figure 4-7 Circumferential Stress at the Horn of a Saddle (Without Stiffener Rings)

The Zick approach⁹ for the analysis of shell stresses is an accepted method. The Zick method determines the shell stresses (items 1 to 5) or forces and moments in the saddle or ring stiffeners (items 6 and 7) as follows:

- (1) Beam bending stresses over the saddle support and at midspan

At midspan the full circumferential section modulus ($\pi R^2 t$) is available to resist bending. R is the mid-surface radius, and t is the penstock wall thickness. Both maximum compression and maximum tension must be checked. Maximum compression usually occurs with all loads acting and pressure at its minimum value. The allowable compressive stress must be limited to the allowable specified in the ASME Code, Section VIII, Division 1, UG-23(b), for axial compression in a thin-walled cylinder. Alternatively, rules in API 620¹⁰

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may be used. As specified in API 620, for values of t/R less than 0.00667, the compression stress (in psi) must not exceed $1.8 \times 10^6 (t/R)$. (See also Section 4.1.2.3 (3).)

Over the saddle, the bending stress is assumed to be resisted by an arc of shell less than the full circumference (for saddle supports that do not utilize stiffener rings). The effective shell arc for resisting longitudinal beam bending over the saddle equals twice the angle delta (Δ) (see Figure 4-6(A)). If stiffener rings are used to stiffen the penstock at saddle supports, the full cross section may be used.

(2) Tangential shear stress at the horn of the saddle

The horn of the saddle is the edge defined by the angle θ shown in Figure 4-4.

Beam shears at the saddle are assumed to be resisted by tangential shear stresses lying in the midsurface of the shell and acting over an arc of shell slightly larger than the saddle contact angle. In Figure 4-6(B), the angle α defines the effective arc for resisting shear; the shear stresses vary in magnitude with the sine of the angle measured from the vertical centerline.

(3) Circumferential compression stress directly over the invert of the saddle

See Figure 4-6(C) for the method of checking the circumferential compression stress in the penstock directly over the invert of the saddle.

(4) Circumferential stress at the horn of the saddle

The circumferential stress at the horn of the saddle is a combination of $P_L + Q$ stresses; the stress limit is held to $1.5KS$ by Zick.⁹ Circumferential bending and thrust forces are assumed to have the same values as would result for a fixed-ended circular arch as shown in Figure 4-7. The calculated bending moments are distributed over a fairly long length of shell to give results consistent with strain gage testing. Generally, pressure stresses are not combined with stresses from the calculated bending and thrust stresses because pressurizing the penstock stiffens and rounds it out.

(5) Additional head stress

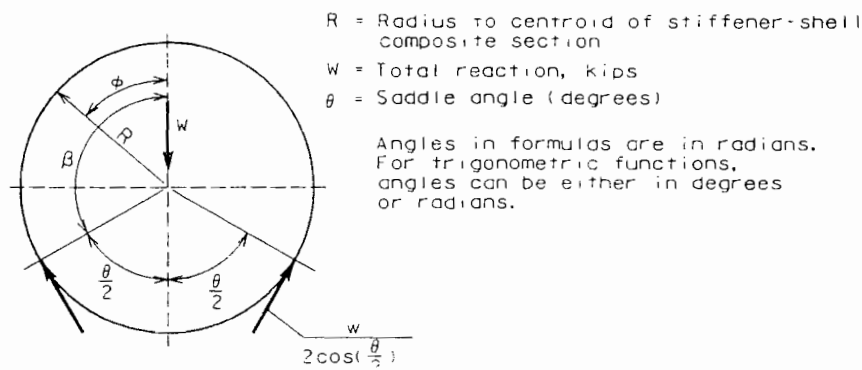
Additional head stress generally is not applicable to long conduits supported on multiple supports, such as a penstock.

(6) Stiffener ring forces and moments

If stiffener rings are used over or adjacent to the saddle, they must be designed for the circumferential bending moments, shears, and thrust loads resulting from the loading. A conservative method for calculating these forces and moments is shown in Figure 4-8 (when stiffener rings are adjacent to the saddle) and Figure 4-7 (when the stiffener ring is in the same plane as the saddle web).

(a) Ring stiffener integral with saddle (lies in same plane as saddle web plate). Use bending moments and thrusts developed in Figure 4-7.

(b) Ring stiffeners adjacent to saddle (stiffeners lie outside plane of saddle web plate). Use bending moments and thrusts as follows:



BENDING MOMENT:

$$M_{\phi} = \left(\frac{WR}{2\pi} \right) \left[\frac{(\pi - \beta)}{\sin \beta} - \phi \sin \phi - \cos \phi \left[\frac{3}{2} + (\pi - \beta) \cot \beta \right] \right]$$

M_{ϕ} maximum at $\phi = \frac{\pi}{2}$

THRUST:

$$P_{\phi} = \frac{W}{\pi} \left[\frac{\phi \sin \phi}{2(1 - \cos \phi)} - \cos \phi \right] + \frac{\cos \phi}{R(1 - \cos \phi)} M_{\phi} - M_t$$

Where $M_t = M_{\phi}$ when $\phi = 0^\circ$

Figure 4-8 Saddle Design - Stiffener Rings Adjacent to Saddle

(7) Saddle forces

A tension force in the saddle is needed to balance forces from the assumed cosine distribution of pressures applied radially to the saddle at the contact between saddle and penstock. Refer to Section 4.3.6.3 and Figure 4-9.

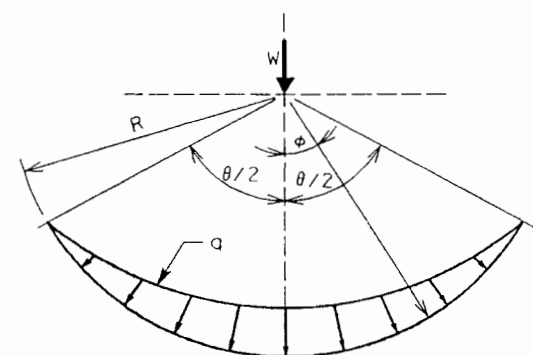
The horn circumferential bending stress plus direct stress generally governs selection of the saddle contact angle and may govern shell thickness. For the circumferential check of stresses, it is assumed the penstock is not under pressure. It is assumed the penstock rounds out when pressure is applied, thereby reducing the bending stress over the horn of the saddle. It is necessary to check only the horn circumferential stress with internal pressure at zero.

4.3.6.2 Analytical Method

The Zick method⁹ is recognized as good practice. However, today's broad use of computers and the availability of software tools for the finite element method of analysis give very good results if applied by experienced designers. These analytical methods are accepted for checking shell stresses near the saddle proper.

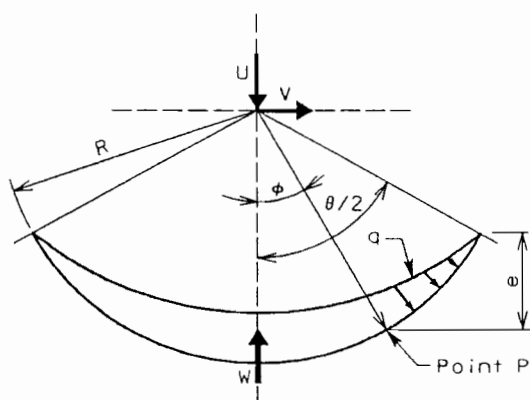
4.3.6.3 Design and Analysis of Saddles

Design of the saddle itself should follow good structural engineering practice. For steel saddles, requirements specified in the AISC "Specification for the Design, Fabrication, and Erection of Structural Steel for Buildings"¹¹ must be met. Figure 4-9 provides formulas for bending moment, shear, and thrust in the design of the saddle proper. Concrete saddles should meet ACI 318 building code¹² requirements for concrete. Load distribution on the saddle as shown in Figure 4-9 is theoretically correct and is used for design.



(A) DISTRIBUTION ON SADDLE
DUE TO LOAD W

A = Area (in.^2) of circular sector subtended by angle $\theta = (R^2/2)(\theta - \sin \theta)$
 γ = Pseudo unit wt. (lbs./in.^2) = (W/A)
 q = Unit loading (lbs/in.) = $R [\cos \phi - \cos(\theta/2)] \gamma$
 R = Radius of saddle.
 Angles in formulas are in radians.
 For trigonometric functions, angles can be either in degrees or radians.



(B) RESULTANT FREE-BODY
FORCES AT POINT P

(Act through center as shown)
 Free-body diagram angle ϕ to angle $\theta/2$

$e = R(\cos \phi - \cos \theta/2)$ in inches
 $A_1 = \frac{R^2}{2} (\cos \phi - \cos \theta/2) (\sin \theta/2 - \sin \phi)$
 $A_2 = \frac{R^2}{2} [(\theta/2 - \phi) - \sin(\theta/2 - \phi)]$
 A_1 and A_2 are areas (in.^2)
 $U = \gamma (A_1 + A_2)$ (lbs.)
 $V = \frac{R^2 \gamma}{2} (\cos \phi - \cos \theta/2)^2$
 U and V are horizontal and vertical load resultants at pt. P.

Resolving forces U and V about point P gives:

Horizontal force, $V = \frac{R^2 \gamma}{2} (\cos \phi - \cos \theta/2)^2$ (lbs.)

Shear, $U = \gamma (A_1 + A_2)$ (lbs.)

Bending moment, $M_\phi = R(V \cos \phi - U \sin \phi)$ (in.-lbs.)

Note: Saddle reaction to balance load W is assumed to be at $\phi = 0^\circ$

Figure 4-9 Forces and Moments for Design of Saddle Proper

4.3.7 Detailing

Various configurations have been used successfully in penstock construction. Details illustrated in Figure 4-4 are typical of those used and should not be construed as the only acceptable details.

The sliding surface at the base of the steel saddle shown in Figure 4-4 (A) also could be located at the saddle plate/wear plate interface. In that case, the sliding bearing at the base of the saddle would not be necessary, and the base of the saddle would be anchored to the foundation with anchor bolts.

The saddle plate of the saddle proper shown for the steel saddle in Figure 4-4 (A) and (B) should be formed at a radius that accounts for the theoretical radius of the penstock when the penstock is under design pressure.

According to Zick,⁹ the saddle width at the interface with the penstock does not control proportioning the design. A minimum saddle width (dimension *a* in Figure 4-5) of 12 inches for steel saddles and 15 inches for concrete saddles is recommended.

4.4 Stiffeners

4.4.1 Circumferential Stiffening

Full circumferential stiffener rings should be provided on exposed penstocks when required to resist external pressure such as vacuum.

4.4.2 Stiffener Spacing

To determine spacing of stiffener rings and required shell thickness, the procedure in UG-28 of the ASME Code, Section VIII, Division 1 must be followed.

4.4.3 Moment of Inertia Requirements

The size of stiffener rings must meet the moment of inertia requirements of UG-29 of the ASME Code.

Alternatively, the moment of inertia (*I*) of intermediate stiffener rings must satisfy the formula⁹:

$$I \geq \frac{pLD_o^3}{77,300,000(N^2-1)} \quad (\text{Equation 4-24})$$

Where:

- I* = moment of inertia of the composite stiffener ring and participating part of the penstock shell (the shell length must not exceed $1.1 (D_o t)^{1/2}$), in.⁴
- p* = external pressure, psi

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- L = stiffener spacing, in.
 D_o = outside diameter, in.
 N = number of complete waves into which a circular ring will buckle
$$N^2 = \frac{0.663}{\left(\frac{H}{D_o}\right) \left(\frac{1}{D_o}\right)^{1/2}} \leq 100$$

 H = distance between ring girder supports or end stiffeners, in.
 t = penstock shell thickness, in.

4.4.4 Welding

Welds attaching stiffener rings to the penstock must be in accordance with UG-30 of the ASME Code, Section VIII, Division 1, which gives acceptable sections and structural shapes for use as stiffener rings and methods of attachment.

Stiffener ring splices should be full-fusion butt joints designed to develop the full section of the stiffener ring.

4.4.5 Tolerances on Roundness

The roundness tolerances specified in UG-80 of the ASME Code must be met.

4.4.6 Line-of-Support

Line-of-support for purposes of determining stiffener spacing may be a cone-cylinder junction, provided the moment-of-inertia of the junction meets the rules for stiffener rings specified in UG-29 of the ASME Code. Ring girder supports are often adequate to serve also as stiffeners to resist external pressure but must meet Section 4.4.3 moment-of-inertia requirements.

4.4.7 Factor of Safety

Although the ASME Code rules are based on a theoretical factor of safety of 3 for external pressure, the tolerances were established to limit the buckling pressure to not less than 80% of that for a perfectly circular vessel. Implicitly, the true factor of safety is $3 \times 0.8 = 2.4$. This factor of safety is not considered overly conservative for exposed penstocks constructed to normal fabrication tolerances. A factor of safety of 3 is used in the moment of inertia equation for stiffener rings in Section 4.4.3.

4.4.8 Attachment of Stiffener Rings

Fillet-welded attachment of stiffener rings is permitted. The maximum size of fillet welds used to attach stiffener rings to a penstock made of heat-treated material, such as ASTM A517, which is quenched and tempered, should be limited to 3/8 inch and the welds must be continuous. Stitch welding is not recommended for any material. With the exception of stitch welding, the rules specified in UG-30 of the ASME Code are recommended.

4.5 Bends

Changes in direction of flow are accomplished with curved pipe sections commonly called bends. Bends up to 24 inches in diameter may be smooth, wrought, or steel fittings, as specified in Section 2.4 or fabricated from mitered sections of pipe. Bends greater than 24 inches in diameter commonly are fabricated from mitered sections of pipe. The mitered pipe sections must be joined by full-penetration, single- or double-welded butt joints.

4.5.1 Fabrication

The radius of bends must be equal to or greater than 1 pipe diameter but need not be greater than 3 pipe diameters. Special situations, such as a bend immediately upstream of a turbine or a free discharge valve, may warrant larger radius bends. Bends may be fabricated by mitering segments of a cone to produce a reducing bend or by mitering straight pipe segments to produce a constant diameter bend. An analytical stress investigation as recommended in Section 4.5.3 should be undertaken. The maximum deflection angle of the mitered segment must not exceed 22.5 degrees. Other methods that consider discontinuity stresses may be used.

4.5.2 Compound Bends

Compound bends are required where it is desired to change flow direction in both plan view and profile. Trigonometric calculations are necessary to determine the true angle of the bend and end rotations.⁷

4.5.3 Stress Analysis

Bends with a radius less than 2.5 pipe diameters must be designed with consideration given to concentration of hoop tension stresses along the inside edge of the bend. The following formula¹³ is used to check stresses:

$$t = \frac{PD}{Sf} \left(\frac{D}{3} \tan \frac{\theta}{2} + \frac{S}{2} \right) \quad (\text{Equation 4-25})$$

Where:

- t = required elbow wall thickness, in.
- P = design pressure, psi
- f = allowable tensile stress at design pressure, psi
- D = outside diameter of elbow, in.
- S = segment length along inside of elbow, in.
- θ = segment deflection angle, degrees

Stresses calculated according to this formula must not exceed the limits specified in Section 3 for P_m . To satisfy the formula, short radius bends may require thicker plate than adjacent sections of straight pipe.

4.6 Welded Joints

4.6.1 Types and Configurations

Typical welded joint configurations for longitudinal and circumferential main welded joints are shown in Figure 4-10.

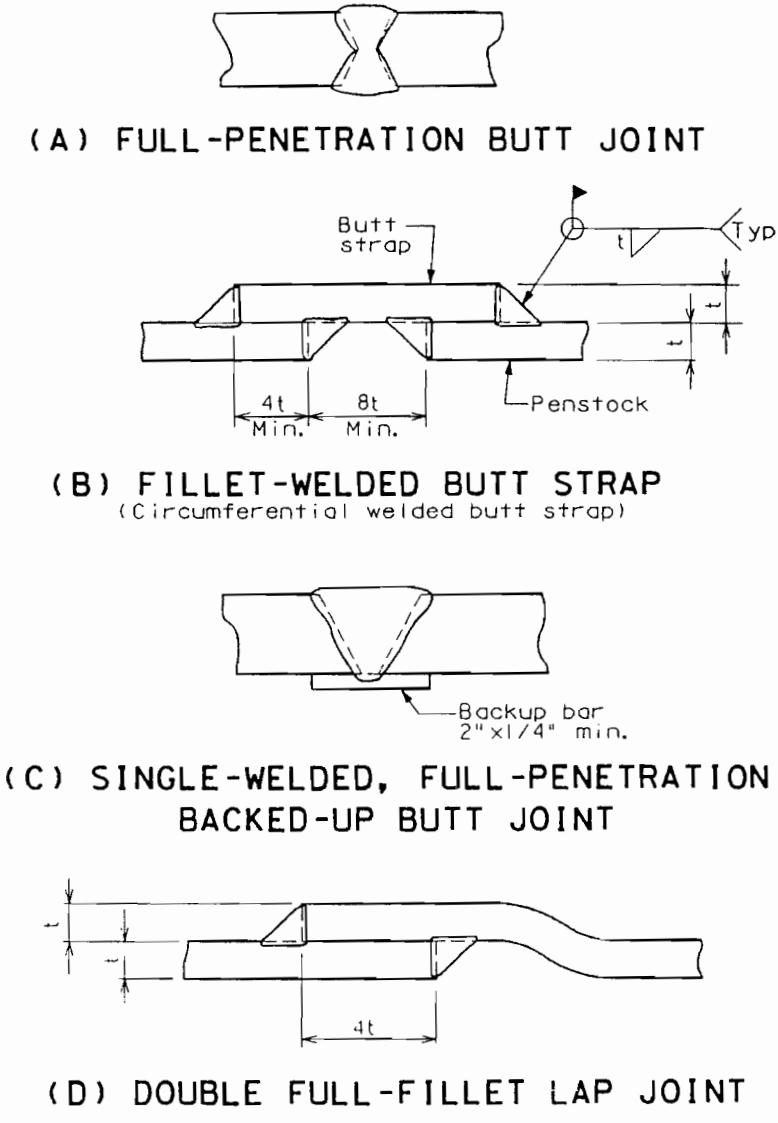


Figure 4-10 Welded Joint Configurations (I)

Configurations for nozzle and manway penetrations, attachments, and corner joints are shown in Figure 4-11.

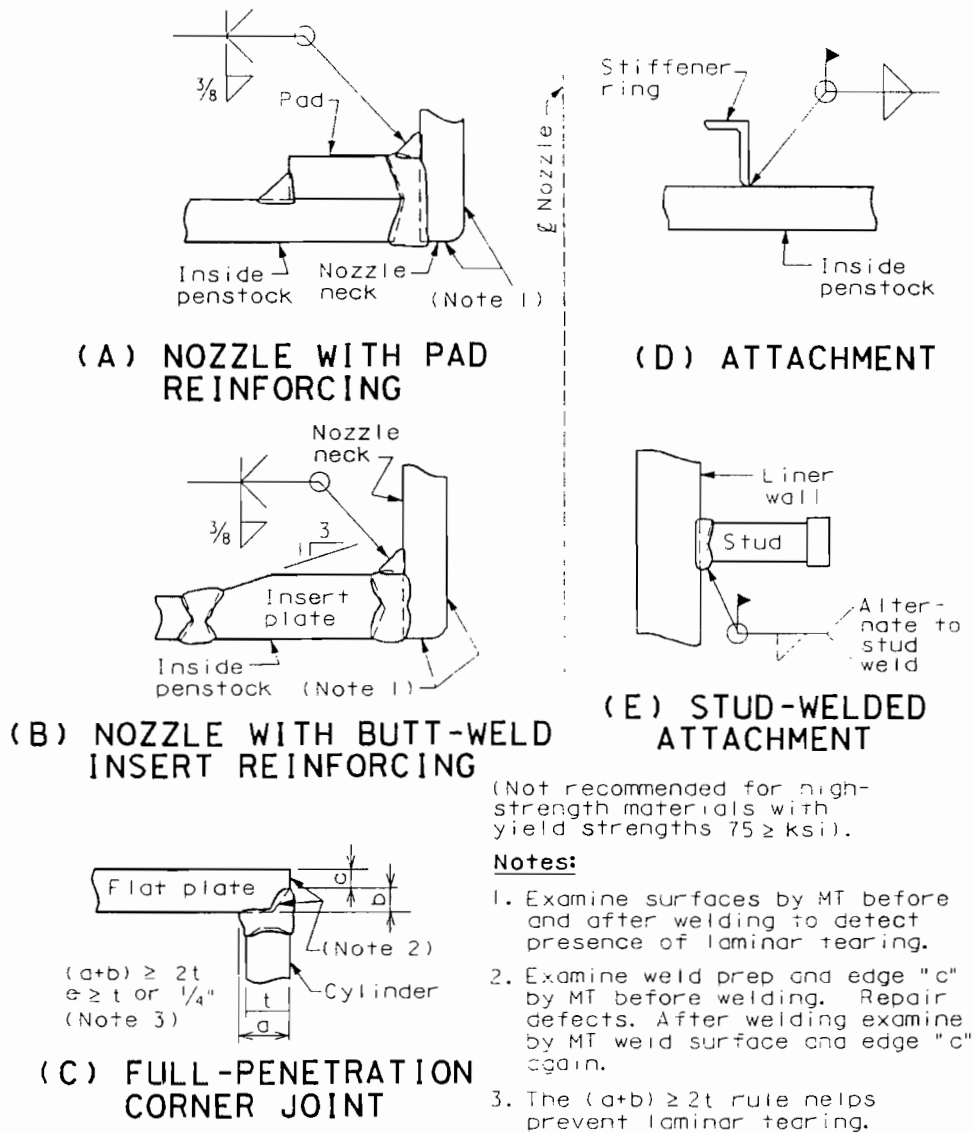


Figure 4-11 Welded Joint Configurations (II)

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Configurations for bifurcation joints are shown in Figure 4-12.

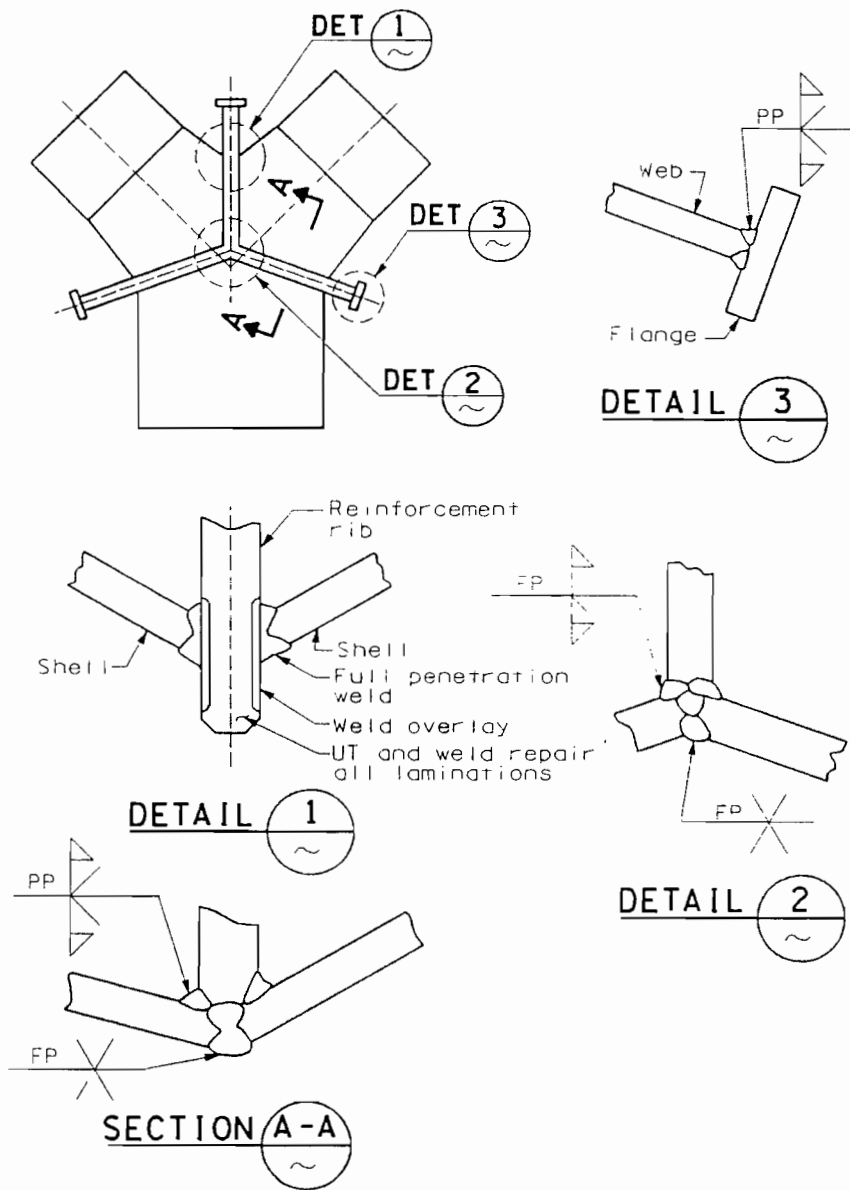


Figure 4-12 Welded Joints—Bifurcations

Detail 1 of Figure 4-12 shows an acceptable detail at the junction of a bifurcation shell (or skin) to the center reinforcing girder. Other details have been successfully used; however, the one shown is a good illustration of such a joint. Special precautions are necessary to avoid or minimize susceptibility to lamellar tearing. These special precautions include: use of weld overlay and ultrasonic examination before and after welding; use of material conforming to low sulfur practice; manufacture of the material by forging (to ultrasonic-tested quality); and the requirement that the material meet specific tensile properties in the three orthogonal directions

(e.g., specific through thickness properties of elongation, impact energy, and tensile strength). The joints between the bifurcation shell and center reinforcement are full-fusion, full-penetration welds, and are 100% examined either ultrasonically or by radiography and also by the magnetic-particle method.

4.6.2 Butt Joints

Welded joints required to be examined by radiography and ultrasound must be full-penetration butt welds.

4.6.3 Double-Welded Lap Joints

Double full-fillet lap joints must be limited to thicknesses not greater than 3/8 inch for longitudinal joints and 5/8 inch for circumferential joints.

4.6.4 Joint Qualifications

Prequalified welded joints for complete penetration groove welds that meet the AISC Specification for Structural Steel Buildings¹⁴ and the AWS Structural Welding Code (D1.1-92)¹⁵ are acceptable for penstock construction.

4.6.5 Pipe Welds

Circumferential joints in pipe 24NPS and smaller in diameter may be joined by full-penetration, single-welded groove butt welds without backing strips.

4.6.6 Bolted Joints

Bolted flanged joints meeting ANSI B16.5 requirements are acceptable for joints in pipe 24NPS and smaller in diameter. Bolted joints that meet the ASME Code, Section VIII, Division 1 requirements are acceptable for all diameters of penstocks.

4.6.7 Single-Welded Lap Joints

Single-fillet, welded lap joints are acceptable for circumferential joints in penstocks 24NPS and smaller in diameter provided the joint is prequalified and the thickness does not exceed 3/8 inch.

4.6.8 Backed-Up Butt Joints

For a number of reasons, backed-up joints generally are not used if joints are required to undergo radiographic or ultrasonic examination. First, it is difficult to interpret radiographic or ultrasonic indications that invariably show up on the film (RT) or scope (UT). Second, if access for installing film to the backed-up side is available, welder access also usually is available and it is preferable to double weld the joint. Backed-up joints for circumferential joints are common for water pipes and carbon steel penstocks where radiography or ultrasound is not required and

4 Exposed Penstocks

thicknesses are 1¼ inches or less. For quenched and tempered high-strength material, such as A517, backed-up joints should not be used.

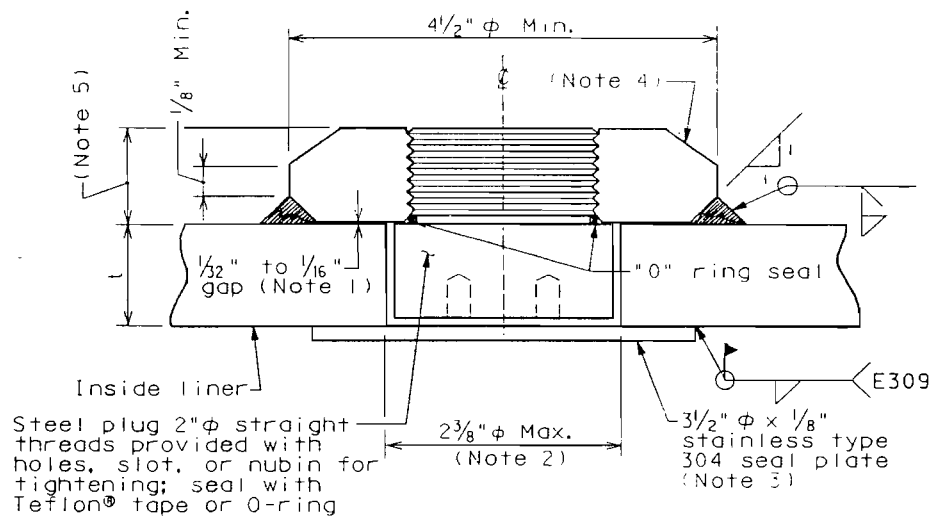
If backed-up joints have the back-up bar removed at the completion of welding and are examined by RT or UT, they are acceptable if a prequalified detail.

4.6.9 Back-Up Bar Splices

Where backed-up joints are permitted, splices in the back-up bar must be full-penetration butt welds.

4.6.10 Grout Connections

Figure 4-13 shows a weld detail for a welded grout connection with a pad.



Notes:

- (1) Before welding.
- (2) Maximum diameter that does not require 100% replacement.
- (3) Stainless for ductility and corrosion/erosion resistance (1/8" thick to minimize projection into waterway).
- (4) Chamfer corners to prevent bonding to concrete.
- (5) Minimum thickness 3/4" for thread engagement.

Figure 4-13 Grout Connection - Weld Detail

Figure 4-14 shows a grout connection without a pad (see Figure 4-13 for details not shown).

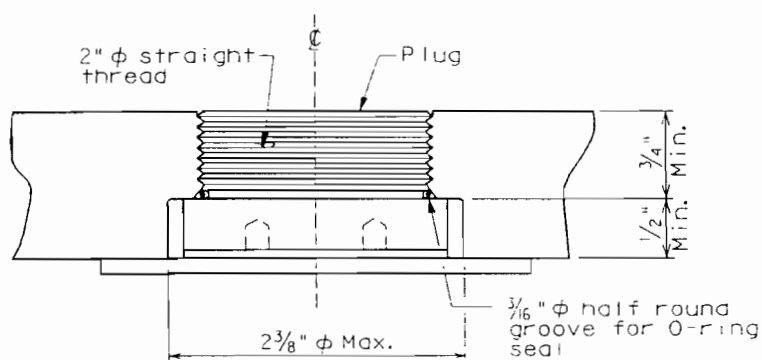


Figure 4-14 Grout Connection Without Pad

4.7 Transitions

4.7.1 Geometric Transitions

4.7.1.1 Diameter Changes

Changes in diameter are usually accomplished by the use of right conical shells placed in the straight tangent portion of a line or combined with a mitered bend as shown in Figure 4-15. Cone angles, diameter ratios, and miter angles shown in these sketches are recommended but not mandatory. At angle changes in the profile of the penstock (cone-cylinder or miter-to-miter) greater than those recommended, special discontinuity analysis is necessary. One method for this analysis is given in the ASME Code, Section VIII, Division 1, Appendix 1-5. Stiffener rings sometimes are required at these junctions. Geometric layout of mitered conical sections bends is given in AWWA Manual M11.¹⁶

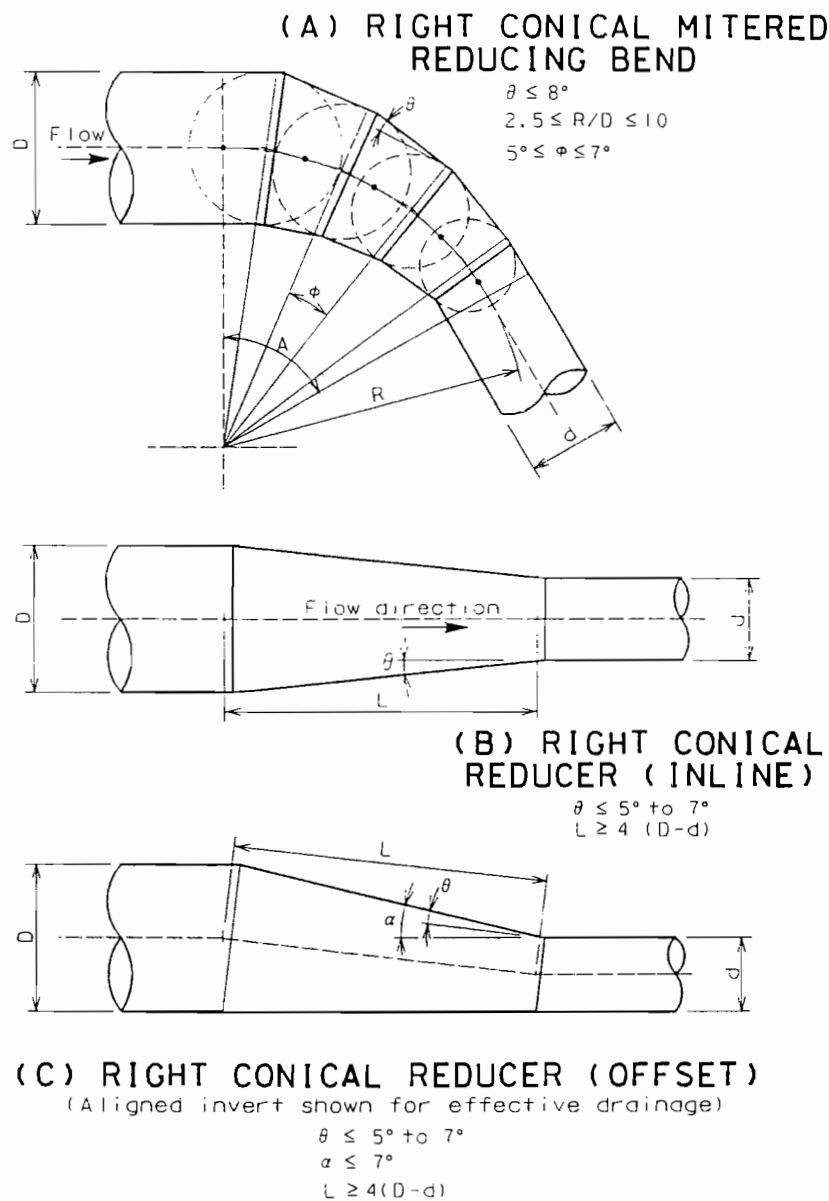


Figure 4-15 Typical Geometric Transitions

Figure 4-16 illustrates a change in diameter effected by a contoured transition of parabolic form. Such transitions are often used at inlets to the penstock and at the inlet/outlet of surge tanks.

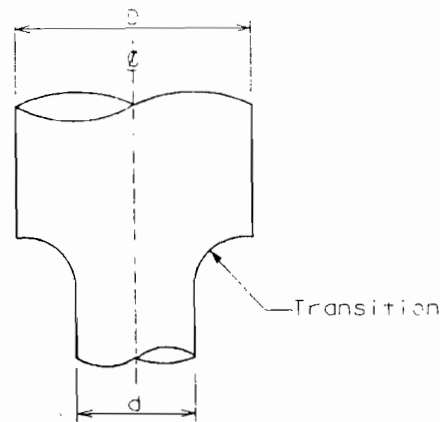


Figure 4-16 Contoured Change in Diameter

4.7.1.2 Cross-Section Shape Changes

Figure 4-17 illustrates a square-to-circular transition. The transition is made up of flat triangular sides and oblique conical quarter sections at the four corners. Such shapes cannot resist the pressure of water forces with membrane type stresses only and, therefore, must be stiffened. Moment resisting frames can be used for stiffening. Each frame must be continuous around the perimeter and spaced to limit bending stresses in the flat plate liner between stiffeners. If the transition is embedded in structural reinforced concrete or mass concrete, anchor studs welded to the exterior side of the transition can be used in place of stiffener frames.

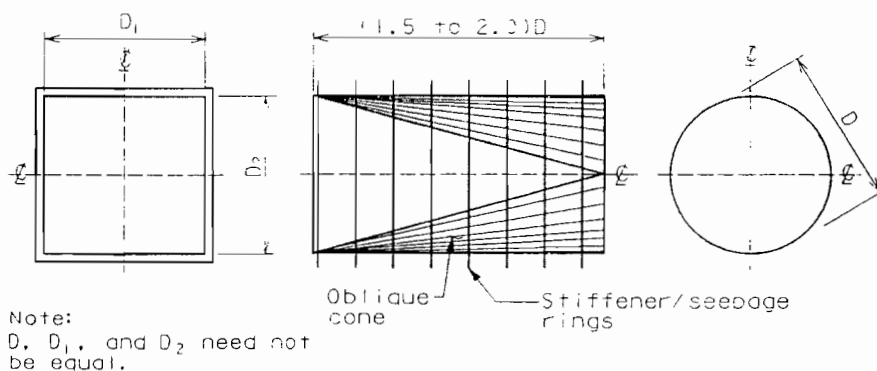


Figure 4-17 Square-to-Circular Transition

Other geometric forms can be used for geometric transitions provided the form can contain pressure with essentially membrane forces only. The form must be stiffened if it cannot contain pressure with essentially membrane forces only. The ASME Code, Section VIII, Division 1, Appendix 13 contains rules for the design of noncircular cross-section conduits, both stiffened and unstiffened.

4.7.2 Thickness Transitions

4.7.2.1 Tapers

A tapered transition having a length not less than three times the offset between the adjacent surfaces of abutting sections, as shown in Figure 4-18, must be provided at butt joints between sections that differ in thickness by more than one-fourth of the thickness of the thinner section, or by more than 1/8 inch, whichever is less. The transition may be formed by any process that will provide a uniform taper. When the transition is formed by removing material from the thicker section, the minimum thickness of that section, after the material is removed, must be not less than that required to resist pressure (such as hoop stress). When the transition is formed by adding additional weld metal beyond what would otherwise be the edge of the weld, such additional weld metal buildup must undergo examination by MT or PT and be included in the examination for the butt weld. The butt weld may be partly or entirely in the tapered section or adjacent to it.

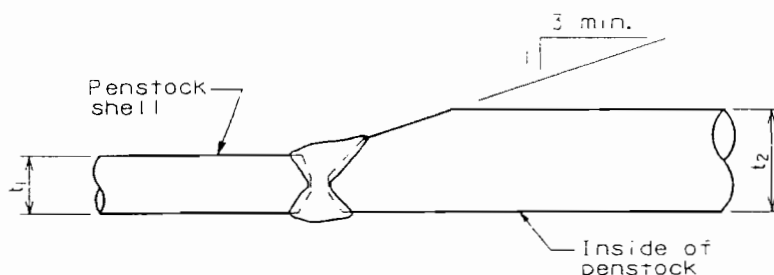


Figure 4-18 Change-in-Thickness Transition (Butt Weld)

4.7.2.2 Nozzle/Branch to Penstock Transitions

A transition between a forged nozzle or branch and the penstock is illustrated in Figure 4-19. Note the detailing feature of using a radius at the otherwise sharp corner to keep stress concentrations to reasonable levels. For some materials, details utilizing fillets and chamfers at sharp corners are sufficient. Generally, the ASME Code details, which reflect good practice, should be used.

Figure 4-19 also illustrates a transition from the thick nozzle neck to a thinner branch pipe. In this case a steeper taper of 1:1 is shown but a radius is used at the end of the taper where it meets the thinner pipe.

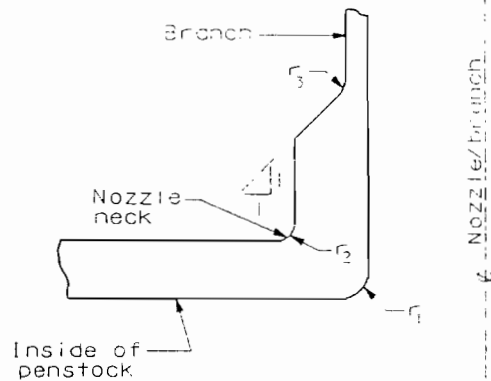


Figure 4-19 Contoured Transition

4.7.2.3 Other Transitions in Thickness

The ASME Code specifies other types of transitions in thickness, in joint misalignment, at changes in direction (such as at a mitered joint), at flanges, and at contouring of forged fittings. The ASME Code limitations and requirements apply to any transition not specifically covered in this manual.

4.7.3 Material Transitions

Figure 4-20 indicates an acceptable transition in material, where the transition is made up of a spool piece of higher strength material (material 2).

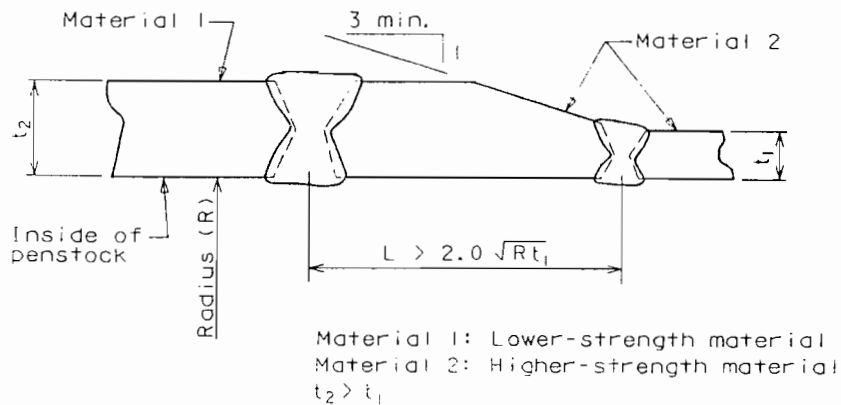


Figure 4-20 Typical Change in Material Transition

References*

1. Roark, R.J. and Young, W.C. *Formulas for Stress and Strain*. McGraw-Hill Book Co., New York, NY (1975).
2. Schorer, H. "Design of Large Pipelines." *Transactions*. ASCE. 98:101 (1933).
3. Zick, L.P. "Stresses in Large Horizontal Cylindrical Pressure Vessels on Two Saddle Supports." *Welding Journal* (Welding Research Supplement). 30:435-S (Sept. 1951).
4. Donnell, L.H. and Wan, C.C. "Effects of Imperfections on Buckling of Thin Cylinders and Columns Under Axial Compression." *Journal of Applied Mechanics* (1950).
5. Baker, E.H., Kovalsky, L. and Rish, F.L. *Structural Analysis of Shells*. McGraw-Hill Book Co., New York, NY (1972).
6. "Penstock Analysis and Stiffener Ring Design." Bulletin No. 5, Part 5. Boulder Canyon Project Final Design Report. U.S. Bureau of Reclamation, Denver, CO.
7. Bier, P.J. "Welded Steel Penstocks—Design and Construction." Engineering Monograph No. 3. U.S. Bureau of Reclamation, Denver, CO (1986).
8. *Manual of Steel Construction, Allowable Stress Design*. American Institute of Steel Construction, Chicago, IL (9th Ed., 1989).
9. Zick, L.P. "Useful Information on the Design of Plate Structures." *Steel Plate Engineering Data—Volume 2*. American Iron and Steel Institute and Steel Plate Fabricators Association, Inc. (Feb. 1979).
10. *Recommended Rules for Design and Construction of Large, Welded, Low-Pressure Storage Tanks*. API Standard 620. American Petroleum Institute, Washington, DC.
11. *Specification for the Design, Fabrication, and Erection of Structural Steel for Buildings*. American Institute of Steel Construction, Inc., Chicago, IL (Nov. 1, 1978).
12. *Building Code Requirements for Reinforced Concrete*. (ACI 318). American Concrete Institute, Detroit, MI.
13. *Standard for Dimensions for Fabricated Steel Water Pipe Fittings*. ANSI/AWWA Standard C208-83(R89). AWWA, Denver, CO.
14. *Specification for Structural Steel Buildings—Allowable Stress Design and Plastic Design*, June 1, 1989. American Institute of Steel Construction, Inc., Chicago, IL.

* The most current version of a standard, code, or specification should be used for reference.

15. Structural Welding Code—Steel. ANSI/AWS D1.1-92. American Welding Society, Miami, FL.
16. *Steel Pipe—A Guide for Design and Installation*. AWWA Manual M11. AWWA, Denver, CO (1989).

The following references are not cited in the text.

Parmakian, J. "Minimum Thickness for Handling Steel Pipe." *Water Power & Dam Construction* (June 1982).

Pirok, J.N. "Some Problems of a Penstock Builder." *ASCE Journal of the Power Division* 83, (PO 3). Paper 1284 (June 1957).

Seely, F.B. and Smith, J.O. *Advanced Mechanics of Material*. John Wiley & Sons, Inc. (2nd Ed., 1952).

"Test of Cylindrical Shells." *University of Illinois Bulletin No. 331*. (Sept. 23, 1941).

"Welded Steel Pipe." *Steel Plate Engineering Data—Volume 3*. American Iron and Steel Institute and Steel Plate Fabricators Association, Inc. (1989).

Wilson, W.M. and Newmark, N.M. "The Strength of Thin Cylindrical Shells as Columns." *University of Illinois Engineering Station Bulletin No. 255* (Feb. 1933).

Zick, L.P. and St. Germain, A.R. "Circumferential Stresses in Pressure Vessel Shells of Revolution." *ASME* (Sept. 1962).



5.1 Design of Buried Penstock Shells

When a steel penstock installation requires the pipeline to be buried below ground or within fill material, the penstock shell must be analyzed to resist not only internal pressure and other hydrodynamic loads, but also external loads due to earthfill, loads resulting from excessive deflections, potential movements of the ground, and numerous live loads.

Shell design for internal pressure and other hydraulic and hydrodynamic conditions is performed in the same manner as for an exposed penstock. Shell design for external soil and other loads must take into consideration the flexibility of the pipe and the shell thickness needed to resist the external loads acting on the pipe prior to pressurization.

External dead and live loads on flexible pipe produce compressive stresses in the pipe wall, which tend to reduce the hoop stress due to internal pressure. The two key elements for shell design are the flexibility of the pipe and the type of soil and its characteristics.

5.1.1 Pipe Flexibility

Three groupings of pipe flexibility are generally considered for design. These are defined by the AWWA as:

- (1) Rigid pipe, whose cross-sectional shapes cannot be distorted sufficiently to change their vertical or horizontal dimensions more than 0.1% without causing damage
- (2) Semirigid conduits, whose cross-sectional shapes can be distorted sufficiently to change their vertical or horizontal dimensions more than 0.1% but not more than 3.0%, without causing material damage
- (3) Flexible conduits, whose cross-sectional shapes can be distorted sufficiently to change their vertical or horizontal dimensions more than 3.0% before causing damage.

Steel pipe, normally considered for penstock applications, usually functions as a flexible conduit within the flexibility limits imposed by the requirement that there be no damage to the lining and coatings used in the applications.

Pipes with D/t ratios less than 288 will not display excessive deflections if the backfill is compacted to a density greater than that of the zone of critical void ratio. The critical void ratio is the density (or void ratio) at which no volume change occurs. A standard density of 80% is considered above the critical void ratio. Stulling and increased compaction efforts should be applied if the ratio is greater than 288.

Acceptable deflections of the pipe shell, based as a percentage of the outside pipe diameter for buried pipe, are as follows:

- (1) Flexible lined and coated pipe: 5%
- (2) Flexible coated and mortar-lined pipe: 3%
- (3) Mortar-lined and coated pipe: 2%

5.1.2 Load Types

Two types of earth loads are applicable to buried pipe design:

- (1) Trench loading, in which the pipe is laid in an excavated trench and backfilled
- (2) Fill earth loading, in which the pipe is laid on a graded or prepared ground surface and fill is placed around and over the pipe.

5.1.3 Analyses for Buried Pipe

Excellent and detail analyses and sample examples for the analyses of buried pipe are presented in AWWA Manual M11¹ and "Welded Steel Pipe", *Steel Plate Engineering Data, Volume 3*, by the American Iron and Steel Institute.²

5.1.4 Special Considerations

For an underground penstock installation, the following must be considered:

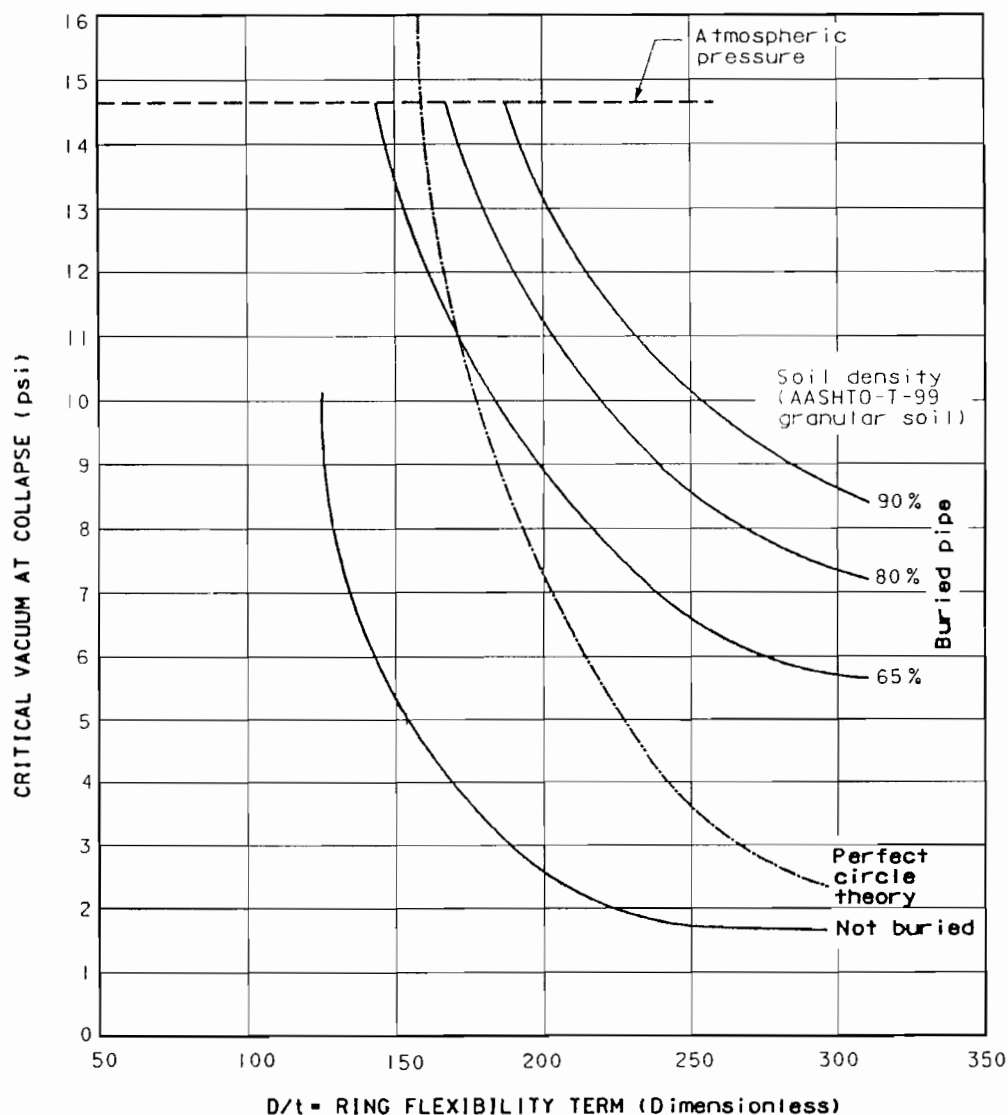
- (1) Protection against corrosion by coating
- (2) Cathodic protection
- (3) Vacuum design
- (4) Flotation and drainage
- (5) Specialty construction relating to trenching, pipe laying, and backfilling
- (6) Welding and construction requirements
- (7) Inspection, hydrotesting, and maintenance requirements.

5.2 External Pressure

Generally, buried penstocks do not require stiffener rings to resist vacuum or earth pressures. The design for external pressures and backfill forces is dependent on the diameter-to-thickness ratio, backfill material and density, and built-in elongation of the vertical penstock diameter. The penstock wall is designed thick enough to resist external pressure, external backfill forces, and vacuum.

5.2.1 Method

References 1 and 2 address external pressure and vacuum design of buried pipelines. Figure 5-1 presents a series of curves which plot critical collapse pressure versus diameter-to-thickness ratio for a given ring deflection (5%). Each curve is for a particular soil compaction or backfill friction angle. A factor of safety must be applied and a ring deflection determined from the appropriate curve. It is preferable to relate the charts to a diameter-to-thickness ratio instead of a vertical diameter elongation and to extend the curves to a critical pressure of at least 50 psi so that applying a factor of safety of 3.0 would cover the case of full vacuum. Vertical deflections, established before backfilling, should be no more than 1% (preferably 0.5%). Note that vertical elongation expressed as a percentage of diameter is equivalent to one-half the out-of-round expressed as a percentage of the difference of diameters. The maximum out-of-round is 1%; therefore, the maximum vertical diameter elongation after backfilling should be 0.5%.



Summary of test for the critical vacuum at collapse if the pipe ring is deformed approximately 5%.

Source: AISI Research, Utah State University, Logan, Utah.

Figure 5-1 Critical Collapse Pressure vs. Ring Deflection

AWWA Manual M11² gives a procedure and formula for calculating allowable external pressure independent of out-of-round. It can be assumed the permissible out-of-round is 1% of the diameter (which conforms to requirements in this manual).

External loadings that can contribute to buckling include external water pressure, soil pressure surcharge (using the Marston theory above the penstock), live loads on top of the surcharge (where the effect on the penstock is determined by the Boussinesq method), and vacuum or net external pressure.

5.2.2 Factor of Safety

The factor of safety (referred to as *design factor* in AWWA Manual M11¹) should be 3.0 for cover-to-diameter ratios less than 2, and 2.5 for cover to diameter ratios equal to or greater than 2.

5.2.3 Stiffener Rings

If external pressure exceeds the factored allowable external pressure permitted by the method specified in AWWA Manual M11,¹ stiffener rings can be used instead of increasing the penstock shell thickness. Stiffener ring design and spacing should follow the procedure given in Section 4.4.

An alternative to adding stiffener rings is specifying a denser backfill or decreasing the elongation of the vertical diameter. This will increase the factored allowable external pressure. These methods are preferable to the use of stiffener rings. In addition, consideration should be given to a means for reducing the magnitude of vacuum, i.e., by devices such as vacuum relief valves.

5.3 Bends

Bends in buried penstocks must meet the same requirements as for bends in exposed penstocks (see Section 4.5).

5.4 Buried Joints

Buried field joints similar to exposed joints for aboveground penstocks provide either rigidity or flexibility in the penstock according to the design requirements.

Rigid joints that must provide longitudinal restraint along the penstock to resist axial and bending forces caused by temperature and pressure can be either full-penetration, butt-welded joints, which provide the greatest strength, or lap-welded joints, which also may provide sufficient strength and are less expensive. Single- or double-welded lap joints often are used for thinner wall thicknesses and pressures up to 250 psi. These joints provide some flexibility and are generally less costly than butt-welded joints. When this type of joint is used, longitudinal stresses at the joint should be checked carefully.

Flanged and bolted joints and mechanically tied joints may be used to join buried penstocks; however, their use is less common than for exposed penstocks. Such joints generally are placed in vaults.

5 Buried Penstocks

Welded and bolted joints must meet the requirements specified in Section 4.6 for welded and bolted joints in exposed penstocks.

Since thermal expansion and contraction of a buried penstock is less severe than for an exposed penstock, buried joints that must provide penstock flexibility may be either mechanical sleeve-type couplings or joints with rubber gaskets. Flexible joints allow angular joint deflection but cannot transfer appreciable longitudinal stresses. Rubber gasketed O-ring joints are used for diameters up to 72 inches and for pressures up to 250 psi. Such joints may provide the least costly joint and may help reduce installation costs. Bell-and-spigot joints are used for circumferential field joints up to 5/8 inch in thickness. Buried flexible joints must be restrained from penstock thrust forces with suitable thrust ties or anchor blocks. Alternatively, joints can be welded to make up lengths sufficiently long to resist thrust forces through soil-pipe friction.

Bolted sleeve-type couplings also are used for closures, roll out sections, and other points where articulating joints are required. Expansion joints for buried penstocks typically are not required. When provided, bolted sleeve-type couplings and mechanically tied joints are placed in vaults for accessibility, and installed in pairs with a spool piece between them to accommodate differential settlement. Proper installation procedures must be observed.

5.4.1 Coated Field Joints and Jumper Straps

During penstock construction special care is required to ensure that field joints are properly protected with a rust inhibitive coating to prevent corrosion problems. Also, for field joints that are not connected by field welding, "jumper straps" often are installed after the field joint is assembled to maintain electrical continuity. Coatings, repairs, and cathodic protection are discussed in Section 10.

5.4.2 Installation Conditions

During installation and before backfilling, buried penstocks are exposed to the same atmospheric and solar conditions as aboveground penstocks. Care must be taken to maintain alignment, and consideration must be given to expansion and contraction due to temperature changes. Some precautions include shading penstock sections already joined together and completing the backfilling on completed sections before closure welds are made.

Closure welds often are made at night to minimize temperature-induced stresses. Welding at night also eliminates all temperature differentials caused by the sun. Special welding procedures often are employed, such as the use of several welders equally spaced around the circumference of large-diameter penstocks and uninterrupted welding from the first to the last welding pass. Preheat and postheat are carefully controlled, as are the gaps around the joint. Often a zero gap procedure is used. Butt-welded closures are preferred.

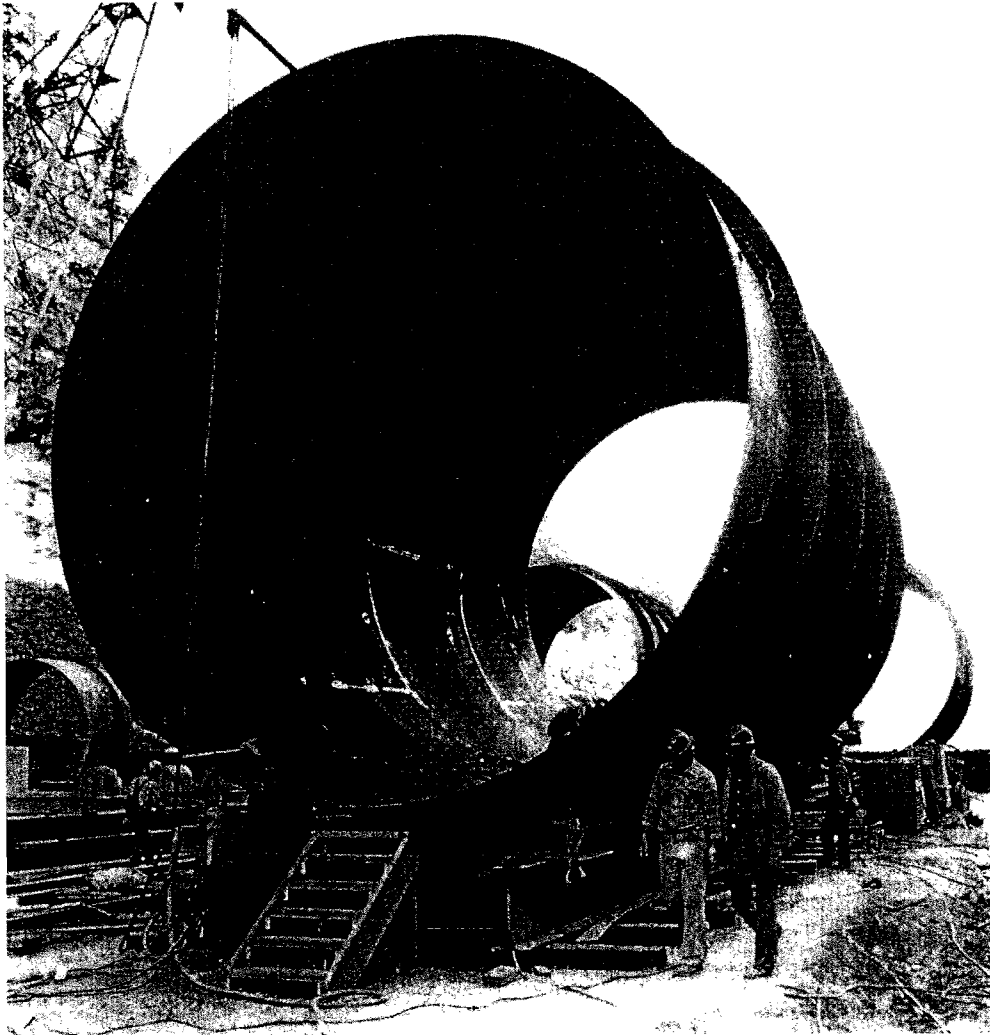
5.5 Transitions

Transitions in buried penstocks must meet the same requirements as for transitions in exposed penstocks (see Section 4.7).

References*

1. *Steel Pipe—A Guide for Design and Installation*. AWWA Manual M11. AWWA, Denver, CO (1989).
2. "Welded Steel Pipe." *Steel Plate Engineering Data—Volume 3*. Revised Edition. American Iron and Steel Institute and Steel Plate Fabricators Association, Inc. (1989).

* The most current version of a standard, code, or specification should be used for reference.



Steel tunnel liners often are required to prevent the migration of tunnel leakage due to unfavorable geologic conditions in the surrounding rock mass. When a steel liner is installed in a tunnel, it must be analyzed to determine that it can resist internal operating pressures as well as resist buckling from external water pressures when the tunnel is dewatered infrequently for inspection and maintenance.

This section discusses procedures for the analysis and design of steel tunnel liners for:

- (1) Internal pressure
- (2) External pressure (steel tunnel liners without stiffeners)

- (3) External pressure (shell between stiffeners when stiffeners are used)
- (4) External pressure (stiffener and contributing contiguous portion of shell).

The magnitude and conditions of internal design pressure loading to be considered are discussed in Section 1.2; similarly, the sources and magnitude of external pressure when a tunnel is dewatered are discussed in Section 1.3.

Further considerations for the magnitude and conditions of internal design pressure loading and the sources and magnitudes of external pressure loading when the tunnel is dewatered are presented within this section.

6.1 Design of Steel Liners for Internal Pressure

A pressurized steel tunnel liner will interact elastically with the contiguous concrete backfill and surrounding rock continuum to share its load. In the interest of obtaining an economic design, the steel tunnel liner should be analyzed for only that portion of the internal pressure that it alone will be required to carry. The depth of rock cover necessary to consider load sharing for various rock conditions is discussed in Figure 3-22 (Chapter 3, "Tunnels and Shafts"; Volume 2, Waterways) of the ASCE/EPRI *Civil Engineering Guidelines*.¹ When rock cover and surrounding rock conditions are inadequate for load sharing, the steel liner must carry the full internal pressure. Rock moduli for various rock types and graphs showing how rock modulus affects load sharing also are presented in the ASCE/EPRI *Civil Engineering Guidelines*.

When rock cover and rock conditions are favorable, the ratio of the pressure carried by the steel liner to the total internal pressure can be determined in accordance with an elastic analysis, where the radial displacement (Δ_r) of the circumference of the steel tunnel liner is determined as shown in Figure 6-1. A more detailed development is presented in Appendix B (Chapter 3, "Tunnels and Shafts"; Volume 2, Waterways) of the ASCE/EPRI *Civil Engineering Guidelines*. In Figure 6-1, the radial temperature gap (Δ_K) results from concrete shrinkage plus the temperature differences between the temperature reached during erection, including the effect of the temperature rise during hydration of cement and the lowest operating water temperature; Δ_C is the radial deformation in the concrete backfill behind the steel liner; Δ_D is the radial deformation in the cylinder of any destressed, fissured rock around the tunnel excavation; and Δ_E is the radial deformation at the inner radius of the unfissured or more competent rock surrounding the tunnel.

In tropical and semitropical zones, $\Delta_K \cong 0$ because the tunnels and construction materials reach ambient air temperature after they are open to forced ventilation for a long period, and the average water temperature in the reservoir is at or near the average air temperature. In addition, the final concrete temperatures can be expected to be in the order of the same magnitude if properly controlled during placement. In northern climates, consideration should be given to the possible existence of a gap and its being properly accounted for.

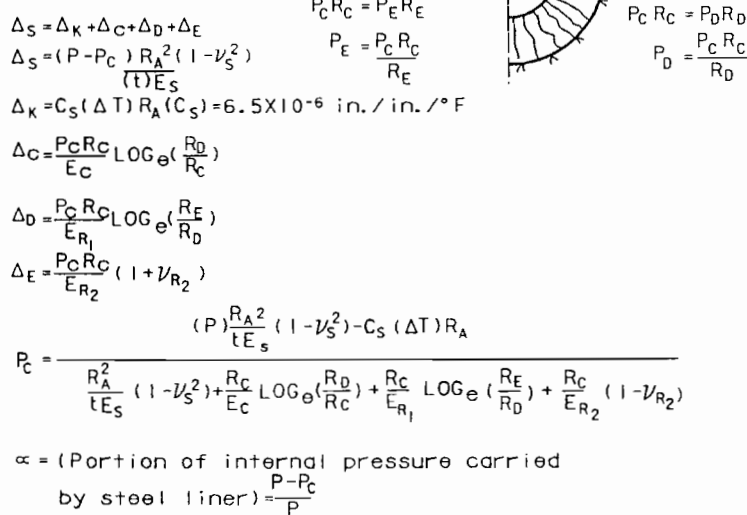


Figure 6-1 Elastic Interaction Between Steel Liner, Concrete Backfill, and Surrounding Rock Continuum

The concrete backfill is assumed to be radially fractured during loading so that strain in the radial direction is not affected by the Poisson ratio for homogeneous, continuous material. Similarly, blasting effects during excavation usually are the cause for a distressed, fissured zone of rock around the excavated tunnel periphery, so that strain in the radial direction in that material also is

unaffected by the Poisson ratio. In Figure 6-1 the pressure transmitted to the face of the concrete (P_c) can be assumed to be distributed radially outward through the fissured materials inversely proportional to the radius. The thickness of the zone of destressed, fissured material can vary from little or nothing in a machine-bored tunnel to several feet in a conventional drilled and blasted tunnel in jointed rock.

The moduli of rock deformation should reflect the effects of overlying rock mass at tunnel depth. Investigations should be carried out accordingly. Geophysical measurements can be made in the tunnel to determine the depth of fissured rock and its relative modulus of deformation with respect to the undisturbed, nondestressed parent rock.

The elastic interaction method presented here is valid only when the vertical and horizontal in-situ stresses are approximately equal. A computer program is presented in Figure 2 of E. T. Moore's "Designing Steel Tunnel Liners for Hydro Plants"², which can be used to determine the portion of the internal pressure carried by the steel liner. When significant differences exist between the vertical and horizontal in-situ stresses, a three-dimensional Finite Element Analysis should be used.

On some occasions the steel liner must be designed and fabricated for installation prior to completing the excavation of the tunnel. The moduli of deformation would need to be determined very conservatively on the basis of tests performed in exploratory drill holes and/or exploratory adits in the various rock units encountered. The values of the moduli can then be considered representative. However, during the actual tunnel excavation, zones of material may be encountered locally over short reaches that have a lower modulus than that determined for use in the design. One way to resolve this problem, short of substituting a heavier plate, is to provide circumferential hoop reinforcement on the outside of the steel liner throughout the local zone of lower resistant rock. The use of circumferential reinforcing, however, is not very effective, especially when the liner has an appreciable thickness. In addition, if a temperature reduction will exist between that at the liner installation and subsequent operation conditions, a gap could exist behind the liner when the tunnel is filled and the liner subjected to internal pressure. Circumferential reinforcement can be significantly less beneficial under such circumstances. Closely spaced, multi-layer reinforcement can then only account for small reductions in the value of the rock modulus of deformation.

6.2 Design of Steel Liners for External Pressure

Experience has shown that buckling occurs in a single lobe. Both E. Amstutz and S. Jacobsen developed their analyses in a very similar manner based on the theory that a single lobe is formed.

The critical external buckling pressure for an unstiffened steel liner is determined considering a gap between the steel liner and the concrete backfill surround due to concrete shrinkage and a temperature difference. The gap can realistically vary from 0 to 0.001 times the radius.

The radial temperature gap results from concrete shrinkage plus the temperature differences between the temperature reached by materials during construction including the effects of the temperature rise during hydration of cement and the rise in ambient temperatures due to forced ventilation into the tunnel using warmer outside air over a long period during construction.

The steel liner is the form for the concrete backfill. Under certain conditions the steel liner might reach temperatures near 80°F when the concrete sets, due to a combination of ambient air temperature and hydration effects. If the tunnel is later dewatered during winter when the water temperature is 34°F, a maximum 46°F temperature difference could exist for a short duration. A 46°F temperature difference produces a gap equal to $0.0003 \times R_A$ (see Figure 6-1).

Permissible tolerances during fabrication and erection allow an out-of-roundness creating an elliptical shape with a 1% difference between the measured maximum and minimum diameters. This difference should not be considered in arriving at the critical radial design gap. A simple analysis will demonstrate that the equivalent increase in curvature on the flattest curvature created for permitted ellipticity is to a circular radius only 1.5% larger than the originally specified radius (see Appendix C (Chapter 3, "Tunnels and Shafts"; Volume 2, Waterways) of the *ASCE/EPRI Civil Engineering Guidelines*¹). The development of a lobe in an unstiffened liner usually involves a sector of the shell subtended by an angle less than 80 degrees. The indented lobe would be expected to develop at the weakest point, which would be at the flattest curvature produced by the ellipticity of the penstock shell.

During concrete placement, the concrete backfill is introduced behind the steel tunnel liner from a steel pump line at the top of the tunnel. The concrete flows down around the steel liner. Due to a combination of frictional drag along the lower external surface of the liner, poor venting, and settlement occurring during consolidation, a void can be created under the bottom invert of the steel liner. This void can subtend an angle of approximately 40 to 60 degrees (0.70 to 1.05 radians). The void is usually shallow and very little grout is needed to fill it. When correctly executed, contact grouting between the steel liner and the concrete backfill eliminates these voids. The determination of the critical radial gap can assume that contact grouting has been properly executed when careful inspection and quality control has been exercised during construction. Therefore, temperature and shrinkage effects alone should govern the selection of an appropriate radial gap for purposes of analysis and design.

6.2.1 Amstutz Formulation

Amstutz assumes that a single lobe or indent will form at one particular spot.³ A considerable extent of the indent always occurs parallel to the axis of the pipe because only the small resistance of the plate to bending has to be overcome. The Amstutz analysis, therefore, has been limited to a circumferential ring of unit width. A new mean radius is developed for the indent or lobe, with two outward and one inward half-waves forming around the mean arc line subtended by this new mean radius (see Figure 6-2).

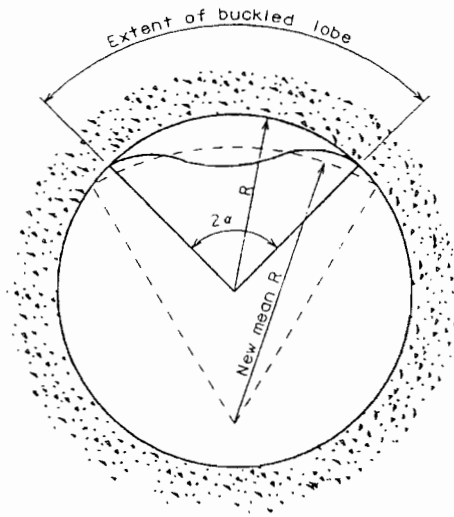


Figure 6-2 Amstutz Buckling Pattern

Amstutz developed his buckling theory for the forces and displacements on the pipe wall element represented by the mean arc line subtended by a corresponding new mean radius. The Amstutz formulas are presented in his paper "Buckling of Pressure-Shaft and Tunnel Linings."³ The Amstutz formulas for determining the stress condition in unstiffened cylindrical steel pipe are:

$$\frac{\sigma_N - \sigma_v}{\sigma_F^* - \sigma_N} \left[\left(\frac{r}{i} \right) \sqrt{\frac{\sigma_N}{E^*}} \right]^3 \cong 1.73 \left(\frac{r}{e} \right) \left[1 - 0.225 \left(\frac{r}{e} \right) \frac{\sigma_F^* - \sigma_N}{E^*} \right] \quad (\text{Equation 6-1})$$

$$p_{cr} \cong \left(\frac{F}{r} \right) \sigma_N \left[1 - 0.175 \left(\frac{r}{e} \right) \frac{\sigma_F^* - \sigma_N}{E^*} \right] \quad (\text{Equation 6-2})$$

Where:

- $i = \frac{t}{\sqrt{12}}, e = \frac{t}{2}, r = \frac{D}{2}, F = t$
- $\sigma_v = - \left(\frac{k}{r} \right) E^*$
- $k/r =$ gap ratio, for gap ratio between steel and concrete = γ (for Figures 6-4 and 6-5)
- $r =$ tunnel liner radius
- $D =$ tunnel liner diameter
- $t =$ plate thickness
- $E =$ modulus of elasticity
- $E^* = E/(1 - \nu^2)$
- $\sigma_F =$ yield strength
- $\sigma_N =$ circumferential axial stress in plate liner ring
- $\mu = 1.5 - 0.5 [1/(1 + 0.002 E/\sigma_F)]^2$

$$\sigma_F^* = \mu \sigma_F / \sqrt{1 - \nu + \nu^2}$$

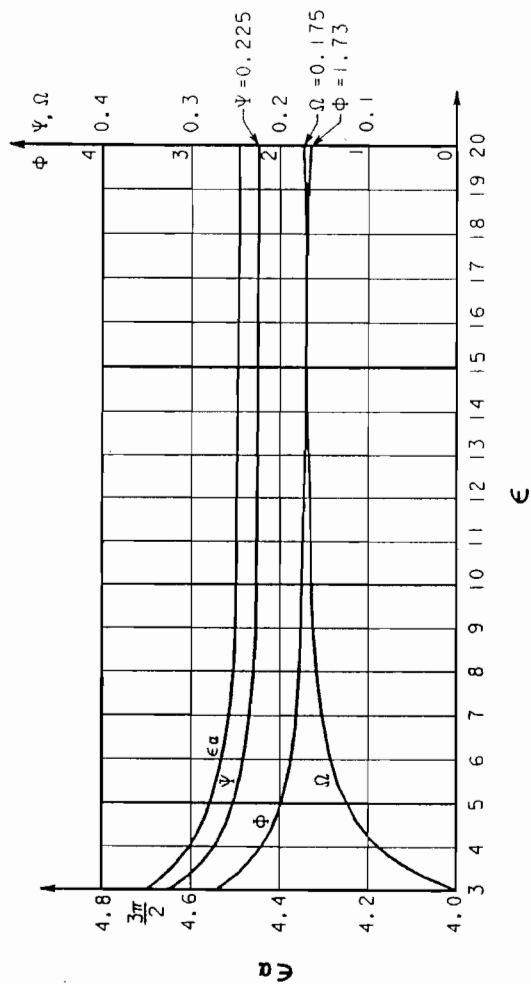
ν = Poisson's ratio

Amstutz's formulas have been reworked (see page 78 of E. T. Moore's paper²) for ease of solution by computer. Figure 6 of Moore's paper presents a computer program to solve Amstutz's equations for the critical buckling pressure on a cylindrical pipe without stiffeners.

For very small gap ratios and relatively thick liners, the value of σ_N approaches σ_Y at the critical buckling pressure, and some caution must be exercised. When the axial stress in the liner approaches the yield stress from a position of low stress, the strain is proportional to stress and the ratio is linear due to the constancy of E , the modulus of elasticity. However, as the stress approaches close to the yield stress, the linearity no longer holds, and strain begins to increase faster than the stress increases until finally when yield stress is reached, the strain increases with no further increase in stress. This, of course, can occur only with a decreasing value of E . E has been assumed constant in the Amstutz formulation. If the axial stress were to exceed about 80% of the yield stress (σ_Y), the Amstutz formulation would need to be modified to reflect the phenomena of a decreasing E by adopting a reducing modulus. However, in addition to complicating the computational process, integrating in a formula for reducing the modulus that is conservatively dimensioned will show such a significant reduction in buckling pressure when $\sigma_N \geq 0.8 \sigma_Y$ that the result will be unacceptable and only results for $\sigma_N < 0.8 \sigma_Y$ will have utility.

Another limitation also must be observed. When Amstutz developed his equations, personal computers were not available. To make his formulation tractable and reduce the number of unknown variables to be solved, he introduced a number of coefficients. He determined that within certain limits some variables could be represented by constants, i.e., coefficients, without affecting the results. ϵ is an expression in the differential equation for the inward deformation of the shell at any point. In his 1970 paper, Amstutz indicated that an acceptable range for ϵ is $5 < \epsilon < 20$. However, from inspection of Figure 6-3, a more acceptable range would be $10 < \epsilon < 20$. Substituting the actual values of F , W , and Y of Figure 6-3 when $\epsilon < 10$ results in critical buckling pressures that are actually higher than the buckling pressures computed using the respective values of 1.73, 0.175, and 0.225 recommended by Amstutz for use in his equations. However, in very few design cases is $\epsilon \geq 10$. The use of the Amstutz equations therefore appears questionable when Jacobsen's procedure is available and is not subject to the Amstutz limitations.

When computing the critical buckling pressure according to the Amstutz equations, the axial stress (σ_N) in the steel liner must be determined along with the corresponding value of ϵ . The results may be considered satisfactory when $\epsilon > 5$ according to Amstutz's recommendation, subject to the limitation that $\sigma_N < 0.8 \sigma_Y$.



Note: At $\epsilon = 2$, $\alpha = 180^\circ$ ($\epsilon \rightarrow \infty$), and ϕ and $\psi \rightarrow \infty$

ϵ	ϵ°	α°	$\tan \epsilon_\alpha$	$\tan \alpha$	$\epsilon \tan \alpha$	$\cos(\epsilon \alpha)$	$\sin(\epsilon \alpha)$	$\sin \alpha$	$\epsilon \alpha$	β (25)	γ (29)	b (35)	ϕ (39)	ψ (40)	Ω (48)
3	270° 00'	90° 00'	∞	∞	∞	0	-1.00000	-1.0000	4.71239	-2.6667	28.3	8.00	2.88	0.33	0
4	263° 37', 2	65° 54', 3	8.9446	2.2360	8.9440	-0.11112	-0.99381	0.91287	4.60104	-1.8095	32.7	16.67	2.21	0.27	0.100
5	261° 11', 6	52° 14', 3	6.4550	1.2910	6.4550	-0.15310	-0.98821	0.79056	4.55868	-1.3953	38.7	27.67	2.00	0.25	0.133
10	258° 19', 7	25° 50'	4.8409	0.48413	4.8413	-0.20231	-0.97932	0.43575	4.50868	-0.6650	71.4	119.03	1.78	0.226	0.168
20	257° 40', 2	12° 53'	4.5749	0.22873	4.5746	-0.21357	-0.97693	0.22297	4.49719	-0.3286	143.4	484.2	1.73	0.225	0.175

Figure 6-3 Amstutz Acceptable Range for ϵ

Figures 6-4 and 6-5 show curves for unstiffened liners for yield stresses of 38,000 psi and 50,000 psi, respectively, based on the Amstutz equations. The upper left-hand portion of the curves defined by dashed lines is where $\epsilon < 5$ or $\sigma_N > 0.8 \sigma_Y$. Results within the solid line reach of the curves can be considered satisfactory. If the Amstutz results were further limited to $\epsilon > 10$, only a small portion of the curves could be considered to provide satisfactory results. The dashed line for $2 \sigma_Y/(D/t)$ indicates pressures at which the lining fails in general yielding.

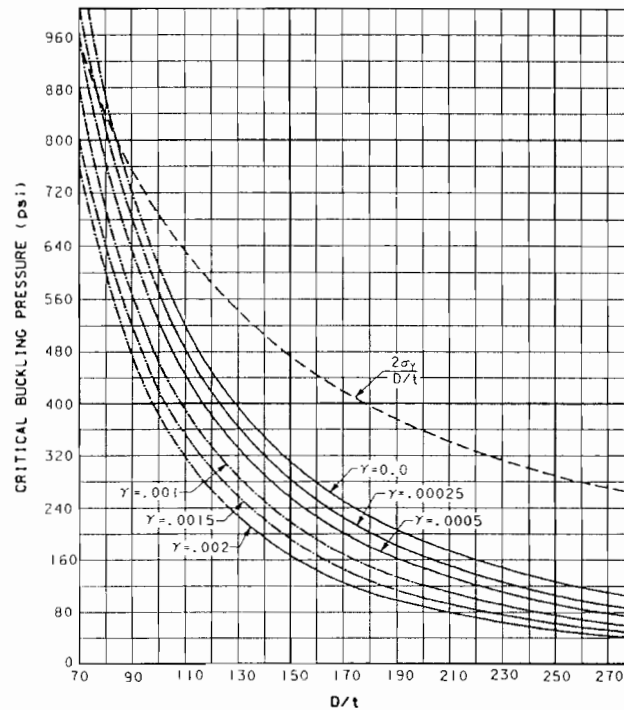


Figure 6-4 Amstutz Curves for Unstiffened Liners (Yield Stress = 38,000 psi)

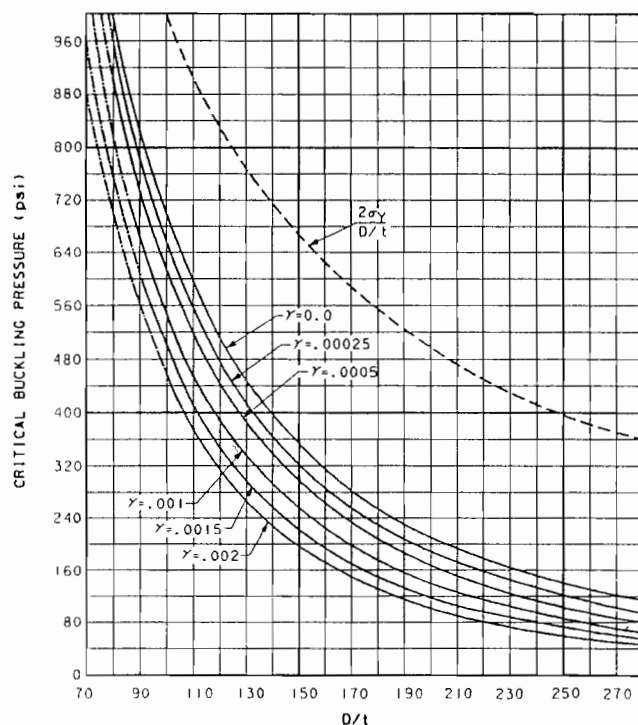


Figure 6-5 Amstutz Curves for Unstiffened Liners (Yield Stress = 50,000 psi)

6.2.2 Jacobsen Formulation

The Jacobsen paper "Buckling of Circular Rings and Cylindrical Tubes Under External Pressure"⁴ gives equations for determining critical buckling pressure for cylindrical pipe without stiffeners.

The three equations that must be solved simultaneously for the three unknowns α , β , and p are:

$$r/t = \sqrt{\frac{\left[(9\pi^2/4\beta^2) - 1 \right] \left[\pi - \alpha + \beta (\sin\alpha/\sin\beta)^2 \right]}{12 (\sin\alpha/\sin\beta)^3 \left[\alpha - (\pi\Delta/r) - \beta (\sin\alpha/\sin\beta) \left[1 + \tan^2(\alpha - \beta)/4 \right] \right]}}$$

(Equation 6-3)

$$p/E^* = \frac{(9/4) (\pi/\beta)^2 - 1}{12(r/t)^3 (\sin\alpha/\sin\beta)^3}$$

(Equation 6-4)

$$\sigma_y/E^* = (t/2r) [1 - (\sin\beta/\sin\alpha)] + (pr \sin\alpha/E^*t \sin\beta) \left[1 + \frac{4\beta r \sin\alpha \tan(\alpha-\beta)}{\pi t \sin\beta} \right]$$

(Equation 6-5)

Where:

- α = one-half the angle subtended to the center of the cylindrical shell by the buckled lobe
- β = one-half the angle subtended by the new mean radius through the half waves of the buckled lobe
- p = critical external buckling pressure, psi
- Δ/r = gap ratio, for gap between steel and concrete
- r = tunnel liner internal radius, in.
- σ_y = yield strength of liner, psi
- t = plate thickness, in.
- E^* = modified modulus of elasticity of steel liner, in psi = $E/(1 - \nu^2)$
- ν = Poisson's ratio for steel.

Jacobsen's formulas have been reworked by E.T. Moore (page 82²) for ease of solution by computer. The critical pressure for a particular D/t ratio and gap ratio is obtained by solving the three simultaneous nonlinear equations using numerical methods.

Figures 6-6 and 6-7 show curves for unstiffened liners for yield stresses of 38,000 psi and 50,000 psi, respectively, based on Jacobsen's equations. Within the practical range of D/t ratios from 70 to 280, the axial stress at buckling

$$\left(\sigma_N = \frac{p_{cr}}{2} \times \frac{D}{t} \right) \quad \text{(Equation 6-6)}$$

does not exceed 80% of the material's yield strength, so it is not necessary to consider a reducing modulus of elasticity to replace E in the Jacobsen analysis. The Jacobsen method gives results approximately 20% lower than the Amstutz method; however, because it does not require the consideration and application of limitations in its use, it is recommended that only the Jacobsen formulation be used for the analysis and design of cylindrical pipe to resist external buckling pressures.

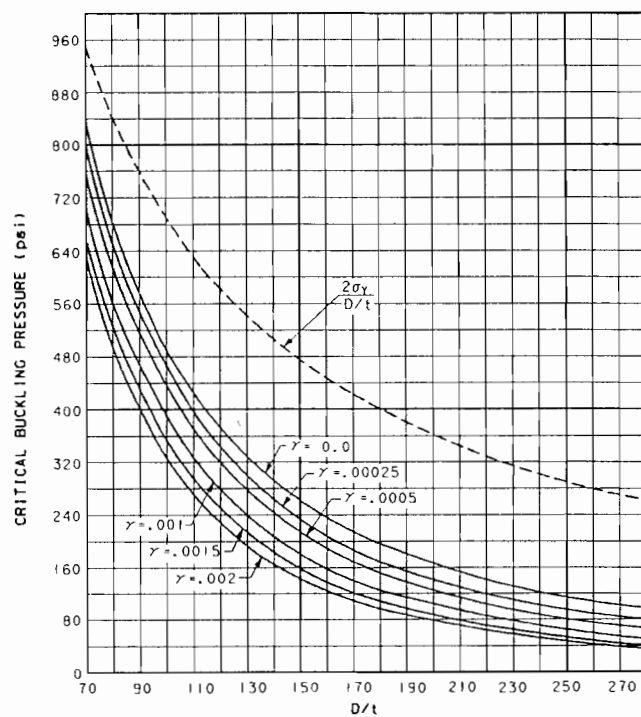


Figure 6-6 Jacobsen Curves for Unstiffened Liners (Yield Stress = 38,000 psi)

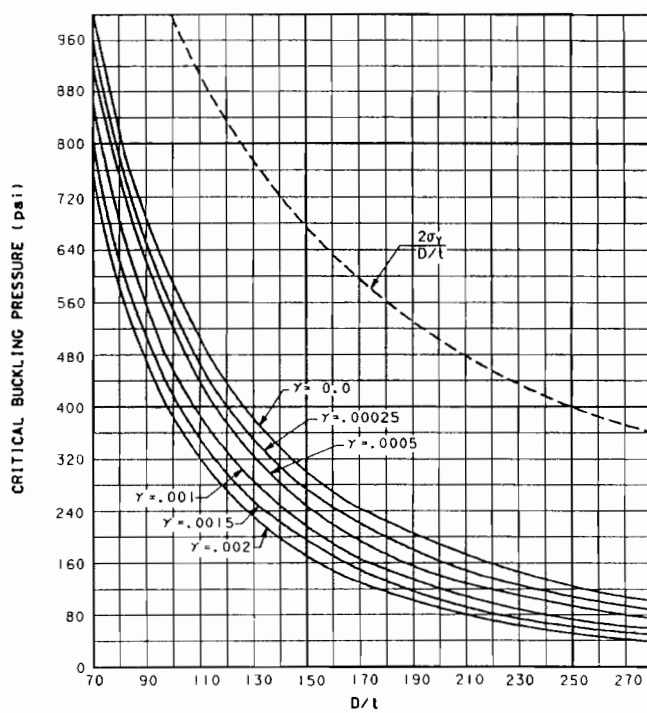


Figure 6-7 Jacobsen Curves for Unstiffened Liners (Yield Stress = 50,000 psi)

6.3 Design of Steel Tunnel Liners with Stiffeners

6.3.1 Introduction

A designer should consider using external circumferential ring stiffeners when the external pressure analysis indicates that the critical external pressure requires an unstiffened liner thickness greater than the thickness required for the internal design pressure. Economic considerations will determine whether the liner shell thickness should be increased, circumferential ring stiffeners should be added, or a combination of increasing the shell thickness and adding ring stiffeners should be adopted.

Stiffener dimensions have been designed and dimensioned in accordance with several different procedures. They include the rotary symmetric formulations, the Bureau of Reclamation general yielding formulation, and the single lobe theory.

6.3.1.1 Rotary Symmetric Formulations

Rotary symmetric theories assume that the circumferential stiffener ring will buckle into an even number of half waves around the circumference. Comparison of these formulations with those of Amstutz, for example, showed that the rotary symmetric theories gave higher critical buckling pressures. Failures occurred at external pressures below those predicted by the theories. Also, failures initiated in practice in one single lobe as depicted in Figure 6-2.

6.3.1.2 General Yielding Formulation

The Bureau of Reclamation's procedure for the design and dimensioning of stiffeners has been used for relatively low external pressures. The collapse mechanism considers a nonembedded pipe shell. The stiffener ring is designed to carry all of the external load over the length (L) between rings. The minimum yield strength and the area of the external ring, plus the area of the effective flange, control the design. Bureau of Reclamation criteria treat the combined section stiffener ring similarly to a very short column. The formulation was developed for relatively low external pressure considerations. It allows relatively wide stiffener spacing and is conservative.

6.3.1.3 Single Lobe Formulation

As previously discussed, both Amstutz and Jacobsen have derived their formulations considering the formation of a single lobe and the confinement offered by the backfill concrete. In actual practice, stiffeners have been observed to fail by buckling in a single lobe. Buckling failure of a stiffener begins when the stress in the extreme, outside fiber of the external stiffener ring element reaches yield stress. This buckling occurs due to a combination of both axial compression and a compressive bending stress at the middle of the indent, as shown in Figure 6-2.

6.3.1.4 Shell Between Stiffeners

When ring stiffeners are provided at spaced intervals, the portion of the pipe between stiffeners will be able to sustain the external pressure acting on it as long as the ring stiffeners maintain the circularity of the pipe shell at their selected locations. The pipe shell between the stiffeners can then be considered a free tube that can buckle inward, free of any constraint provided by the surrounding backfill concrete. The buckling resistance of the free tube between external ring stiffeners is analyzed as a separate analysis. When the external pressure is less than the critical buckling pressure of the shell, the total reactive force of the shell or free tube on the stiffener is only a small portion of the total external radial pressure on the shell between stiffeners. The equivalent force per unit length of the circumference will be less than the acting external pressure multiplied by the "carrying width" of the pipe shell at the stiffener.

6.3.2 Design of Shell as Free Tube Between Stiffeners

The critical external pressure on the shell between stiffeners can be determined by the R. von Mises corrected instability formula for tubes loaded with radial pressure only⁵:

$$p_{cr} = \frac{E(\frac{t}{r})}{1 - \nu^2} \left[\frac{1 - \nu^2}{\left(n^2 - 1 \left(\frac{n^2 L^2}{\pi^2 r^2} + 1 \right) \right)^2} \right] + \frac{E(\frac{t}{r})^3}{12(1 - \nu^2)} \left(n^2 - 1 + \frac{2n^2 - 1 - \nu}{\frac{n^2 L^2}{\pi^2 r^2} - 1} \right)$$

(Equation 6-7)

Where:

- r = radius to neutral axis of shell in original formulation but for practical purposes can also be taken as the radius to outside of shell, in.
- L = length of tube between stiffeners, i.e., center-to-center spacing of stiffeners, in.
- t = thickness of shell, in.
- p_{cr} = collapsing pressure for FS = 1.0, psi
- E = modulus of elasticity, psi
- ν = Poisson's ratio
- n = number of lobes or waves in the complete circumference at collapse

The resulting graph for collapse of a free tube (from the von Mises instability formula) is illustrated in Figure 6-8. This figure provides assistance in determining the collapse, or buckling, of a free tube. Similar formulas and graphic presentations to determine buckling of a free tube have been developed by S. Timoshenko⁶ and W. Flügge.⁷

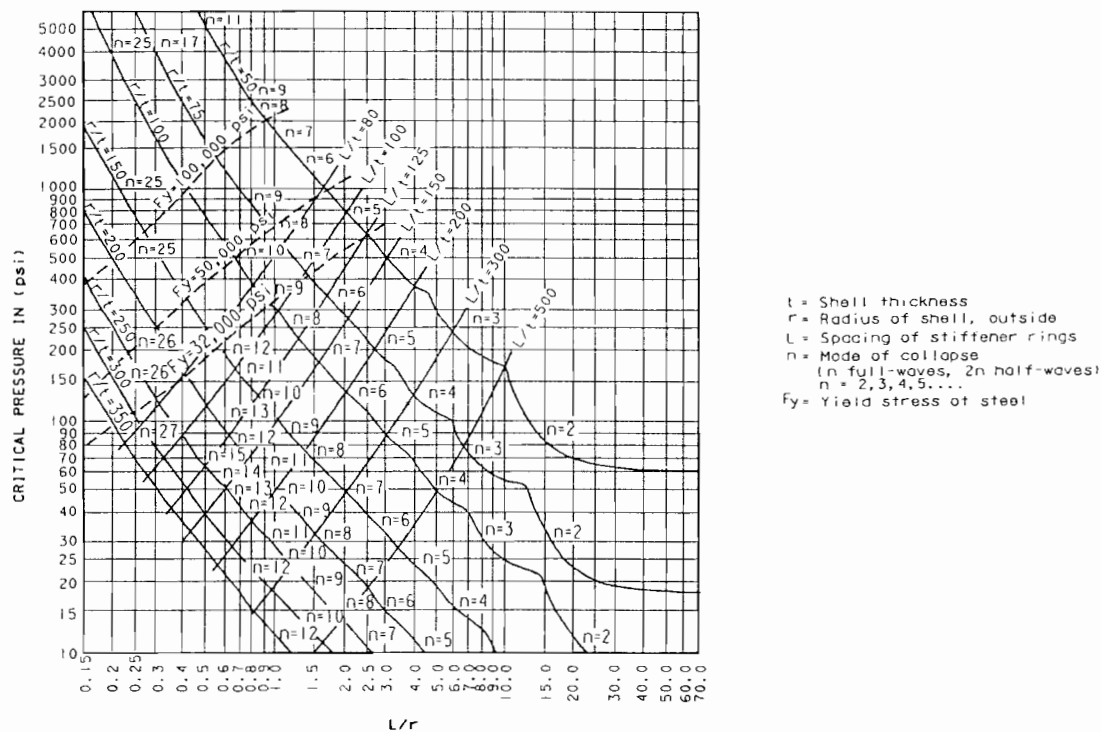


Figure 6-8 Collapse of a Free Tube (von Mises)

n is the integer number for which p_{cr} is a minimum, and is not an independent variable. n can be determined entirely by trial and error substitution; however, it is usually faster to estimate an initial n that is close to or equal to the final n by using available graphs and formulas. For most practical design cases, $6 \leq n \leq 12$. The following formula from D. F. Windenburg and C. Trilling's paper, "Collapse by Instability of Thin Cylindrical Shells Under External Pressure,"⁵ can be used to estimate n :

$$n = \sqrt[4]{\frac{(3/4)(\pi^2)/4(1 - \nu^2)^{1/2}}{(L/D)^2(t/D)}} \quad (\text{Equation 6-8})$$

Where:

- ν = Poisson's ratio
- L = length of tube between stiffeners, in.
- D = tunnel liner diameter, in.
- t = thickness of shell, in.

For steel penstocks with $\nu = 0.3$ the formula reduces to:

$$n = \sqrt[4]{\frac{7.061}{(L/D)^2 (\nu/D)}}$$

(Equation 6-9)

n can also be estimated using Figure 6-9.

When the value of n is calculated, it is rounded to the nearest whole integer.

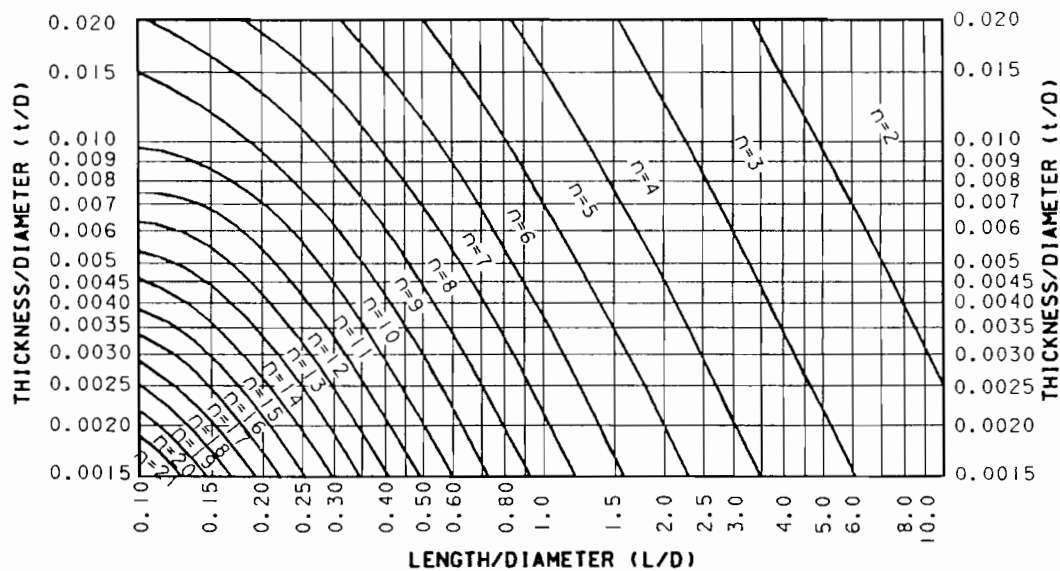


Figure 6-9 Estimation of n

The shell between stiffeners should be dimensioned to have a critical buckling pressure slightly in excess of the stiffeners. The same safety factor should be used for both the shell between stiffeners and for the stiffeners.

6.3.3 Design of Stiffeners

Circumferential ring stiffeners as analyzed by both E. Amstutz and S. Jacobsen provide the required circularity at the stiffener locations to prevent buckling of the shell. The Amstutz procedure for analyzing stiffeners is similar to the procedure he developed for unstiffened cylindrical shells. However, the value of ϵ at the critical external pressure is always determined to be well below the value of 3 at which 2α is greater than 180 degrees (see Figure 6-2), and Amstutz's equations are no longer valid (see Figure 6-3). The Jacobsen equations do not have this limitation, and a reliable answer is obtained with $2\alpha > 180$ degrees. For this reason it is recommended that the Jacobsen procedure be used to analyze and design stiffeners.

The Jacobsen paper "Buckling of Circular Rings and Cylindrical Tubes Under External Pressure"⁴ gives equations for determining critical buckling pressure on stiffeners. The three equations that must be solved simultaneously for the three unknowns α , β , and p are:

$$r/\sqrt{(12J/F)} = \left\{ \frac{[(9\pi^2/4\beta^2) - 1] [\pi - \alpha + \beta(\sin\alpha/\sin\beta)^2]}{12(\sin\alpha/\sin\beta)^3 [\alpha - (\pi\Delta/r) - \beta(\sin\alpha/\sin\beta)[1 + \tan^2(\alpha - \beta)/4]]} \right\}^{1/2} \quad (\text{Equation 6-10})$$

$$(p/EF)\sqrt{(12J/F)} = \frac{[(9\pi^2/4\beta^2) - 1]}{(r^3 \sin^3 \alpha) / [(J/F)\sqrt{(12J/F)} \sin^3 \beta]} \quad (\text{Equation 6-11})$$

$$\sigma_y/E = \frac{h\sqrt{(12J/F)}}{r\sqrt{(12J/F)}} \left(1 - \frac{\sin\beta}{\sin\alpha} \right) + \frac{p\sqrt{(12J/F)} r \sin\alpha}{EF\sqrt{(12J/F)} \sin\beta} \left[1 + \frac{8\beta h r \sin\alpha \tan(\alpha - \beta)}{\pi\sqrt{(12J/F)} \sqrt{(12J/F)} \sin\beta} \right] \quad (\text{Equation 6-12})$$

Where:

- α = one-half the angle subtended to the center of the cylindrical shell by the buckled lobe, degrees
- β = one-half the angle subtended by the new mean radius through the half waves of the buckled lobe, degrees
- p = critical radial external buckling pressure, psi
- J = moment of inertia of external stiffener and contributing portion of shell, in.⁴
- F = cross-sectional area of external circumferential ring stiffener and pipe shell between stiffener rings, in.²
- h = distance from neutral axis of stiffener to outer edge of ring stiffener element, in.
- r = radius to neutral axis of ring stiffener, in.
- σ_y = yield strength of liner and stiffener material, psi
- E = modulus of elasticity of liner and stiffener material, psi
- Δ/r = gap ratio, i.e., gap/liner radius

The contributing width of the shell is given by $1.57\sqrt{(rt)} + t_s$

Where:

- r = radius of liner shell, in.
- t = thickness of liner shell, in.
- t_s = thickness at base of external ring stiffener, in.

Jacobsen's formulas can be reworked to obtain three simultaneous, nonlinear equations as functions of α , β and p (or p_{cr}) which are set equal to zero for solution by numerical methods (see Moore's paper, which also presents a computer program for solving Jacobsen's equations for the critical buckling pressure on circumferential ring stiffeners²).

Some general rules or guidelines can be followed to help a designer initiate selection of an appropriate stiffener configuration. Sample guidelines are presented at the end of Moore's paper.

6.3.4 Dimensioning Weld Between Shell and External Ring Stiffener

The welds connecting the external stiffeners to the tunnel liner shell must be dimensioned to carry the shear at the section. The following expression from Jacobsen's "Steel Linings for Hydro Tunnels"⁸ is used to determine the maximum total shear (V) carried by the stiffener:

$$V = E(JJ) \left(\frac{\pi a}{l} \right) \left(\frac{1}{\rho^2} - \frac{\pi^2}{l^2} \right) \quad (\text{Equation 6-13})$$

Where:

- V = maximum total shear, psi
- E = modulus of elasticity of liner and stiffener material, psi
- JJ = moment of inertia of external ring stiffener and contributing portion of shell, in.⁴
- a = $\left(\frac{l}{\pi} \right) \tan(\alpha - \beta)$, in.
(α and β are determined by the computer program of Figure 13, of Moore's paper.)
- r = radius of tunnel liner, in.
- ρ = $r \frac{\sin \alpha}{\sin \beta}$ = new mean radius of lobe, in inches
- l = $\frac{\rho \beta}{1.5}$, where β is in radians and l is in inches

The unit shear for designing the welds at the base of the external stiffeners is then determined from the expression:

$$v = \frac{V Q}{(JJ) b} \quad (\text{Equation 6-14})$$

Where:

- v = unit horizontal shear at the base of external circumferential ring stiffener where it is welded to the shell, psi
- V = maximum total shear, psi
- Q = static moment of the area of external portion of stiffener alone about the axis of the total stiffener which includes a contributing portion of shell.
- JJ = moment of inertia of external ring stiffener and contributing portion of shell, in.⁴
- b = width of external stiffener at connection to shell, in.

It is recommended that the weld size adopted be capable of developing the full tensile strength of the connecting leg if angles or tee sections are used as external ring stiffeners.

6.4 Tunnel Liner Bends

Bends for tunnel liners have the same requirements as for exposed penstocks, as discussed in Section 4.5.

6.5 Seepage Rings

Seepage rings are used primarily to minimize seepage behind the steel liner section, and secondarily to create a longer seepage path for the potential reduction of external pressure (see Figure 6-10).

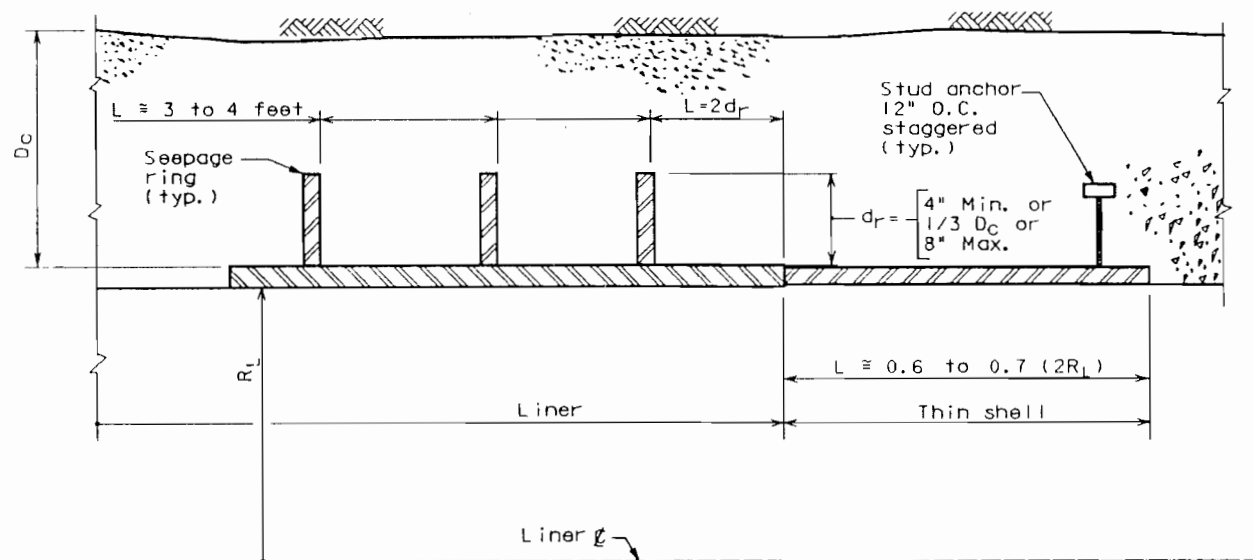


Figure 6-10 Seepage Ring and Thin Shell Configuration

Seepage rings usually are installed at or near the upstream end of the tunnel liner. When deemed appropriate by the designer, they can be used at other locations, such as near the powerhouse where the penstock enters the plant, along with separate seepage drainage below the penstock.

Although the primary purpose of the rings is to minimize potential seepage, it is recommended that consideration be given to the ring acting in conjunction with the shell and that a shell-ring design be done. Allowance should be made for higher acceptable stresses up to 90% of yield, based on the liner design condition. Low or intermediate strength material can be used, unless compatibility of welding material with the liner is a requirement.

One, two, or three seepage rings may be used, depending on designer preference. It is suggested that at least one ring be used for static head pressures up to 400 feet, two rings for pressures up to 800 feet, and three for pressures above 800 feet.

The radial depth of the stiffener ring must be about one-third the nominal thickness of the concrete encasing the steel liner, but not less than 4 inches or more than 8 inches. Ring thickness will depend on the ring/shell design analyses, but should be not less than 3/8 inch.

Seepage rings can be fabricated from plate sectors, bent structural angle sections, or bent structural "T" sections. Full-strength fillet welds customarily are used to attach the ring to the shell. However, full-penetration welds should be considered for attaching rings formed from plate sections.

Ring location and spacing may be dictated by the ring-shell analyses. As a rule of thumb, the first most upstream ring is located approximately two times the radial height of the ring from the end of the steel liner. The second and subsequent rings are located downstream approximately three to four feet away from the first ring and each other.

Recent steel liner designs for tunnels consider the installation of a thin liner shell as the most upstream shell. This thin shell is welded to the liner and provides the end flexibility needed in the liner to prevent the creation of a leakage/pressure gap at the liner-to-concrete interface because of potential radial movement of the concrete liner due to internal pressure. This shell, with an axial length approximately 0.60 to 0.70 times the nominal liner diameter, is designed to act as an impervious membrane to the concrete lining (just upstream of the steel liner). The thin shell is made of the same material as the steel liner and is attached to the liner with a butt weld. The thickness of the thin shell is a matter of designer preference and can range from 3/8 inch to 1/2 inch. Because it is thin, the shell must be anchored extensively to its encasing concrete using studs, hooked bars, "U" bars, or spirals. This ensures against a buckling failure from external pressure and also ensures that the shell will perform as a thin watertight membrane to the concrete lining as it expands radially outward due to the internal pressure. If a thin-shell concept is incorporated in the design, the seepage rings for the liner still should be located at the last liner section and not on the thin shell.

6.6 Tunnel Liner Joints

6.6.1 Welded Joints

Welded joints for tunnel liners are subject to the same requirements as for exposed penstocks, as discussed in Section 4.6.

6.6.2 Backed-Up Joints

Where clearance between the liner and tunnel wall is too restricted to permit adequate working access for welders and NDE technicians, the use of a backed-up weld detail is permissible (see Figure 4-10(C)). Backed-up weld joints are undesirable because they cannot be radiographed or ultrasonically inspected with precision, and the joint requires better fit-up and alignment than a joint welded from both sides. To avoid notches that could trigger a brittle fracture, splices in back-up bars must be full-fusion butt welds.

6.6.3 Grout Connections

Most liners require connections or penetrations for rock consolidation grouting and for contact grouting between rock and concrete, and between concrete and liner. These penetrations are sealed off after completion of all grouting operations. The actual details used vary considerably (see Figures 4-13 and 4-14). However, the $2\frac{3}{8}$ -inch-diameter hole in the penstock wall is the maximum that can be considered as self reinforced (ASME Code, Section VIII, Division 1, UG-36). Also, sealing the plug with a stainless steel plate is very effective in producing a permanent leak-proof seal.

6.7 Tunnel Liner Transitions

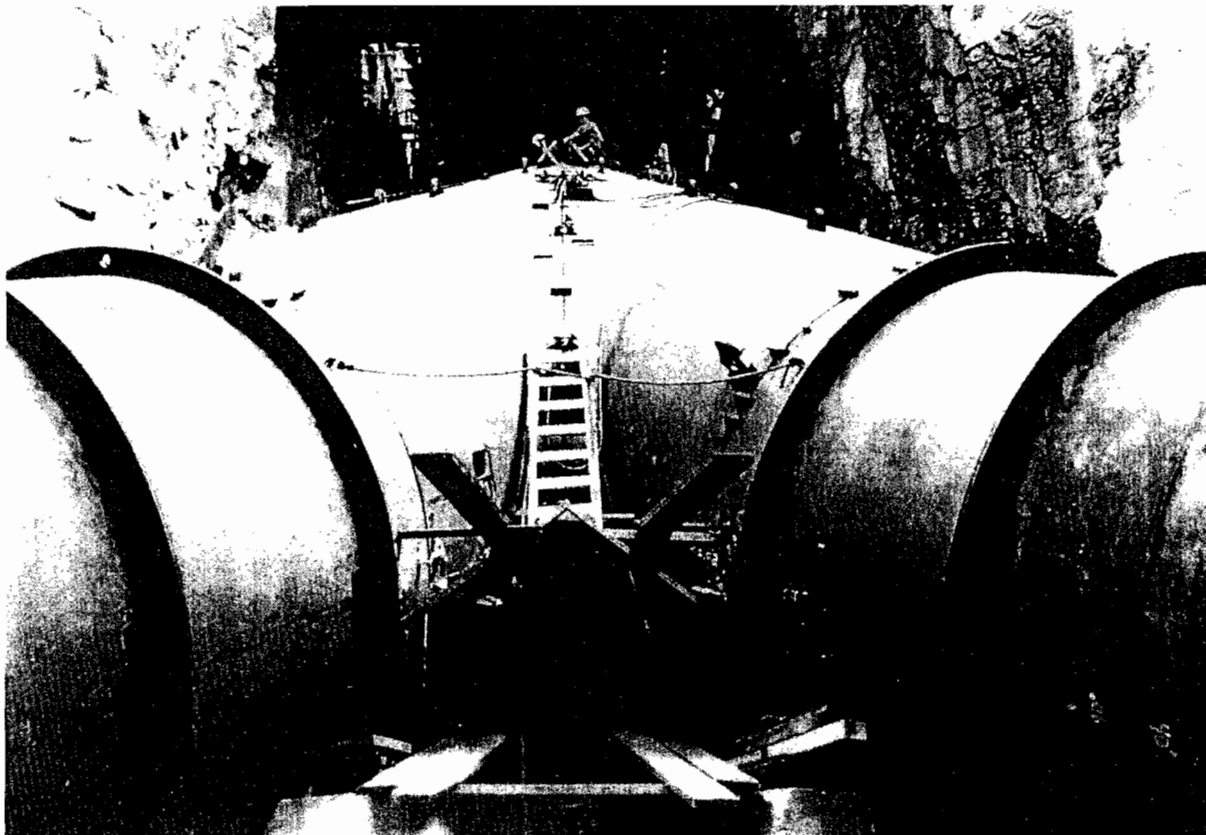
Transitions for tunnel liners are subject to the same requirements as for exposed penstocks, as discussed in Section 4.7.

References*

1. "Tunnels and Shafts." *Civil Engineering Guidelines for Planning and Designing Hydroelectric Developments, Volume 2*. ASCE/EPRI (1989).
2. Moore, E.T. "Designing Steel Tunnel Liners for Hydro Plants." *Hydro Review* (Oct. 1990).
3. Amstutz, Ing. E. "Buckling of Pressure-Shaft and Tunnel Linings." *Water Power* (Nov. 1970).
4. Jacobson, S. "Buckling of Circular Rings and Cylindrical Tubes Under External Pressure." *Water Power* (Dec. 1974).
5. Windenburg, D.F. and Trilling, C. "Collapse by Instability of Thin Cylindrical Shells Under External Pressure." *Transactions. ASME* (April 1934).
6. Timoshenko, S. *Theory of Elastic Stability*. McGraw-Hill Book Co., New York, NY (1936).
7. Flügge, W. *Stresses in Shells*. Springer-Verlag, Berlin (1960).
8. Jacobsen, S. "Steel Linings for Hydro Tunnels." *Water Power & Dam Construction* (June 1983).

* The most current version of a standard, code, or specification should be used for reference.

7 *BIFURCATIONS (WYE BRANCHES)*



Wye branches or bifurcations are installed to split or divide the tunnel or penstock flow for hydroelectric facilities with multiple units.

For underground facilities, wye branches are constructed of steel or reinforced concrete cast against a solid rock mass where the internal loading is transferred to the rock. For outdoor powerhouses with surface penstocks, wye branches usually are fabricated of steel and encased in a concrete anchor block to transfer the hydraulic thrusts to the surrounding foundation. Only steel wye branches are discussed in this section. When underground penstocks are steel lined, it is preferable to locate the steel wye branches so that they can be encased in concrete.

Wye branches must be suitably reinforced so that no substantial stress concentration or deformation occurs.

There are two major categories of bifurcating geometries: the straight symmetrical wye and the nonsymmetrical wye.

7 Bifurcations

The symmetrical wye may be a single symmetric bifurcation or a series of bifurcating pipes in which the branch pipes are parallel to the direction of the main pipe. Generally, when the bifurcating pipe is the straight symmetrical wye type (see Figure 7-1), the internal angle between the two branching pipes should range between 60 and 90 degrees.

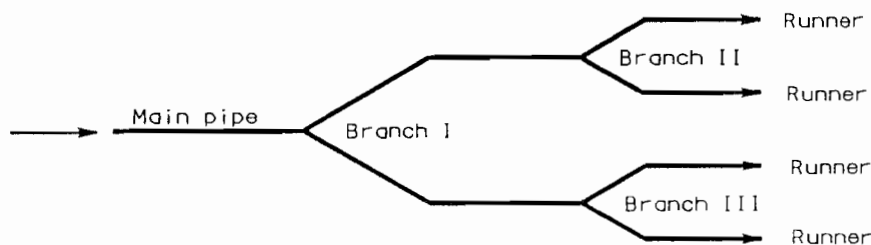


Figure 7-1 Straight Symmetrical Wye

Nonsymmetrical wyes distribute several branch pipes in the same direction from the straight main pipe, as shown in Figure 7-2. To reduce the head loss, it is advantageous to decrease the bifurcating angle. As the bifurcating angle decreases, the more reinforcing material is required at the bifurcating point.

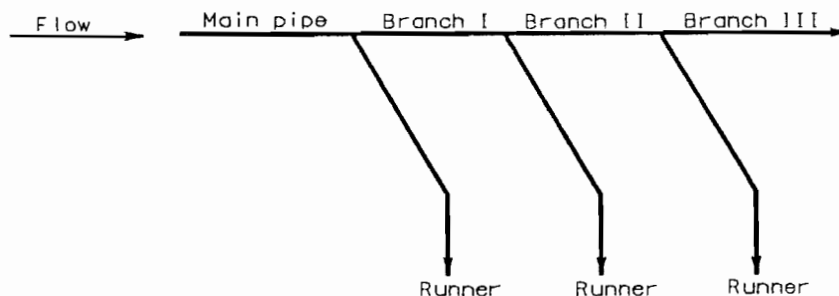


Figure 7-2 Nonsymmetrical Wye

7.1 Hydraulics

Wye branches must be designed for smooth hydraulic flow to avoid excessive head losses, vibration, and cavitation. They must be geometrically detailed to evenly proportion the flow distribution to eliminate acceleration or deceleration of flow in the adjoining branches and thus minimize head loss.

For example, if a wye branch were to branch into a 1/3 and 2/3 distribution, the 1/3 branch would have 1/3 the main penstock cross-sectional area and the 2/3 branch would have 2/3 the main penstock area (see Figure 7-3).

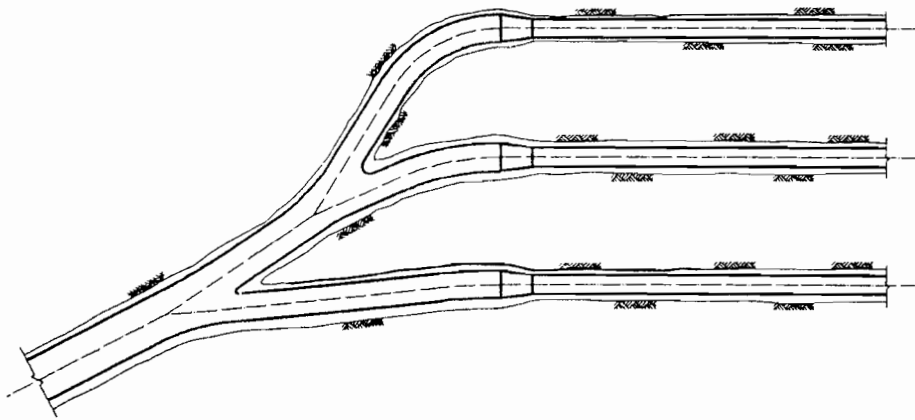


Figure 7-3 Wye Branch Proportion

Wye branches also must be detailed to direct the flow from the approaching wye branch leg to match the design flow capacity of the adjoining branches. For example, in a wye branch where the branches split into a $1/3$ and $2/3$ distribution, a plane projected from the crotch of the wye branch and parallel to the approaching leg centerline will divide its cross-sectional area into a $1/3$ and $2/3$ proportion (see Figure 7-4).

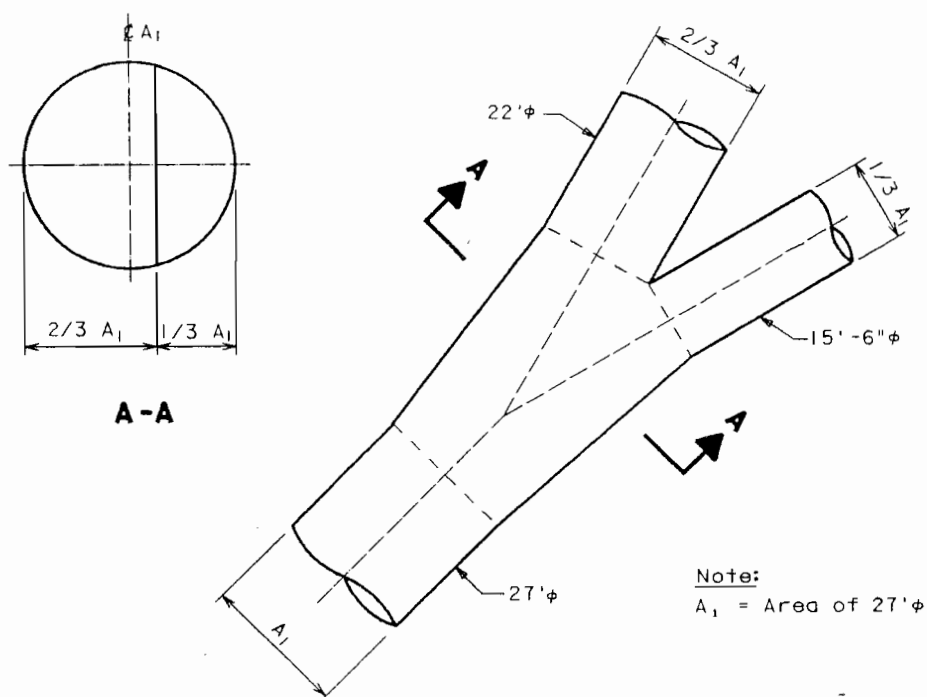


Figure 7-4 Wye Branch Approach Angle

7 Bifurcations

In case of manifold nonsymmetrical bifurcation, it is advisable to use a conical-shape design for the entrance into the branching pipes (see Figure 7-5 (B), (C) and (E)), rather than a true cylindrical shape (see Figure 7-5 (A) and (D)), because the head loss using the conical pipe is about 1/3 that of cylindrical pipe.

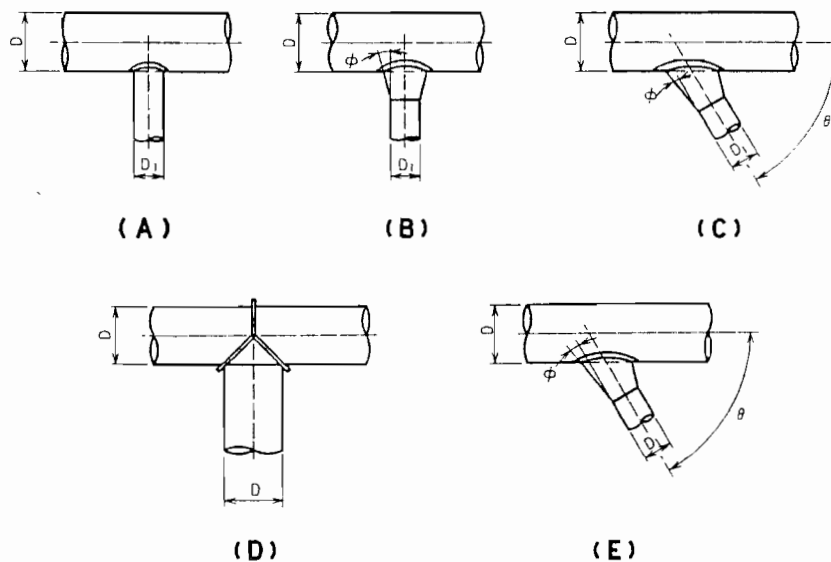


Figure 7-5 Externally Reinforced Wyes

The manifold type is subject to the possibility of different head losses for each of the branch pipes, while the combination of single or multiple symmetrical bifurcating pipes reduces this possibility. In either case, particular attention must be given to vortex formation and vibration for various combinations of the operating conditions of water turbines. Abrasion is reduced proportionally by the degree of absence of vortex formation.

In addition to the bifurcation configurations shown in Figures 7-1 and 7-2, it is possible to design trifurcated wye branches. This design makes the construction more complex. Although the head loss is negligible at the center pipe, it is substantial at the pipes on each side, causing uneven head loss.

Several different branches should be detailed, initial costs estimated, and the value of energy due to head loss calculated. The wye branch with the lowest total cost should be evaluated in light of hydraulic adequacy.

7.2 Stress Level Considerations

The shell of the wye and the reinforcing members should be designed for general membrane stress levels equal to the lesser of 2/3 of yield strength or 1/3 of ultimate strength to account for uncertainty in the action of the curved beams and stress concentrations. Surface stress allowables at points of geometric discontinuity are limited to 3 times the allowable membrane stress. Local membrane stresses may be 1.5 times the allowable membrane stress provided

they attenuate within $1\sqrt{rt}$ to a value equal to 1.1 times the general membrane allowable stress (where r is the inside radius of curvature of the wye at the point being analyzed, and t is the thickness of the wye at the point being analyzed).

Wye branches should be of welded construction, radiographically tested (RT) where possible, and stress relieved. When RT examination is not possible, ultrasonic testing (UT) should be performed. Post-weld heat treatment is required because the reinforcing members usually are thicker than $1\frac{1}{2}$ inches. Before acceptance, the wyes should be hydrostatically tested in accordance with Section 13.

7.3 Types of Wye Branches

The design of wye sections may include external stiffening only or may incorporate several modes of internal stiffening and supports.

7.3.1 Externally Reinforced Wyes

Externally reinforced and stiffened bifurcations utilizing a single reinforcing pad or insert plate should be limited to branches in which the branch diameter is less than $1/3$ the main pipe diameter. For cases in which the branch pipe intersects the main pipe at an angle other than 90 degrees, the longest dimension of the opening (b) in the main pipe must be less than $1/3$ of the main pipe diameter (see Figure 7-5 (A), (B), and (C)).

A single curved reinforcing plate may be used provided the cutout area is replaced in the reinforcing plate and the out-of-plane bending stresses at the elongated opening are properly absorbed in the reinforcing plate section (see Figure 7-5 (E)). In addition, $2/3$ of the cutout area must be replaced within $.5\sqrt{rt}$ of the edge of the opening (where r is the inside radius of curvature of the wye at the point being analyzed, and t is the thickness of the wye at the point being analyzed). The limit of reinforcement normal to the longitudinal axis of the penstock is two times the shell thickness.

The two- and three-plate reinforcement designs (see Figure 7-5 (D)) use external stiffeners or a combination of external stiffeners and internal tie rod. The discontinuity loads as determined by the geometry of the bifurcation must be absorbed by the external stiffener space frame. The design of the space frame must incorporate the appropriate curved beam factors and must absorb the axial load introduced by the geometric discontinuity of the wye.

For the external stiffeners alone, the circumferential stress due to hydrostatic pressure must be taken by the bending in the stiffeners. For this design, the stiffeners act to resist the internal pressure and do not penetrate the shell.

The longitudinal forces in the bifurcation must include both the test case value of $.5 Pr$ and the operating case value of Poisson's ratio, $.3 Pr$ (where P is the appropriate pressure, and r is the radius of the wye). No credit is permitted for the encasing concrete.

For the system that uses a combination of external stiffeners and an internal rod, a single tie rod is added. The rod penetrates the shell section and acts as a tension member. The rod is used to reduce the stiffener size that would be required if only external stiffeners were used. The effects of corrosion and erosion must be factored into the selected cross-sectional shape.

However, this combination system is not recommended because of flow-induced vibration due to vortex shedding off the tie rod. These vibrations can result in fatigue failures at points of stress concentrations, especially at the point where the tie rod penetrates the wye branch shell.

For certain high-pressure applications, the dimensions of the external stiffening become restrictive. The required depth and thickness of the stiffeners results in an uneconomical or impractical configuration. Also, the size of the excavation for the wye is influenced by the dimensions of the external stiffening.

A recommended alternative is the spherical type of bifurcation, which may be advantageous for high-pressure applications. A spherical header branch system is shown on Figure 7-6. The head loss at the branch depends upon the ratio of cross-sectional area of main and branch pipes, branching angle, ratio of flow distribution, flow, and other factors.

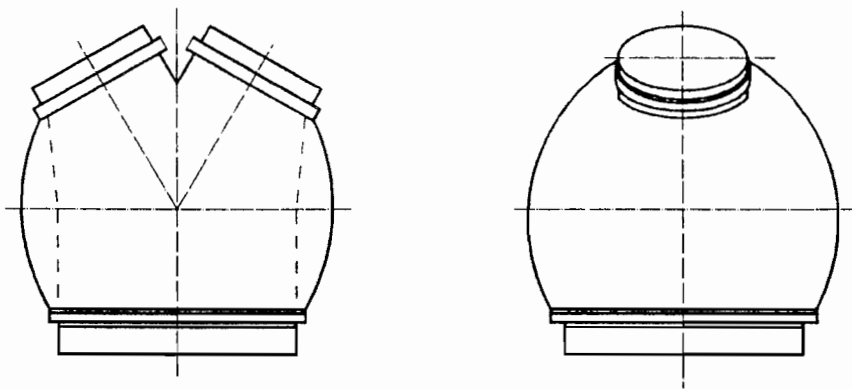


Figure 7-6 Spherical Wye

For a spherical header branch system, flow-rectifying plates are provided inside the spherical shell interior to reduce the head loss resulting from the suddenly increased inner volume at the sphere part. The flow-rectifying plates are provided with sufficiently large water-pressure equalizing holes to transmit the impact of flowing water pressure to their exterior. The flow-rectifying plates should not be rigidly welded to the spherical shell, because such welding would restrict the deformation of the sphere due to water pressure.

Circumferential reinforcing rings are required at the joining part of the branch and the spherical shell. The reinforcement of the opening must be designed under the ASME Code, Section VIII, Division 1 requirements for large-diameter openings. Limits of reinforcement and compact reinforcement rules also must meet the same ASME Code requirements.

A common variation of the spherical header incorporates conical and knuckle transitions between the main pipe and the spherical header, as well as similar transitions between the branch pipes and the spherical header. This configuration eases the discontinuity stresses at the openings in the spherical element by use of transition members. This shape generally carries most of the stresses in membrane tension and substantially reduces surface bending stresses. An example of this configuration is shown in Figure 7-7. Finite element analyses techniques are recommended for this type of configuration.

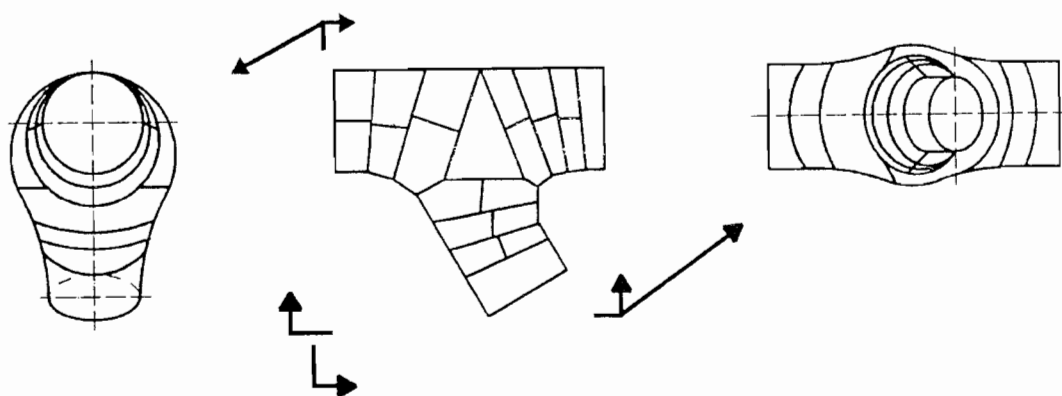


Figure 7-7 Spherical Wye Variation

7.3.2 Internally Reinforced Wyes

In the one-plate reinforcement design, the reinforcement extends completely through the interior of the shell section and acts as a member that carries both bending and tensile loads. This type of system may be hydraulically deficient and cause a pulsing flow with acceleration and deceleration as the water passes the internal crotch plate. This in turn may create an excessive amount of head loss.

There are, however, two direct solutions to the problem. One is the well-known internal single-member solid splitter plate in the form of a crescent. The other is the composite double-hollow splitter wye branch.

7.3.2.1 Internal Single-Member Solid Splitter Plate

For the internal single-member splitter in the form of a crescent, the crescent must be shaped so that it is in tension only (see Figure 7-8). The centroid of the adjacent load must be coincident with the centroid of the crotch plate to justify the use of the tensile-stress-only concept.

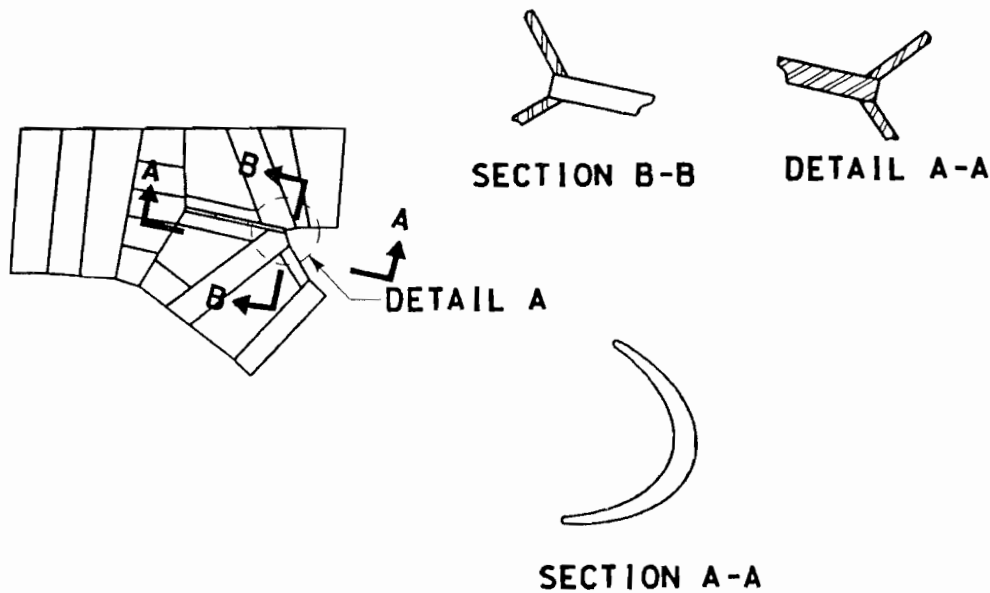


Figure 7-8 Internally Reinforced Wye

The inward projection of the splitter plate allows the designer to minimize the eccentricity between the centroid of the load and the centroid of the reinforcing cross section splitter plate. Ideally, this eccentricity will be reduced to zero, resulting in a tensile-stress-only condition in the splitter plate since the bending moment has been eliminated. In most cases, the eccentricity is not completely eliminated and some bending occurs. In these cases, the bending stress must be multiplied by the appropriate curved beam factor in establishing the true total stress.

To achieve this condition, the inward projection of the splitter plate substantially affects the head-loss flow characteristics. To compensate for this undesirable flow constriction, the main pipe may be flared to increase the cross-sectional area in the region of the splitter. The use of conical sections in both the main pipe and conical reducers extending to the branch pipes ensures minimal splitter plate design coupled with favorable flow and head-loss characteristics.

7.3.2.2 Composite Double-Hollow Splitter Wye Branch

The geometric considerations described above for the single-member solid splitter plate also apply to the composite double-hollow splitter wye branch (see Figure 7-9). The inward projection of the splitter reduces the eccentricity of the load and reinforcement centroid. The double-hollow composite splitter enables the designer to minimize the steel plate thicknesses, thereby eliminating costly post-weld heat treatment. Reinforced concrete may be used in the hollow-splitter space and may act in conjunction with the steel elements. The splitter plate must incorporate the longitudinal loads in the branches for both the test case and the operating case. The curved beam factors must be used for the bending component of stress.

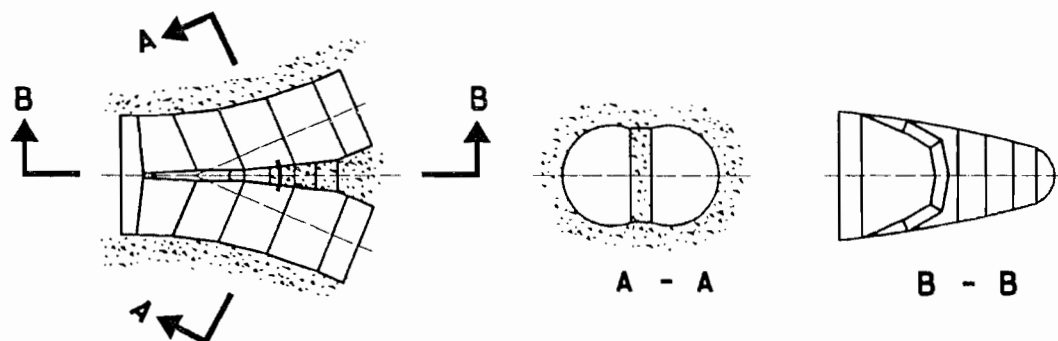


Figure 7-9 Internally Reinforced Wye (Double-Hollow Splitter Type)

As with the single-element splitter, the inward projection may minimize the splitter cross-section requirements, but at the expense of restricting the flow and inducing undesirable head losses. To eliminate this situation, the main pipe may be flared by use of conical sections to increase the cross section at the splitter location. Hydraulic studies are recommended to determine the head loss and flow characteristics on an individual design basis.

7.4 Design Methods

The quasi-space frame concept, in which a portion of the shell acts with the reinforcing ribs, is prevalent in the current published literature. Most complicated wye configurations warrant an analysis by use of the finite element method. Element size, integration points, and node points must be selected to represent the stress as accurately as possible. The shell plate to reinforcing rib junction should have a fine mesh.

Current design practices use one of the existing design guides, a more detailed type of analysis such as three-dimensional finite element analysis or a combination of both.

The design method developed for the two- and three-plate wye reinforcing clamp section involves estimating the amount of plate needed for the reinforcement and applying the resultant circumferential and longitudinal forces from the pipe shells on the stiffener section.

Detailing of wyes must be a consideration in the design of the reinforcing clamps. When dealing with a combination of large diameters and high pressures, a single reinforcing web plate may not yield a moment of inertia sufficient to control stress levels, or the height-to-thickness ratio of the single stiffener may be excessive. A flange must be installed on the reinforcing web member for excessive ratios. The increased moment of inertia required may be significantly offset by installing flanges on the outside of the stiffener. Conversely, fabrication becomes more difficult by installing flanges as insert plates within the shell.

In a tee section, the curved-beam equations then can be used to calculate the critical plate stresses. It is imperative that the appropriate curved beam factor (K) be incorporated into this design. In addition, the internal pressure membrane stress in the shell at the shell-to-web intersection must be added to the tensile and bending C-girder stresses.

The technique for C-girder analysis incorporates an idealization of the web-to-shell intersection as a tee configuration throughout the periphery. However, to be more correct, this configuration should be calculated on the basis of the true geometry, i.e., a sweptback V. The resulting moment of inertia, section modulus, and curved beam factor may be significantly affected at the critical horizontal plane cross section if this refinement is not reconciled. Naturally, if a true tee configuration is used, the above-described refinement need not be considered.

When the wye branches intersect at angles other than 90 degrees from the pipe axis, one plate will have a larger load than the other. Therefore, by compatibility, the calculations must include balancing the deflections of both plates at the junction to determine the amount of load distributed to each plate.

In dealing with larger diameter pipe, the economic design for controlling deflections is the three-plate design in which a half-ring plate is added to the two-plate design to hold down the free ends. The third plate is not attached to the shell, as are the other two plates, but is free except at the top and bottom intersection points with the other two plates.

7.5 Additional Considerations

Careful attention should be given to the reinforcing rib to shell plate details to satisfy the design requirements and reduce overall cost of the wye branch. The external clamp system is probably the most efficient structurally and hydraulically for low- to medium-head large-diameter wyes. High-head and large-diameter wyes are better served with internally projecting crescent plates or composite internally reinforced wyes.

References*

The following references are not cited in the text.

Bier, P.J. "Power Penstocks." *Water Power* (June, July, and August 1958).

Bier, P.J. "Welded Steel Penstocks—Design and Construction." *Engineering Monograph No. 3*. U.S. Bureau of Reclamation, Denver, CO (1966).

Gaylord, E.H., Jr. and Gaylord, C.N. *Structural Engineering Handbook*. McGraw-Hill Book Co., New York, NY (5th Ed., 1975).

"Welded Steel Penstock and Tunnel Liners." *Steel Plate Engineering Data—Vol. 4*. American Iron and Steel Institute and Steel Plate Fabricators Association, Inc. (1984).

* The most current version of a standard, code, or specification should be used for reference.



8.1 General

8.1.1 Types of Penstock Support

A penstock can be supported in a variety of ways, depending on the initial design selected, existing geologic conditions, and penstock profile. The penstock can be totally buried, partially buried, or supported aboveground.

Generally, a penstock is totally buried only when either access or drainage is required over the penstock, when the penstock requires protection, such as from falling trees, or when dictated by economics. The soil cover protects the penstock against the effects of temperature variations, and no expansion joints are required. A partially buried penstock can be very economical if the slope is stable and not more than 30 degrees. However, temperature loads on the penstock must be considered. For both totally and partially buried penstocks, corrosion due to contact with the soil can be a problem and must be looked at very carefully. Additionally, penstocks that

are buried are difficult to inspect or repair. An aboveground support system is the most commonly encountered installation and allows a better handling of inspection and repair access and corrosion problems. However, the design must take into consideration the accumulation of longitudinal loads at the supports, and expansion joints usually are required.

8.1.2 Locations and Purpose of Anchors and Piers

Penstocks installed aboveground normally are supported using piers with a spacing of 20 to 100 feet. The spacing depends on the diameter of the penstock, the subsurface bearing strength at the support, the type of end supports when not continuous, allowable coupling deflection, and the constructibility of the penstock at the site. It is preferable to span as far as practical, but the cost of the additional steel to handle bending stresses and concentrated loads at supports may exceed the cost of installing piers. These two expenses must be balanced to obtain the most economical penstock. Piers are designed to support the dead load of pipe and contained water, and resist the longitudinal forces resulting from temperature change, friction, and circumferential stress. Also, lateral loads from wind or earthquake may need to be considered.

Anchors are installed at bends to resist hydrostatic loads and the accumulation of longitudinal loads, to prevent shifts in the pipeline during installation, and to resist vibrational forces that tend to cause displacement in the penstock. Generally, expansion joints in the penstock are installed to eliminate temperature loads and longitudinal stress. If no expansion joints are used, these longitudinal loads will be high and must be considered in both the pipe stress and anchor design.

When expansion joints are used, a maximum spacing of 500 feet between anchors normally is used because of the accumulation of longitudinal forces, the need for fixed points during erection, and the need to control vibration during operation.

8.1.3 Geologic Conditions

Geological considerations and the remedies for adverse conditions are outlined in Section 1.11.

Movement of the foundation at the piers and anchor blocks is not desirable and can cause excessive vibration and penstock overstress. Sound, stable rock provides the best foundation for anchor blocks and piers, and if economically feasible, the penstock alignment should be located at or near surface rock during the initial layout phase. If sound rock cannot be found or reached for the foundations, and "softer" earth materials (e.g., weak or weathered rock and soil) will comprise the foundation, settlements must be calculated and appropriate provisions made in the penstock design. Provisions for ground improvement, e.g., piles, may be necessary to attain suitable foundation conditions.

The surface and subsurface geologic conditions at the pier and anchor block sites should be investigated thoroughly by experienced geologists to verify that the natural foundation satisfies the design requirements or that foundation improvements are necessary to obtain satisfactory pier and anchor block installations. Weathering profiles can be very erratic and must be carefully evaluated. Misjudgments about the weathering profile can result in over- or under-excavations for the foundations and may necessitate redesign of the supports. The bottom of the foundations should be below the frost line. The geology should be clearly described and

portrayed in the construction document drawings to provide the contractor with all available information about the geologic characteristics of the foundation.

At the inlet of the penstock, provisions must be made to prevent leakage of water, which can change geologic foundation conditions over time. For example, acidic water can dissolve carbonate and sulfate rocks, and continuous seepage can cause surface and subsurface erosion of soil materials. This may require underdrains to divert leakage, slope protection to prevent erosion, and a regular inspection program to check for changing conditions.

8.1.4 Foundation Stability

For sliding and overturning stability analysis, the factors of safety shown below are recommended minimums. The foundation material (soil, rock) resistance values used in this analysis should be those determined by a qualified geotechnical engineer to be appropriate allowables for design use for the specific site. Higher factors of safety are required when difficult foundation conditions are encountered or when limited subsurface investigation has been performed.

8.1.4.1 Sliding

Following are values for the minimum factor of safety against sliding for a particular condition.

- (1) Static pressure in-service conditions: 1.5
- (2) Dynamic conditions, transient pressures resulting from load rejection: 1.3
- (3) Seismic loading (DBC) combined with static pressure or test pressure: 1.2
- (4) Extreme conditions, Dam Safety Criteria earthquake: 1.05

The resistance against sliding is a function of the sliding coefficient of friction times the vertical load, shear key if used, rock anchors if used, and passive soil resistance.

Geologic investigations should be made prior to finalizing the value of the sliding coefficient of friction. See Section 8.1.5 for typical friction values. These investigations also should be used in determining shear values of rock used for shear keys, bonding of rock anchors, and values to be used for passive resistance.

When shear keys are used, considerable care must be taken during excavation to ensure that the rock is not weakened by fracturing the rock. Once final rock excavation is complete, the rock surface should be sealed with either a gunite cover or thin concrete seal slab to prevent contamination. Prior to sealing, loose material should be removed by either an air or water jet.

When using a passive soil resistance, the designer must recognize that movement of the soil is required to develop passive resistance. If this movement cannot be tolerated, then at-rest soil resistance must be used.

8.1.4.2 Overturning

Following are values for the minimum factor of safety against overturning for a particular condition.

- (1) Static pressure in-service conditions: 1.5
- (2) Dynamic conditions, transient pressures resulting from load rejection: 1.3
- (3) Seismic loading (DBC) combined with static pressure or test pressure: 1.2
- (4) Extreme conditions, Dam Safety Criteria earthquake: 1.05

The resistance against overturning is a function of vertical loads, passive resistance if applicable, and rock anchors if used. The limitations on the use of passive resistance noted above in the sliding case also apply here. When rock anchors are used, their total contribution must not exceed one-half the contribution of the gravity loads. A mass that will provide at least a factor of safety of 1.0 is required to control vibrations in the penstocks.

The bearing stresses should be calculated, and the bearing capacity of the foundation material should be verified by geologic investigation (see Section 8.1.5).

8.1.4.3 Hydrostatic Uplift

Uplift forces commonly are not encountered; however, if there is uplift (e.g., anchors under water), the combination of vertical loads and rock anchors must yield a safety factor greater than 1.2. These forces also must be considered when checking sliding and overturning stability.

8.1.4.4 Movement at Anchors and Piers

Normally a penstock system is designed to have no foundation movement. All movements in the penstock should occur at couplings and expansion joints. The potential for movements at the piers and anchors should be considered in the total penstock assembly. If movements and settlements cannot be controlled in the foundations, the anchors and piers should be detailed with bolted connections that will allow penstock realignment.

8.1.4.5 Slope Stability

After designing the piers and anchors, it is essential that an overall slope stability analysis be performed on the supporting medium of the piers and anchors. Proper consideration also must be given to slope protection. A major washout of the slope due to penstock leakage or natural causes may cause the anchor to fail. Armoring of earth backfill with riprap or slope paving may be required. (See Section 1.11.)

8.1.5 Foundation Strength

Table 8-1 gives the estimated foundation strengths of various materials. These values should be checked by testing before final design is completed.

Table 8-1 Foundation Strength of Material

MATERIAL	LOAD CAPACITY IN BEARING (TONS/FT ²)	SHEAR CAPACITY ON KEYS (TONS/FT ²)	STATIC FRICTION COEFFICIENT "f", CONCRETE ON FOUNDATION MATERIAL
SOFT CLAY	1.0	0.5	0.30
HARD CLAY	1.5	1.0	0.33
LOOSE SAND AND GRAVEL	2.0	1.5	0.40
CONFINED GRAVEL	3.0	2.5	0.50
SOFT ROCK	10.0	5.0	0.60
HARD ROCK	40.0	20.0	0.65

8.1.6 Codes, Allowables, and Methods

Reinforced concrete supports should be designed in accordance with ACI 318,¹ using the ultimate strength method. For this method the design strength of the support must be that obtained using the load factors specified in the ACI Code for the loading conditions in Sections 8.2.2 and 8.3.2.

Where it is deemed economical by the designer, the ACI 318 alternate design method (working stress) may be used.

8.2 Anchors for Penstocks

8.2.1 General Theory

Anchorage are major components of a penstock system and are usually necessary at bends in the penstock center-line alignment. The functions and dimensions of any anchor depend on how adjacent nonrigid girth joints are distributed along the penstock barrel. Such joints include sleeve couplings, expansion joints, or other rubber-gasketed joints. These joints divide the barrel shell into separate units which are water-tight, but structurally discontinuous. For a penstock in service, directional and dimensional changes along its length will concentrate hydrostatic forces at the nonrigid joints. For both aboveground and buried penstocks, these unbalanced forces must be resisted; an external anchorage on the penstock barrel is the most efficient means of resistance.

When water is in a penstock, the transverse components of internal pressure are simply contained by hoop tension. In contrast, axial components developed by pressure and velocity become vectors, which are described by centerline alignment, the degree of fluid pressure, and cross-sectional area. Unbalanced axial vectors result wherever any of these three factors change, as at penstock bends or at reducer sections and line valves. These unbalanced vectors are assumed fully restrained within the vessel wall wherever the penstock barrel is made

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structurally continuous either by fully welded rigid girth joints or tightly bolted flanged joints. Otherwise, at each nonrigid joint two opposing axial pressure force vectors are mobilized normal to the joint plane. This is analogous to two hydraulic jacks pushing on the joint face and moving the joint faces apart. This jacking force must be resisted by an anchoring device.

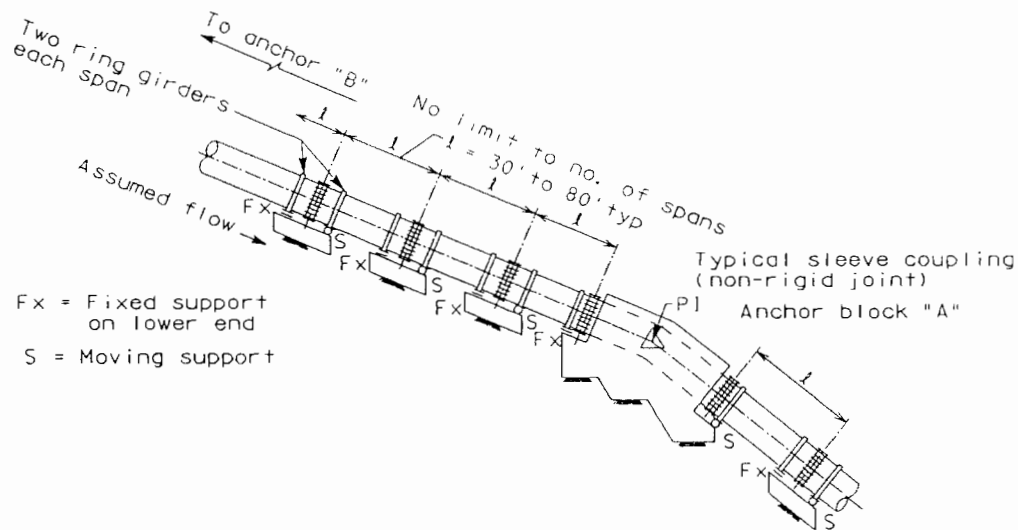
A typical design for a penstock installed aboveground will use sleeve-type couplings or single expansion joints between anchor blocks. Both designs are shown in Figure 8-1. All nonrigid joints effectively sever the penstock into free body reaches extending between the two joints. A typical bend is anchored between two straight penstock legs, and the axial pressure force vectors act normal to the joint planes on each side of the anchor, intersecting at the PI of the bend. Their vector resultant is the major anchor load.

An alternative aboveground design might utilize a penstock barrel which includes a greatly reduced number of nonrigid girth joints (less than one for each bend). This has a cost advantage in reducing the number of fixed concrete anchors. Conceptually, this penstock could absorb thermal and other movements by allowing free flexure of the barrel between major anchorages. This is feasible for smaller diameters, and successful installations have been made. The theory is included in the referenced work by R. Bouchayer.² Situations are limited for which this alternative design may be practical.

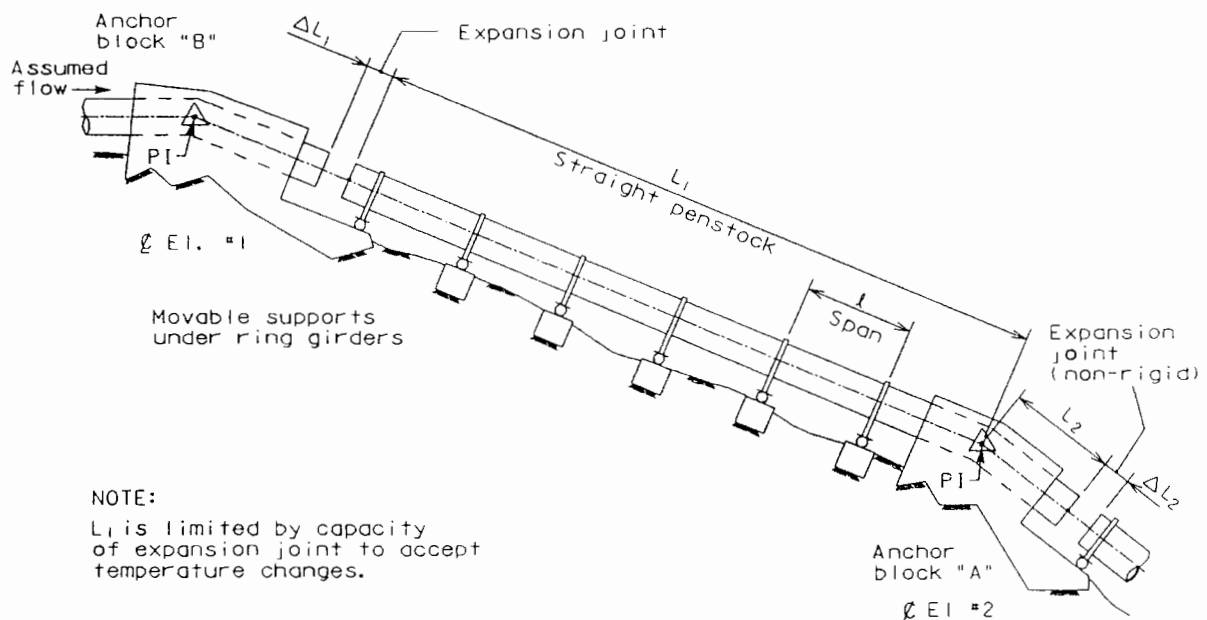
For a buried penstock design, the field joints are often welded and do not require a specific anchorage. nonrigid buried joints are commonly sealed by rubber O-rings, or sometimes by sleeve-type couplings. Restraint of unbalanced vectors at nonrigid buried joints is equally as important as for above-ground situations. Earth friction and earth bearing capacity are used as major anchoring forces on a buried penstock. It is often possible to achieve the necessary anchorage using the referenced thrust restraint design.³ If concrete is used, bend thrusts may be uniformly distributed from the curved barrel into the earth material.

Installation of a true expansion joint (in contrast to a bolted sleeve-type coupling) embedded in earth backfill is unlikely because of the reduced exposure to temperature variations and also due to the impracticality of allowing a pipe to slide longitudinally in the earth backfill.

The above general concepts are illustrated in Figures 8-2 through 8-4. The points of application for pressure vectors are shown along with the gravity and friction loads acting on an anchor.



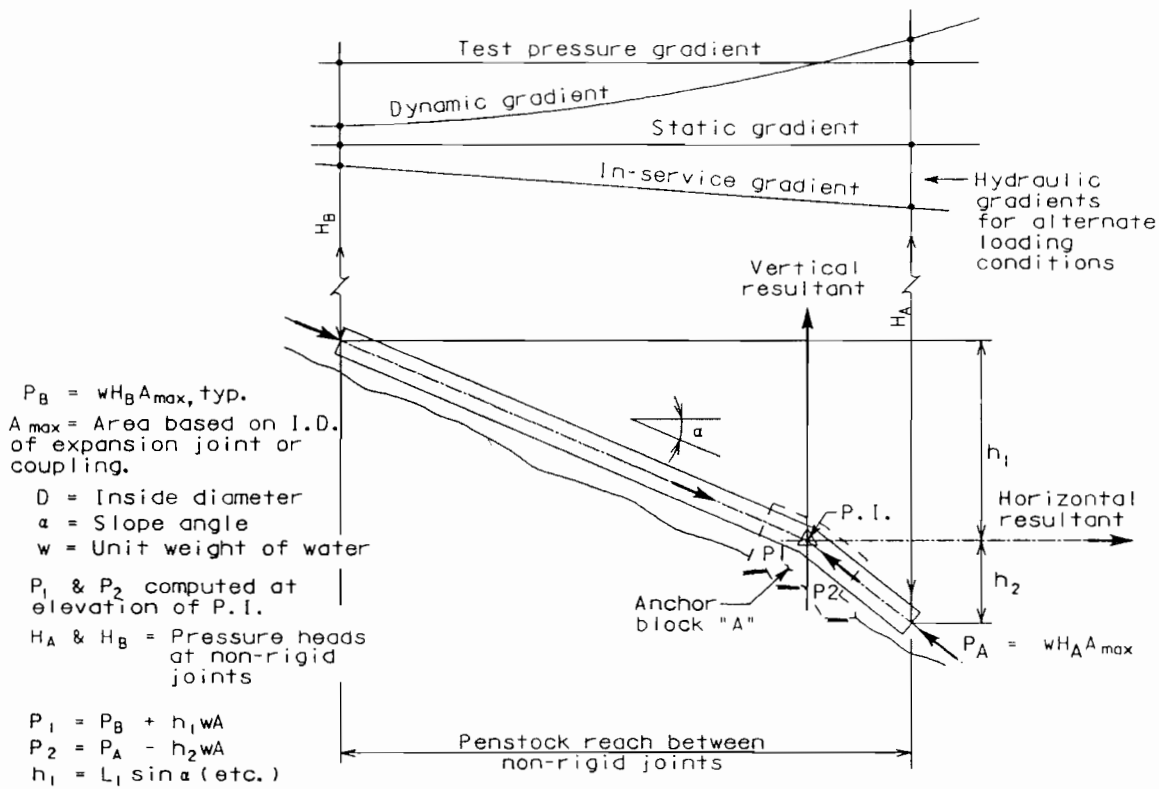
(A) PENSTOCK WITH SLEEVE COUPLINGS (PROFILE)



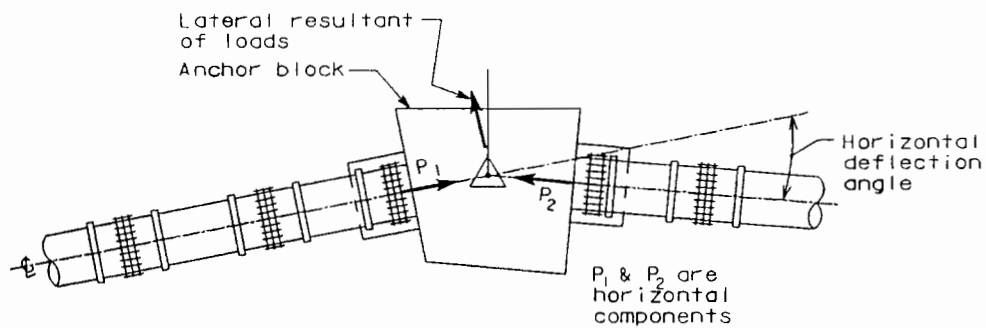
(B) PENSTOCK WITH EXPANSION JOINTS (PROFILE)

Figure 8-1 Anchor Details of Typical Aboveground Penstocks Near Anchors

8 Anchor Blocks and Piers

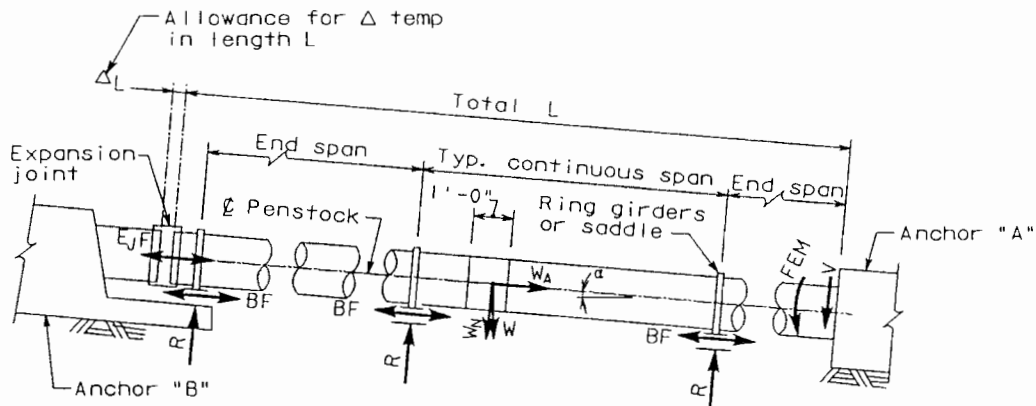


(A) PENSTOCK PROFILE - DETAIL OF PRESSURE LOADS

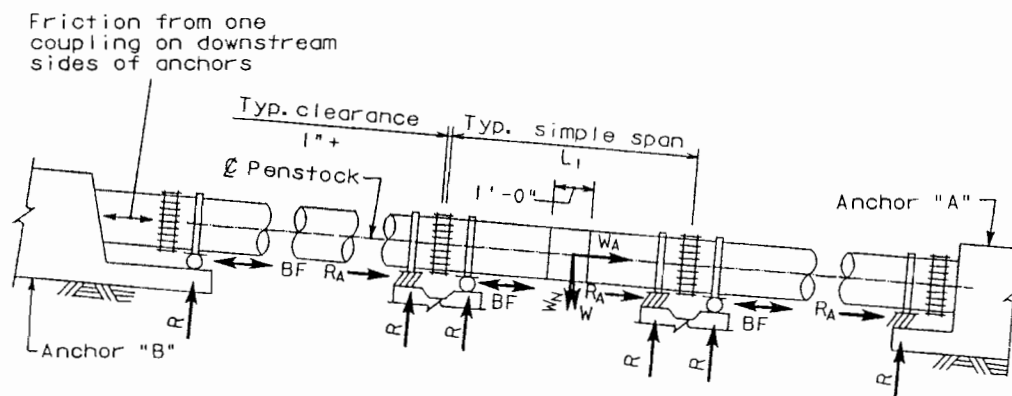


(B) PLAN SHOWING LATERAL RESULTANT

Figure 8-2 Anchor Block Pressure Forces



(A) PENSTOCK WITH EXPANSION JOINTS



(B) PENSTOCK WITH SLEEVE COUPLINGS

NOTATION:

(All loads and reactions shown are axial or normal to ℓ of penstock.)

W = Total gravity weight of water and steel in 1 foot of penstock

W_N = Component of W normal to ℓ of penstock (equals $W \cos \alpha$, etc.)

W_A = Axial component of steel W only

(Water component is absorbed into P_1 & P_2 at P.I.)

R = Load at each support

$R_A = \Sigma W_A$ for one span

BF = Bearing friction

$E_j F$ = Expansion joint friction

FEM & V = Fixed loads on anchor

Notes:

1. BF and $E_j F$ do not depend on Δ temp.

2. Coupling friction assumed same as expansion joint. Apply in most adverse direction.

Figure 8-3 Penstock Loads Affecting Anchors

8 Anchor Blocks and Piers

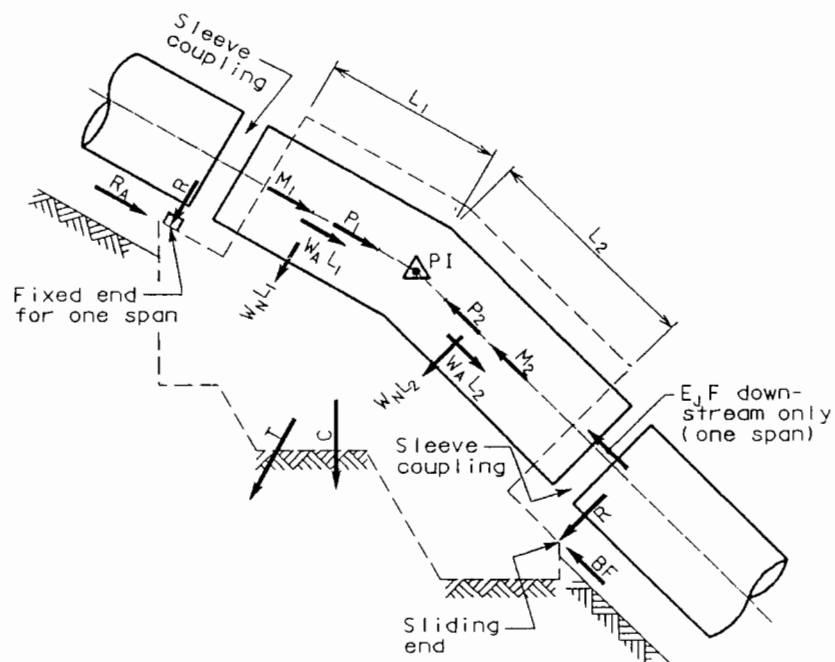
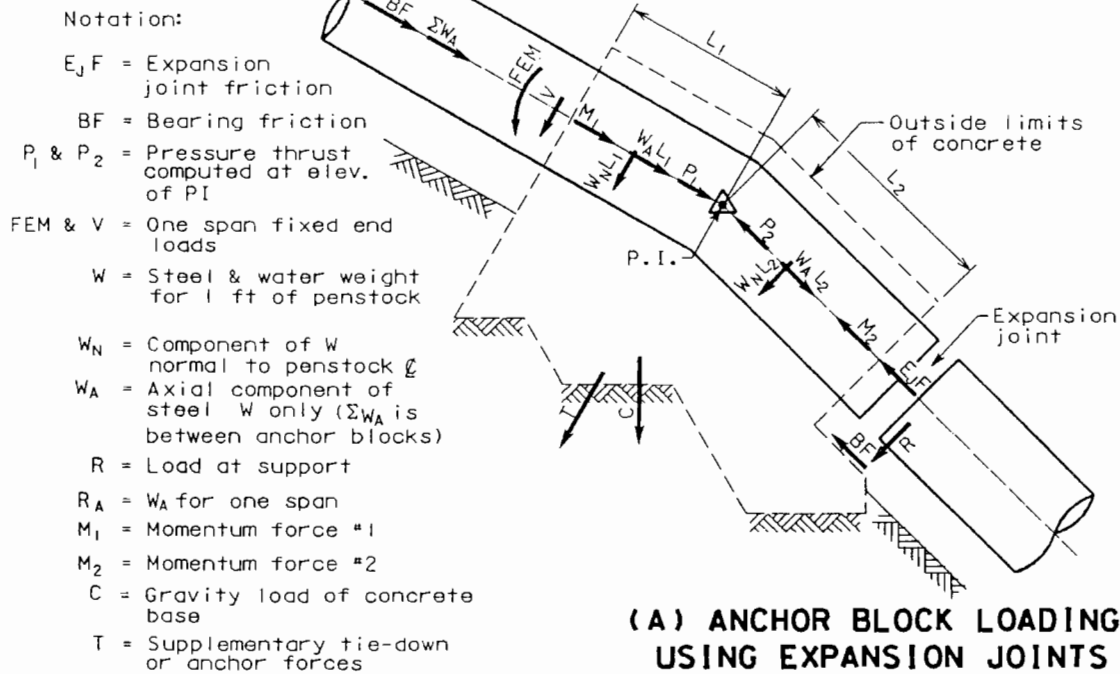


Figure 8-4 Anchor Block Loading

8.2.2 Loads Acting on Anchor Blocks

8.2.2.1 Loading Combinations

Loads that act on anchor blocks are combined in a manner similar to the loading combination system assumed for penstock barrel design (see Section 3.2). For each anchor design, the combinations listed below begin with the most constant and probable loadings. These are then incremented by loads of lower probability.

There are at least four groups of loading combinations for anchor block design that allow an efficient design system (see Table 8-2). Group A comprises the most constant group. The combination of group A with group B is the basic design load for any anchor design. Additional combinations are obtained by separately adding loads from groups C and D to the group A loads. Using the basic anchor design, the factor of safety under each combination of the second group of loads must be greater than the minimum specified in the design criteria. Otherwise, the anchor block must be revised to meet the criteria. Code symbols (DL1, DL2, etc.) are taken from the nomenclature used in Section 3.2.

8.2.2.2 Components of the Loading Groups

(1) Gravity loads

These include moments and beam-type end reactions from adjacent penstock spans along with the weights of all materials embedded within the anchor block. Resolving load components in axial and normal directions simplifies analysis. The designer may add the axial component of the weight of penstock water into the static pressure used at the bend PI. If earth loads are used to assist anchor stability, the effects of erosion or future excavations must be considered. Live loading on an anchor is rare.

(2) Internal pressure loads

Axial pressure vectors are proportional to the array of static pressures described in Section 3.2. The basic design condition includes the largest safety factor because typical power failure in a pumping system or load rejection at a generating plant may occur several times a year. Intermediate and emergency transient pressures represent ultimate situations in which a load is applied to an anchor for a short duration but is repetitive during the surging time. This may be significant for loading on soil as contrasted to a rock foundation. Hydrostatic tests during construction may cause major loading changes that do not occur during normal service. Another important distinction is that the penstock cross-sectional area must be based on outside shell diameter when calculating the axial load force at expansion joints or sleeve couplings because the pressure also effectively acts on the ends on the shell. Reducing or expanding sections and line valves also develop substantial axial pressure vectors.

Table 8-2 Anchor Block Loading Combinations

TYPE OF LOAD	DESCRIPTION
GROUP A: CONSTANT TYPE LOADING, UNIQUE AT EACH ANCHOR	
GRAVITY LOADS (DL1) (DL2)	DEAD LOAD COMPONENTS OF STEEL, CONCRETE, AND WATER; ALSO FIXED-END BEAM MOMENTS AND REACTIONS FROM AN ADJACENT, CONTINUOUS PENSTOCK BARREL
FRICTION LOADS (EJL) (SFL)	SLIDING FRICTION LOADS INDUCED BY TEMPERATURE CHANGES IN THE BARREL; THESE ORIGINATE AT BEARINGS AND EXPANSION JOINTS OR SLEEVE COUPLINGS.
OTHER CONSTANTS (DL3) (DL4)	EARTH BACKFILL LOADING (BACKFILL MAY BE A CONSTANT LOAD) GROUND WATER PRESSURE OR UPLIFT (BUOYANCY)
THERMAL LOADS (TL1)	ONLY IF PENSTOCK IS RIGID BETWEEN ANCHORS
STATIC PRESSURE LOADS (P _{N1})	AXIAL PRESSURE VECTORS DETERMINED BY MAXIMUM STATIC WATER PRESSURE IN THE PENSTOCK AT EACH BEND PI
GROUP B: VARIABLE LOADS, ADDED TO GROUP A	
TRANSIENT PRESSURE (P _{N1})	POSITIVE TRANSIENT PRESSURE DETERMINED BY HYDRAULIC OPERATING CRITERIA
MOMENTUM LOAD	DUE TO THE VELOCITY OF WATER
GROUP C: SPECIAL VARIABLE LOADS, ADDED SINGLY TO GROUPS A AND B WITHOUT THE TRANSIENT PRESSURES, AND AS APPLICABLE	
SEISMIC LOADS (EQ1)	DESIGN BASIS CRITERIA EARTHQUAKE FOR THE PENSTOCK LOCATION
WIND LOADS (LL1)	NORMALLY ONLY ON AN EMPTY PENSTOCK
SNOW LOADS (LL2)	ONLY AT SPECIAL LOCATIONS (INCLUDES ICING AND TRANSIENT PRESSURE)
CONSTRUCTION LOADS (PT1)	FROM HYDRO-TESTING AND ANY SIMILAR STATIC LOADING
GROUP D: LOADS FROM EXTREME PRESSURE OR SEISMIC CONDITIONS, ADDED TO GROUPS A AND TO B WITHOUT THE NORMAL TRANSIENT PRESSURE	
(P _{EM1})	WATER HAMMER FROM GATE CLOSURE IN 2L/a SECONDS
(P _{EX1})	WATER HAMMER FROM MOST ADVERSE GATE CLOSURE
(EQ2)	APPLICATION OF DAM SAFETY CRITERIA EARTHQUAKE

(3) Momentum loading

This is a vector load applied at the PI of the bend. Equal vectors, upstream and downstream, occur and the resultant is easily found by assuming the upstream vector constant and reversing the axial direction of the downstream vector.

(4) Friction loads

These result from longitudinal penstock movements due to temperature changes after original installation. Expansion joints and bolted sleeve-type couplings induce longitudinal forces by friction between the sliding materials. Similar forces result from friction in the bearings under ring girders or at saddle supports. All of these forces are almost constant and are unaffected by the degree of temperature changes.

(5) Seismic loading

Seismic loads are very important because these forces could cause a disastrous anchorage failure. During an earthquake, the foundation is accelerated in a vibratory manner and penstock structures respond according to their inertia characteristics. An anchor block is very rigid and generally responds in the same period of vibration as the foundation material. A penstock barrel is more flexible and its seismic response applies loads to the anchor by beam action. See Section 1.8 for details on estimating seismic loadings. Analyses of earthquake effects also show that the pseudostatic seismic load vectors should be applied in two directions. A coordinate system of three prime dimensions is convenient for expressing the loading combinations. When a maximum load is applied in one prime direction, a load of lesser intensity is applied 90 degrees to the first load. This means that the smaller load may be applied in either of the other two directions to give the most adverse condition. Often the smaller load is one-half of the larger. Descriptions of this method are provided by the Seismology Committee of the Structural Engineers Association of California.⁴

(6) Miscellaneous loadings

Other loadings include:

- (a) All wind loads that act on the penstock and are ultimately passed into some anchorage
- (b) Loading due to ice formation at sites subject to low temperatures, especially dead-load effects of icing should water leak out of the penstock; designers also should consider the effects of ice formation at saddles or around other sliding supports
- (c) Erection loads and effects of hydrotesting; these require consideration and/or analysis at each anchor
- (d) Buoyancy forces on buried anchor blocks
- (e) Anchor loads during filling or draining; these loads are usually low
- (f) Thermal loading, which applies only to penstock barrels rigidly confined between anchors. This condition must be carefully evaluated because large thermal strains occur in the steel shell during filling or draining of the penstock due to the difference in the temperature between the water and steel.

8.2.3 Anchor Block Design**8.2.3.1 General Theory**

Anchorage for an aboveground penstock must accept, without movement or deflection, the combined effect of the three-dimensional, nonplanar, combined loading applied to each bend. For a truly rigid anchorage, all the diverse loads must be effectively transferred through the concrete anchor block into the foundation. Ultimately, foundation resistance is from the bearing

and shearing strength of the earth material under the anchor block. The foundation material dictates the shape and configuration of an anchor block as well as its total mass weight.

For efficient anchor block design, all load systems, including the dead load of the concrete, may be systematically analyzed by dividing all the loads into three-dimensional vector components. Analysis is simplified if all loads and gravity forces acting at an anchor are considered with one component in the vertical direction and the other two in fixed horizontal directions, such as the prime compass bearings. This procedure is well suited to spread sheet analysis and tabulations by computer. If all loads on an anchor are combined without including the weight of the concrete anchor, the vertical resultant gives an estimate of the minimal required dead load of the anchor to balance uplift forces. Similarly, the horizontal resultant estimates the minimal shearing forces applied to the foundation material. However, a major force remains due to the seismic loads on the mass of the anchor.

The magnitudes of specific loads applied to an anchorage will show significant variations in accuracy. Even such parameters as the internal pressure value may be an approximation. Conservative values must be used for the loads controlling the design in order to ensure adequate anchorage design.

8.2.3.2 Anchor Block Configuration

Reinforced concrete is the most efficient medium for transferring the anchorage load to the earth foundation. A common design configuration is complete encasement of the bend in concrete, but this is not absolute. An effective alternative anchor design may use tie-down straps or rings, anchored in turn to a lower mass of concrete. This design has the advantage of reducing the mass weight of the concrete under seismic loading. Weight and dimensions of an anchor block should hold the resultant vector of all foundation loadings to a position which will not overstress the earth foundation materials. Rocking is not allowed, and tie-down tendons may help by pretensioning an anchor block onto a foundation. Even drilled piles or caissons have been used to increase the shearing resistance of a weak foundation. These additional components are effective in increasing the lateral resistance of a foundation under seismic loading and also lowering the mass center of anchor blocks to an elevation at which overturning effects are more manageable.

Seismic loading will produce the greatest lateral loads at the base of the anchor, at the contact with the earth foundation. If the sliding friction resistance of the earth under the concrete block is inadequate for such a loading, then other devices such as shear keys, anchor tendons, and concrete caisson piles may be used. These supplemental devices, which increase the lateral resistance of soil foundation, often are needed because resisting lateral forces by bearing against soil must allow for the large relative displacements needed to mobilize passive soil forces, hardly acceptable for an anchor. Lateral soil pressures used to counter sliding loads cannot exceed the value of the at-rest pressure of the soil; preferably only the value of the active lateral pressure of the soil should be used.

Whenever a penstock pipe is completely encased at an anchor, sufficient reinforcing steel must be used to bind the concrete into one unit, transferring the weight of the anchor upper half to the lower half. In this situation the concrete also is loaded by the effects of radial expansion of the

pipe shell under the internal pressure loads. Unsightly cracking of the concrete may result from this condition. Usually a gap space forms between the outside of the shell and the cured concrete. The gap allows small expansions of the shell before its contact with the concrete. Unfortunately, the clearance distance is an uncertain value.

The best practice is to encase a penstock shell with concrete while the shell is internally pressurized to a major portion of the normal service pressure. This ensures that the shell is fixed within the encasement during normal service. Additional reinforcement may be provided for the exceptional increased pressures. However, concrete encasement often must be placed around a dry pipe for practical reasons. In this case, the addition of extra reinforcing to control cracking is advised. Hydro-testing also may create conditions for cracking; checks should determine if the radial strain of the shell could crack the encasement. A minimal action is wrapping of the final 6 inches of pipe at the encasement edge with resilient material to absorb the deformation at this weak part of the concrete block.

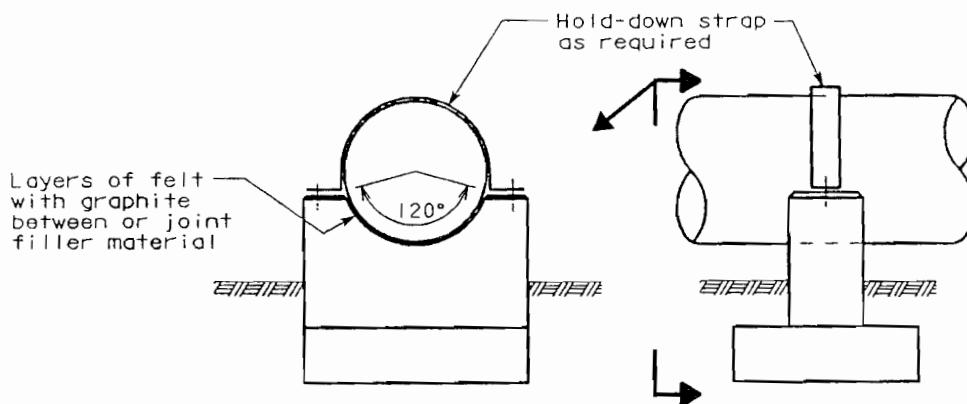
8.3 Piers

8.3.1 General

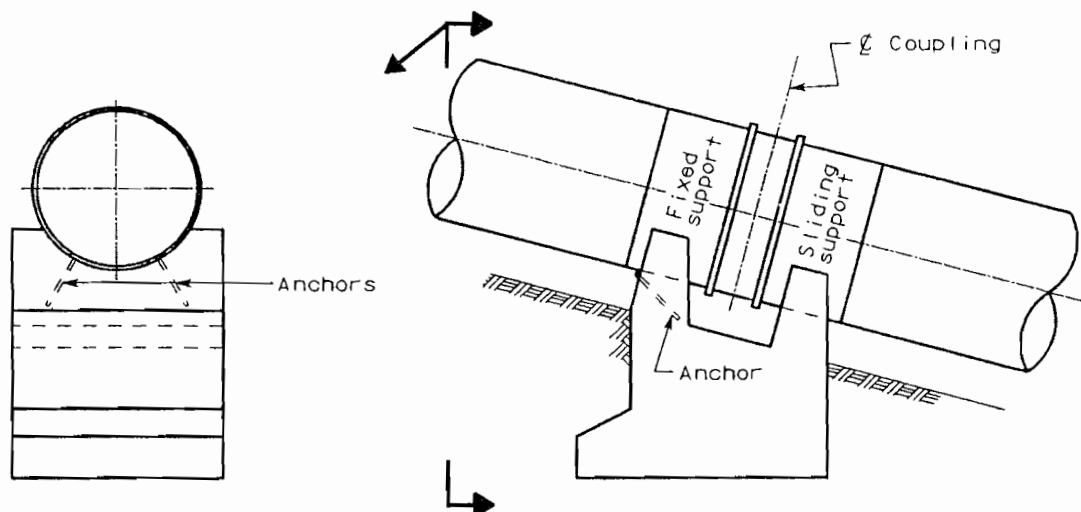
Concrete piers support the penstock at intermediate points between the anchor blocks. Pier spacing is determined by optimizing the penstock span length after all combined hoop and longitudinal shell stresses are determined. Pier spacing usually ranges from 20 to 100 feet. Spacing also is influenced by other factors in the pipe configuration such as locations of expansion joints and/or coupled joints. The two concrete pier configurations commonly used are saddle supports (see Figure 8-5) and ring girder supports (see Figure 8-6). Where steel saddles are used, their concrete piers are similar to those used at ring girders.

Concrete saddles generally are used for the smaller pipe diameters and shorter spans (usually 20 to 40 feet) where the pipe span end reaction at the support does not cause overstress of the pipe shell and thus require a ring stiffener. To provide the most ideal condition for pipe and concrete saddle stress distribution at the reaction point, the saddle contact angle is usually 120 to 180 degrees, with the smaller angle giving the higher steel stress but lower lateral load concrete stress at the edges. Often a fabric pad is placed between the pipe and concrete surface. Also, wooden wedges are often placed in the top edge of the saddle (saddle horn) to reduce stress concentration. Saddles can be detailed to allow the pipe to slide or to anchor the pipe, as illustrated in Figure 8-5.

3 Anchor Blocks and Piers



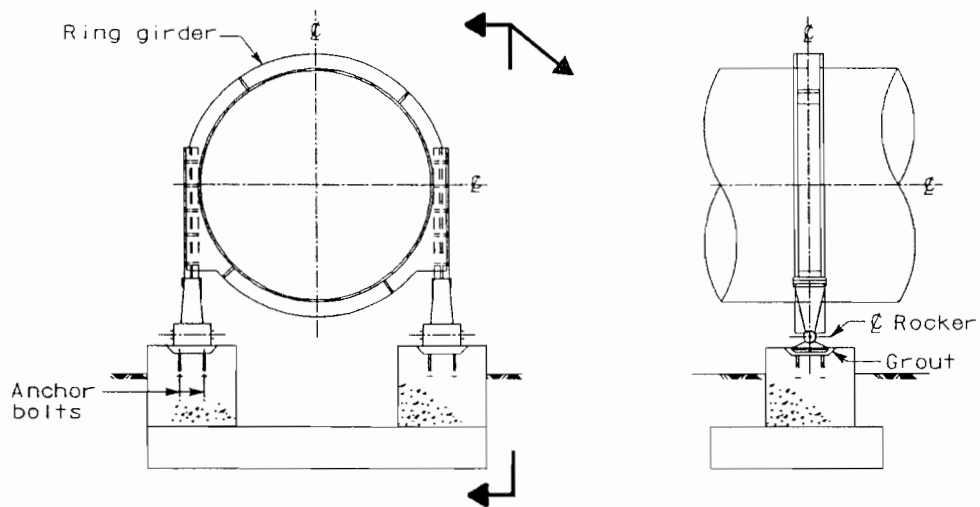
(A) SIMPLE SADDLE SUPPORT



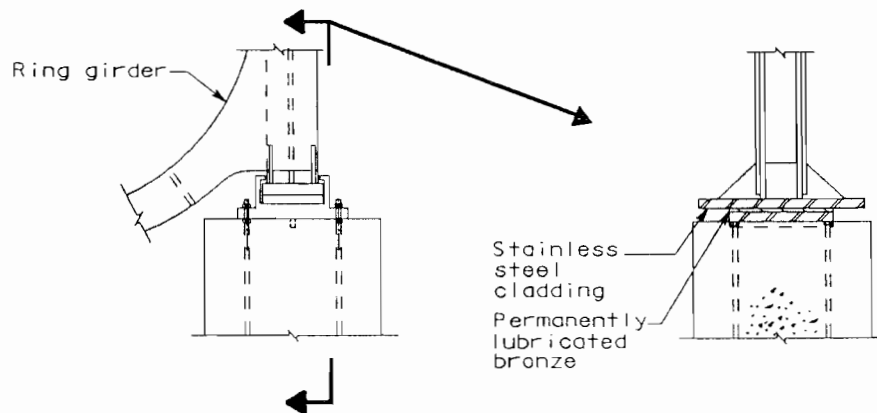
(B) PAIR OF SADDLES AT SLEEVE COUPLING

Figure 8-5 Typical Saddle Supports

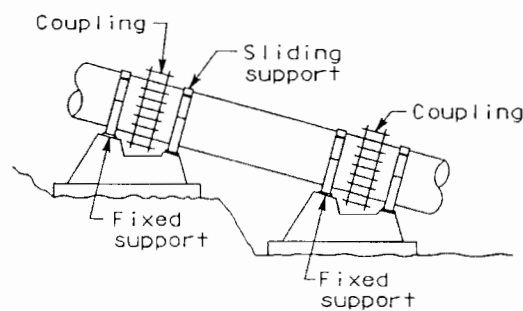
Ring girder supports are used for larger pipe diameters where the reaction load at the support is too large to be reasonably accommodated by a plain pipe shell. The pipe reaction loads are transmitted from the ring girder to the concrete piers either by sliding bearing plates or rollers or rocker assemblies. These details are developed to prevent lateral movement of the pipe but allow longitudinal movement or rotation. Where it is desired to prevent longitudinal movement of the pipe, the piers can also be detailed with anchor bolts through the ring girder base plates to form a fixed support point (see Figure 8-6).



(A) ROCKER SUPPORT



(B) SLIDING SUPPORT



(C) RING GIRDER SUPPORTS AT COUPLING SLEEVE

Figure 8-6 Typical Ring Girder Supports

The final overall dimensions of saddles, piers, and their base mats are determined by the underlying soil or rock properties and by the stability evaluation described in Section 8.1.4. In most cases the concrete reinforcement steel required will be based on the minimum temperature steel required. However, specific details must be checked such as the narrow cross-section of concrete near the top of the saddles (saddle horns) subjected to lateral reaction loads plus radial pipe expansion. These same lateral loads require checking of the reinforcing across the center of the saddle under the pipe.

8.3.2 Loads and Combinations

All concrete supports must be designed for the dead load of the pipe and contained water (and ice and snow if applicable), longitudinal forces resulting from frictional resistance due to longitudinal strain (Poisson-ratio effect) and temperature movements, and lateral forces caused by wind and earthquake forces. In addition to the loads that act on the pipe and therefore on the concrete support, the support also may experience additional loads, including buoyancy due to groundwater and prestressing load if rock anchors are used.

The load combinations and service level categories used in the concrete support design must be consistent with those used in the pipe design. The most critical combinations must be determined separately for the overall stability evaluation (see Section 8.1.4) and then for the detailed reinforced concrete design. The loading conditions can be categorized as follows:

(1) Construction conditions

Dead load with pipe empty or full, buoyancy, wind, prestress load, and temperature

(2) Test conditions

Dead load, buoyancy, wind, prestress load, internal test pressure, and temperature

(3) Operating conditions (static and dynamic cases)

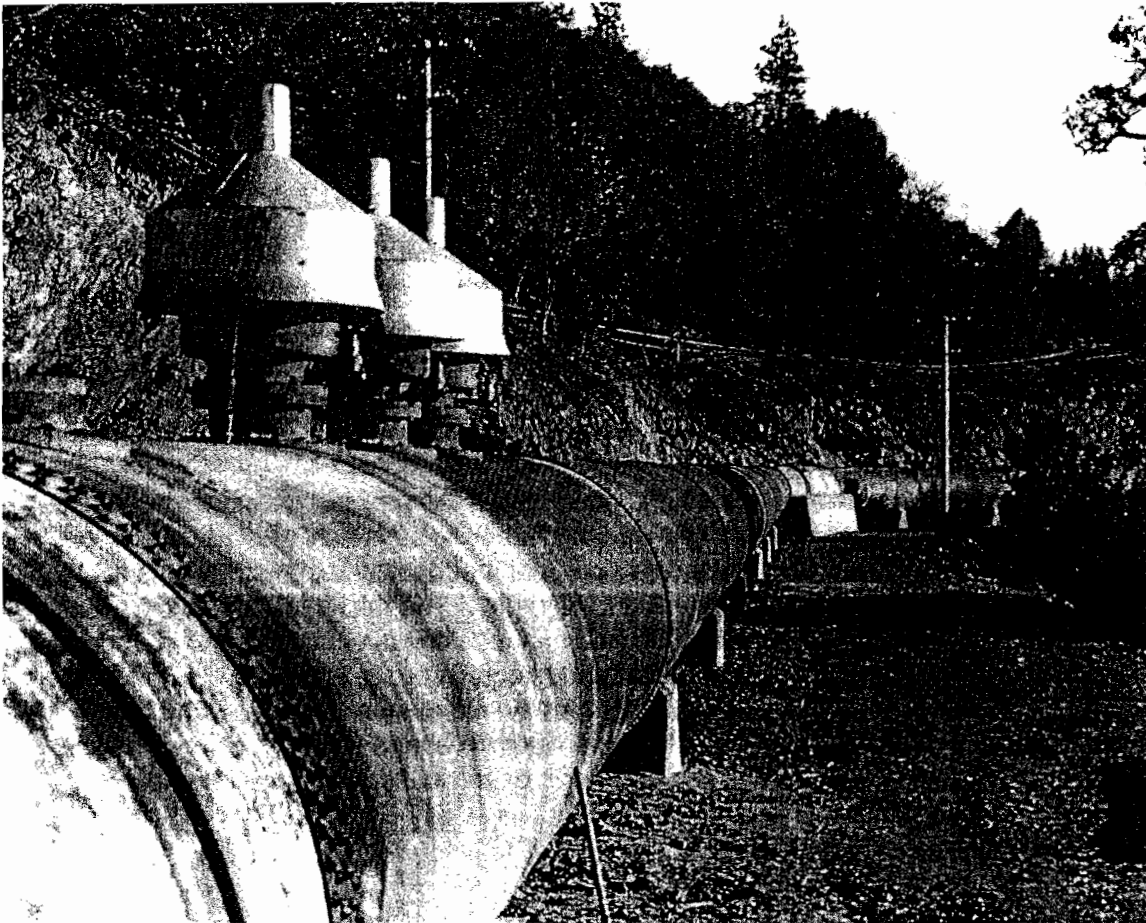
Dead load, buoyancy, wind or seismic, prestress load, internal pressures (steady state and transient), and temperature.

For each condition the worst combinations of values of each component and associated friction effects should be evaluated. Also, if there are no expansion joints, Poisson-ratio effects must be included in the loading.

References*

1. Building Code Requirements for Reinforced Concrete (ACI 318). American Concrete Institute, Detroit, MI.
2. Bouchayer, R. "Stresses on Anchor Blocks." *Transactions*, ASCE, Paper No. 3390, 127:546 (1962).
3. "Thrust Restraint Design for Ductile Iron Pipe." Ductile Iron Pipe Research Association, Birmingham, AL (1989).
4. "Recommended Lateral Force Requirements and Commentary." Seismology Committee of the Structural Engineers Association of California, San Francisco, CA (1988).

* The most current version of a standard, code, or specification should be used for reference.



9.1 Mechanical Expansion Joints

Mechanical expansion joints frequently are installed in exposed penstocks between anchor blocks to accommodate longitudinal movements caused by thermal conditions. Using a conventional design, the joints can accommodate up to 10 inches of total movement. Mechanical expansion joints can be installed in vaults but cannot effectively be installed in buried penstocks. Figure 9-1 shows a typical mechanical expansion joint.

The mechanical expansion joint consists of a body, slip pipe, packing chamber, and end rings, with proper bolting to compress the rubber gaskets and lubricating rings to withstand internal design pressure. If mechanical expansion joints are installed in series without anchor blocks, limit rods are required to restrict the movement of each expansion joint. Expansion joints can be manufactured for any diameter pipe. If required, flanged ends can be supplied to implement

9.2 Mechanical Bolted Sleeve-Type Couplings

The mechanical bolted sleeve-type coupling (BSTC) consists of a center sleeve, two end rings, two wedge-shaped elastomeric sealing rings, and bolting hardware to compress the end rings against the gaskets to form a leak proof seal between the penstock shell and center sleeve. The center sleeve may be of the same material and thickness as the penstock. The end rings must be a single-piece, contoured, hot-rolled steel section and must not roll more than 3.5 degrees from the unbolted position during installation. The center sleeve and end rings must be cold expanded 1% of the diameter to proof test the butt weld and parent metal and to size the parts.

ANSI/AWWA Standard C-219-91¹ should be consulted for the design of bolted, sleeve-type couplings.

Bolted sleeve-type couplings allow approximately 3/8 inch of longitudinal displacement of the pipe ends, per joint, and allow deflection (see Figure 9-2) within the diameter and sleeve-length limits specified in Table 9-1. Laying deflection is indicated by ϕ . Bolted sleeve-type couplings can accommodate some differential settlement only where two (2) or more couplings are installed in a series between properly supported pipe sections.

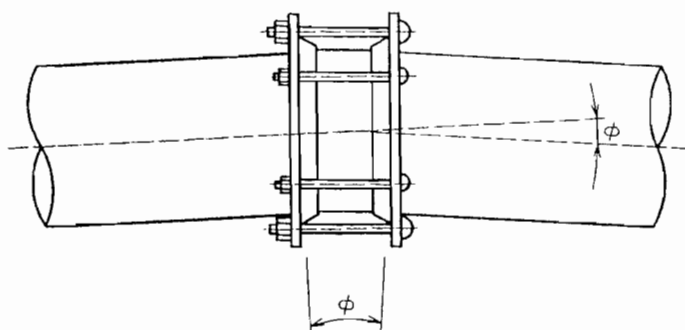


Figure 9-2 BSTC Laying Deflection (ϕ)

Table 9-1 Recommended Maximum Laying Deflection

PIPE O.D. (IN.)	MAXIMUM LAYING DEFLECTION (DEGREES)	
	7-IN. SLEEVE LENGTH	10-IN. SLEEVE LENGTH
30-37	3	3.5
37-42	2.5	3.5
42-49	2	3
49-54	2	3
54-66	-	2.5
66-78	-	2
78-90*	-	1.5

*: For diameters over 90 inches, consult the BSTC manufacturer for deflection limits.

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Figure 9-3 shows various types of bolted sleeve-type couplings.

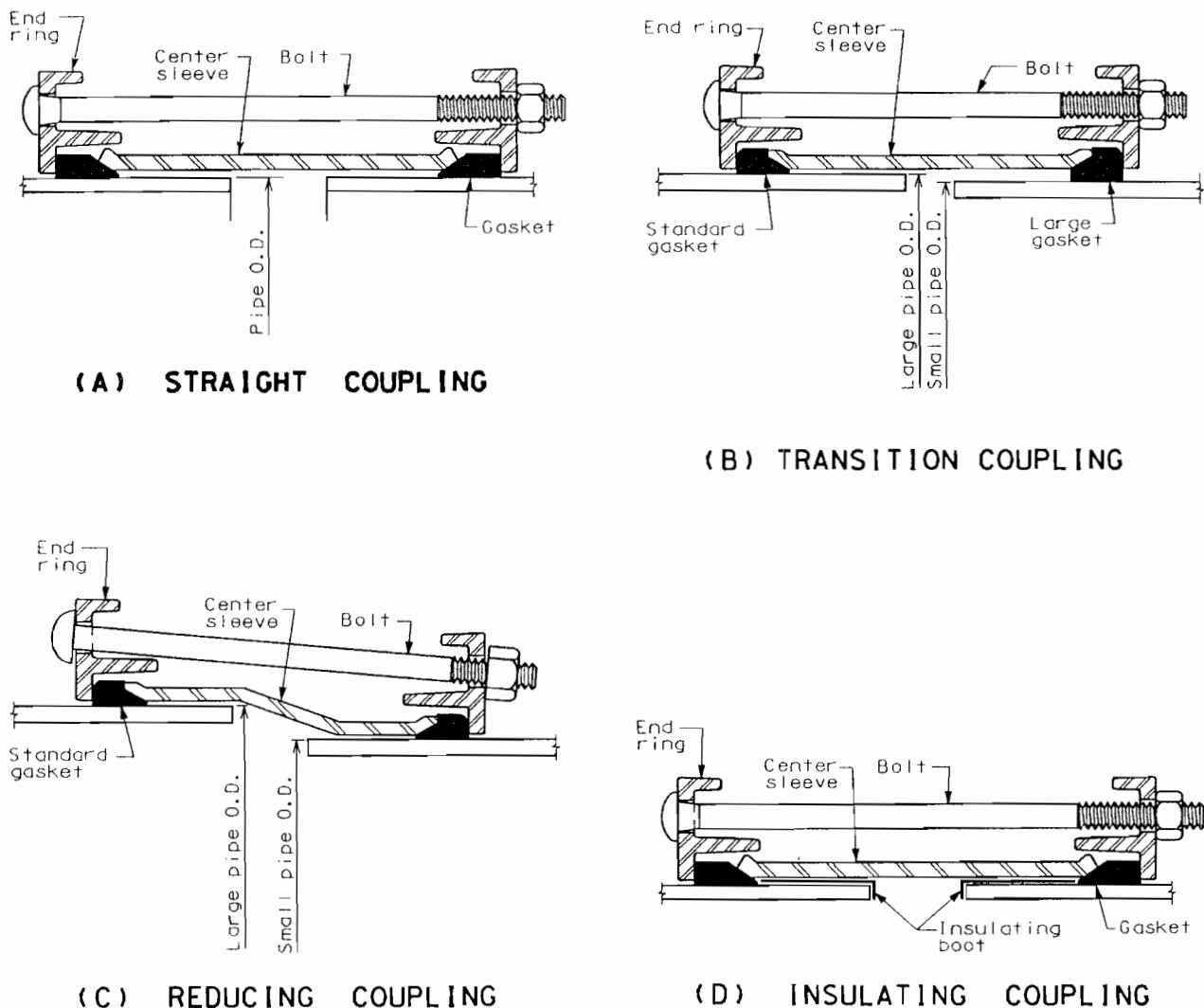


Figure 9-3 Types of Bolted Sleeve-Type Couplings

BSTC may be buried in earth; however, a hydrotest of the completed penstock must be performed prior to backfilling so that any leakage can be corrected by tightening (according to the installation instructions).

Where BSTC are buried, proper corrosion protection must be considered to protect the coupling components and the gasket sealing area on the pipe ends.

When designing penstocks where BSTC are required, consult AWWA Standard C-219.¹

Figure 9-4 illustrates deflections with couplings.

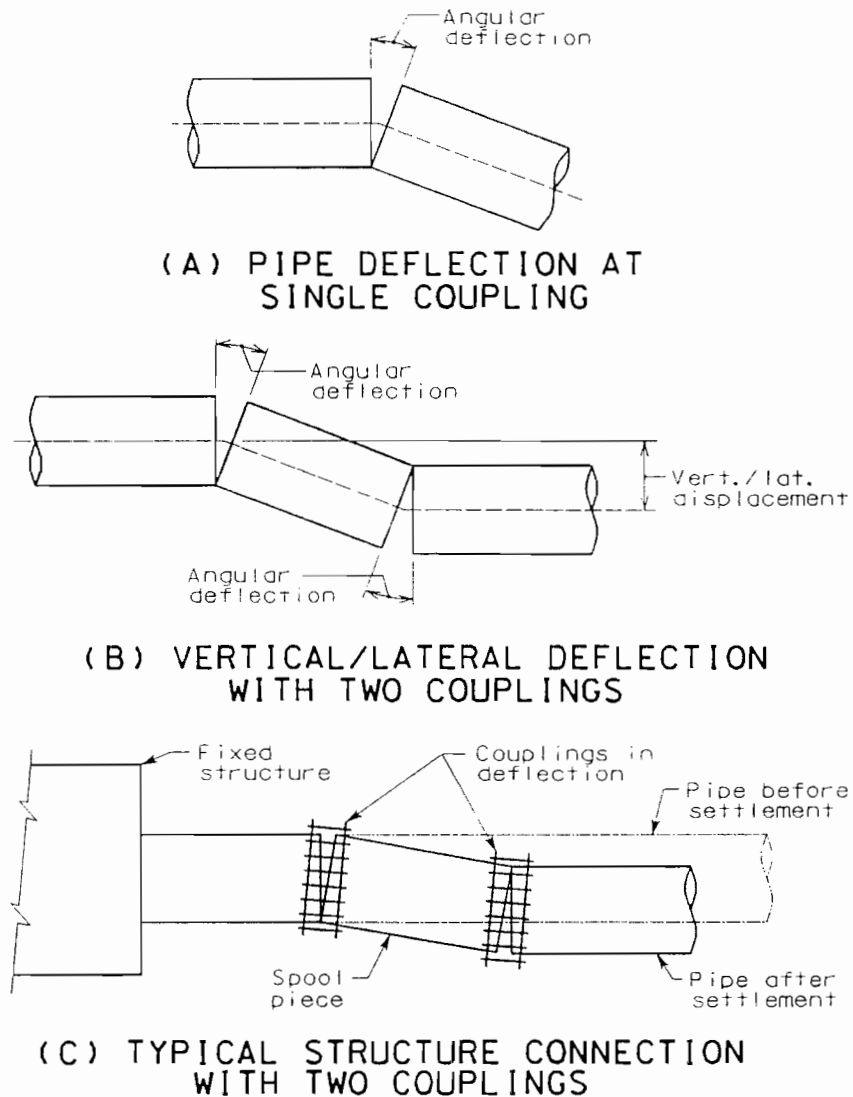


Figure 9-4 Deflections with Couplings

Table 9-2 specifies recommended pipe gaps.

Table 9-2 Recommended Pipe Gaps

SLEEVE LENGTH (IN.)	OPTIMUM GAP (IN.)*	DEFLECTION LIMITED MAXIMUM GAP WITH RESTRAINTS (IN.)
7	1/2 TO 3/4	3
10	1/2 TO 1-1/4	4.5

*The optimum gap must be observed to achieve the indicated laying deflection. For actual field conditions, some added angularity may be practical.

9.2.1 Bolting Formulas and Design

Selection of the proper bolted sleeve-type coupling depends on the line pressure, temperature, and other service conditions. The pressure or design rating of the pipeline is one of the most important considerations in selecting the proper coupling. The pressure rating of a coupling is determined by the sleeve thickness, the minimum yield strength of the material used in its manufacture, and the gasket pressure developed by the tension on the bolts.

The Barlow hoop stress formula, revised for open-ended pipe, is adequate for designing most couplings. The revised formula is expressed as:

$$S = \frac{pD}{2t} \quad (\text{Equation 9-2})$$

Where:

- S = stress
- p = internal design pressure
- D = outside cylinder diameter (rather than inside diameter)
- t = thickness of cylinder wall

Gasket pressure is determined by an empirical formula and may be assumed to be at least 1.5 times the line pressure. The basic formula is

$$\text{Gasket pressure} = \frac{(\text{Number of bolts})(5,000 \text{ lbs})}{\text{Gasket area}} \quad (\text{Equation 9-3})$$

Five thousand pounds is the average tension developed by a 5/8-inch by 11-inch UNC-2 bolt tightened to 70 foot-pounds of torque. The 5/8-inch bolt is the standard size used by most manufacturers. Where bolts other than the 5/8-inch standard bolt are required, obtain the proper stress values from the bolted sleeve-type coupling manufacturer. The gasket area is the area at the heel of the gasket for the pipe size involved.

To assist in the design of sleeve thickness and bolt size and number for a particular internal design pressure, see the design examples for mechanical expansion joints and BSTC in Section 17.

A minimum number of bolts is necessary to provide uniform gasket pressure. The minimum number of bolts is determined by the equation:

$$\text{Minimum number of bolts} = (\text{Pipe O.D.}) (.4384) \quad (\text{Equation 9-4})$$

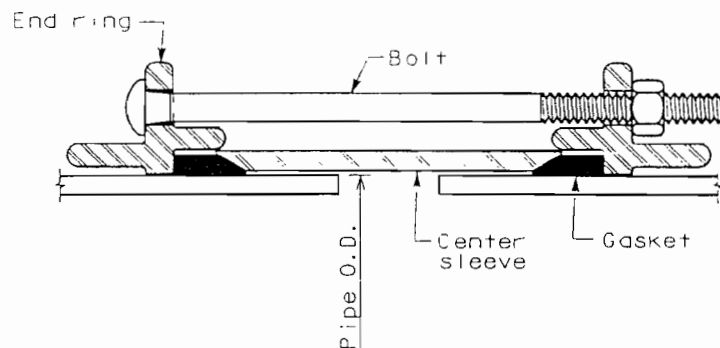
If BSTC are used on very steep slopes, the designer should consider installing pipe stops on the middle of the center sleeve (see Figure 9-3). Pipe stops prevent the coupling from creeping downstream as the penstock sections move with thermal changes. One disadvantage with pipe

stops is that they make installation and removal of an adjacent section of penstock difficult because the complete coupling cannot be moved onto one of the pipe ends. In actual practice, pipe stops rarely are necessary.

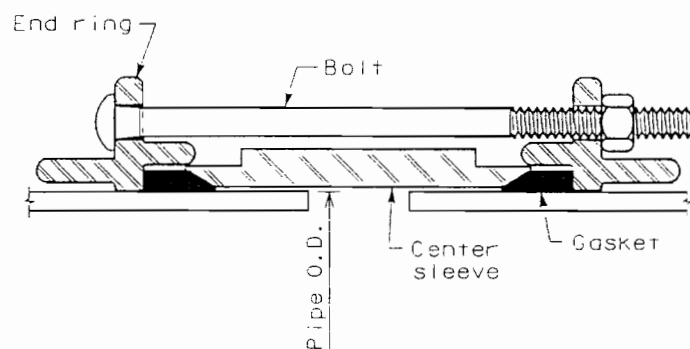
For steep-slope installations, the penstock cans should be installed from the bottom of the slope upward. This prevents the cans from creeping downhill as they are installed. As the cans are installed upward, the gap between the cans can be held constant by the use of temporary wood, nylatrile, or steel wedges. It should be noted that, where practical, BSTC should be installed loosely on the corresponding pipe ends prior to transport to the job site. By doing this, final installation is simplified, and any out-of-roundness of the pipe ends or coupling parts can be corrected under factory or shop conditions.

9.2.2 Large-Diameter and High-Pressure Bolted Sleeve-Type Couplings

Large-diameter (over 96 inches) and high-pressure (over 450 psi) penstocks may require BSTC as shown in Figure 9-5.



(A) LARGE-DIAMETER COUPLING



(B) LARGE-DIAMETER,
HIGH-PRESSURE COUPLING

Figure 9-5 Large-Diameter and High-Pressure Couplings

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The end ring and center sleeve design may be altered to provide a leak-proof seal at higher pressures. The design of the penstock determines the appropriate BSTC design. Transportation considerations and commercially available tooling and technology are the limiting factors for the diameters of penstock couplings. One-piece, solid-ring couplings have been furnished up to 217 inches in diameter; however, the generally accepted limitation is 144 inches. The designer must consider overall stress limits in the end rings and center sleeve, along with stresses introduced to the bolting patterns required to compress the wedge shaped elastomeric rings to provide a leak-proof design.

9.2.3 Segmented Bolted Sleeve-Type Couplings

Where diameter, transportation, and erection considerations do not permit one-piece BSTC, the designer should consider the use of segmented bolted sleeve-type couplings (see Figure 9-6). Segmented BSTC may be manufactured for any diameter of penstock. The design pressure of the penstock determines the design of the segmented BSTC. Sleeve thickness is determined by the Barlow formula (Equation 9-2). The bolting formula (Equation 9-4) gives the number of segments required to properly compress the elastomeric sealing rings to provide a leak-proof design. Center sleeves and follower rings for segmented BSTC are manufactured in three 120-degree segments.

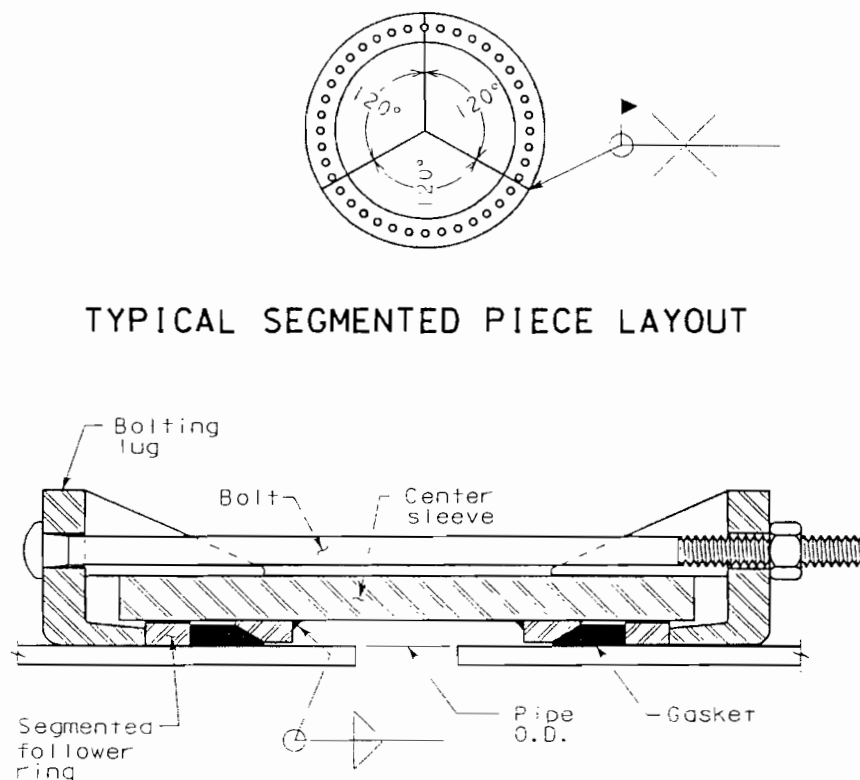


Figure 9-6 Segmented Bolted Sleeve-Type Coupling

Any machining or welding of anchor rings is performed in the flat or rolled position. The sleeve then is manufactured in three equal segments. These segments are matchmarked with ends beveled for welding in the field. Struts or ties must be provided to hold the segments to the designed arc during transportation to the job site.

At the erection site, the segments are positioned for welding into a single sleeve. The segmented BSTC manufacturer must furnish the information for the proper welding procedure, including any fit-up drawings required to assist the installing contractor. Welds on single-piece pressure rings or sleeves that were not cold expanded during the manufacturing process must undergo NDE.

After the completion of all field welding and NDE of the welds, the sleeve is positioned over the penstock ends. The procedures for installing mechanical BSTC gaskets may be followed. The three-piece followers (or bolting segments serving as followers and bolting segments) are then positioned against the gaskets. Then, the installation instructions furnished by the manufacturer should be followed.

9.3 Manholes and Smaller Openings

Manholes provide access to the penstock shell interior by personnel and equipment. During construction, manholes allow access by the contractor; after installation, they permit maintenance inspections, relining, or removal of rocks and sediment.

Manholes allow inside air circulation. This is vital for safe working conditions in conformance with OSHA regulations. Maximum spacing for manholes is 500 feet along the exposed length of barrel. Manholes placed at local crests on the pipe profile will facilitate use of hoisting equipment needed for painters' scaffolding. For adequate maintenance, manholes must be located adjacent to all line valves and should be near all air-release and/or vacuum valves. Figure 9-7 shows recommended manhole positions.

The dimensions of all openings or manholes should relate to the size of the maintenance personnel, minimum dimensions of components of ladders, carts, hoses, hoists or scaffolding, and to the expected volume of sand-blasting grit or other debris removed through the opening. Small openings that accommodate the optimum lengths of welders' cables or other hoses are common. Reductions in clearance due to mortar-lining thickness also must be considered. It should be noted that OSHA requirements may supersede any of these requirements.

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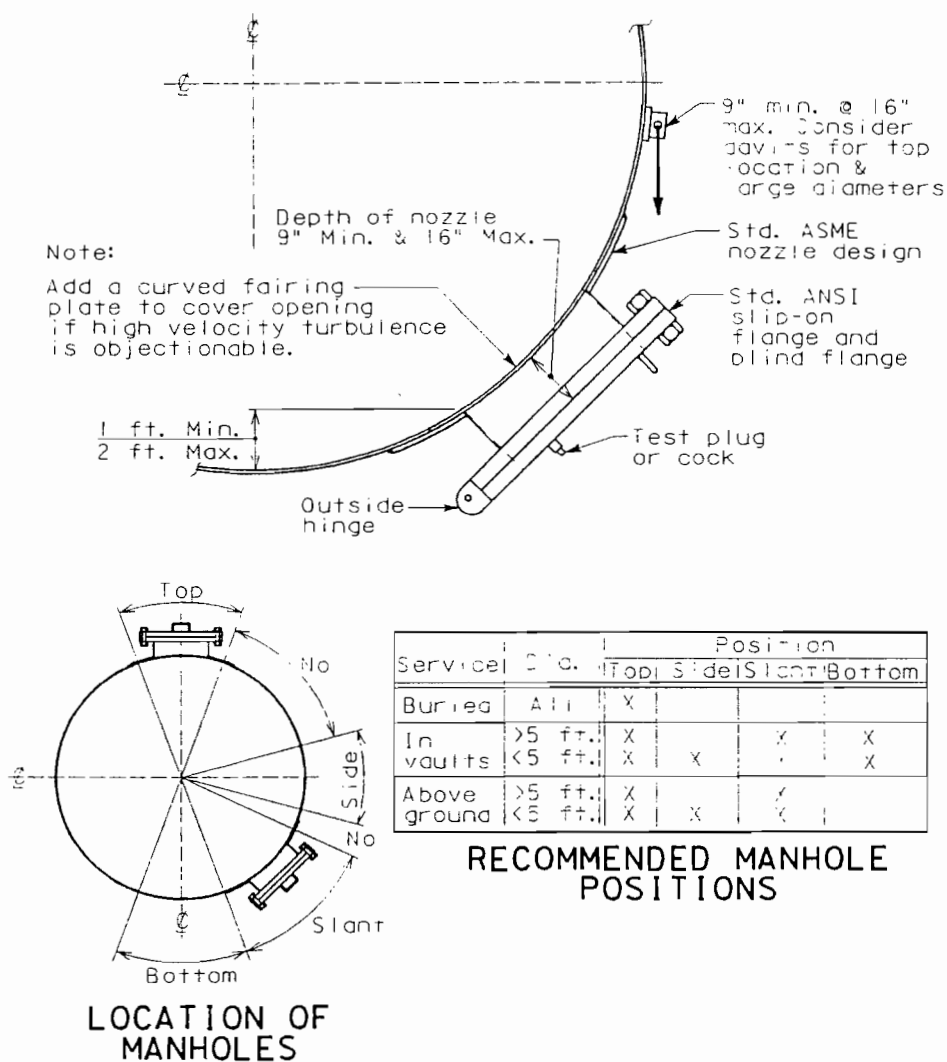


Figure 9-7 Manhole Locations and Positions

The recommended nominal diameter for a manhole opening is 24 inches, which is the largest size for which standard flanges are fabricated (see Figure 9-8). Manholes or other openings must be installed 90 degrees to the main penstock shell, and have the shortest possible shaft length.

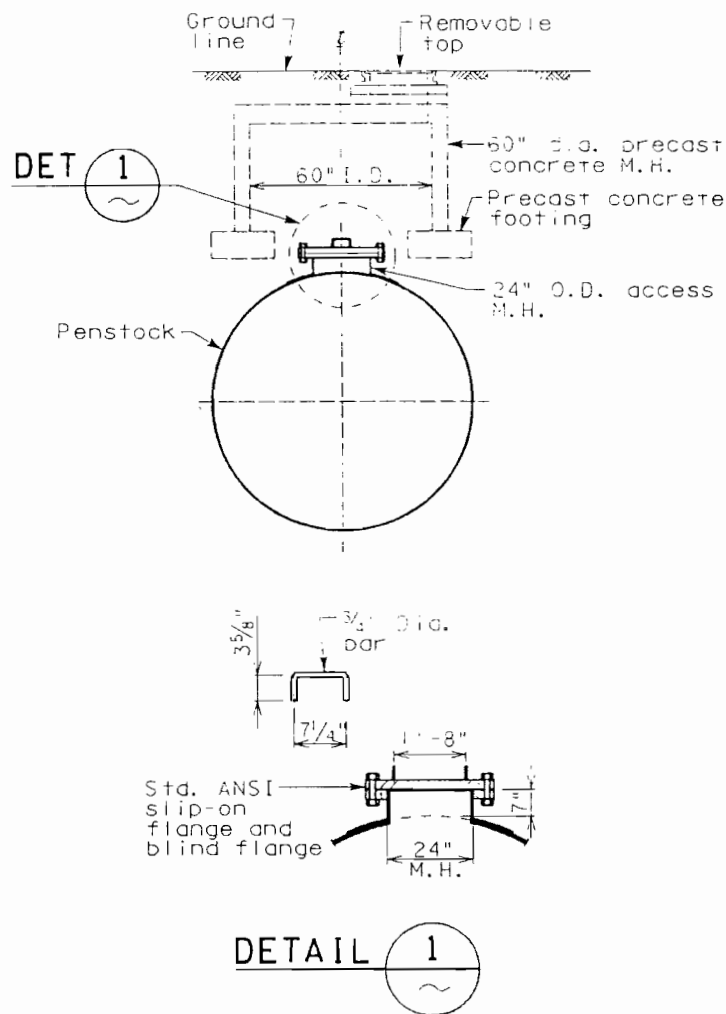
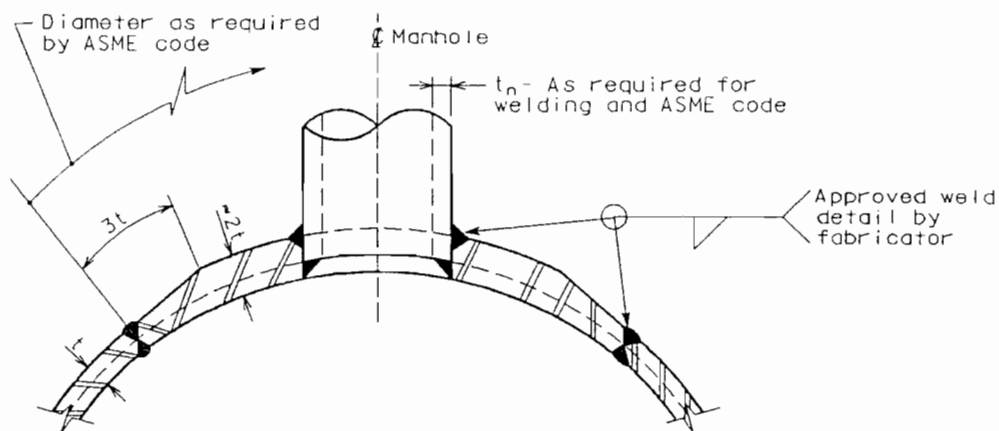


Figure 9-8 Typical 24-Inch Access Manhole for Buried Penstocks

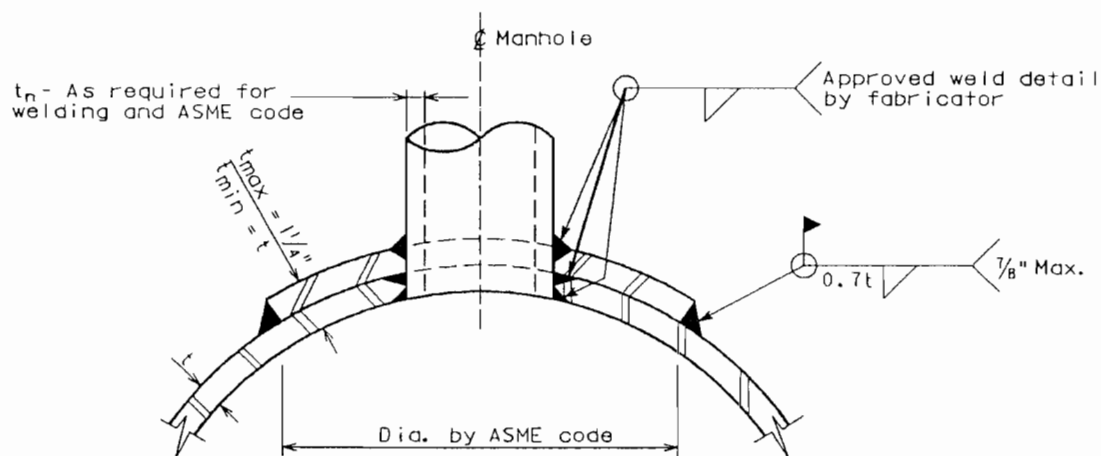
A penstock shell must be reinforced at most openings by collar or saddle-like plates welded around the opening. The controlling specification for this reinforcement is described in the ASME Code, Section VIII, Division I, Paragraphs UG 36 and 37. Figure 9-9 shows a typical but generalized basic design according to the ASME Code. The maximum size of opening that may be reinforced in this manner depends on the ratio of the diameter of the opening to the diameter of the main barrel, as shown in Table 9-3.

Openings with diameters larger than those given in Table 9-3 require special reinforcing of the penstock. Opening ratios up to 0.6 D may be accommodated under Appendix 1-7 of the ASME Code, but over this limit, special designs described in Section 7.4 of this manual must be used.

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(A) INSERT MANHOLE REINFORCEMENT



(B) LIMITS OF SADDLE PLATE REINFORCEMENT

Figure 9-9 Manhole Reinforcement

Table 9-3 Maximum Allowable Opening Diameter

INSIDE DIAMETER (D) OF MAIN PENSTOCK (IN.)	MAXIMUM DIAMETER OF OPENING ALLOWED FOR BASIC DESIGN MANHOLE (IN.)
40	1/2 D
41-60	20
61-120	1/3 D
OVER 120	40

Smaller penstocks also are affected by the criteria of Table 9-3. For example, the smallest diameter for practical personnel access through a penstock manhole is approximately 18 inches. Table 9-3 shows that 36 inches is the smallest diameter penstock that can accept an 18 inch typical manhole. Access to a penstock barrel less than 36 inches in diameter is best provided by installation of short, removable spool sections between two sleeve-type couplings.

The ASME Code requires collar reinforcement for other openings smaller than manholes. An exception usually is made for openings smaller than $2\frac{3}{8}$ inches where the controlling factor passes from diameter to shell thickness.

Fairing plates will prevent disruption of high-velocity flow in a penstock if this is objectionable. A design method that holds the curved plate in position and yet allows rapid plate removal without major damage to protective lining material is required.

Steel pipe flanges and gaskets must conform to the requirements given in Section 9.8.

9.4 Standpipes and Air Valves

9.4.1 General

To ensure proper operation and protection, standpipes and air valves are required on penstocks for the following purposes:

- (1) To admit air at the proper rate to accommodate the closure sequence of valves or gates and prevent collapse of the penstock; this is ordinarily the design criterion for sizing the diameter of standpipes and the large orifice of air valves. They must be located wherever the hydraulic gradient may fall low enough to cause collapsing pressure in the penstock.
- (2) To admit air in the event of an uncontrolled drainage for release of water from the penstock; this involves uncertainties that must be evaluated for each penstock, but is not normally used as a criterion for sizing.
- (3) To release air during watering-up of the penstock; this requires placement of standpipes and air valves at high points on the profile or at other locations where air may collect during filling. Filling rates must be set based on the size of the standpipe or air valve.
- (4) To release small amounts of accumulated air that come out of solution during normal operation; this requires the release of air accumulated at high points and at abrupt changes in pipe slope.

Adequate drainage and erosion protection must be provided at standpipe and air valve installations to accommodate accidental spillage during filling or normal operation.

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9.4.2 Standpipes

Each standpipe provides a nonmechanical vent that allows the most rapid air flow into the penstock and a path for air release. A typical standpipe attachment to a penstock is shown in Figure 9-10.

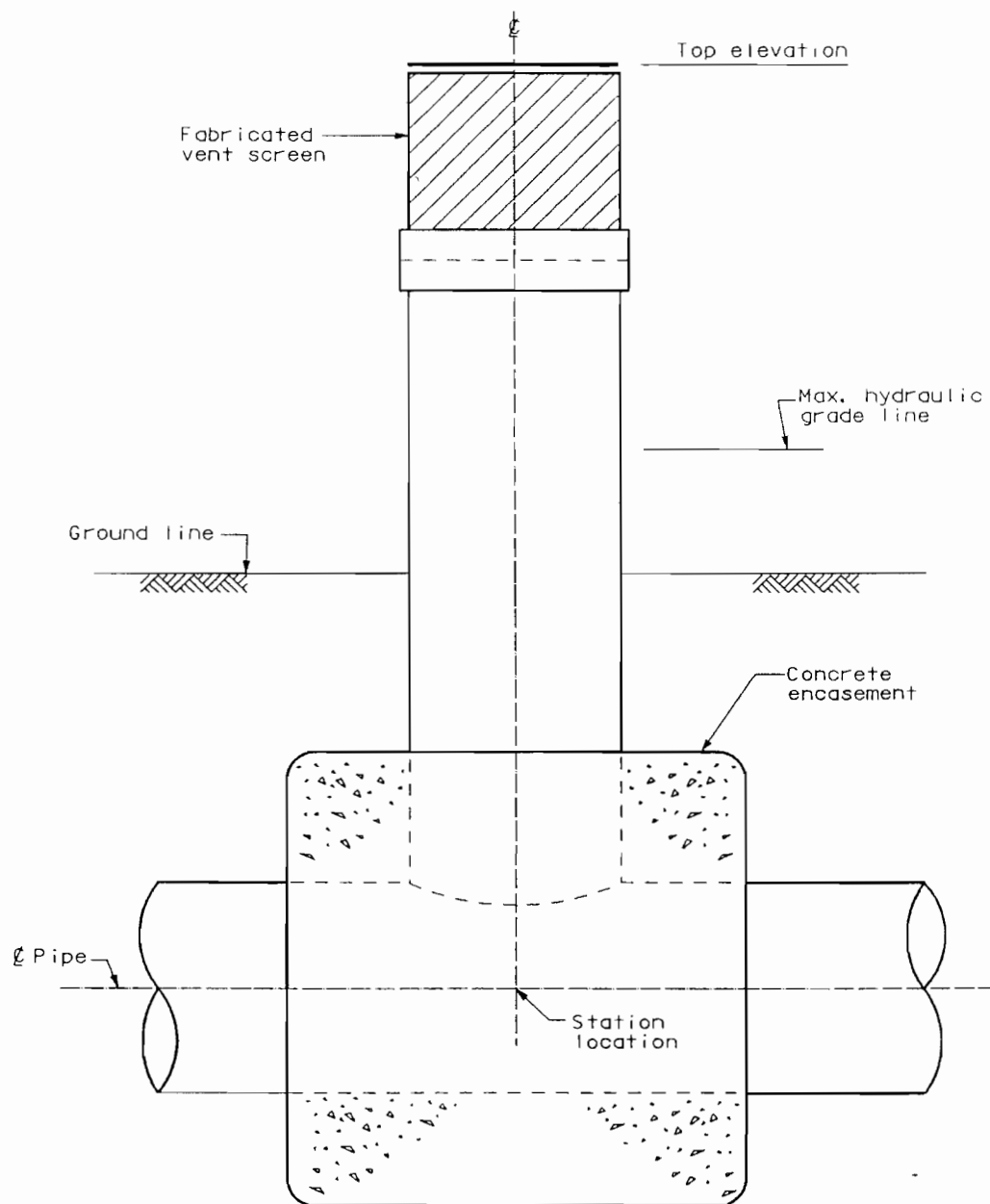


Figure 9-10 Typical Standpipe

Standpipes that vent above ground must be protected from freezing if water stands in the pipe. Standpipes may be provided with bubblers or heaters, or placed in a heated valve house for protection from snow and ice. Standpipes must be provided with hoods, appropriate grillwork, or screens to keep out debris. A tall standpipe is subject to large bending moments from wind and seismic forces and must be reviewed for resonance by the methods used for steel stacks. Normal design for seismic forces considers the water at normal operating level. Allowable stresses with water at maximum level are given in Table 3-5.

9.4.3 Air Valves

Air valves provide a mechanical vent that allows smaller air flows in and out of a penstock. The three basic types of air valves are: (1) air release, (2) air/vacuum, and (3) combination air release/vacuum. Air release valves have either large orifices designed for discharging air during filling or small orifices designed for releasing accumulated air during normal operation. The location of air valves is important to their operation, and they must be placed properly to provide the desired service.

Air/vacuum and combination air release/vacuum valves must be sized to allow the release of air without pressurizing the penstock. The filling rate during watering-up must be considered. Manufacturers of air valves should be consulted for more specific recommendations on sizing.

As with standpipes, air valves must be protected from freezing. They are best installed below the frost line in valve houses or underground vaults that are properly ventilated and drained. Ventilation must be designed to prevent the vault or enclosure from becoming pressurized during the discharge of air and to allow air inflow during negative pressure situations.

The sizing of air valves depends on the operation of the penstock and the location, function, and individual characteristics of the air valve. Individual valve manufacturers should be consulted. Additional information is provided in the referenced works by J. Parmakian² and J. E. Lescovich.³

Air valve body and trim materials and coatings must be selected to provide proper corrosion resistance to the water transported in the penstock and the environmental conditions in which the air valves are installed.

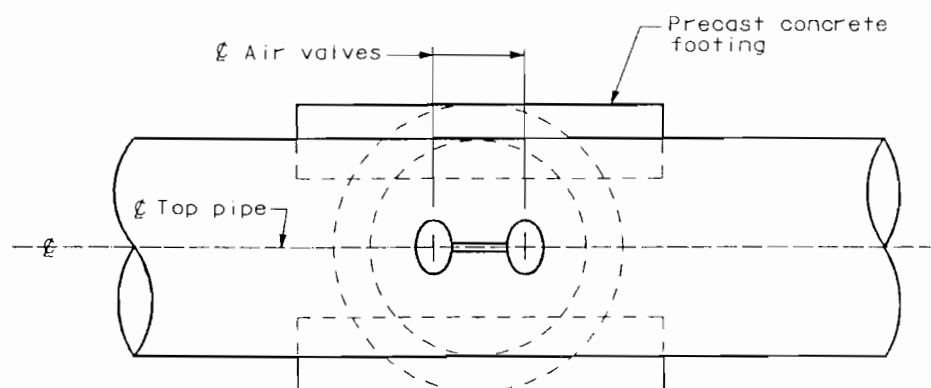
A minimum of two air valves must be used at any location to increase reliability. An isolating valve must be placed below the air valve to permit inspection and maintenance.

A typical air valve installation for a buried penstock is shown in Figure 9-11.

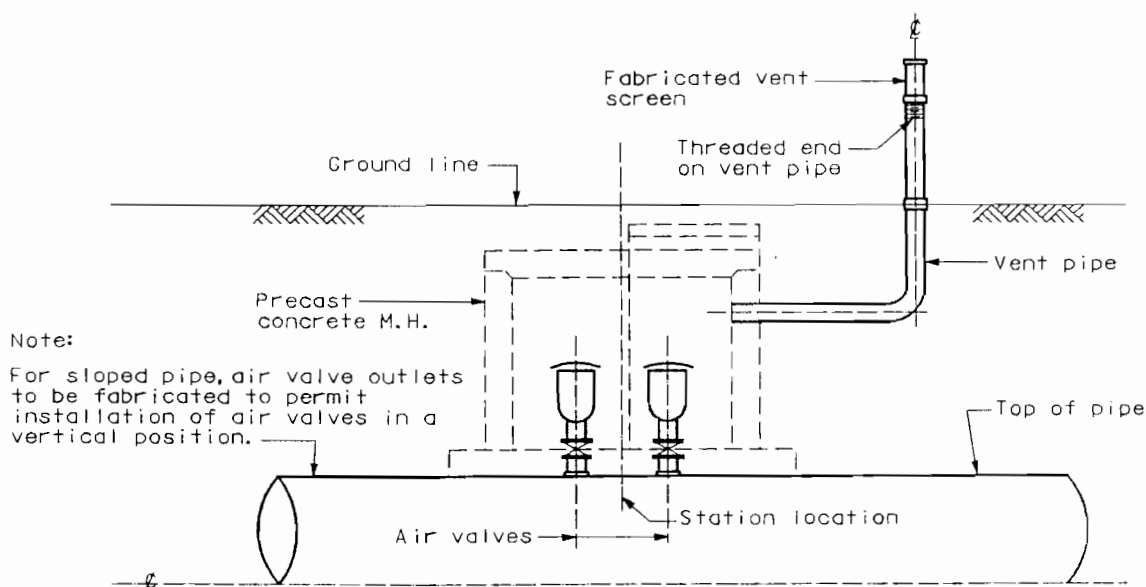
9.5 Anchor Bolts and Fasteners

A variety of anchor bolt sizes and lengths are used on penstocks. The largest types are tie-down tension anchors used to transfer bend forces downward into anchor block concrete. Smaller anchor bolts and fasteners provide tie-down forces that secure the components of penstock supports and bearings to individual concrete footings or smaller anchor blocks.

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(A) PLAN



(B) ELEVATION

Figure 9-11 Typical Air Valve Installation

9.5.1 Tie-Down Anchors

Tie-down anchors are discussed generally in Section 8.2, which describes a penstock pipe partially encased in concrete at an anchorage. At these locations restraint is always necessary without regard to bend direction. At a downward-to-upward bend, the net thrust is directed into the ground and the tie-down anchors prevent lateral movement or slippage. For an upward-to-downward bend, net thrust is upward and must be opposed by the tie-down bolts extending from a concrete anchor block.

Tie-down tension anchors and related members are attached around the penstock shell as shown in Figure 9-12. Bolt diameters are likely to exceed 1½ inches, and the embedded length of such tension members must extend into the concrete a distance at least 40 times the diameter of the anchor bolt. Nuts and washers or shear plates are placed on the lower or embedded end bolt and surrounded with reinforcing steel.

Threaded connections are superior to welded ones for field assembly of the tie-down anchors, especially if reinforcing bars or rods of high-strength material are used for anchor bars.

Good construction practice requires that the completed tie-down assemblies be attached to the penstock bend in its final position as fresh concrete is cast under the penstock. This effectively secures unified action between the tension ties and the anchor block concrete, especially if the tie-down anchors are shipped in several pieces.

The use of penstock bend anchors as described above implies that large forces will be transferred from the penstock shell to the concrete. To prevent buckling of the shell under this exterior loading, stiffeners commonly are used around the pipe at the points of tie-down attachment. For lighter loading, only wide flat straps may be suitable.

9.5.2 Anchor Bolts

Anchor bolts are used at supports for penstock rocker bearings, base plates for valves, and bearings under ring girder bearings and saddles. Usually anchor bolts are under 1½ inches in diameter. Except when loaded under seismic forces that cause overturning, anchor bolts are most likely loaded by shear forces rather than tension. Shear load on these bolts may be reduced if lugs or shear plates are welded under base plates to better interact with the concrete of the base structure.

Since shear loading is most likely on the anchor bolts, the specifications for bolt material may vary. For most loading conditions, mild steel bolts conforming to ASTM A-307 specifications are adequate. When shear load on a bolt reaches 0.17 of ultimate strength for the bolt material, other steel bolt material should be considered because of lower stress limitations on mild steel bolts loaded by single shear in the threaded area. As an example, the AISC Specification for the Design, Fabrication and Erection of Structural Steel for Buildings⁴ limits the shear value for bearing on A-307 bolts to 10.0 ksi, while similar shear allowed for A-325 bolts is 21.0 ksi. A-325 bolts may be specified, not because they are tensioned to capacity, but because they are easily obtained.

If accuracy of bolt or fastener position is necessary to match shop fabrication, the usual practice is to install the anchor bolts inside pipe sleeves that extend at least 30 inches into the structural concrete. Only the bottom ends of the bolts initially are attached to the plates under the ends of the pipe sleeves (see Figure 9-13). The bolts may be adjusted somewhat in the field if the installed bolt location differs from the shop fabrication location. The void around a bolt in a pipe sleeve is filled with concrete grout after erection work has passed that point.

It is preferable to use leveling nuts on all anchor bolts. Leveling nuts will transfer erection dead loads onto the threaded portions of the bolts. Each bolt in a sleeve will act as a long column before grouting; often each bolt must be laterally supported by shims in the sleeve before grouting.

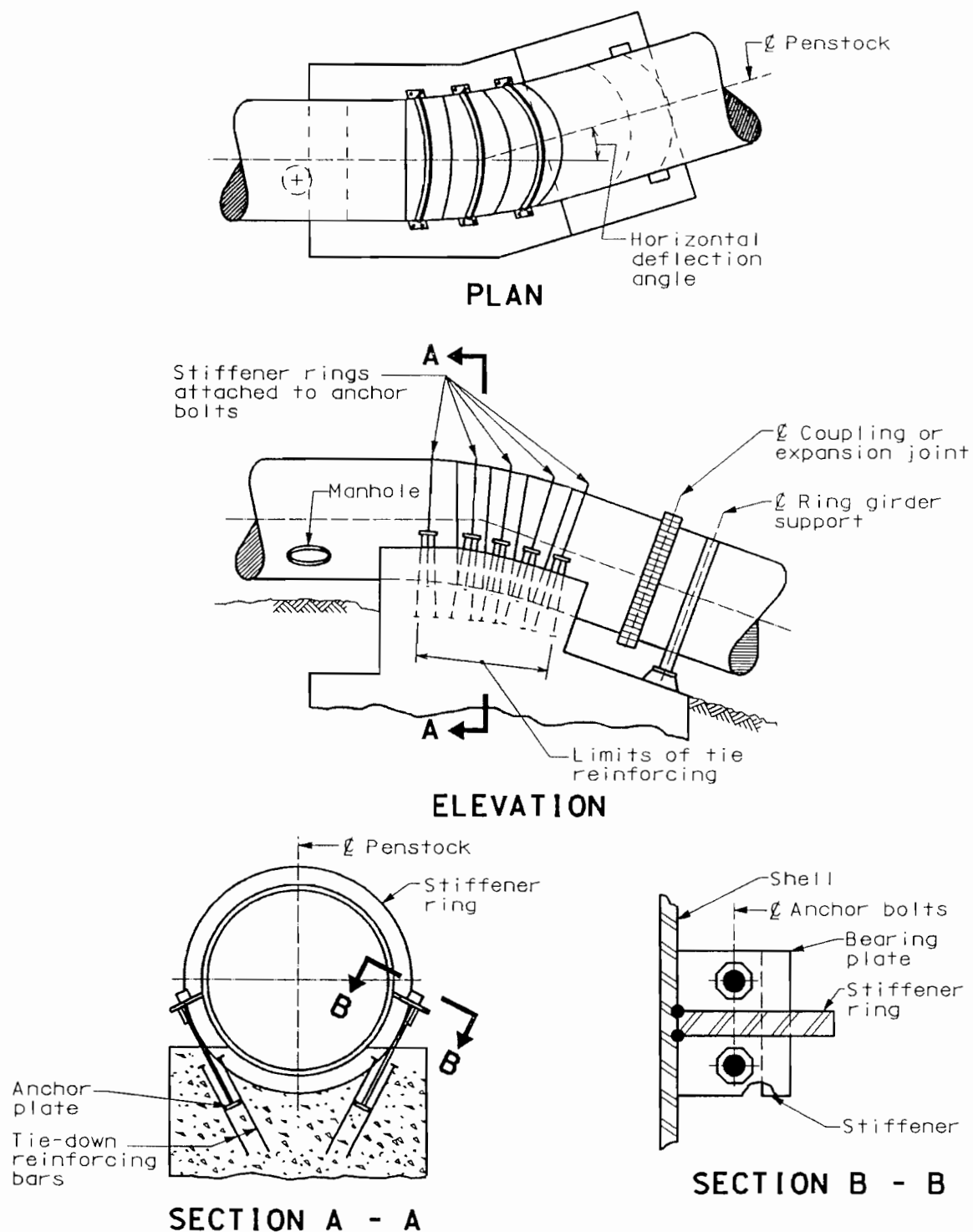
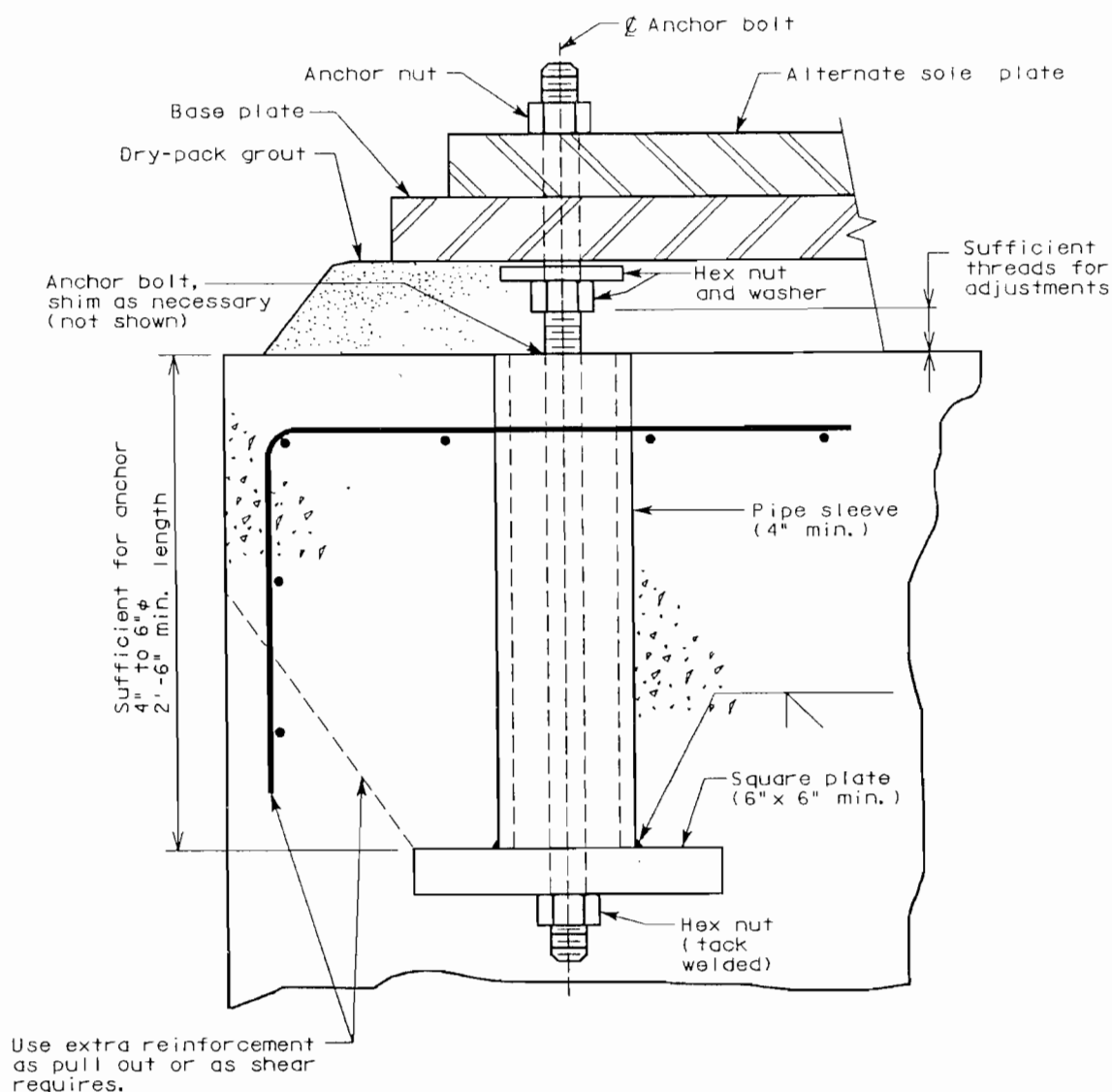


Figure 9-12 Tie-Down Anchors



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Figure 9-13 Typical Anchor Bolt in Pipe Sleeve

In addition, bolt or sleeve diameters are selected so that the projected diameters under the lateral loading will bear on the concrete at less than 500 psi. Further, pull-out forces on the sleeves may require additional reinforcement and embedment cover on the adjacent outside faces of the concrete anchor block.

For lightly loaded anchor bolts less than 1¼ inches in diameter, it may be practical to set the penstock component in place and drill the anchor holes through the base plates, assuming space is available for drilling. Expansion anchors or epoxy-grouted anchorages may be used as required for adverse vibration conditions.

9.6 Anchor Rings, Thrust Collars, and Joint Harnesses

9.6.1 Scope

Anchor rings, thrust collars, and joint harness lugs are fabricated from steel plates that are attached to the penstock circumference by continuous fillet welds. Special conditions may warrant the use of groove welds. Anchor rings and thrust collars provide resistance to thrust forces and are typically used for penstocks embedded in concrete. Joint harnesses provide a means of restraining penstocks from movement due to thrust forces at joints that are not capable of providing restraint.

9.6.2 Anchor Rings and Thrust Collars

Anchor rings and thrust collars that are embedded in concrete must be designed to resist thrust loads in accordance with established engineering principles. The average bearing stress of the ring against concrete encasement must not exceed 0.3 times the specified 28-day compressive stress of the concrete. Anchor rings and thrust collars must be proportioned to transfer thrust loads from the penstock to the encasement primarily through shear. Anchor rings and thrust collars must be dimensioned and detailed according to Table 9-4 and Figure 9-14, respectively. Consideration must be given to minimizing penstock wall thickness and welding requirements. It may be necessary to thicken the penstock shell or add wrapper plate reinforcement to reduce stresses in the penstock wall at anchor rings and thrust collars. Also, the thickened penstock shell or wrapper plate, if required, must extend beyond the limit of concrete encasement to limit secondary bending stress. "Punching shear" resistance of the concrete encasement also must be investigated.

Table 9-4 Anchor Ring Dimensions

PENSTOCK O.D. (D) (IN.)	ANCHOR RING WIDTH (A) (IN.)	RING THICKNESS (B) (IN.)	PENSTOCK THICKNESS (T _y) (IN.)	FILLET WELD SIZE (T _w) (IN.)	PERMISSIBLE LOAD ON ANCHOR RING (F) (LB.)
ANCHOR RINGS FOR 150-psi PENSTOCK PRESSURE (NORMAL CONDITION) (1.0 MPa)					
48	4	0.875	0.670	0.250	271,434
54	4	0.875	0.720	0.250	343,534
60	5	1.000	0.840	0.250	424,116
66	5	1.125	0.890	0.250	513,180
72	5	1.125	0.940	0.250	610,727
78	6	1.250	1.060	0.375	716,756
84	6	1.250	1.100	0.375	831,267
90	7	1.500	1.230	0.375	954,261
96	7	1.500	1.270	0.375	1,085,737
ANCHOR RINGS FOR 250-psi PENSTOCK PRESSURE (NORMAL CONDITION) (1.723 MPa)					
48	6	1.250	1.070	0.375	452,390
54	6	1.375	1.150	0.375	572,557
60	7	1.500	1.300	0.375	706,860
66	8	1.750	1.460	0.375	855,301
72	8	1.750	1.530	0.500	1,017,878
78	9	2.000	1.690	0.500	1,194,593
84	10	2.125	1.840	0.500	1,385,445
90	11	2.250	1.920	0.625	1,590,435
96	11	2.500	2.070	0.625	1,809,562

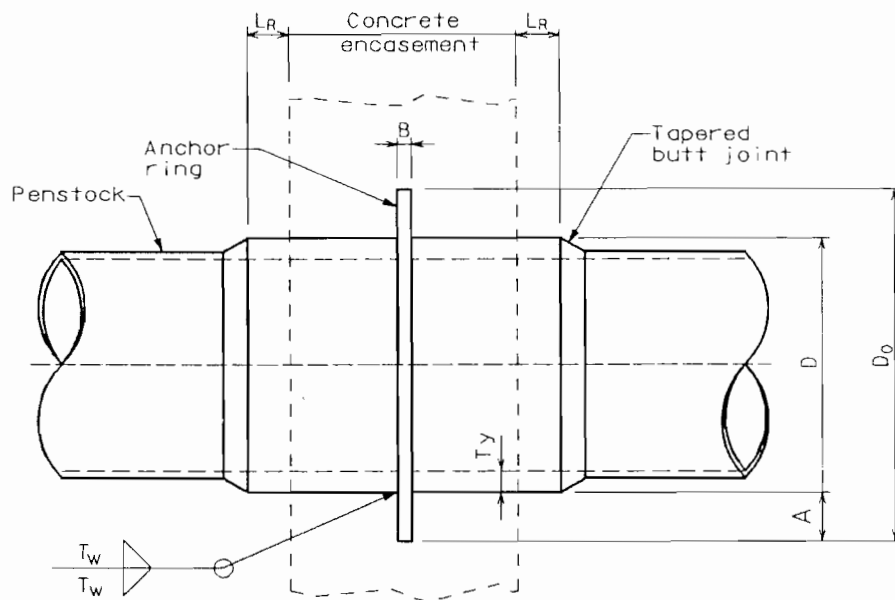


Figure 9-14 Anchor Ring and Thrust Collar Detail

The table values assume that anchor rings are fabricated from A-36 steel, the penstock is fabricated from ASTM A516, Grade 60 steel, and E70XX welding electrodes are used.

A procedure for designing rings and collars to accommodate thrust loads has been proposed by Gary Redmond and others. The procedure assumes the following:

- (1) The ring transfers loads to the concrete encasement with a contact pressure that varies linearly from zero at the tip of the ring to 1,000 psi at the ring/penstock connection (triangular pressure diagram with an average allowable ring contact pressure of 500 psi).
- (2) The ring will resist a full "dead end" thrust force as the result of pressure (thrust = pressure times area of penstock cross section).
- (3) Thrust restraint from forces other than internal pressure (thermal, for example) has not been included.
- (4) Maximum bending stress in the ring is limited to 60% of yield or approximately 22 ksi for a ring fabricated from A-36 steel.
- (5) Maximum principal and effective stresses in the penstock shell at the shell/ring junction, according to the Hencky-von Mises theory, are limited to the lesser of 1/3 tensile or 2/3 yield or approximately 20 ksi for a penstock shell fabricated from ASTM A516, Grade 60 steel.
- (6) Since thrust rings are embedded in concrete, the penstock will be fully restrained from longitudinal and circumferential growth due to internal pressure; thus, stresses in the penstock wall are affected by Poisson's ratio (γ) which is assumed to be 0.3.

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- (7) The depth of anchor ring embedment in the concrete encasement must be sufficient to resist punching shear forces when thrust load is transferred from the ring to the encasement.

The anchor ring dimensions given in Table 9-4 (for penstocks 48 inches through 96 inches in diameter) were developed using this procedure.

9.6.2.1 Equations

Following are equations for determining the allowable stresses and other parameters for anchor rings and thrust collars.

(1) Average bearing stress

The average bearing stress (σ_a) of the anchor ring on the surrounding concrete encasement is limited to allowable concrete bearing and is given by:

$$\sigma_a = \frac{P_n D^2}{D_o^2 - D^2} \leq 0.3 f'_c \quad (\text{Equation 9-5})$$

Where:

- σ_a = average bearing stress of the anchor ring on surrounding concrete, psi
- P_n = internal pressure under normal conditions, psi
- D = penstock outside diameter, in.
- D_o = ring outside diameter, in.
- f'_c = concrete encasement specified 28-day compressive stress, psi

The ring, which resists thrust forces from the penstock, applies a bending moment to the penstock wall. If a triangular bearing pressure distribution, varying from zero pressure at the ring tip to maximum pressure at the ring/penstock wall connection is assumed, then the unit moment (M_o) at the base of the ring of width A is:

$$M_o = \left(\frac{\pi P D^2}{4} \right) \left(\frac{1}{\pi D} \right) \left(\frac{A}{3} \right) = \frac{P D A}{12} \quad (\text{Equation 9-6})$$

Where:

- M_o = unit bending moment in ring at ring/penstock wall connection, in.-lb/in.
- P = normal design pressure, psi
- A = anchor ring width, in.
- D = penstock outside diameter, in.

$$\text{Use } \leq \phi (0.85 f'_c A_1) \quad \phi = 0.85$$

from N25 3101

$$0.85 \times (0.85 f'_c A_1) \times 2 \quad \text{clause 16.3}$$

(2) Bending stress

The bending stress (σ_r) in the ring, assuming a ring of rectangular cross section, is given by:

$$\sigma_r = \frac{6M_o}{B^2} \quad (\text{Equation 9-7})$$

Where:

- σ_r = bending stress in the ring/penstock wall connection, psi
- M_o = unit bending moment, in.-lb/in.
- B = anchor ring thickness, in.

The sum of the moments at the ring/penstock wall connection must equal zero; thus, the internal moments in the penstock wall, on either side of the ring, are one-half of the ring moment in Equation 9-6 above, or:

$$M_1 = \frac{M_o}{2} = \frac{PDA}{24} \quad (\text{Equation 9-8})$$

Where:

- M_1 = unit bending moment in penstock wall on each side of ring, in.-lb/in.
- M_o = unit bending moment, in.-lb/in.

(3) Longitudinal stress

The longitudinal stress (σ_1) in the penstock wall, which is completely restrained from movement by the surrounding concrete encasement, is given by:

$$\sigma_1 = \frac{PD}{4T_y} + \gamma \frac{PD}{2T_y} + \frac{6M_1}{T_y^2} \leq \sigma_n \quad (\text{Equation 9-9})$$

(4) Circumferential stress

The circumferential stress (σ_2) in the penstock wall, which is completely restrained from movement by the surrounding concrete encasement, is given by:

$$\sigma_2 = \frac{PD}{2T_y} + \gamma \left(\frac{PD}{4T_y} + \frac{6M_1}{T_y^2} \right) \leq \sigma_n \quad (\text{Equation 9-10})$$

Where:

- σ_1 = longitudinal stress in the penstock wall, psi
- σ_2 = circumferential stress in the penstock wall, psi
- P = normal design pressure, psi

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- D = penstock outside diameter, in.
 T_y = penstock thickness, in.
 γ = Poisson's ratio (0.3 for carbon steel)
 M_1 = unit bending moment in penstock wall on each side of ring, in.-lb/in.
 σ_n = allowable bending stress for normal conditions, psi

(5) Equivalent stress

The combined Hencky-von Mises equivalent stress (σ_e) is given by:

$$\sigma_e = \left(\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2 \right)^{1/2} \leq \sigma_n \quad (\text{Equation 9-11})$$

(6) Secondary bending stress

The secondary bending stress (σ_b) in the penstock shell at the penstock/concrete encasement interface is given by:

$$\sigma_b = 1.82 \left(\frac{PR}{T_y} \right) \quad (\text{Equation 9-12})$$

Where:

- σ_b = secondary shell bending stress, psi
 P = normal design pressure, psi
 R = penstock radius, in.
 T_y = penstock thickness, in.

(7) Extension of penstock shell beyond concrete encasement

The length that the thickened penstock shell must extend beyond the concrete encasement limits is given by:

$$L_R = 2.33 \left(RT_y \right)^{1/2} \quad (\text{Equation 9-13})$$

Where:

- L_R = length that thickened penstock shell must extend beyond concrete encasement limits, in.
 R = penstock radius, in.
 T_y = penstock thickness, in.

(8) Fillet weld size

The size (T_w) of the fillet welds that attach the ring to the penstock shell, assuming use of E60XX electrodes, is given by:

$$T_w = \frac{f_r}{\left[(0.3) (60,000) \left(\frac{\sqrt{2}}{2} \right) \right]} = \frac{f_r}{12,728} \quad (\text{Equation 9-14})$$

The following equations define f_r , f_b , and f_v :

$$f_r = (f_b^2 + f_v^2)^{1/2} \quad (\text{Equation 9-15})$$

$$f_b = \frac{M_o}{\left(B + \frac{T_w}{2} \right)} \quad (\text{Equation 9-16})$$

$$f_v = \frac{PD}{8} \quad (\text{Equation 9-17})$$

Where:

- T_w = fillet weld size, in.
- f_r = resultant shear force in fillet weld, lb/in.
- f_b = unit shear force in fillet weld resisting anchor ring bending, lb/in.
- f_v = unit shear force in fillet weld resisting direct shear from anchor ring, lb/in.
- M_o = unit bending moment in ring at ring/penstock wall connection, in-lb/in.
- B = anchor ring thickness, in.
- P = normal design pressure, psi
- D = penstock outside diameter, in.

9.6.3 Joint Harnesses

Joint harnesses are used to transfer thrust loads across couplings or joints that are not capable of resisting thrust loads along the penstock. The design of joint harnesses must comply with specifications given in AWWA Manual M11⁵ (Paragraph 13.10, Tables 13-6 and 13-7, and Figure 13-17). Figures 9-15 through 9-17 show typical joint harness configurations and details.

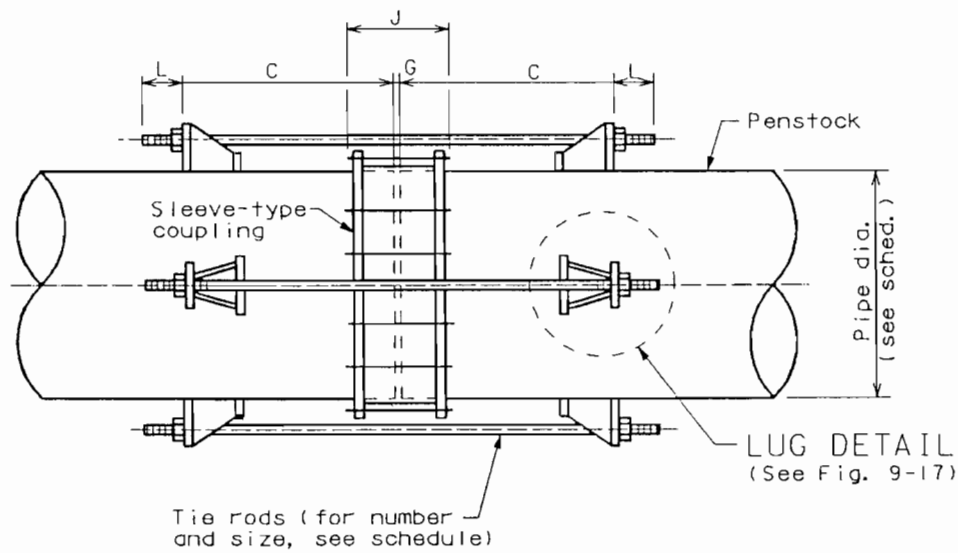


Figure 9-15 Joint Harness for Single-Bolted, Sleeve-Type Coupling Installation

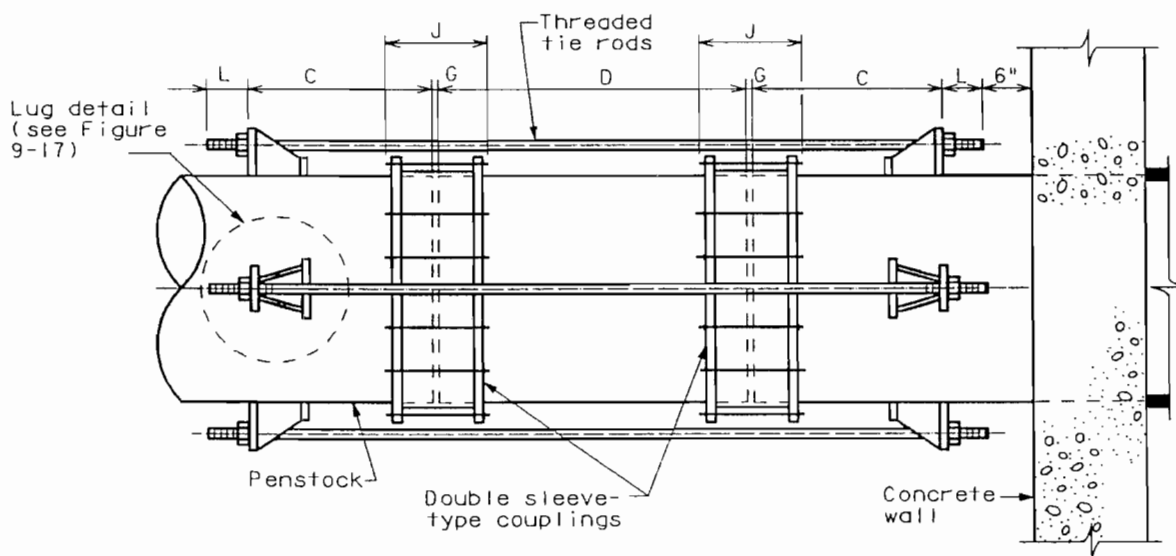


Figure 9-16 Joint Harness for Double-Bolted, Sleeve-Type Coupling Installation

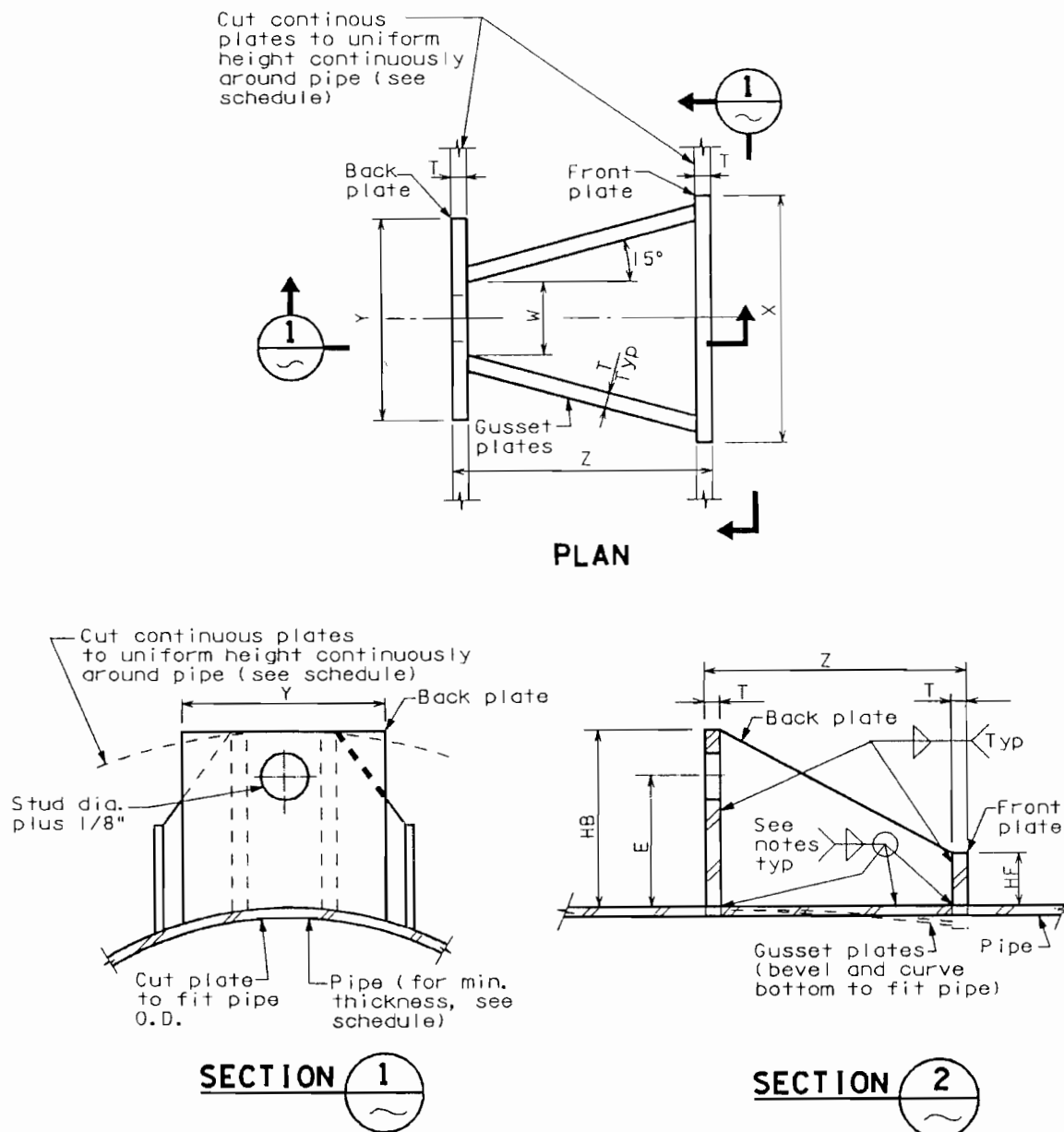


Figure 9-17 Lug Detail for Joint Harness

For penstocks larger than 96 inches in diameter, a procedure for designing joint harnesses to accommodate thrust loads has been proposed by R. L. Brockenbrough.⁶

9.7 Penstock Roll-Out and Lift-Out Sections

For larger penstocks, especially those connecting into tunnels, full-diameter access to the interior is desirable through either a roll-out or lift-out penstock section. This is a good alternative to the severe dimensional limitations of manholes or mandrels. Removable sections provide space for wheeled and powered equipment to enter into the waterway for maintenance and repairs.

For the opening, a short spool piece of penstock is provided and connected into the main conduit with sleeve-type couplings. Sleeve-type couplings are much more flexible than flanges and less costly. The spool always must be supported by ring girders or some equivalent method and the coupling middle rings furnished without pipe stops. A manhole provides short time access. Some designs include rails on which to roll a heavy section or a simple lift-out piece with crane hooks.

Typical designs for roll-out sections are shown in Figures 9-18 and 9-19.

9.8 Flanged Connections

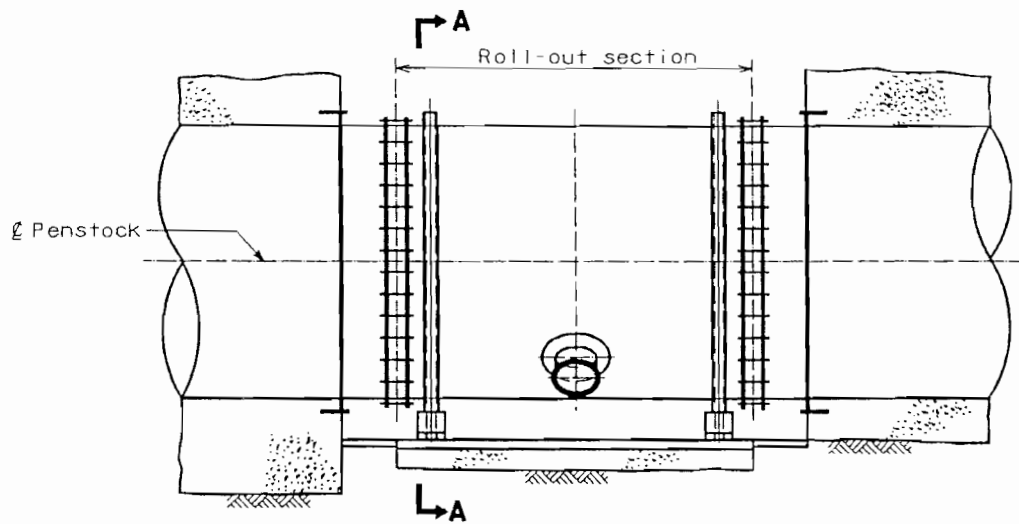
9.8.1 Scope

This section covers steel pipe flanges for use on penstocks with reference to the following published flange standards.

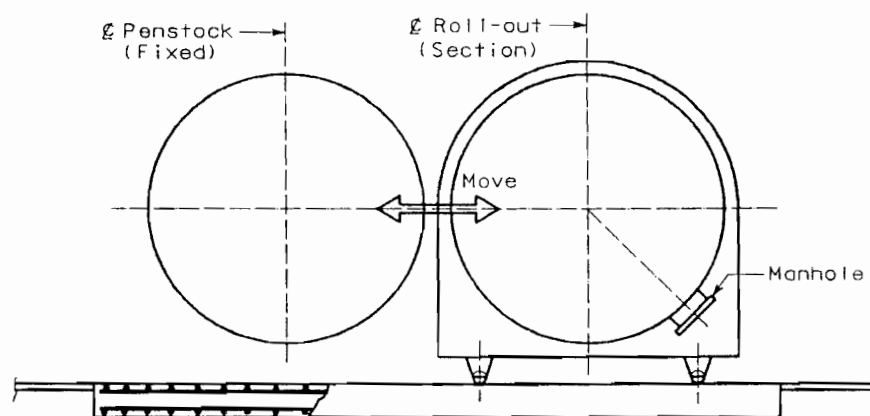
- (1) ANSI/AWWA C207, Steel Pipe Flanges for Waterworks Service—Sizes 4 inches through 144 inches⁷
- (2) ASME/ANSI B16.5, Pipe Flanges and Flanged Fittings⁸
- (3) API Standard 605, Large-Diameter Carbon Steel Flanges⁹
- (4) MSS SP-44, Steel Pipe Line Flanges.¹⁰

The primary differences between the standards involve size and pressure rating limitations, flange type, and flange facing. The standard used for a particular project should be selected based on penstock diameter and design pressure, and by mating requirements with existing valves and fittings.

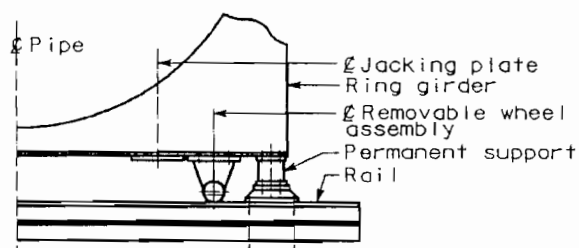
The following subsections provide a summary of basic flange considerations. In all cases, flanges must conform to the specifications of the applicable flange standard.



ELEVATION

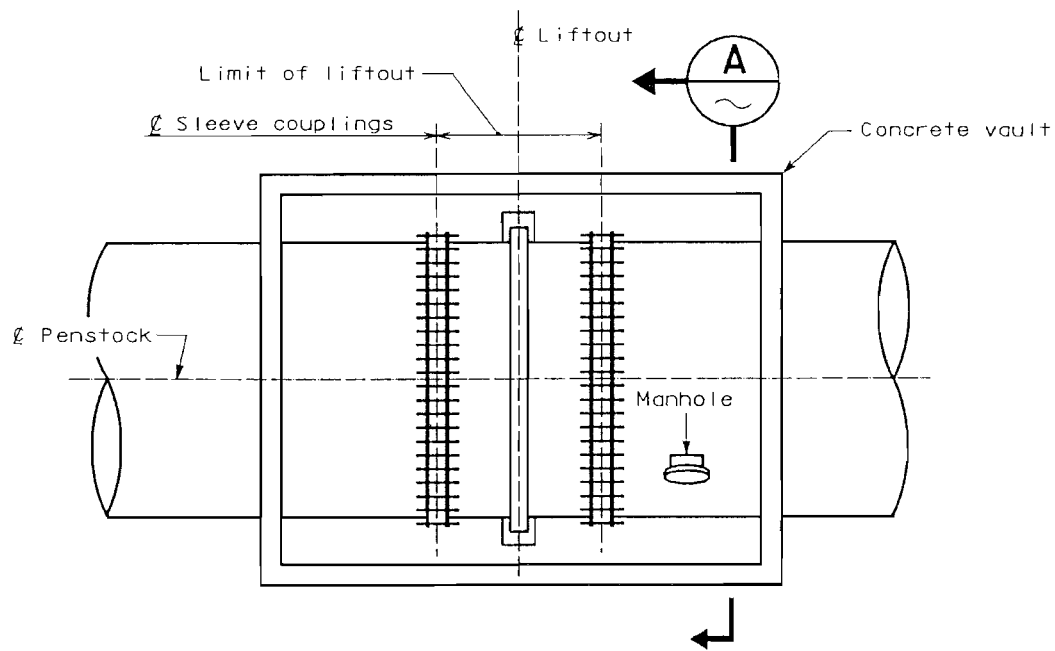


SECTION A-A

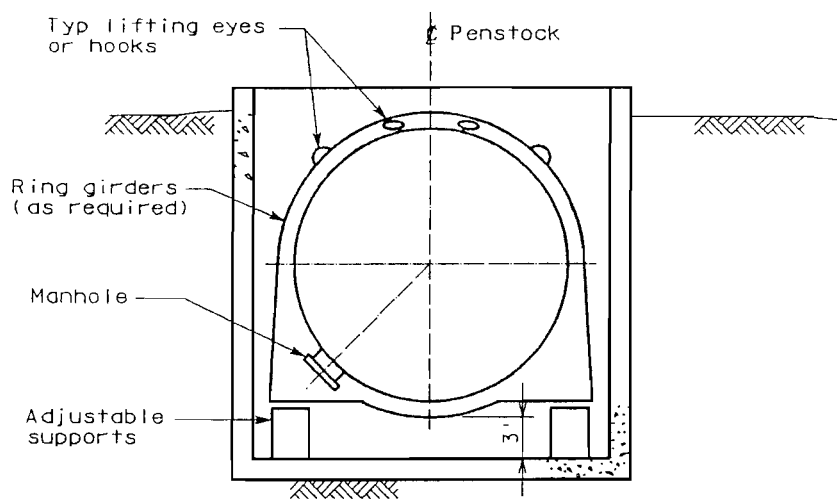


ALTERNATIVE BASE

Figure 9-18 Penstock Roll-Out Section



PLAN



SECTION A

Figure 9-19 Penstock Lift-Out Section

9.8.2 Pressure Ratings

9.8.2.1 ANSI/AWWA C207 Standard

The AWWA C207 standard applies to flanges and blind flanges ranging in size from 4 through 144 inches with pressure ratings up to 275 psi, and from 4 to 48 inches with a pressure rating of 300 psi, at atmospheric temperature. Pressure ratings are for customary conditions and temperatures in waterworks service. The pressure rating is based on the design of the maximum operating pressure plus the anticipated surge and waterhammer pressure. Test pressures for flanges conforming to this standard must not exceed 125% of the ratings. Pressure ratings identified in AWWA C207 are given in Table 9-5.

Table 9-5 ANSI/AWWA C207 Pressure Ratings

CLASS	NOMINAL PIPE SIZE (IN.)	PRESSURE RATING (psi)
B	4-144	86
D	4-12	175
D	14-144	150
E	4-144	275
F	4-48	300

9.8.2.2 ASME/ANSI B16.5 Standard

The ASME/ANSI B16.5 Standard generally is used for high-pressure, high-temperature mechanical piping; however, it also may be used for small diameter penstocks. It applies to flanges and blind flanges in sizes up to 24 inches in rating classes 150, 300, 400, 600, 900, 1500, and 2500. The allowable pressure for each rating class, defined as the maximum allowable nonshock working gage pressure, depends on the alloys used and the temperature of the penstock shell, which is generally the same as that of the fluid. Pressure ratings identified in ANSI B16.5 for carbon steel (Material Group 1.1) with a temperature between -20°F and 100°F are given in Table 9-6.

Table 9-6 ASME/ANSI B16.5 Pressure Ratings

CLASS	PRESSURE RATING (psi)
150	285
300	740
400	990
600	1,480
900	2,220

9.8.2.3 API Standard 605

The American Petroleum Institute API Standard 605 applies to flanges ranging in size from 26 through 60 inches with pressure ratings up to 1,480 psi, and from 26 through 48 inches with a pressure rating of 2,220 psi. Pressure ratings are the maximum allowable nonshock working gage pressure, at the specified temperature. API Standard 605 pressure ratings for temperatures less than 100°F are given in Table 9-7.

Table 9-7 API Standard 605 Pressure Ratings

CLASS	NOMINAL PIPE SIZE (IN.)	PRESSURE RATING (psi)
75	26-60	140
150	26-60	285
300	26-60	740
400	26-60	990
600	26-60	1,480
900	26-48	2,220

9.8.2.4 MSS SP-44 Standard Practice

The MSS SP-44 Standard Practice applies to flanges and blind flanges ranging in size from 12 to 60 inches with pressure ratings up to 1,480 psi, and from 12 to 48 inches with a pressure rating of 2,220 psi, at atmospheric temperature conditions. Atmospheric temperature, for purposes of the standard practice, is defined to range from -20°F to 250°F. Pressure ratings are the maximum allowable nonshock working gage pressure. MSS SP-44 pressure ratings are given in Table 9-8.

Table 9-8 MSS SP-44 Pressure Ratings

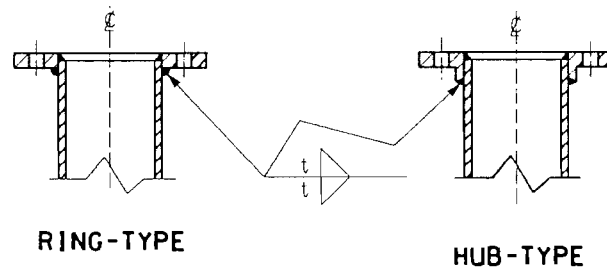
CLASS	NOMINAL PIPE SIZE (IN.)	PRESSURE RATING (psi)
150	12-60	285
300	12-60	740
400	12-60	990
600	12-60	1,480
900	12-48	2,220

For flanged connections outside the pressure class limits of the referenced standards, flanges should be designed according to the ASME Boiler and Pressure Vessel Code,¹² Section VIII, Division 1, Appendix 2, "Rules for Bolted Flange Connections with Ring-Type Gaskets," or Appendix Y, "Flat Face Flanges with Metal-to-Metal Contact Outside the Bolt Circle."

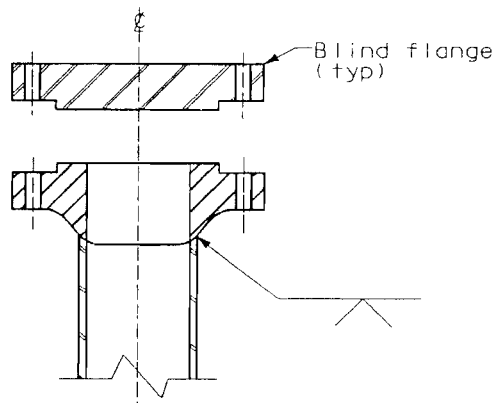
9.8.3 Flange Types

Depending on the standard applied, flanges may be ring-type or hub-type slip-on flanges, or welding neck type, as shown in Figure 9-20. Slip-on type flanges are attached to the penstock or appurtenance by two fillet welds. Welding neck flanges are butt welded to the penstock.

Flanges conforming to AWWA C207 may be either ring-type or hub-type slip-on flanges. Additionally, welding neck flanges may be used instead of slip-on flanges for pressure Class E, at the purchaser's option. Flanges conforming to ANSI B16.5 may be of the slip-on or welding neck type. Flanges conforming to API Standard 605 or MSS SP-44 are welding neck type flanges.



(A) SLIP-ON FLANGES



(B) WELDING NECK FLANGES

Figure 9-20 Flange Types

9.8.4 Facing

Flanges conforming to AWWA C207 are flat faced, including welding neck flanges used for pressure Class E service. Flanges conforming to ANSI B16.5, API Standard 605, and MSS SP-44 are of the raised-face type for all pressure classes. A 0.06-inch raised face is regularly furnished for pressure classes 300 and lower, whereas a 0.25-inch raised face is regularly furnished for pressure classes 400 and higher. Connecting a raised-face flange to a flat-faced flange causes cracking or distorting of one of the flanges, resulting in leakage. When this situation cannot be avoided, the raised face should be machined off and a full-face gasket used, subject to the provisions of the flange standard.

For proper gasket seating, the flatness of the machined facing surface of the flange can become critical for large, high-pressure flanges. The tolerances given in Table 9-9, relative to a plane located perpendicular to the penstock axis, are recommended.

Table 9-9 Flatness Tolerances

PENSTOCK DIAMETER (IN.)	FLATNESS TOLERANCE (IN.)
LESS THAN 24	± 0.005
24 - 48	± 0.100
50 - 144	± 0.015
GREATER THAN 144	± 0.020

9.8.5 Gaskets

Gasket materials should be selected in accordance with the provisions of the flange standard. Gasket design is based on expected bolt loading and service conditions and, where applicable, the need to provide an insulated joint. Rubber or nonasbestos ring-type gaskets generally are used for cold-water penstock service.

9.8.6 Drilling

Drilling templates are in multiples of four so that fittings can be made to face any quarter. Bolt holes are drilled to straddle the centerline, except where special mating conditions exist. When different standards are used for companion flanges, the designer must verify that the drilling patterns are compatible. When companion flanges are required for flanged valves and fittings, the designer may wish to consider having the valve or fitting manufacturer supply the companion flange.

9.9 Blowoffs

9.9.1 General

Blowoffs are provided at the low end of the penstock section to dewater the line. Sometimes, there are intermediate sections of penstock with sags that require additional blowoffs. A blowoff also must be provided just outside a tunnel portal to drain any tunnel leakage flow that may be present during penstock maintenance. In this case, the blowoff should be conservatively sized to account for unknown tunnel leakage. The exact location of blowoffs frequently is influenced by opportunities to dispose of the water. Where a penstock crosses a stream or drainage structure, usually there is a low point in the line; however, if the penstock runs under the stream or drain, it obviously cannot be drained completely into the channel. In such a situation, it is preferable to locate the blowoff connection at the lowest point that will drain by gravity and, if necessary, provide a means for pumping out the section below the blowoff. Under normal circumstances, most of the penstock will be drained through the plant. However, the blowoff components should be designed for full-head conditions in the event that draining through the plant does not occur. Where possible, the blowoff pipeline should be sloped to allow complete drainage.

9.9.2 Blowoff Valves

Blowoffs should be provided with both a shutoff valve and guard valve. If the penstock is above ground, the valves should be attached directly to the outlet of the penstock. For aboveground penstocks, freeze protection may be required for the blowoff valves. A pipe attached to the valves is used to route the discharge to a safe location. The discharge pipe usually requires

installation of an elbow at the blowoff valve assembly, which must be blocked securely to avoid stresses at the attachment to the penstock.

Usually the blowoff is below ground. Because the operating nuts of the valve must be accessible from the surface, the valves cannot be located beneath the penstock, but may be set with the stems vertical and just beyond the side of the penstock. If desired, the blowoff valves may be located within a manhole or valve vault to be more readily accessible for maintenance. The size of the valves and discharge pipe is dependent on the length of penstock to be dewatered and the time required to drain the line. Other considerations are the size of the outlet relative to the penstock size and the need to provide reinforcement.

9.9.3 Outlets

Typically outlets are welded to the penstock with reinforcing collars. Generally this work is done in the shop during penstock fabrication. Shop lining and coating of the outlets is satisfactory and more economical than work done in the field. All outlets should be checked to determine if reinforcement is required; however, for outlets larger than about 1/3 the diameter of the line, reinforcing should be given special consideration, even for small size penstocks. Outlets that are encased in concrete should be wrapped in compressible material to prevent bending stresses.

The end of the outlet should be prepared to receive the valve or fitting to be attached. This may call for a flange, a plain end for a flexible coupling joint, a grooved spigot end for a bell-and-spigot joint, or a threaded end.

9.9.4 Blowoff Details

A typical blowoff detail for a buried penstock is shown in Figure 9-21. In this example, two blowoff valves have been provided. The gate valve functions as a guard valve and is intended for watertight shutoff. The guard valve remains wide open during draining of the penstock. The ball valve also may be operated wide open, but could be used for flow control if necessary. The double-valve system is used because abrasion resulting from washing sediment out of the penstock may cause the ball valve seals to become worn, making tight shutoff impossible. If necessary, an orifice plate could be installed between the valve assembly and outlet structure to further limit flow rate. If the penstock is installed at considerable depth, the blowoff may discharge into a manhole (as shown in Figure 9-21) or to a riser pipe, and then into the drain pipe. In this case, the manhole must be pumped out to fully dewater the penstock. Where topography is favorable, the manhole may be omitted, and the blowoff piping may be routed to drain by gravity to a natural drainage way.

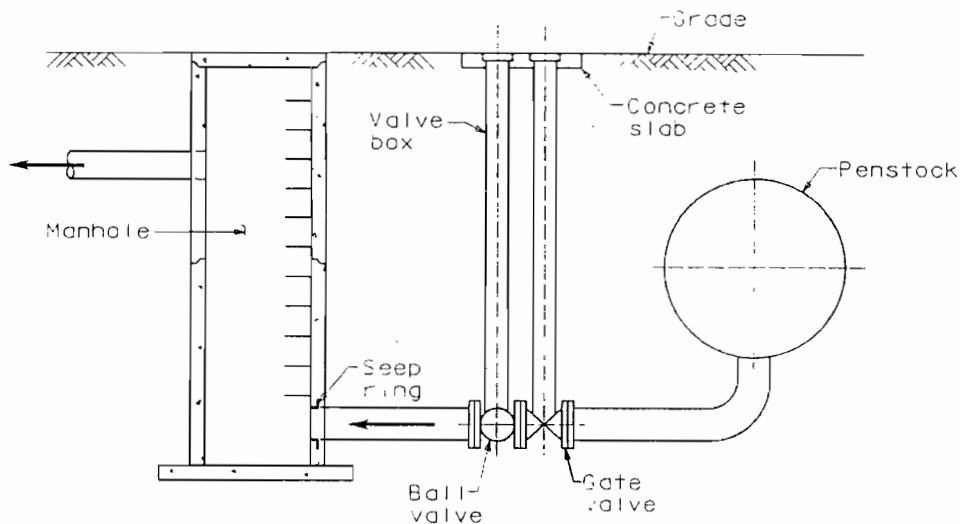


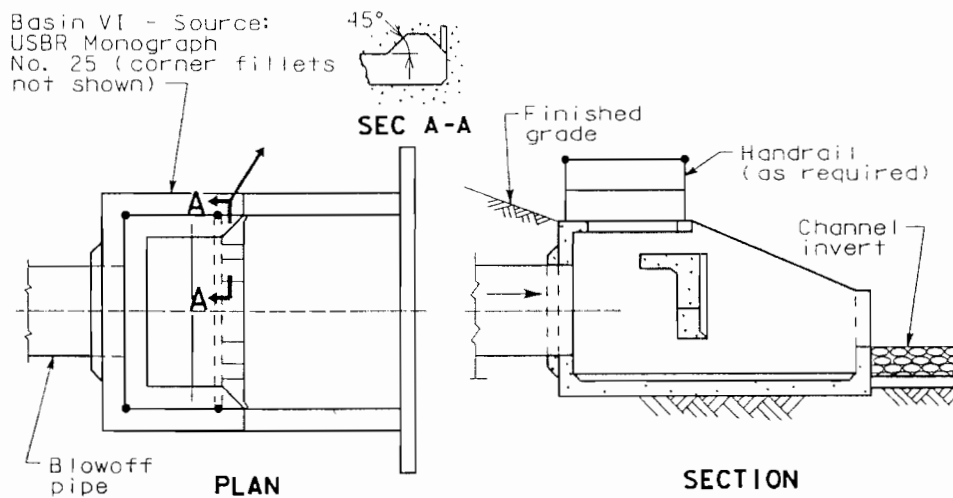
Figure 9-21 Typical Blowoff Detail

A concrete headwall structure typically is provided at the blowoff pipeline termination to dissipate the discharge water energy and prevent localized erosion within the channel accepting the discharge. Numerous headwall designs are possible. Two typical headwall configurations are shown in Figure 9-22.

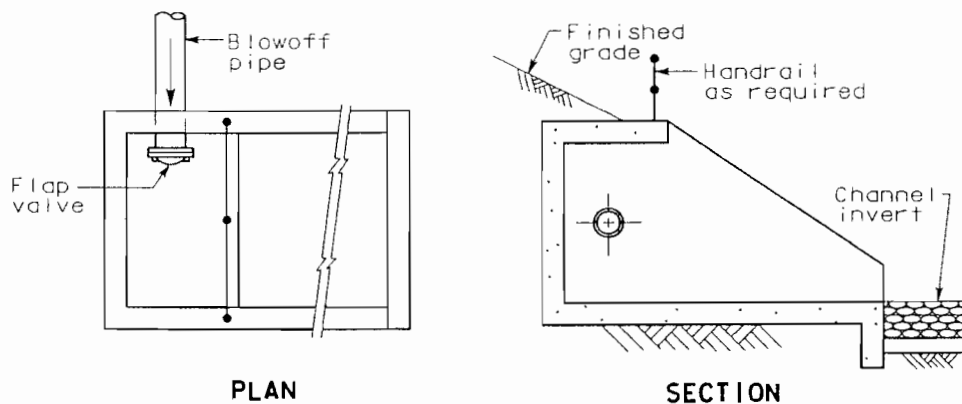
9.10 Makeup/Closure Sections

9.10.1 Field Trim

Makeup or field trim sections may be required in a penstock to adjust for the normal accuracy of surveys and pipe fabrication when a critical point in the pipe laying sequence is reached, such as a major elbow or a bifurcation. To resolve this problem, the shop pipe is furnished in a length sufficiently long to allow for field trim.



(A) USBR IMPACT-TYPE ENERGY DISSIPATOR



(B) TYPICAL HEADWALL

Figure 9-22 Typical Blowoff Headwall

9.10.2 Closure Sections

Closure sections are required when pipe must be laid between two fixed points or when pipe is laid from more than one starting point, as often occurs on long lines. In this case a butt-strap closure, as shown in Figure 9-23, may be used for pressures up to 400 psi.

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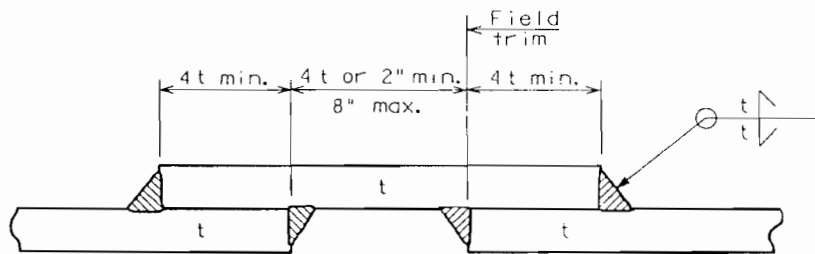


Figure 9-23 Butt-Strap Closure

For higher pressures a butt-weld closure section must be dropped into the line. Because closures usually involve adjustment for some degree of angular and axial misalignment between the two ends of the existing pipe, an outside backup bar may be used to provide a butt-weld joint that can accommodate this problem (see Figure 9-24).

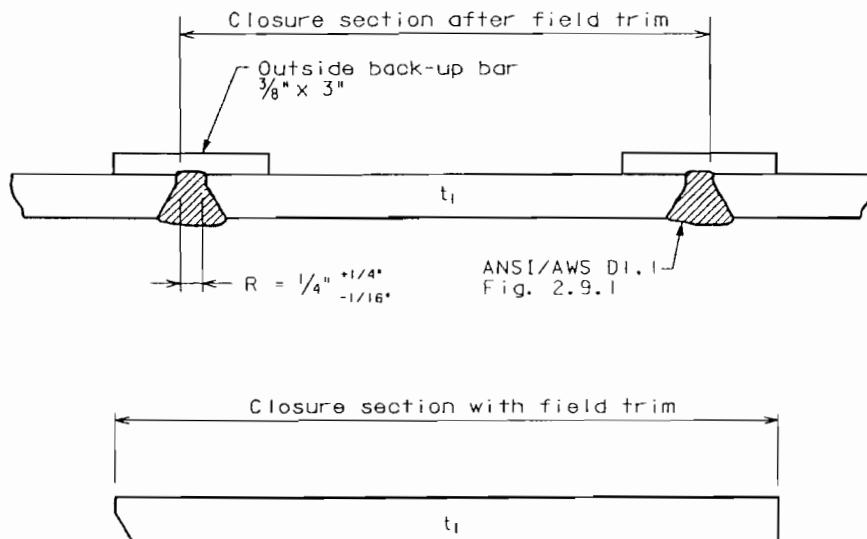


Figure 9-24 Butt-Weld Closure

When making closure welds, temperature stresses must be considered. The closure should be made after backfill has been brought up to the closure section and at a pipe temperature as close to the normal operating temperature as possible. This usually occurs at the coolest part of the day.

9.11 Closure Heads

Closure heads commonly are used to block off the end of a pipe line, either as a temporary measure during hydro-testing and maintenance or as a permanent closure.

Closure heads may have the following construction:

- (1) Flanged-type connections
- (2) Flat-disk closures
- (3) Elliptical, torispherical, and hemispherical dished closures.

Flanged-type closures commonly used for penstocks are slip-on type flanges that are welded to the penstock and a blind flange. Flanges can be of several classes or fabrication methods, including:

- (1) Steel ring, hubless flanges made from rolled plate, billet or curved flat bars
- (2) Cast steel, using a welding hub
- (3) Forged steel, using a hub or a welding neck.

Blind flange-type closure heads can be used effectively with "flanged" pipe connections. This type of closure head can be economical for pipes up to 48 inches in diameter, with a nominal design pressure up to 150 psi.

Section 9.8 covers particular flange data, design criteria, and applications. Reinforced (stiffened) blind flange-type closures can be designed for pipe diameters greater than 48 inches and pressures in excess of 150 psi. This section is limited to the utilization, design, and application of dished closures and special, large, designed flanges.

In considering the selection between the blind flange type or the dished closure, the designer must consider: (1) functional use, (2) pressure considerations, and (3) size limitations.

Dished head closures usually are made of the same material as the penstock. However, different but compatible material should be considered to ensure the availability of these specialty items. Dished heads, depending on diameter and thickness, are formed from single circular flat plate material by spinning or by "drawing" with dies in a press. Large-diameter heads up to 20 feet have been fabricated by the spinning process. Also, head fabrication is done by weldments of sectional head elements. The principal advantage of dished heads over flat plate blind-flange closures is the large reduction of discontinuity between the shape of the head and the penstock cylinder. This results in a substantial reduction of discontinuity stresses at or near the junction of the two members.

Although the hemispherical type of dished head is stronger than either the elliptical or torispherical type, excessive forming required for its fabrication limits its use. Further improvements in the analysis and reduction of stresses can be attained by use of the elliptical

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dished head. Physical tests and measurements of elliptical and torispherical heads show that torispherical heads subject to internal pressure approach an elliptical shape. Shape and stress-strain measurements indicate that the best-suited shape for a dished head on a cylindrical penstock under internal pressure is an ellipse having a 2:1 major-to-minor axis ratio. Strain measurements on such thin-walled elliptical heads show that radial stresses are of minor importance. The principal stresses in the head act at right angles to each other and are identified as the hoop stress and the meridional stress. At the junction of the dished head with the cylinder, these stresses become the hoop (circumferential) stress of the shell and the longitudinal stress of the shell, respectively.

Elliptical heads having a 2:1 major-to-minor axis ratio are recommended for use in penstock design. For these types of heads, the required thickness (t) of the dished head at its thinnest point is given by the equation:

$$t = \frac{PD}{2SE} \quad (\text{Equation 9-18})$$

Where:

- t = minimum required head thickness after fabrication and exclusive of any corrosion allowance, in.
- P = design pressure, psi
- D = inside diameter of the head at the skirt; or inside length of major axis, measured perpendicular to the longitudinal axis, in.
- S = maximum allowable stress intensity value compatible with the design pressure, psi
- E = lowest weld joint reduction factor of any joint in the dished head or at the head-to-shell joint, percentage in decimal form

Tests and stress measurements show that a 2:1 ratio elliptical head made from the same material as the cylindrical shell and of the same thickness as the shell at the point of juncture will be as strong if not stronger than the shell.

Ellipsoidal formed heads, subject to internal pressure, must meet the requirements of Figure UW-13.1 of the ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, as to fabrication and skirt length.

Typical hemispherical and ellipsoidal head installations are shown in Figures 9-25 and 9-26.

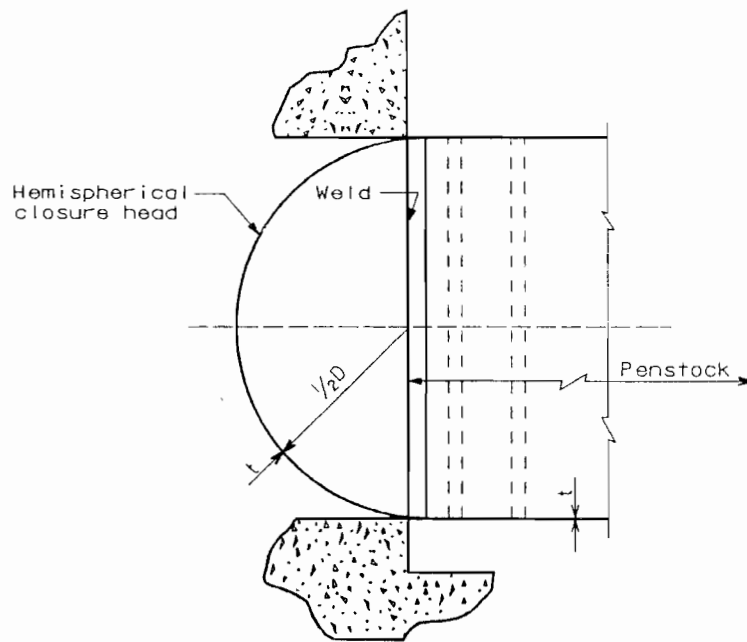


Figure 9-25 Hemispherical Closure Head

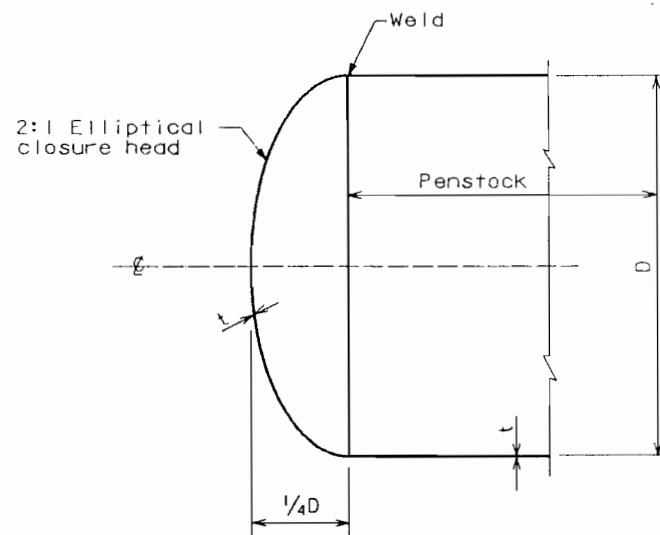


Figure 9-26 Elliptical Closure Head

References*

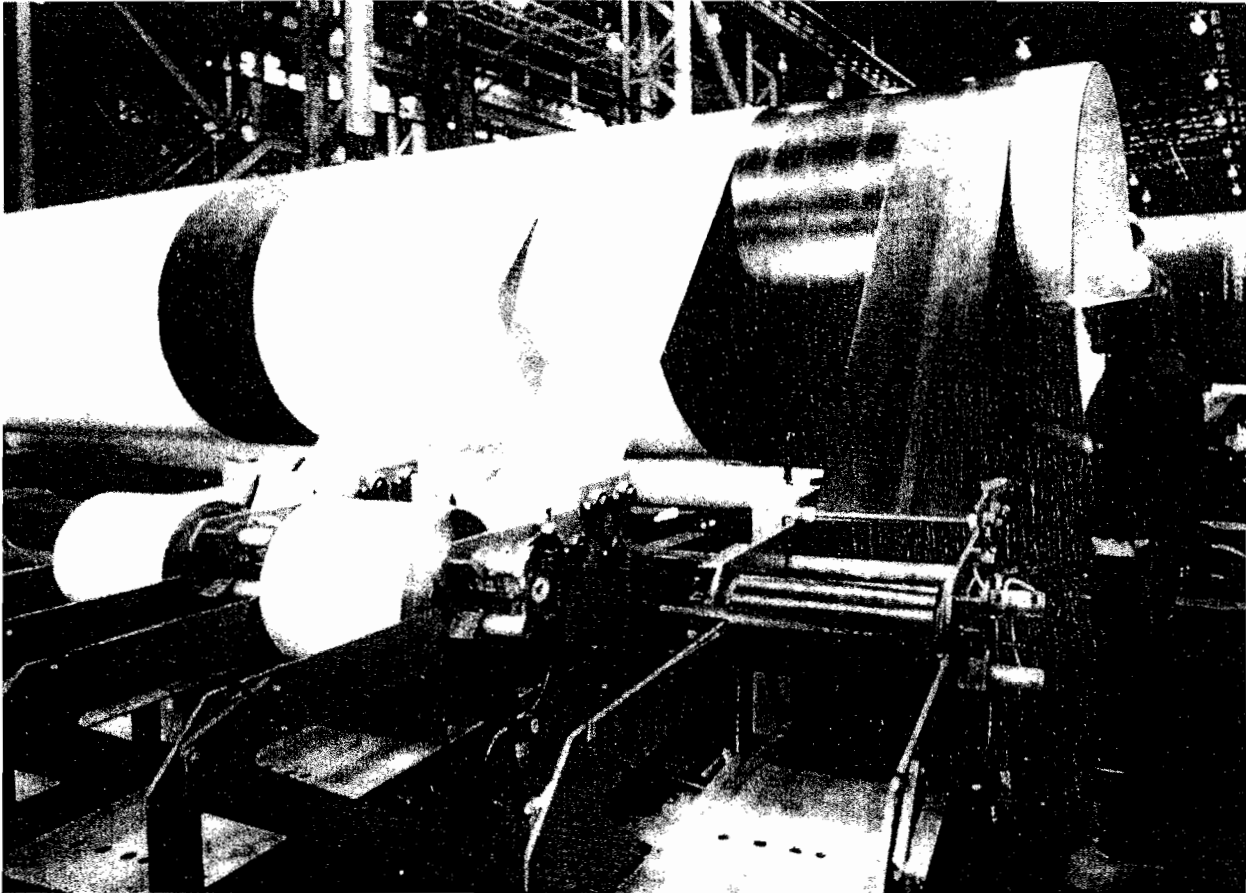
1. Bolted Sleeve-Type Couplings for Plain-End Pipe. ANSI/AWWA Standard C-219. American Water Works Association, Denver, CO.
2. Parmakian, J. "Air-Inlet Valves for Steel Pipe Lines." *Transactions*. ASCE, 115:438 (1950).
3. Lescovich, J.E. "Locating and Sizing Air-Release Valves." *Jour. AWWA*. AWWA, Denver, CO (July 1972).
4. Specifications for the Design, Fabrication and Erection of Structural Steel for Buildings. American Institute of Steel Construction, Inc. Chicago, IL.
5. *Steel Pipe—A Guide for Design and Installation*. AWWA Manual M11. AWWA, Denver, CO (1989).
6. Brockenbrough, R.L. & Assoc. *Steelpipe User's Manual*. Pittsburgh, PA (March 1, 1990).
7. Steel Pipe Flanges for Waterworks Service—Sizes 4 inches through 144 inches ANSI/AWWA C207. AWWA, Denver, CO.
8. Pipe Flanges and Flanged Fittings. ASME/ANSI B16.5. ASME, New York, NY.
9. Large-Diameter Carbon Steel Flanges. API Standard 605. American Petroleum Institute, Washington, D.C.
10. Steel Pipe Line Flanges. SP-44. Manufacturers Standardization Society of the Valve and Fittings Industry, Inc., Vienna, VA.
11. Rules for Construction of Pressure Vessels, ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. ASME, New York, NY.

The following references are not cited in the text.

Brownell, L.E. and Young, E.H. *Process Equipment Design*. John Wiley and Sons, Inc. (1959).

Chase, R. *Pressure Vessels, the ASME Code Simplified*. McGraw-Hill Publications, New York, NY (1977, 5th Ed.).

* The most current version of a standard, code, or specification should be used for reference.



The prevention and control of metallic penstock corrosion are of paramount importance in providing reliable service and long life. This section discusses atmospheric, aqueous, and underground corrosion mechanisms, tolerance for corrosion, applicable corrosion prevention and control measures, and penstock failure due to corrosion. Discussion is limited to metallic corrosion of steel penstocks.

10.1 Corrosive Environments

The corrosive environments to which a penstock is exposed include water, atmosphere, and soil.

10.1.1 Aqueous Corrosion

Metallic corrosion in aqueous solutions is a natural electrochemical process involving both the flow of electrons between metallic components as well as chemical reactions at the anodic (corroding) and cathodic (noncorroding) surfaces.

The driving voltage for corrosion current flow may be derived from galvanic (two metal) coupling, or can be caused by local differences on the surface of a single metal or alloy. These differences or concentration effects include both metallurgical as well as electrolytic (water) influences. Included among the metallurgical causes are differences in metallic composition, heat treatment (including welding), and mechanical stress. Variations in the electrolyte can result from oxygen concentration, pH differences, salt gradients, and temperature heterogeneities.

The formation of potential differences results in anodic and cathodic surfaces. Current flows within the formed electrochemical circuit, and metal is lost. The current flows electronically from the cathode (positively charged) to the anode (negatively charged) and ionically (electrolytically) from the anode surfaces to the cathode surfaces through the electrolyte (any substance that will convey ions).

At the anodic sites, metal is oxidized to a positively charged metallic ion (cation) that enters the solution (corrodes). The electrons liberated at the anode are conducted metallicity to the cathode surface. At the cathode, the electrons reduce oxygen to form hydroxyl ions and hydrogen ions to form hydrogen gas. The hydroxyl ions and hydrogen gas blanket the cathode surfaces and, in this manner, form an insulating film (cathodic polarization).

Relatively high concentrations of hydroxides at the cathode surfaces increase the local alkalinity so that conditions favorable for precipitation of calcium carbonate exist. The higher pH at the cathode is conducive to passivation of underlying surfaces, and corrosion products form a high resistance barrier at the anode surfaces. These reactions increase the circuit resistance and correspondingly decrease corrosion activity. Therefore, in static water, the original oxygen supply is consumed and corrosion rates decelerate. For this reason, metals and alloys exposed in deep water, even in normally highly aggressive seawater, corrode slowly. At the water surface and in the tidal and splash zones where oxygen is readily available, aggressive attack is experienced.

Under dynamic conditions, cathodic films may be swept away and oxygen replenished (depolarization), allowing the cathodic reaction and corrosion to continue. The dynamics of water motion and wetting and drying have similar depolarizing effects.

Thus, the activity at the cathode surfaces controls corrosion in neutral and near-neutral waters. Cathodic activity is limited by oxygen availability. Corrosion products on the anode surfaces increase the corrosion reaction by further restricting oxygen to underlying surfaces. Although oxygen is required for corrosion in neutral waters, contrary to common understanding, corrosion occurs where oxygen concentration is minimal.

A thin oxide layer can be formed on steel in powerful oxidizing environments such as concentrated nitric and sulfuric acids. A similar passive film forms on iron and steel concurrently exposed to oxygen in highly alkaline conditions. The properties of the resulting oxide film, rather than those of the parent iron or steel, govern corrosion behavior.

Any phenomenon that disrupts or removes the products of polarization or passivation increases corrosion activity. If the disruption occurs uniformly and totally on all surfaces, the metal would be uniformly lost (general corrosion) and, if the loss rate were known, sacrificial thickness could

be incorporated during design to provide the necessary service life. Unfortunately, in practice, the protective films are disrupted in localized areas only, resulting in pitting corrosion.

Flowing water can remove polarization products and replenish oxygen, and certain anions (such as chlorides and sulfides) can penetrate and disrupt passive films. Chlorides in water penetrate and disrupt passive films. In this manner, galvanic cells are established, with the intact passive film being the cathode and the parent material exposed at disruptions in the oxide layer serving as the anode. As corrosion progresses, the environment at the anodic sites becomes depleted of oxygen and "acidic," as both oxygen and hydroxyl ions are consumed by reaction with metallic ions to form metal oxide and hydroxide corrosion products. The ensuing oxygen and pH concentration cells further accelerate corrosion.

Other potential but less frequently encountered forms of aqueous attack are dealloying, microbiological corrosion, and stress corrosion. Dealloying is a phenomenon common to cast iron. One of the alloying elements (carbon) is more noble than the other alloying metal (iron). Thus, intergranular galvanic cells are established that corrode the more active metal, iron.

Microbiological corrosion is the deterioration of a metal or alloy as a result of the metabolic activity of micro-organisms. They accelerate corrosion by promoting concentration cells, creating corrosive conditions as a result of their life cycle or decomposition, and behaving as cathodic and anodic depolarizers. The two common corrosion-related forms are the iron and sulfate-reducing bacteria.

The iron bacterium is aerobic (requires oxygen for its metabolic processes). It assimilates ferrous iron in solution and converts it to ferric iron, which precipitates in voluminous deposits (tubercles) on any surface, metallic or nonmetallic. If the deposits are nonuniform over the surface, they promote oxygen concentration cells.

The sulfate-reducing bacterium is anaerobic (thrives in oxygen-deficient environments). It requires sulfates in solution which it reduces to sulfides using nascent hydrogen in the process. Because hydrogen is a cathodic polarizing agent, the sulfate-reducing bacterium, much like oxygen, is a cathodic depolarizer. Since hydrogen sulfide is generated, pH concentration cells also can result.

Stress corrosion cracking and hydrogen embrittlement are two forms of stress corrosion.

10.1.1.1 Stress Corrosion Cracking

Stress corrosion cracking is the result of the synergistic effects of applied or residual tensile stress and corrosion. Corrosion pitting creates a weakened cross section. Cracking initiates at the base of the pit where stresses are concentrated and continues to grow into the cross section. Crack growth is self-sustaining and self-accelerating since it results in progressively increasing stress concentrations. The cause of crack initiation and growth is not clearly understood. Stress concentrations at the roots of pits may result in local strains sufficient to disrupt any passive or protective films formed previously by corrosion. This establishes a new cell in which the pit surfaces behave cathodically to the freshly exposed metal anode. Repetition of this cycle results in progressively increasing stress until the material's strength is reached and

the material fails suddenly. The rapid strain rate occurring when the material strength is exceeded would provide a brittle fracture. In any case, stress corrosion cracking is an anodic phenomenon and cathodic currents are successfully used for its prevention and control.

10.1.1.2 Hydrogen Embrittlement

Hydrogen embrittlement is a phenomenon caused by diffusion of atomic hydrogen into the metal. The absorbed hydrogen atoms, upon entry into the metal, may combine into hydrogen gas at dislocations in the metal microstructure. This results in internally straining the metal and, if sufficient hydrogen is absorbed, brittle fracture of the internally stressed metal. Because hydrogen ion reduction is required, hydrogen embrittlement occurs at cathodes and is more likely in acidic electrolytes.

Both stress corrosion cracking and hydrogen embrittlement can result in failure of components at applied stress levels much lower than metal strength. Both phenomena result in sudden failure with no macroscopically visible warning signals. Susceptibility increases with increases in strength. Since both phenomena result in adding strains to materials that may already be highly strained, the high-strength and ultra-high-strength materials are primarily affected.

Paradoxically, although cathodic protection is applied to control stress corrosion cracking, it can generate hydrogen, thereby increasing the potential for hydrogen embrittlement. Usually this is not a problem when cathodic protection is applied in accordance with recommended practice.

10.1.2 Atmospheric Corrosion

The effect of corrosion on metal surfaces exposed to the atmosphere is the most visually evident form of corrosion. Atmospheric corrosion is a form of aqueous corrosion. Were it not for moisture, such as from precipitation and condensation, atmospheric corrosion would not occur. With atmospheric corrosion, two accelerators of aqueous corrosion are common: an abundance of oxygen, and cyclic wetting and drying activity.

Corrosivity of the atmosphere is controlled largely by the contaminants within the air. Industrial and marine atmospheres are among the most corrosive. Industrial atmospheres contain such contaminants as sulfur oxides and relatively large concentrations of carbon dioxide. Both result in acidic films on metals. Sea air contains chlorides and other salts, which are powerful corrosion accelerators. Acidic and salt films can concentrate from cyclic wetting and drying activity. Rural and inland industrial uses usually are relatively contaminant free and are not generally considered unusually corrosive.

10.1.3 Corrosion by Soils

Soil is an essentially neutral, aqueous electrolyte; thus, corrosion of ferrous alloys in soil is a special case of aqueous corrosion. The general cause of corrosion in neutral soil is attributable to cathodic depolarization or depassivation by the activity of oxygen. In oxygen concentration-promoted corrosion, the combined effect of oxygen and moisture causes corrosion. The driving voltage for the corrosion cell is caused by differences in oxygen available

to all surfaces. The conductivity of the soil controls both the intensity and extent of attack. Oxygen concentrations in soil are attributable to differences in aeration, salt concentrations and their effect on oxygen solubility, soil permeability, and ground-water flow.

Stray currents are the second most prevalent cause of underground pipeline corrosion. In stray current corrosion, current from a foreign source such as a cathodic protection system, electrified railway, or improperly grounded welder is distributed in the earth. Current is collected at some surfaces (cathodes) of the pipeline and discharged from other surfaces (anodes) as it returns to the originating source. Again, as with galvanic corrosion, metal corrodes at the anodic sites. The extent and intensity of stray current corrosion are related to the driving voltage of the foreign power supply, the resistance of its circuit, the geometrical relationship between the source of currents and the penstock, the axial resistance of the penstock, the dielectric properties and continuity of the penstock, and the soil conductivity.

In any corrosion reaction, the magnitude of the current controls the rate of metal loss. The magnitude of current is related to the driving potential and the circuit resistance in accordance with Ohm's law. Although the voltage difference between anode and cathode is normally small (less than 0.5 volt), the circuit resistance also may be low depending on the anodic and cathodic surface areas, their proximity, and the electrolyte conductivity. Larger potential differences, larger anode and cathode surface areas, and higher conductivities lead to *macrocell* corrosion (macro distances between cathodic and anodic surfaces) and localized corrosion. Small potential differences, small anodic and cathodic sites, and low conductivities result in *microcell* corrosion (microscopic distances between anodes and cathodes) and more uniform corrosion. Oxygen concentration and stray currents can produce macrocell corrosion.

The rate of anodic pitting in macrocell corrosion is related directly to the anodic current density, i.e., current discharge per anodic surface area. Large cathodes and small anodes result in high anodic current density and rapid pitting. Small cathodes and large anodes result in low anodic current density and slow, more general attack.

10.2 Vulnerability of Penstock Steel to Corrosion

Exposed surfaces of steel otherwise coated with a bonded dielectric coating are vulnerable to corrosion. The activity is limited to exposed surfaces only. Since coating defects (such as pinholes and holidays) are small, microcell corrosion occurs. Also, the area available to behave cathodically is minimized. This provides a more favorable cathode-to-anode area relationship, low anodic current densities, and reduced corrosion intensity. The magnitude of stray current collected and subsequently discharged is also reduced depending on coating efficiency relative to bare penstock or to penstock coated with conductive coatings such as concrete and mortar.

10.3 Corrosion Prevention and Control Measures

Corrosion activity can be prevented through design and by protective linings and coatings to minimize corrosion, and controlled with cathodic protection.

10.3.1 Design for Corrosion Prevention

Penstocks are very similar to pipelines with three notable exceptions. They may be on steep profiles, convey water at high velocities, and unless proper preventative measures are taken, they may be grounded through the plant grounding system. The penstock and its appurtenances should be designed to minimize corrosion.

10.3.1.1 High Water Velocities

Because of the high water velocities encountered in penstocks, internal surfaces must be as smooth as practicable. Accomplishing smoothness may require the elimination of high weld beads and other protrusions that could potentially upset the flow. Aside from head-loss considerations, elimination of protrusions minimizes potential damage from abrasion due to suspended sediment and cavitation erosion. Because of this consideration, the protective lining surfaces also must be smooth. Spray application results in smoother surfaces than brush or roller application. Because of the potential for turbine runner damage from loose fragments, penstocks normally are not lined with cement mortar. However, on flatter profiles, cement mortar lining may be applicable provided that the velocity does not exceed 20 feet per second and there are not large volumes of transported sands and gravels.

10.3.1.2 Electrical Grounding

Electrical ground mats for hydroelectric power plants are commonly constructed either of copper or copper-clad steel. Coupling the steel penstock with copper or copper-clad steel results in an adverse galvanic corrosion cell for the steel, wherein the steel corrodes and the copper is protected. Therefore, wherever practicable, the penstock should be electrically isolated from the intake gate or valve and the valve at its downstream terminus. A common source of electrical shorts to the plant grounding system is at a penstock's wall penetration. Typically, to facilitate construction, the penstock pipe is welded to the steel reinforcement prior to placement of the concrete. To eliminate this potential, the penetration can be cased, the penstock pushed through the casing, and the annulus sealed using casing insulating seals.

10.3.1.3 Design to Minimize Corrosion

As discussed previously, atmospheric corrosion usually is the result of wetting and drying. Corrosion activity occurs only during the wetted periods. Thus, designs must include provisions to reduce the duration of the wet periods. Such provisions include designs that promote free drainage by elimination of moisture and water traps, and ensure that the bottom of an atmospherically exposed penstock does not contact potentially moist soil. Whether the penstock is exposed to the atmosphere or to soil, any dielectric coating on the pipe should be carried through partial concrete encasements such as supports, anchors, and thrust blocks. Failure to coat interfacial surfaces between the steel and concrete results in accelerated corrosion at the edge of the concrete, attributable to a pH concentration cell. Because corrosion activity tends to be highly localized, i.e., resulting in pitting, the use of a corrosion allowance in the form of sacrificial thickness additions is not applicable.

10.3.2 Protective Linings and Coatings

Corrosive environments to which penstocks are exposed dictate the use of high-performance, industrial coatings and linings in conjunction with premium surface preparation, application, and cure.

10.3.2.1 General

No protective coating provides perfectly continuous coverage of the substrate. Bonded dielectric coatings will be perforated at damaged areas and pinholes. Mortar and concrete coatings will be cracked and spalled. Thus, some ferrous surface is exposed regardless of coating type.

Bonded dielectric coatings protect the substrate from corrosion by forming an electrically insulating barrier between the substrate and the electrolyte. The effectiveness of bonded dielectric coatings is a function of the bond developed between the coating and the substrate, the impermeability of the coating, and its continuity and durability. Although the coating may be perfectly continuous at the application plant, defects will occur during handling, shipping, and installation, and in service after installation. The defects usually are in the form of pinholes and damaged areas.

Surfaces beneath bonded dielectric coatings cannot serve as anodic or cathodic sites. Depending on the degree of continuity (efficiency) of the coating, the reactive surface area can be greatly reduced. The reduction of available surface to serve as cathodic sites greatly reduces the cathode-to-anode surface area ratio, resulting in a concomitant reduction in corrosion activity.

The bonded dielectric coatings specified for steel penstock typically exhibit efficiencies of 99% and greater. That is, only 1% or less of the surface area is exposed to behave anodically or cathodically. This in turn reduces the corrosion activity as discussed previously.

Portland cement concrete and mortar protect steel by a totally different mechanism than the bonded dielectric coatings. Concrete embedment and mortar coatings change the environment to which the steel is exposed. The highly alkaline (12.5 pH) environment provided by the hydrated cementitious materials, in the presence of oxygen, oxidizes (passivates) the steel surface to form a relatively stable hydrous ferric oxide. The passive film ennobles the surface, resulting in potentials very similar to materials such as copper and stainless steel or about 0.5 volt or more positive (cathodic) than steel exposed to neutral pH solutions. Thus, under sound mortar or concrete, the steel normally is protected from galvanic corrosion.

However, mortar and concrete are permeable to water and, being hygroscopic, attract moisture. Permeability of the coating varies, depending on its porosity from location to location. Wetting and drying, due to rising and falling ground water within the pipe trench, result in the accumulation of a greater concentration of salts in the coating than the water from which they were derived.

Similar to dielectric coatings, mortar and concrete coatings of perfect integrity are not achievable in practice. They are brittle (strain-intolerant) materials and develop only minimal bonding to steel. These characteristics lead to cracking and spalling so that all surfaces of the steel are not

exposed to the passivating, high-pH environment. Cracks and spalls allow unobstructed ingress of oxygen and depassivating ions such as chlorides. Complete consolidation of the coating so that all steel surfaces contact the passivating environment is not likely.

A common, unavoidable pH concentration cell occurs at mortared, rubber-gasketed penstock joints. A high water/cement ratio grout is required to fill a diaper form around the joint. Excessive drying shrinkage is experienced, and the mortar cracks.

Cementitious materials also are subject to loss of alkalinity by carbonation. Being ionically conductive, the underlying steel is vulnerable to stray current corrosion; therefore, the common causes of corrosion of steel embedded in concrete or mortar are failure to achieve passivation of all surfaces initially, loss of passivity at localized areas, and stray current corrosion.

Because cementitious coatings are conductive, all underlying steel surfaces are available to behave either cathodically or anodically. Since loss of passivity occurs at localized areas only, the cathode-to-anode area relationship is large, and high anodic current densities and intense corrosion activity result. The corrosion products of steel are more voluminous than the alloy. This phenomenon causes cracking and spalling of the mortar and concrete quite similar to cyclic freezing and thawing. The new exposed surfaces increase the extensity of corrosion.

10.3.2.2 Protective Linings

Protective linings are materials used in the interior of penstocks to protect the metal from corrosion and wear.

Linings may be applied in the field at the time of installation or in the shop after fabrication. It is always necessary to field repair the coating after installation.

Some materials are temperature dependent; coal tar enamel has withstood 80 feet per second with water temperatures below 55°F. Coal tar epoxy may withstand 40 feet per second at temperatures below 50°F. Some lining materials may cause cavitation problems if they have a high degree of friction.

Coatings in contact with potable water also must meet EPA health requirements. Organizations such as the National Sanitation Foundation and Underwriters Laboratories are authorized by the EPA to run specific health effect tests on paints and coatings. See ANSI/AWWA D102-78¹ (AWWA standard for painting steel water storage tanks).

The interior of penstocks can be lined with one of the materials listed in Table 10-1, provided the material is manufactured and applied in accordance with the applicable standard and that water velocities do not exceed 20 feet per second.

Table 10-1 Penstock Lining Materials

MATERIAL	LISTED STANDARD
COAL TAR ENAMEL (USUALLY SHOP APPLIED)	AWWA C203
LIQUID EPOXY COATING SYSTEMS (INCLUDES COAL TAR EPOXY)	AWWA C210
CEMENT MORTAR (SHOP APPLIED)	AWWA C205
CEMENT MORTAR (FIELD APPLIED)	AWWA C602
VINYL RESIN	SSPC 4.04
FIBERGLASS	NO STANDARD

10.3.2.3 Protective Coatings

Coatings for the exterior surfaces of penstocks protect against corrosion and provide a pleasing appearance. The coatings may be applied completely in the shop at the time of fabrication or in the field at the time of installation. Certain coatings should be applied in the shop where there are proper facilities.

Commercial coating materials and manufacturers of specific coating materials should be researched rather than relying solely on one set of standards such as the AWWA standards. Other organizations that provide specifications for coatings include the Steel Structures Painting Council (SSPC), the American Society for Testing Materials (ASTM), the Bureau of Reclamation, the Army Corps of Engineers, the Department of Defense, and the National Association of Corrosion Engineers.

Following are recommended coating materials classified according to their use as (1) atmospheric coatings, (2) coatings for soil burial, and (3) coatings for encasement in concrete.

(1) Atmospheric coatings

The exterior of penstocks exposed to weather can be coated with one of the materials listed in Table 10-2, manufactured and applied in accordance with the listed standard. All paint material, including thinner, for coating must be made by the same manufacturer.

Table 10-2 Atmospheric Coatings

MATERIAL	LISTED STANDARD
THREE-COAT ALKYD	SYSTEM 1-91 AWWA
FOUR-COAT ALKYD	SYSTEM 2-91 AWWA
THREE-COAT ALKYD/SILICONE-ALKYD	SYSTEM 3-91 AWWA
THREE-COAT EPOXY/EPOXY POLYURETHANE	SYSTEM 4-91 AWWA
THREE-COAT INORGANIC ZINC/EPOXY/URETHANE	SYSTEM 5-91 AWWA
TWO- OR THREE-COAT COAL TAR EPOXY	SYSTEM 6-91 AWWA
TWO- OR THREE-COAT WATER REDUCIBLE EPOXY/EPOXY-POLYAMIDE	SYSTEM 7-91 AWWA
THREE-COAT WATER REDUCIBLE ACRYLIC	SYSTEM 8-91 AWWA
EPOXY COATINGS (WILL CHALK WHEN EXPOSED TO SUNLIGHT UNLESS TOP COATED WITH AN ULTRAVIOLET-TOLERANT COATING, SUCH AS AN ALIPHATIC POLYURETHANE)	AWWA C218
VINYL, VR-3 AND VR-6 COATING SYSTEM	SSPC 4.04 BUR M-54 AND M-55

Precautionary Notes:

- (a) For coal tar epoxy, the time limit between coats cannot be exceeded. When exceeded, the coal tar must be preconditioned with a brush-off blast prior to applying additional coats.
- (b) Consideration must be given to the effects of light-colored penstock coatings on ice buildup in cold climates and of dark-colored coatings on penstock expansion and contraction in hot climates.

(2) Coatings for soil burial

The exterior of buried penstocks can be coated with one of the materials listed in Table 10-3, manufactured and applied in accordance with the listed standard.

Table 10-3 Buried Penstock Coatings

MATERIAL	LISTED STANDARD
COAL TAR ENAMEL	AWWA C203
CEMENT-MORTAR	AWWA C205
COLD-APPLIED TAPE COATING (BODY OF PENSTOCK)	AWWA C214
COLD-APPLIED TAPE COATING (SPECIAL SECTIONS ONLY)	AWWA C209
LIQUID EPOXY COATING SYSTEMS (INCLUDES COAL TAR EPOXY)	AWWA C210
FUSION-BONDED EPOXY COATING SYSTEM (CANNOT BE APPLIED IN THE FIELD)	AWWA C213
EXTRUDED POLYOLEFIN COATING (LIMITED TO 20-INCH-DIAMETER PENSTOCKS)	AWWA C215
POLYURETHANE COATINGS	NO STANDARD
ROCK SHIELD	NO STANDARD

(3) Encasement in concrete

The coating applied to the exposed portion of the penstock must extend 6 to 12 inches into the concrete embedment. The use of water stops should be considered.

10.3.2.4 Surface Preparation

The penstock surface must be prepared in accordance with the standard referred to in the specification for the material selected or the manufacturer's instructions if no recognized standard is specified. In addition, the procedures given below must be followed.

- (1) Consult NACE TM-01-70² or TM-01-75³ for surface preparation of bolt coatings encased in concrete.
- (2) Surfaces to be encased in concrete must be cleaned and coated before concreting.
- (3) During or after abrasive blasting, any accumulations of slag, dirt, blisters, weld spatter, metal laminations, or other irregularities must be removed by appropriate mechanical means. Rough edges and welds must be smoothed or rounded and then brush coated.

- (4) Bolts, nuts, and other appurtenances to remain uncoated must be cleaned and masked prior to penstock coating.
- (5) Piezometer and flow meter holes must be plugged prior to the coating application.
- (6) Metalwork to be welded must be primed with a weldable inorganic zinc prior to shipping.
- (7) Metalwork must be coated after welding.

10.3.2.5 Coating Application and Cure

Coating materials must be applied in accordance with the standard referred to in the specification for the material selected or the manufacturer's instructions if no recognized standard is specified. This includes, but is not limited to, material mixing, application, dry film thickness, handling, and potlife, if applicable. Limitations on ambient and substrate temperatures, relative humidity, and dew point must be observed as stated in the referenced standard or in the recommendations of the manufacturer of the coating material. The contractor must be experienced in the application of the type of coating specified. In addition, the following general procedures must be followed (some procedures do not apply to all coatings).

- (1) Until the coating is dry or cured the penstock must be free of airborne dust. Exterior coating must not be applied during winds in excess of 20 mph unless enclosures are provided. Surfaces that become contaminated between applications of coats must be recleaned prior to application of the next coat. All surfaces to be coated or lined must have temperatures 5°F above the dew point.
- (2) A coating system agreed upon by the owner and the contractor must be used on bolts, nuts, and other appurtenances.
- (3) The curing time between coats must be rigidly followed.
- (4) Items that are normally factory coated may be furnished primed or primed and top coated provided such coatings are compatible with subsequent coatings to be applied in the field.
- (5) A one-quart sample representing the coating material must be furnished at the time of shipment for each type, batch, lot, and color of liquid and mastic used when quantities exceed 20 gallons. The constituents of multiple component coatings must be furnished separately.
- (6) When multiple coats are applied, successive coats must be shaded sufficiently to denote color differences between coats. The final coat color must meet the design specifications.
- (7) Penstock sections must be identified to facilitate proper erection without marking the exterior of any exposed part of penstock coatings.
- (8) Samples of coal tar enamel must consist of 25 pounds of the enamel and 12 linear feet of fibrous glass.

10.3.2.6 Inspection

The owner's representative must inspect all phases of the coating and lining work to ensure that standards and specifications are met. Coating and lining material, test reports, surface preparation checks, and inspection procedures must conform with the specified standards. Shop-applied coatings must be inspected prior to shipment. Coating and surface preparation must be documented by inspection reports. Reports must list work done, materials used, batch numbers, date, weather conditions, and any deviations from project specifications or standards.

Adequate illumination intensity, usually at least 100 footcandles, must be provided on all surfaces to be inspected. Reports of tests and inspections must be made available to the owner's representative.

10.3.2.7 Storing Coating and Lining Material

Materials must be stored in compliance with local regulations and manufacturer's recommendations, and must remain in the original sealed containers until used. Materials must be used within the time limit recommended by the manufacturer. The container must indicate the manufacturer's name, stock number, hazardous substances, type of coating, specification number, color, date of manufacture, batch number, and instructions for reduction where applicable. Such information must be plainly legible at the time the material is used in the work. Coatings and primers must be delivered in container sizes ordered by the contractor. Coatings must be thinned with the coating manufacturer's thinner when thinning is permitted.

10.3.2.8 Regulations

Many federal, state, and local jurisdictions have established abrasive, paint, and coating regulations. Coatings and linings and their application procedures must comply with the regulations in effect at the sites where the materials are applied or used.

For all locations, the manufacturer of the material must furnish the applicator and owner a Manufacturer's Material Safety Data Sheet (MSDS) describing any components that may be hazardous. The manufacturer must certify that the materials comply with all pollution, safety, safety equipment, and lighting regulations in effect at the point of application. Abrasives and coatings must be disposed of in accordance with regulations.

10.3.2.9 Manufacturer's Information Submittals

Copies of the manufacturer's recommendations for mixing and applying coating materials must be furnished to the penstock owner's representative prior to application. The instructions must be followed unless otherwise specified in writing. The manufacturer's certificate of compliance (indicating both compliance with the standard to which the material was made and that the sample is from the actual batch or lot to be furnished) must be furnished for each material prior to its use in the work. The certificate must include material identification, quantity, batch number, date of manufacture, certified test results, and specifications (if available) for the material being used.

10.3.2.10 Handling

Coated penstock sections must be protected against damage during transporting and handling in accordance with AWWA C200,⁴ or for shop-coated sections in accordance with the appropriate AWWA coating standard.

10.3.3 Cathodic Protection

Cathodic protection is an electrochemical corrosion prevention and control method. It is the only method with the potential to eliminate totally the initiation of corrosion activity or halt ongoing corrosion. To achieve cathodic protection, a favorable electrochemical circuit is established by installing an external electrode in the electrolyte and passing direct current (conventional) from that electrode through the electrolyte to the structure to be protected. This electrical current polarizes the potential of the cathodic surfaces (relatively positive) on the structure to that of the anodic (more negative) surfaces. When this is accomplished, there is no current flow between the formerly anodic and cathodic surfaces, and corrosion is prevented or eliminated. This represents a balanced or equilibrium condition. In normal practice, sufficient current is passed to the surfaces so that the formerly anodic surfaces receive current from the electrolyte and their potential shifts in the more negative direction. In any event, it is the cathode surfaces that require the majority of the current.

Achieving cathodic protection requires that all surfaces of the structure collect sufficient current to overcome the naturally occurring corrosion potentials. This requires that all surfaces of the structure be electrically continuous. Although welded pipe joints provide for electrical continuity, rubber-gasketed joints do not. Thus, the latter must be fitted with joint jumper bonds.

Cathodic protection anodes must be placed in the same electrolyte as the structure. Since anodes placed inside the penstock would be an obstruction to high-velocity flow, internal surfaces of penstocks usually are not cathodically protected.

Cathodic protection is categorized into two types depending on whether the source of the required current is galvanic or impressed current.

10.3.3.1 Galvanic Anodes

Galvanic anodes commonly used for the protection of steel are made of magnesium or zinc. When coupled to steel, a favorable corrosion cell is established in which the zinc or magnesium corrodes and the steel is protected. Corrosion is not eliminated but is transferred to the galvanic anode, which sacrifices itself to protect the structure and is consumed. Hence galvanic anodes are often referred to as sacrificial anodes. Galvanic-cell-generated voltage differences are relatively small. When coupled to steel, zinc and magnesium develop cell voltages of about 0.5 and 1.0 volts, respectively. Because of these relatively low driving voltages, the cell current is normally very small. Typical output per anode is on the order of 50 to 100 milliamps. The corrosion rates of magnesium and zinc are 17 and 26 pounds per ampere-year, respectively. As galvanic anodes are consumed, they must be replaced. For example, a 17-pound magnesium anode delivering 100 milliamps of current theoretically will last 10 years.

10.3.3.2 Impressed Current

In contrast to the sacrificial anode, the impressed current method requires an external source for DC power. Typically, power is provided by a rectifier, which converts AC power to a pulsating form of DC energy. However, with the impressed current method the anode is not required to provide a favorable voltage relationship with steel. Thus, more noble materials (materials that corrode or are consumed very slowly with the discharge of current) can be installed as anodes. Common impressed current anodes include graphite and high-silicon cast iron, which corrode at rates less than 2 pounds per ampere-year. More exotic materials such as the mixed-metal oxides are consumed at even lower rates. Since the current is provided by an external source, driving voltage and consequently current output are virtually unlimited.

10.3.3.3 Protection Criteria

A potential of at least -0.85 volts relative to a copper/copper sulfate reference electrode (CSE) is the commonly applied minimum criterion for protection. NACE RP0169-83⁵ provides additional minimum protection criteria. To prevent potential damage, such as cathodic disbondment and hydrogen embrittlement from hydrogen effects, the maximum (absolute value) potential is -1.10 volts.

10.3.3.4 System Design

Obtaining cathodic protection within the above potential tolerances not only requires sufficient current, but also proper distribution of the current. The external electrode (either the galvanic or impressed current anode) is referred to as a cathodic protection ground bed. The magnitude of the current, its distribution (number of ground beds required), and, consequently, the energy requirement are related to the coating and its specific leakage resistance. For example, cement-mortar coated steel requires significantly more current than steel with a bonded dielectric coating. This results in substantial differences in costs of the installed systems and their operating costs. For a specific application, one type of protection (galvanic or impressed current) may provide a distinct advantage over the other. Thus, selection of a compatible protective coating, type of system to be utilized (galvanic anode or impressed current), and design of the cathodic protection system, including provisions for electrical continuity and isolation, should be made by an experienced cathodic protection specialist. NACE certifies professionals capable of designing cathodic protection systems.

10.3.4 Corrosion Monitoring

Corrosion monitoring systems provide for electrical continuity and electronic access, thus facilitating the application and monitoring of cathodic protection. Penstock continuity is automatically provided by welded joints. Bonds are installed across rubber-gasketed (including coupled) joints. Electronic access is obtained by installation of surface test stations with leads originating from the pipe metallic components. Typically, test stations are installed on either side of insulating fittings at the upstream and downstream termini of the penstock, at buried line crossings, and at intermediate points along the penstock such that they are no more than 1000 feet apart. Corrosion monitoring systems are designed so that corrosion activity on the penstock can be evaluated through data generated by surface testing.

One commonly used test is the determination of pipe-to-soil potential. To conduct this test, the pipe lead is connected to the positive terminal of a high-impedance voltmeter. The negative terminal is connected to a standard reference electrode (normally CSE) contacting the soil surface directly over the penstock. The soil is saturated at the point of electrode contact to reduce contact resistance, thereby increasing the accuracy of the measurement. The premise for this test is that as the penstock corrodes, sampling over time will result in progressively higher negative potential values.

Complications arise because the reference electrode is remotely located from the penstock (typically a minimum of 5 feet). Use of the remote electrode results in determination of an average potential of underlying metal which may be comprised of both anodic and cathodic surfaces. If the anode surface is large relative to the cathode, the measured potential is dominated by the anode potential and produces the desired effect. However, if the cathode is large relative to the anode (for example, mortar coated or concrete encased), the measured potential is skewed toward that of the cathode.

Thus, potential measurements can be grossly misleading. That is, an increase in corrosion activity may be accompanied by little or no change in observed potential. Thus, monitoring is not as applicable to the mortar-coated and concrete-encased penstock as to dielectric coated steel. Because of the foregoing potential problems in testing and interpretation of test results, monitoring also should be conducted by an experienced corrosion engineer.

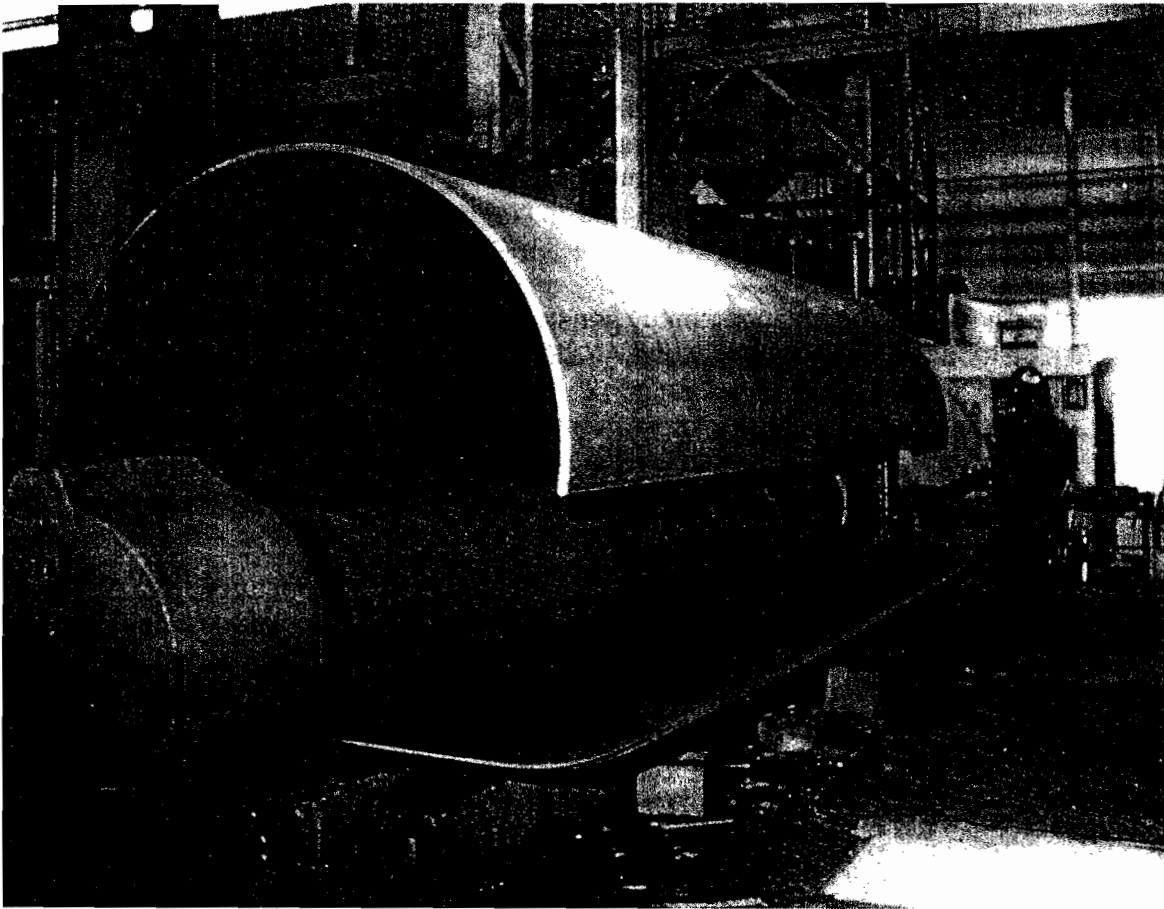
10.4 Failure Due to Corrosion

Typically, corrosion results in wall perforations and leakage. Repair requirements vary from weld repair of a perforation to total line refurbishment or replacement. Added to the cost of repair or replacement is the direct cost of any accompanying flooding. Failure of penstocks could result in serious disruption of electrical service. Penstock failures pose a serious threat to safety and can have serious economic consequences. Failures can be minimized through proper application of sound engineering principles to prevent and control corrosion.

References*

1. Painting Steel Water-Storage Tanks. AWWA Standard D102-78. AWWA, Denver, CO.
2. Visual Standard for Surfaces of New Steel Airblast Cleaned with Sand Abrasive. NACE TM-01-70. National Association of Corrosion Engineers, Houston, TX (1970).
3. Visual Standard for Surfaces of New Steel Centrifugally Blast Cleaned with Steel Grit and Shot. NACE TM-01-75. National Association of Corrosion Engineers, Houston, TX.
4. Steel Water Pipe 6 Inches and Larger. AWWA Standard C200. AWWA, Denver, CO.
5. Recommended Practice Control of External Corrosion on Underground or Submerged Metallic Piping Systems. NACE Standard RP0169-83, Item No. 53002. National Association of Corrosion Engineers, Houston, TX.

* The most current version of a standard, code, or specification should be used for reference.



11.1 General

The following manufacturing standards and rules apply specifically to penstocks and penstock parts that are fabricated by welding and constructed of carbon steel, low-alloy steel, high-alloy steel, or heat-treated steel.

11.1.1 Fabrication Standard

Fabrication of penstocks must conform to the provisions of Section VIII, Division 1 of the ASME Code¹ and also the following specific requirements, additional requirements, and the special provisions for spiral seam penstocks given in Section 11.4.

11.1.2 ANSI/AWWA C200 Standard

The ANSI/AWWA C200 standard² covers the manufacture of electrically butt-welded, straight-seam or spiral-seam pipe intended for water use. For quality assurance, this standard relies on production weld testing and shop hydrostatic testing of each section by stressing the pipe wall to 75% of the specified minimum yield of the steel used.

The ANSI/AWWA standard also covers pipe end preparation and tolerances for field joints. Pipe manufactured to this standard has been used primarily in buried lines with O-ring joints to working pressures exceeding 250 psi, and bell-and-spigot field-welded and butt-welded joints to working pressures up to 400 psi. The allowable design stress on these lines usually has been limited to 21,000 psi.

When carefully evaluated by the engineer, many penstock installations will fall within this service condition, and pipe fabrication conforming to the ANSI/AWWA C200 standard² is acceptable.

11.1.3 Manufacturing Methods

11.1.3.1 Straight Seam

Straight-seam penstocks may be shop or field fabricated. They are manufactured from plates that are edge-broken (crimped) and then rolled or formed, resulting in a continuous and uniform curvature using a plate bending roll or a U-ing and forming press. Several types of bending rolls are available, including: (1) pyramid bending rolls, (2) pinch bending rolls, and (3) initial pinch bending rolls (see Figure 11-1). For the pyramid and initial pinch bending rolls the plate edges must be crimped or edge-broken to the proper radius prior to rolling. Figure 11-2 shows U-ing and O-ing presses used for forming. All plates for shell sections must be formed to the required shape by a process that will not unduly impair the physical properties of the material.

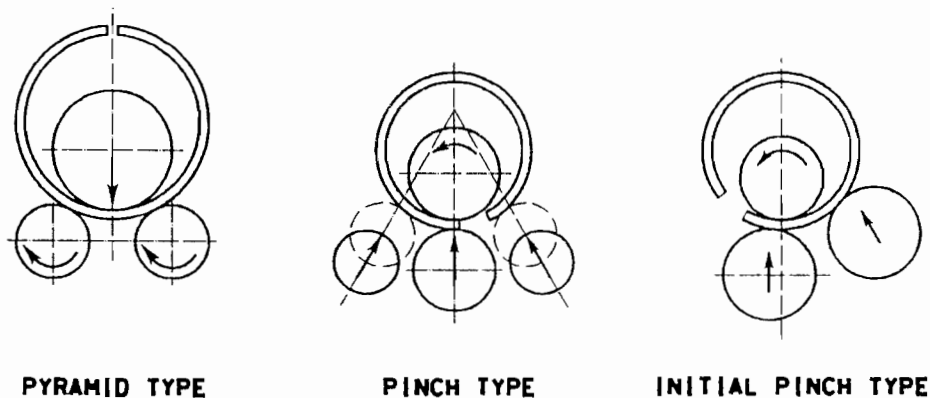


Figure 11-1 Bending Rolls

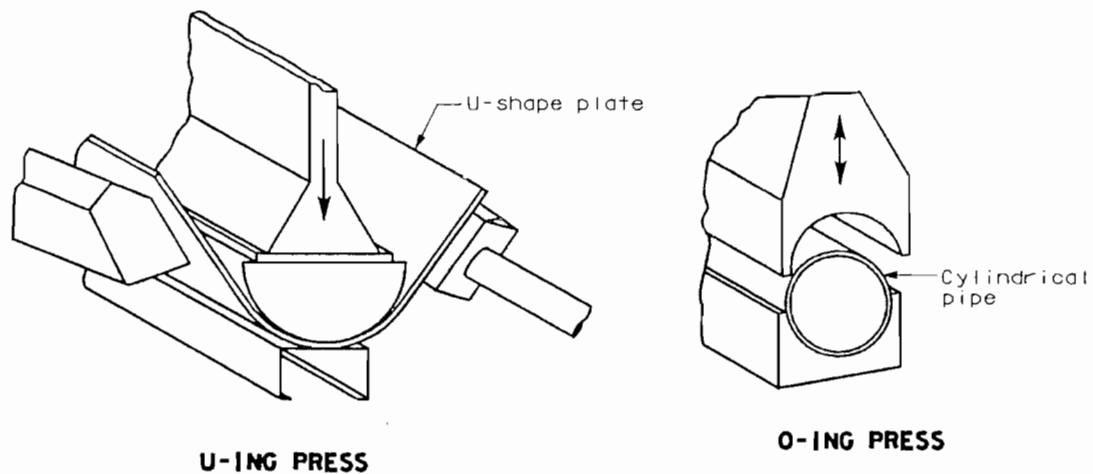


Figure 11-2 Forming Presses

Plates for penstock shells must be sized to minimize the number of longitudinal and circumferential seams in the completed section. Penstock courses must be assembled with the longitudinal seams staggered a minimum of 15 degrees.

11.1.3.2 Spiral Seam

Spiral-seam penstocks are manufactured in the shop using steel in coiled form. Currently, wall thicknesses through 3/4 inch are available. The diameter of the penstock is determined by the angle of the coil feed to the axis of the penstock in relation to the width of the coil (see Figure 11-3). In this process, the coil is unrolled, the coil edges are prepared as required, and the coil is helically formed to a true circular shape and welded in a continuous operation. The penstock is cut to length as required by an automatic cut-off device.

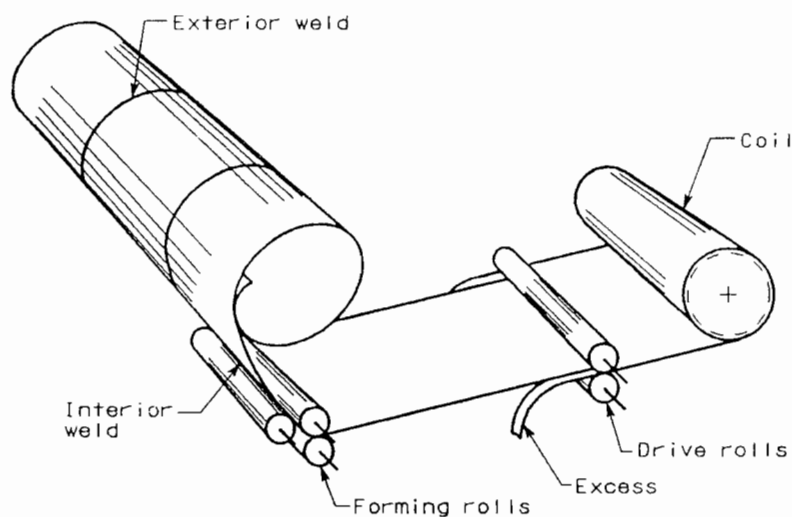


Figure 11-3 Process for Making Spiral-Seam Pipe

Pipe ends for the required field joints may need expansion or sizing to meet the tolerances specified in Section 11.3.3.1.

11.2 Qualification of Manufacturer

The manufacturer must demonstrate that it has the shop facilities and also has had fabrication experience with specialty work of similar size and type within the past 5 years, as well as an internal quality control system similar to the SPFA Plant Quality Certification Program.³

11.3 Manufacturing Specifications

Fabrication of the penstock and penstock parts must be performed in accordance with the provisions of the ASME Code,¹ Section VIII, Division 1. However, hydrostatic testing is not mandatory. The fabricator is not required to stamp any part of the penstock with the official ASME Code symbol applicable to pressure vessels. Fabrication in accordance with the ASME Code may not be required for low-head or nontraditional penstocks.

11.3.1 Material

Materials used for steel penstocks and penstock parts must be furnished in strict accordance with the proper ASTM standard and specifically in accordance with the requirements of ASTM designations A6⁴ and A20⁵ as appropriate.

11.3.1.1 Material for Pressure Parts

Material subject to stress due to pressure must conform to one of the specifications given in Section 2.4 or 2.5.

11.3.1.2 Material for Nonpressure Parts

Material for nonpressure parts, such as supports, thrust rings, lugs, and clips, need not conform to the specifications for the material to which they are attached or to the material specifications described in Sections 2.4 and 2.5. However, if nonpressure parts are attached to the penstock by welding, they must be of weldable quality.

11.3.2 Welding

Welding of shells and assemblies classified as pressure parts must be in accordance with the ASME Code, Section VIII, Division 1, Part UW. The welding processes used in the construction of penstocks are restricted to those outlined in Section 11.3.2.1. No production welding can be undertaken until the welding procedures have been qualified. Only welders and welding operators who have been qualified in accordance with Section IX of the ASME Code⁶ can be used in production.

11.3.2.1 Welding Process

Arc-welding and gas-welding processes are restricted to shielded metal arc (SMAW), flux cored arc (FCAW), submerged arc (SAW), gas metal arc (GMAW), and gas tungsten arc (GTAW).

11.3.2.2 Welding Materials

Welding materials used for production must comply with the requirements of the ASME Code, Section VIII, Division 1, and Section IX, and the applicable welding procedure specification.

11.3.2.3 Edge Preparation

Shell plate edges must be prepared for welding as required by the applicable welding procedure specification. Edges of plates prepared for welding must be examined visually for signs of lamination, shearing cracks, and other imperfections. Defects must be removed by mechanical means or by thermal gouging processes.

11.3.2.4 Preheating

The minimum preheating temperature for all welded joints is 50°F (mandatory). A metallurgical or welding engineer should be consulted about mandatory or recommended preheat requirements. The nonmandatory guidelines of the ASME Code, Appendix R of Section VIII, also should be reviewed. The procedure specification for the material being welded specifies the minimum preheating temperature under the weld procedure qualification requirements of Section IX of the ASME Code.

11.3.2.5 Postweld Heat Treatment

Weld procedures must conform to Section IX of the ASME Code,⁶ including conditions for postweld heat treatment. All penstock sections and parts must be given a postweld heat treatment at temperatures not less than those specified in the ASME Code,¹ Section VIII, Division 1 (Tables UCS-56, UCS-56.1, UHA-32, and UHT-56) when the nominal thickness of any welded joint in the penstock section or part exceeds the limits in those tables.

11.3.3 Field Joint Ends

Ends of supplied penstock sections must be of the type specified by the purchaser. ANSI/AWWA C200² describes the following types of ends:

- (1) Plain-end pipe
- (2) Beveled ends for field butt welding
- (3) Ends fitted with flanges
- (4) Ends fitted with butt straps for field welding

- (5) Ends for mechanical couplings
- (6) Lap-joint pipe ends for field welding
- (7) Bell-and-spigot ends with rubber gaskets.

11.3.3.1 Manufacturing Tolerances at Ends

- (1) Out-of-roundness

The difference between the major and minor outside penstock shell diameters must not exceed 1%. In addition, the penstock shell must meet the requirements of UG-80(b) of the ASME Code¹ Section VIII, Division 1, with regard to plus-or-minus deviation from true circular form.

- (2) Outside diameter

The outside diameter tolerance, based on circumferential measurement (πD), is $\pm 1/4$ inch for the following:

- (a) Plain-end pipe
- (b) Beveled ends for field butt welding
- (c) Ends fitted with flanges
- (d) Ends fitted with butt straps for field welding
- (e) Ends for mechanical couplings (must have tolerances within the limits required by the coupling manufacturer)
- (f) Lap joints for field welding (circumferential difference between inside of bell and outside of spigot must not exceed 2/5 inch).

- (3) Squareness of ends for butt welding

Ends of penstock sections must not vary by more than 1/16 inch at any point from a plane normal to the longitudinal axis and passing through the center of the section at the end.

11.3.4 Tolerances

11.3.4.1 Out-of-Roundness

The difference between the major and minor outside diameters must not exceed 1%. In addition, the penstock shell must meet the requirements of UG-80(b) of the ASME Code,¹ Section VIII, Division 1 with regard to plus-or-minus deviation from true circular form.

11.3.4.2 Outside Diameter

The outside diameter tolerance, based on circumferential measurement (πD), is $\pm 1/4$ inch.

11.3.4.3 Alignment

Using a 10-foot straight edge placed so that the ends are in contact with the pipe, alignment cannot exceed $1/8$ inch.

11.3.4.4 Length

Lengths must be within $\pm 1/2$ inch of the specified length unless otherwise agreed upon by the manufacturer and purchaser.

11.3.4.5 Minimum Wall Thickness

The wall thickness (in inches) at any point in the penstock must not be less than the specified nominal thickness minus $1/100$ inch.

11.4 Supplemental Requirements for Spiral-Seam Penstocks

Spiral seam penstocks generally are manufactured in accordance with ANSI/AWWA C200² and ASTM A139⁷ standards. Spiral-seam pipe for traditional penstocks must be manufactured in accordance with the requirements of Section 11.3, with the exception of the following special provisions.

11.4.1 Steel Coil

Steel coils must be con-cast killed steel, with fine austenitic grain size, and must conform to one of the specifications given in Sections 2.4 and 2.5, including physical tests as supplemented by Section 2.2.2.

11.4.2 Forming

The spiral-seam manufacturing process must be set up to the correct feed angle for the coil width used and the penstock diameter required. The forming process must be continuous. The initial start-up section up to the point at which diameter control has been obtained must be discarded.

11.4.3 Tolerances

The mandatory pipe barrel circumference tolerance is $\pm 0.35\%$, but cannot exceed $\pm 3/4$ inch from the nominal outside circumference based on the diameter specified. Pipe ends must be sized, if necessary, to meet field joint tolerances required by Section 11.3.3.1. All other tolerances must conform to those specified in Section 11.3.4.

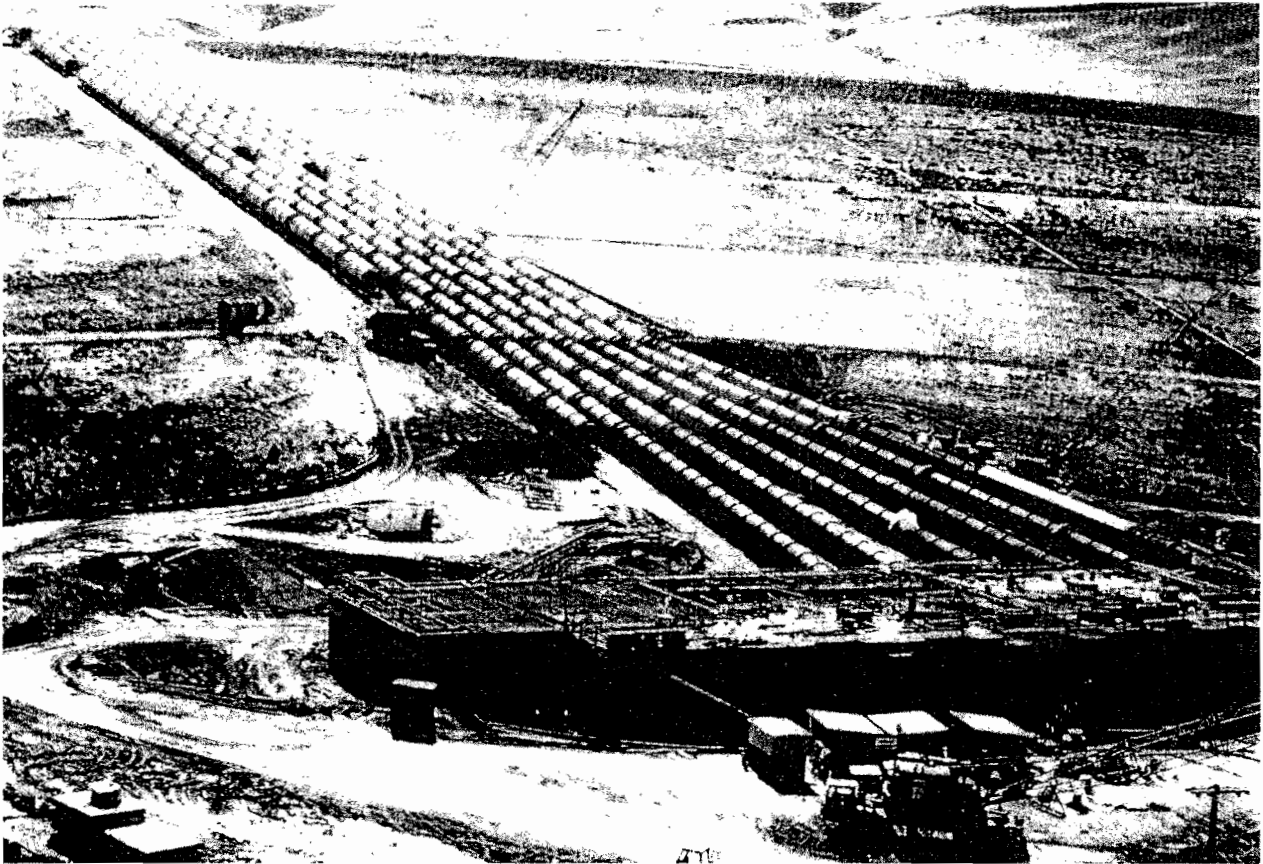
11.4.4 Weld Inspection

When design stresses exceed 21,000 psi, consideration must be given to weld inspection beyond the requirement of the manufacturing standards. Some manufacturers offer 100% continuous and uninterrupted ultrasonic inspection of helical weld seam and skelp-end weld seam in accordance with methods and acceptance criteria outlined in API Specification 5L,⁸ coupled with shop hydrostatic testing of all penstock sections at 75% of the material yield strength.

References*

1. Rules for Construction of Pressure Vessels, ASME Boiler and Pressure Vessel Code, Section VIII, Division 1. ASME, New York, NY.
2. Steel Water Pipe 6 Inches and Larger, ANSI/AWWA Standard C200. AWWA, Denver, CO.
3. SPFA Plant Quality Certification Program, Steel Plate Fabricators Association, Inc., Westchester, IL .
4. Standard Specification for General Requirements for Rolled Steel Plates, Shapes, Sheet Piling, and Bars for Structural Use. ASTM Designation A6. ASTM, Philadelphia, PA .
5. Standard Specification for General Requirements for Steel Plates for Pressure Vessels. ASTM Designation A20. ASTM, Philadelphia, PA .
6. Qualification Standard for Welding and Brazing Procedures, Welders, Brazers, and Welding And Brazing Operators. ASME Boiler and Pressure Vessel Code, Section IX. ASME, New York, NY.
7. Standard Specification for Electric-Fusion (Arc) Welded Steel Pipe (NPS 4 and Over). ASTM Designation A139. ASTM, Philadelphia, PA.
8. Specification for Line Pipe. API Specification 5L. American Petroleum Institute, Washington, DC.

* The most current version of a standard, code, or specification should be used for reference.



No two penstock installations are truly identical. Subtle variations require unique approaches best suited to a particular installation. There are, however, general installation requirements and procedures common to all penstock systems. This section discusses those general requirements and procedures, leaving the consideration of unique requirements and solutions to the designer and installation contractor.

12.1 Handling, Supports, and Tie-Downs

Compared to the fabrication site, the harsher environment at the construction site requires greater care in the unloading, storing, and handling of penstock sections. This is particularly true for pre-coated penstock sections.

To avoid damage to coatings, coated pipe must not be placed directly on the ground. It must be supported properly when in storage so as to spread the supporting load over a large area. If necessary, painted penstock may be rolled only along supports on unpainted areas. When

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lifting penstock for unloading or installation, fabric or fabric-protected slings must be used for exterior painted pipe and wire rope slings for unpainted sections. If available, sling pick-up lugs welded to stiffener rings or ring girder supports may be used. Every effort must be made to sling the penstock for pickup either at its balancing point or points of least pick-up stress. Penstock sections must not be dragged along the ground or bumped. While being fitted up, penstock sections should be sling-supported.

Both temporary and permanent supports may be needed during penstock installation. Supports include temporary wood blocking, steel saddles, internal stiffeners or spiders, external lugs, and anchor ties.

Also, steps must be taken to prevent excessive deformation of the penstock sections, particularly after the lining and coating have been applied. Internal bracing may be used to prevent over-stressing the shell and also to prevent excessive deflection and out of roundness. Padding supports and tie-down straps are necessary to prevent damage to the lining and coating.

Internal bracing (pipe stulling) may be necessary for buried penstock under backfill conditions. Table 12-1 lists pipe stulling criteria. The stulling criteria apply to all pipe, whether bare, painted, or cement-mortar lined. Wood stulls must be cut to a length equal to the pipe inner diameter less 1/4 inch. Cedar shingles can be used as wedges to provide a snug fit. Framing anchors are used to tie posts together (as radii of a circle). Steel stulls must be cut to a length equal to the pipe inner diameter less 1/8 inch and inserted in sleeves at one end. Sleeves and the other end of the stull must be tack-welded to the pipe shell. Based on common practice, wood stulls with felt or carpet padding are used for painted pipe with inner diameters up to 85 inches. The pipe stulling must not be removed until the compacted backfill is placed to a depth that provides adequate lateral support for the pipe.

Table 12-1 Pipe Stulling Criteria (for 40- /45-foot lengths)

DIAMETER-TO-THICKNESS RATIO (D/t)	O.D. ≤ 23 IN. (2 x 6-IN. WOOD)	24 - 29 IN. O.D. (3 X 3-IN. MINIMUM, WOOD)	30 I.D. - 49.5 IN. O.D. (3 X 3-IN. MINIMUM, WOOD FOR 30 IN.; 4 X 4-IN. WOOD FOR GREATER THAN 31 3/8 IN. O.D.)	> 49.5 - 59 IN. O.D. (4 X 4-IN. WOOD)	> 59 O.D.- 85 IN. I.D. (4 X 4-IN. WOOD)	> 85 IN. I.D. (STEEL TUBING WITH SLEEVE)
$D/t \leq 120$	NO STULLS	NO STULLS	NO STULLS	NO STULLS	NO STULLS	NO STULLS
$120 < D/t \leq 160$	BRACE BETWEEN BUNKS	2 STULLS VERTICAL	2 STULLS CROSSED	2 STULLS CROSSED	2 STULLS 3 LEGS	2 STULLS 3 LEGS
$160 < D/t \leq 200$	BRACE BETWEEN BUNKS	2 STULLS VERTICAL	2 STULLS CROSSED	2 STULLS 3 LEGS	3 STULLS 3 LEGS	2 STULLS 3 LEGS
$200 < D/t \leq 230$	—	2 STULLS VERTICAL	2 STULLS CROSSED	3 STULLS 3 LEGS	3 STULLS 3 LEGS	3 STULLS 3 LEGS
$230 < D/t \leq 288$	—	—	2 STULLS 3 LEGS	2 STULLS 3 LEGS	3 STULLS 3 LEGS	3 STULLS 3 LEGS

NOTES:

- (1) For 40 ft lengths, end stulls should be approximately 6 ft, 8 in. from pipe end.
- (2) For 45 ft lengths, end stulls should be approximately 9 ft, 0 in. from pipe end.
- (3) For 20 ft and 22.5 ft lengths, 2 stulls per pipe should be approximately 5 ft from pipe end.
- (4) Flanged ends and stiffner rings may replace stulls.
- (5) Shipping bunks and tie-down straps are to be located at stulls.

Table 12-2 gives steel and wood stulling specifications.

Table 12-2 Steel Stulling Specifications

SHELL I.D. (D) (IN.)	SHELL THICKNESS (t) (IN.)	D/t	REQUIRED r (IN.)	NOMINAL DIAMETER (IN.)	O.D. (IN.)	MINIMUM WALL THICKNESS (IN.)
120	1	120	1.00	3	3.50	.188
120	.75	160				
120	.52	230				
108	.90	120	.90	2-50	2.875	.203
108	.675	160				
108	.47	230				
102	.85	120	.85	2	2.375	.154
102	.64	160				
102	.44	230				
96	.80	120	.80	2	2.375	.154
96	.60	160				
96	.42	230				
90	.75	120	.74	2	2.375	.154
90	.56	160				
90	.39	230				

NOTE:

$L/r \leq 120$ is required for steel stulling material.

Pipe stulling configurations are illustrated in Figure 12-1.

Temporary bracing can be applied internally or externally to the shell.

Internal bracing, also called spider bracing, may consist either of a simple cross pattern attached directly to the steel interior by welding or a complex system of adjustable rods attached to a central ring with steel lugs welded at the interior of the liner. This bracing is used when the steel conforms to specified tolerances for out-of-roundness and it is necessary to maintain these tolerances during shipment and handling.

The adjustable rod type (tension spider) allows adjustment of the out-of-roundness tolerances in the shop or in the field. However, this is a difficult procedure with thick-walled penstocks.

Bracing and attachments also can be provided externally to the penstock in the form of stiffener rings. These may be required under design and, as such, perform a double function. External ring bracing can be used effectively for penstocks designed for saddle support, for buried construction, or for tunnel liners subjected to external pressures.

Bracing, when required to support the penstock sections for hydrostatic testing and concreting, should be minimized to the greatest extent practical or designed in a manner that does not restrain the shell. Bracing resulting in potential restraint should be located at low-stress shell areas of complex structures, such as wyes.

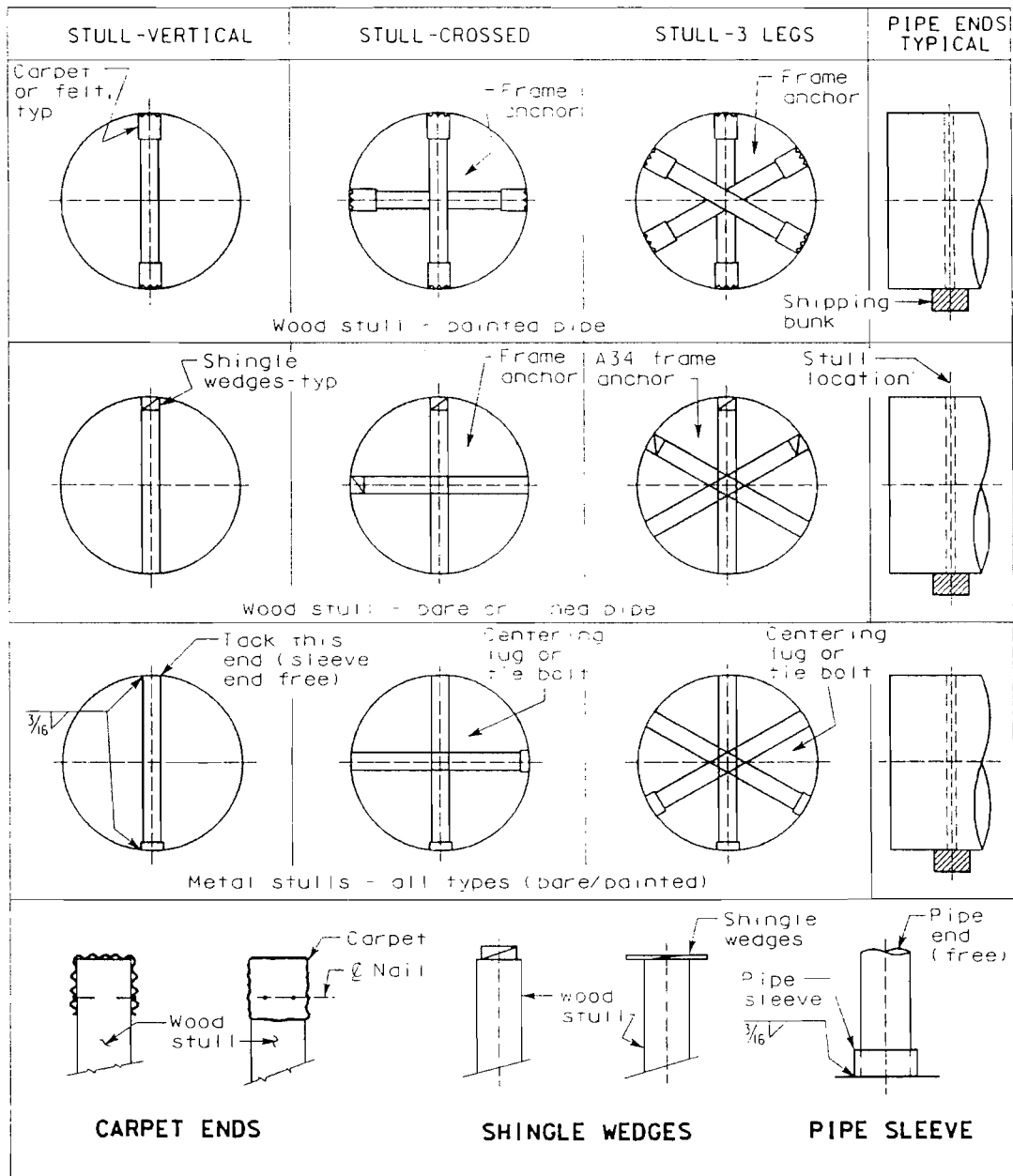


Figure 12-1 Pipe Stulling Configurations

When no longer needed, all bracing and attachments within the shell must be completely removed, the areas of weldments ground flush with the shell plate material, the surfaces inspected by NDE, and then coated as specified.

Temporary ties may consist of simple lug pieces welded to the shell surface with provisions for fastening a rope shackle or tie rod, or may consist of complex ring girder type tie-downs, if

needed to prevent shell movement during construction. Plate straps, anchor bolt footings, and lifting sling lugs, or other external attachments are used to accommodate working platforms for construction and for positioning and adjusting the shells into alignment with adjacent structures.

As with bracing, temporary ties are required primarily for shipment and to expedite construction. Ties that have served their intended use should be removed and the weldment ground smooth with the shell surface, inspected by NDE, and protected by painting.

Tie downs consisting of strap anchors can be used to hold down an aboveground penstock to each of its supports. These straps should be wide enough and suitably padded to limit stress applied to the pipe. They are intended to limit pipe movement. Tie downs also are required for penstocks fully encased in concrete as a means of preventing flotation.

Temporary bracing and ties can be fabricated of structural shapes and rolled stock material. This is true also for those components welded directly to the shell, unless material metallurgy dictates otherwise.

Penstocks or tunnel liners to be encased in concrete must be designed for the buoyant and concreting loads resulting from concrete placement. Pipe stulling may be necessary. In an installation through reinforced concrete, the reinforcing bars must not be used as temporary support for the penstock during installation. The penstock must not contact the reinforcing bars because of the risk of galvanic corrosion to the penstock.

During the installation of the penstock, consideration must be given to ensure that permanent supports, such as thrust blocks or thrust-resisting saddles, are installed in conjunction with the penstock to accept any gravitational or temperature-related loads. This is particularly true for exposed penstocks on steep terrain and penstocks using flexible field joints such as bolted sleeve-type couplings and expansion joints.

12.2 Temperature Conditions During Installation

During penstock design and installation, the effects of temperature often are not given sufficient consideration and are seriously underestimated. The effects of temperature variations are most severe for an aboveground penstock installation exposed to direct sunlight, less severe for a buried penstock during its installation, and least severe for a penstock installed free in an open tunnel or embedded in concrete, i.e., a steel liner. The main temperature effect is movement during construction. Temperature-induced movement plagues every penstock builder because of its potential to cause both temporary problems as well as permanent damage to the penstock and its supporting structures. Ideally all installation should be done at a temperature that is constant and close to the mean temperature of the site. The effects of temperature must be investigated for both the dewatered and watered conditions.

Temperature changes affect alignment, straightness, and length of penstock cans. Direct sunlight on an exposed penstock results in one side being hot and the shaded side cooler. Temperature differentials can be as high as 70°F, with the side exposed to direct sunlight reaching temperatures up to 140-150°F. These conditions cause penstock deflection and/or curling, resulting in large side-lurch forces on the supports. This problem is especially critical on

2 Installation

long-span, ring-girder-supported installations. Even for cantilever type installations, the end of the cantilever can fluctuate several inches off alignment with the variation in night and day temperatures. Temperature side-lurch, expansion/contraction, and curling conditions are calculable for both the dewatered and watered conditions. The designer should prepare an adjustment table for all three axes to be used by the constructor during the installation process. Resulting loads and stresses should be considered in the design of the shell and its supports.

Side-lurch forces and uplift conditions may be greater during construction and the dewatered condition because there is no water to act as a heat sink. High lurch or uplift forces have been known to cause support displacement and damage. The dewatered penstock line and its supports should be inspected for potential damage before refilling.

It is important to make frequent checks of penstock alignment and length during the construction period. These checks are best done before sunrise (dawn) or after sunset (dusk) when the installation has reached a uniform temperature over the entire shell surface. These checks then should be compared with the design assumption criteria and alignment for the mean temperature at the site; corrections should be made as needed. Should there be increases or decreases in length that exceed allowable tolerances, adjustments to the penstock length must be made either by cutting off any excess length or by adding a filler piece of pipe. Decreases in length due to temperature effects may be aggravated by weld-related shrinkage, which can reach as high as 1/8 inch for each butt-welded joint.

Long, continuous runs of exposed penstock are subject to greater temperature effects. Because concrete footings or piers usually are placed prior to the penstock erection, adjustments in the base plate and ring girder support/fastening design are necessary. Allowance should be made for misalignment in all directions at each footing or pier. The degree of allowance for directional movement will vary with the installation and could be as high as 2 inches from a theoretical reference axis. By providing this type of flexibility, the number of field adjustments for shrinkage, length increase, or misalignment can be reduced greatly on long penstock runs. Expansion joints designed to allow as much as 6 inches of change in length also help protect the installation against temperature effects. These expansion joints can be located so that the penstock length upstream and downstream of the joint undergoes an essentially equal thermal expansion/contraction or deflection/rotation condition. On steep slope installations sliding of coupling joints can be prevented by attaching external restraints to the shell.

The temperature effects on an exposed penstock can be reduced by the application of a reflective type paint or by insulation. Insulation may need to be considered in very cold regions where there have been cases of freezing of the water within the conduit, even under flow conditions. To prevent freezing, the only safe measure is to maintain the water temperature above 32°F, with a margin (if possible) of 1°F or 2°F, even though water may not turn into troublesome ice even at 28°F to 29°F. Troublesome ice is water containing ice particles (frazzle or needle ice). The ice particles may cause serious problems because they can quickly block a pipeline by adhering to valves or any minor obstructions. Calculations relating to the prevention of freezing in pipelines are based on the same principles of heat transmission and loss that govern other similar structures. The designer must determine frost depth and the factors that influence this depth, i.e., soil type, soil condition, land contouring, land exposure, and snow

cover. Similarly, warming water may have detrimental effects due to temperature-related stress conditions at shell restraints.

Also requiring consideration in the design of footings, piers, anchor blocks, or other concrete supporting structures are the freeze-thaw effect and temperature heat cracking that occur at these concrete structures. Further, the thermal-related loads imposed by the penstock on these structures must be recognized in the design process. The designer must consider a design temperature condition and then calculate, for construction use, the adjustment that may have to be made in the field for temperature variations. This will not only provide valuable information for use in the installation, but also compel the designer to give consideration to temperature effects.

During cold weather, lowered temperatures affect the penstock welding and protective coatings. For certain types of coatings (for example, coal tar enamel) it is inadvisable to store unprotected pipe through the winter season prior to installation. Also, to prevent damage to the coating, special care is needed in handling or transporting coated pipe during cold weather.

Pipelines installed on slopes, particularly above ground, tend to slide downslope and, under certain conditions, resist uphill creep. Large pipes at the bottom of the slope have crushed under direct axial compression when, upon expansion due to temperature change, they did not creep uphill as expected. Use of frequent anchoring on long slope intervals reduces to safe values the weight of pipe supported at each anchorage.

Steel liners embedded in concrete also experience the effects of temperature variations. Axial length changes due to temperature changes (15°F to 35°F temperature differentials) produce shearing loads on external fasteners and attachments. Also, temperature variations cause radial shell growth or shrinkage and can influence the stress conditions in the shell. Usually axial and outward radial movements are not considered critical to the design. For axial movements, it is reasonable to assume that temperature changes affecting the steel shell will be gradual and, as such, the shell and surrounding concrete will react similarly, without the presence of relative differential movements. This assumption holds even with heavy thrust rings or seal rings at the shell exterior. The assumption does not apply if the temperature of the water flowing within the liner changes rapidly.

12.3 Welding, Preheat, and Postweld Heat Treatment

Many different types of steel can be used for penstock work. Variations of these materials in terms of form, thickness, and weld joint configuration make it difficult to present specific welding criteria or parameters.

Material physical properties and chemical characteristics for the parent and weld filler metals, shop capabilities and experience, the selected welding procedure, preheat and postweld heat treatment, and the environment in which the weld is formed all play key roles in the formation of a welded structure. The metallurgist, fabricator, plate supplier, and filler metal supplier are key members in a well-executed welded penstock.

Weld preparation of joints is done by flame cutting, grinding, planing, crimping, or shearing. All edges to be welded should be clean or cleaned by grinding and should form the required angle for the joint. Torch cutting requires special attention to keeping edges linear and preventing distortion. All torched surfaces must be ground to remove impurities.

Shop and field welding of the penstock and steel liner shells (and also assemblies subject to high internal or external pressure) must be in accordance with the ASME Code, Section VIII, Division 1, Part UW. Special requirements for a particular installation may necessitate deviation from ASME Code specifications. Consideration also must be given to welding requirements and techniques for field welding, as given by AWWA standards and AWS D1.1.¹ Bend, Charpy V-notch, and other nondestructive weld tests should be considered as warranted by the installation or as required by standards and codes.

Shielded metal arc welding (SMAW) or flux core arc welding (FCAW) can be used for field welds and welding of intricate joints. In using these welding methods, the designer should ensure that acceptable Charpy V-notch test results are obtainable in both the weld metal and the heat-affected zone (HAZ). Submerged arc welding (SAW) is used for longitudinal and circumferential welds on cylindrical and conical pipe sections welded in the shop and, in some instances, in the field. Inert-gas-shielded arc welding has seen some use as has the electroslag process for thick-walled pipes. Use of these two methods requires special precautions and treatment of welded members; the metallurgist and shop fabricators must be consulted prior to their use.

Edges of plates prepared for welding should be examined for signs of laminations, shearing cracks, and other imperfections before and after forming. Pieces showing signs of laminations (cracks or visible seams) may need to be rejected. Ultrasonic examination must be used to ensure that laminations do not extend 3 inches or more from the weld edge as specified in the ASME or ASTM codes. When rolled into a cylindrical or conical section, the abutting surfaces to be welded must form only straight lines for welding on both longitudinal and circumferential joints. Patches or re-entrant shapes must not be welded into locations where laminated areas have been replaced. The repair of lamellar tears appearing during welding of longitudinal or circumferential welds must be in accordance with special requirements. Welds located on highly stressed areas must be given special consideration relative to formation, orientation, welding technique and approach, restraint, and nondestructive testing requirements.

During the welding process, attention must be given to minimizing distortions due to welding. When welding circumferential joints of a large-diameter penstock, adequate stiffening must be provided, because flexing during rotation of the pipe may lead to transverse cracks in the weld filler metal. High-strength, thick-wall steel penstocks must be welded, as far as possible, in the same heat without interruption. This applies especially to field welds that need to be made under restrained conditions.

Successful welding also must consider other essential variables such as preheat. Preheat temperatures must be at least 50°F, but are also dependent on the type of parent material selected, its carbon equivalent, plate thickness, electrode diameters, and amperage used during welding. Higher preheats are beneficial in removing hydrogen, thus minimizing the potential for hydrogen embrittlement. Preheat equipment should be selected for its portability, good heat regulation, and uniform heat distribution.

Postweld heat treatment (PWHT) and/or annealing also may be desirable for some steels. Local heating of welds causes considerable expansion of material, with additional stresses resulting from contraction during cooling. This is especially true for circumferential joint welds in which postweld heat treatment must be uniform over a width of approximately 2 feet on either side of the joint to prevent additional weld bending stresses due to expansion and contraction. Heat also must be applied an additional 4 feet from the 2-foot limit on each side to create a temperature gradient decreasing away from the weld at a rate of 250°F per foot of length.

Postweld heat treatment normally is required when butt-welded plate thickness exceeds 1½ inches or as may be required by Section VIII, Division 1 of the ASME Code. If this type of treatment is not feasible or economical, then consideration should be given to using higher strength steel to thin down the plate thickness to less than 1½ inches. If postweld heat treatment is specified, test specimens of the welded plates should be prepared and heat treated to simulate the PWHT used during fabrication. This allows testing to be correlated with the material properties of the completed structure.

12.4 Longitudinal and Circumferential Joints

When calculating the shell thickness, the engineer must look not only at the properties of the plate material but, more importantly, at the properties of the weld. A transverse crack on a circumferential joint is equally as dangerous as a weld defect in a longitudinal seam. Although the longitudinal stress on a closed-end penstock liner is one-half the hoop stress, the hoop stress component is the same for a circumferential weld as for a longitudinal weld.

Field-welded joints must satisfy the same or even more stringent requirements specified for shop-fabricated joints. Several joint configurations can be used for penstock design. The single "V" butt, asymmetrical double "V" butt, and the symmetrical double "V" butt-type joints are preferred for penstock work. Similarly detailed "U" type joints have been used with thicker plates. The square butt-type joint is used for thin plates and penstocks subjected to low pressures. Back-up bar type welds also are used for circumferential joints but are not recommended on thick plates or on high-strength, highly-stressed material. They must not be used on longitudinal joints. Back-up type joints must be avoided because of the dangerous notch at the back-up bar splices and the flaws that can be expected to develop in the root weld. If used, back-up type joints must be welded on both sides to prevent corrosion at the root of the weld. The selected joint detail depends on several factors, including whether welding will be in the shop or field, the method of welding (automatic or manual), and the accessibility of the joint for welding. Appropriate joint reduction factors as specified in this manual must be used in the design.

Careful consideration must be given to the detailing of joints at angle changes and for plates of different thicknesses. The guidelines given by the appropriate sections of the ASME Code can be used. Weldments of plates must meet certain tolerances relative to angle or flattening of welds, displacement of longitudinal welds, displacement of circumferential welds, and crowning or undercutting of welds. Again, appropriate ASME Code guidelines can be used for tolerance specifications.

When detailing plates for can sections, longitudinal welds should be located within 45 degrees above or below the horizontal axis of the circular section. Similarly, longitudinal weld joints on mitered bends should be placed on the outside radius of the joint within 45 degrees above or

below the horizontal center line. These locations place the joint at less stressed areas. It should be noted that for mitered bends, the shell area on the inside (smaller) radius of the bend is more highly stressed than on the long radius side. The smaller the radius, the greater the degree of weakening. Also, when joining pipe sections at circumferential joints, rotate the sections about their longitudinal axis by 15 to 20 degrees, so as to offset adjacent longitudinal joints and prevent the formation of a cross. This procedure should be followed for both shop and site fabrication.

Where field-welded joints are designated, the pipe should be left bare or coated with a weld-compatible primer about 4 to 6 inches back from the joint to prevent damage to the final protective coating due to welding.

12.5 Concreting

Concreting of structures for exposed penstock and buried pipeline such as footings, piers, and anchor blocks should follow the criteria, guidelines, and working practices established by the American Concrete Institute. For the loads usually encountered in these installations, concrete having a minimum of 3,000 psi, 28 day strength, is usually satisfactory. Good material, admixtures, and mix practice are necessary to achieve this strength. Other desirable characteristics include workability, durability, and freeze-thaw and alkali-reactivity resistance.

Concreting of steel liners presents more difficult installation conditions compared to readily accessible structures. Concreting of steel liners requires:

- (1) Preparation of structures for concrete placement
- (2) Clearances for concrete placement around the liner
- (3) Installation of slick lines for placement of concrete
- (4) Supports and tie-downs to maintain shell alignment and to prevent flotation during concreting.

Preparation of the foundation and excavated tunnel rock surfaces for the placement of concrete depends on the purpose of the concrete. For exterior aboveground and buried penstocks, concrete provides structural support. For steel liner embedment, the concrete serves not only as a structural support for the filled penstock, but also distributes the internal pressure to the surrounding rock; the concrete also maintains the physical stability of the steel liner against external pressure and thermal-related forces. Although the steel lining provides the desired hydraulic flow and water-sealing attributes and is the predominant load-carrying member against internal pressure, it is desirable to provide a mass of solid, continuous concrete in close contact with surrounding rock. The rock surface must be cleaned and all timber or other degradable and nonessential support material must be removed. All possible voids should be filled by grouting.

Water seepage must be controlled to prevent damage to the fresh concrete before it sets. Side wall flow guides and invert drains can be used for this purpose.

High water flows may require grouting, damming and/or pumping to remove water prior to concrete placement. Drains and other water control features not required as part of the

permanent operating system should be marked for location and later grouted. Excessively high inflows of water must be controlled by grouting the rock face as the tunnel is advanced.

Concrete of 1½ inches maximum size aggregate may be used. The maximum size depends on the concrete pump used, the thickness of the concrete lining around the steel liner, and the existence of obstructions such as stiffener rings, external shell drain pipes, and thrust rings around which the concrete must flow. It is necessary to select concrete placement equipment that does not involve high-velocity discharge or cause concrete separation.

The concrete mix for backfill must be, as a minimum, 2,000 psi, 28-day strength. Low heat of hydration cement should be used to prevent overheating. The inside of the steel liner should be spray-cooled continuously with water during setting. Fly-ash, superplasticizers, an increase in sand content, or air entrainment can be used to enhance concrete placement, if there are vibration difficulties or any external stiffeners and appurtenances that may inhibit good distribution of the concrete. Whenever possible, concrete placement should begin at the crown and at both lower quadrants and slowly worked around the sides of the liner. Pipes used for concrete placement must be well submerged in fresh concrete. This embedment may vary from 3 to 5 feet for thin concrete linings in tunnel-boring-machine (TBM) tunnels to 5 to 10 feet for the thicker linings of drill-and-blast tunnels. Sloping cold joints may be permissible. For a vertical joint the can sections being concreted may be bulkheaded at the downstream end. Bulkhead forms must be stripped 72 hours after placement (sooner if field conditions allow). Green cutting of cold joints is preferred, depending on the installation.

Tie-down is very important in steel liner installation, not only to prevent flotation but also to prevent the formation of *hard spots* on the exterior of the shell. Cable tie-downs and adjustable steel struts with some compressible wedging against the shell can be used. Wood wedging should be avoided because it could lead to corrosive action from chemicals within the wood. Pin connected tie-downs that permit some outward radial movement at the shell also are acceptable. Hard-welded or braced holddowns must be avoided.

12.6 Grouting

Grouting requirements discussed in this manual apply primarily to steel tunnel liners embedded in concrete. Grouting applications for steel liners include: (1) contact grouting of the voids between the concrete backfill and rock intersurface, and (2) void or skin grouting of voids between the steel liner and surrounding concrete.

12.6.1 Contact Grouting

Contact grouting generally is performed in the upper 180 degrees of the tunnel and must be incorporated in the design and detailing of the steel liner. Grouting plugs are provided in the shell during the shop fabrication process. Customarily, contact grout plugs are located at the top shell surface, 15 and 60 degrees on either side of the vertical central axis of the shell. These plugs can be located either at intervals of 10 feet along the same plane that is perpendicular to the liner axis or can be staggered at intervals of 5 to 10 feet on either side of alternate transverse planes. The preferred contact grouting plug details are those using a stainless steel

sealing disk (see Section 9.11). The welded stainless steel disk has proven to be an excellent detail and helps prevent the cracking associated with seal welding around threaded plugs.

Contact-grouting installation and implementation procedures during construction are the same as those imposed for the concrete-lined section of the tunnel. Grout holes are drilled through the concrete at a depth of about 1 to 2 feet into the surrounding rock. Drilling is done through the grout plug or through pre-placed grout sleeves.

The grout pipe connection is made to the threaded grout boss plate, and contact grouting is performed using a sand-cement grout mix. Grout pressures may range from 30 to 100 psi. Higher pressures have been used. The higher pressure range should be checked for the potential for steel liner buckling due to external local pressure.

12.6.2 Void (Skin) Grouting

Void or skin grouting fills voids that develop between the steel liner and concrete backfill because of shrinkage due to concrete curing and from temperature adjustments through the steel shell and surrounding media. There are two schools of thought relating to void grouting. One, as expected, is to require void grouting as an enhancement to the resistance of the steel shell against buckling from external pressure and for the better transfer (sharing) of load between the steel shell and surrounding media. The second school advocates that void grouting is not a cost-effective or safer approach, because of additional field construction costs, cleaning work, schedule impacts, in-situ painting delays, and problems in ensuring a good weld sealment detail for the void grout hole. This second school's approach requires that the shell be designed to handle external pressure assuming a conservatively sized gap and higher external pressure, up to water levels at reservoir maximum static head levels or at a water pressure equal to ground level. It is alleged that this approach results, in most cases, in a safer and more economical installation. Should void grouting be elected, however, sounding of the steel shell is required several days or even weeks after concreting. This period of time permits good setting and allows concrete shrinkage and temperature stabilization to occur. Void grouting should be done at pressures not exceeding 30 psi.

By sounding the steel shell, the peripheral extent of voids can be established and marked on the shell interior. After a void area is located, 1/2-inch to 3/4-inch holes are drilled through the steel at the lower and upper void area extremities. A flowable, nonshrink, noncorrosive type grout is pumped through the lower hole and allowed to vent through a riser pipe installed at the upper hole. After the grout has set, both holes are plugged with threaded plugs and capped with a welded stainless steel plate, similar in detail to the grout plug plate.

Another method is to provide grout holes along the invert of the shell at a maximum of 10 feet on centers, or one each between stiffeners. Vent holes also are provided. Grouting between the steel liner and concrete is done first, followed by grouting between the concrete lining and rock (followed by high-pressure consolidation grouting of the rock, if required).

12.7 Trenching, Bedding, and Backfill

Trenching, laying, bedding, and backfilling methodology, approach, and techniques depend on factors such as character and purpose of the penstock; its size, operating pressure, and operating conditions; the location of the penstock relative to city/urban areas; the type of terrain to be traversed by the installation; the depth of the trench; and the character of the trench soil and backfill.

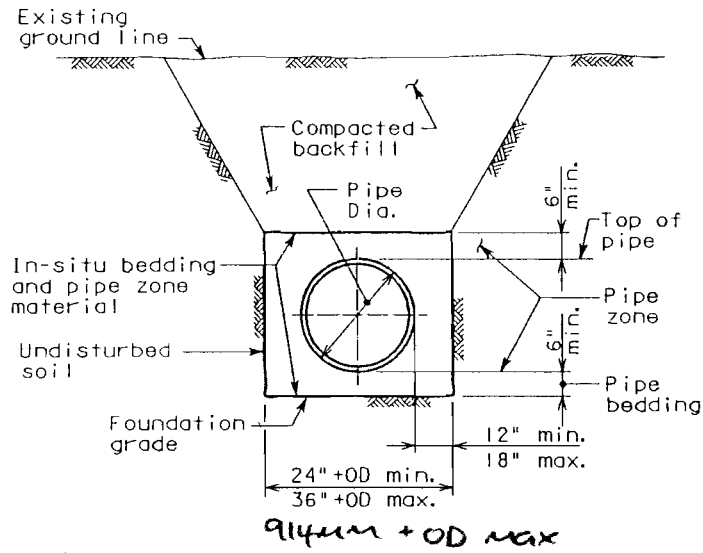
Trenching details for buried penstock designs are as numerous and different as for aboveground installations. Two typical examples are given in Figure 12-2. Factors to consider in the trenching, bedding, and backfill of buried penstock include:

- (1) Surface and subsurface exploration
- (2) Soil/rock classification
- (3) Excavation and/or blasting techniques
- (4) Effects of backfill on penstock coatings
- (5) Material sources and disposal
- (6) Refurbishing and/or reconstruction of disturbed grounds, facilities, and structures
- (7) Safeguarding and bracing of excavations
- (8) Closing of abandoned conduits
- (9) Specification for and care of backfill material
- (10) Backfill operations for pipe and structures
- (11) Rock refill and sand backfill criteria.

The engineering properties of the excavated soils and the backfill are very important. The use of soil mechanics will produce safer and more cost-effective results.

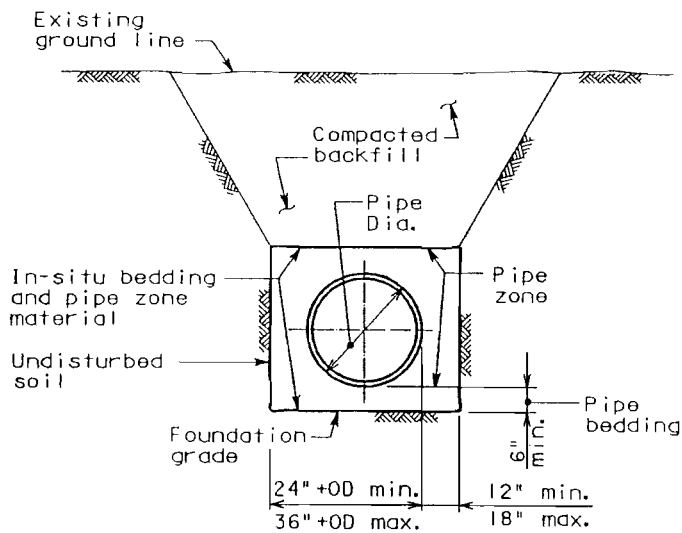
Trenching depth must be based on design requirements that have taken into consideration frost line depth and any existing civil government guidelines and conditions. Trench width should provide for minimum, safe working clearances, with consideration given to the need for the working space that may be required during tamping operations or for local installation of joints and/or appurtenances. It is important to grade the bottom of the trench as accurately as practicable. Backfilling is not permitted until bad, local foundation conditions have been treated.

The bottom of the trench should be free of stones and hard lumps and well graded. For large pipes, the trench bottom may be shaped for arc contact instead of line contact. If rigid blocks are used for pipe support (not recommended), they should be removed prior to backfill. Rigid



Note:
Trenches must conform
to applicable OSHA
requirements.

(A) FOR PIPE DIAMETERS ≤ 90 IN. (2.286 m)



(B) FOR PIPE DIAMETERS > 90 IN. (2.286 m)

Figure 12-2 Typical Trench Sections

blocks form hard spots that can damage pipe coatings. Also, they can create a galvanic cell and encourage corrosion. If blocking of pipe is necessary, bags filled with sand should be used; they can be broken at the appropriate time to prevent load concentration. Alternatively a layer of pea gravel or sand 3 inches or thicker can be placed on the bottom.

Important in the design of buried penstocks is a good understanding of the earth loads on steel pipe and the principles of how vertical and lateral loads resulting from trench fill or from fill cover correlate with the flexibility/rigidity of the pipe. General backfill requirements include: (1) use of a material that is free of boulders, rock, or other unsuitable substances, and (2) the deposit of the material in the trench along both sides of the pipe to an elevation of not less than 6 inches above the top of the pipe. Material for backfill may be taken directly from the excavation unless the material is unsuitable or of insufficient quantity. Backfill must not be dropped directly on the pipe. The material must be placed along each side so that the surface elevation at the sides is even, with the differential on each side not exceeding 6 inches. Backfill containing hard material over 1 inch in diameter must not be placed within 6 inches of the penstock. The backfill must be tamped in horizontal layers of thicknesses varying from 4 to 12 inches, with 6 inches being recommended. Layer thickness depends on the fill properties, degree of compaction required, and method of compaction. Hand or power tampers can be used to compact cohesive material to a density of not less than 95% of laboratory dry density as determined by the Proctor compaction test (ASTM D-698). Free-draining, cohesionless material, such as sand or gravel, may be compacted by tampers, rollers, surface vibrators, or internal vibrators. Free-draining material also must be deposited in horizontal layers; after compaction, layers must not exceed 6 inches in thickness.

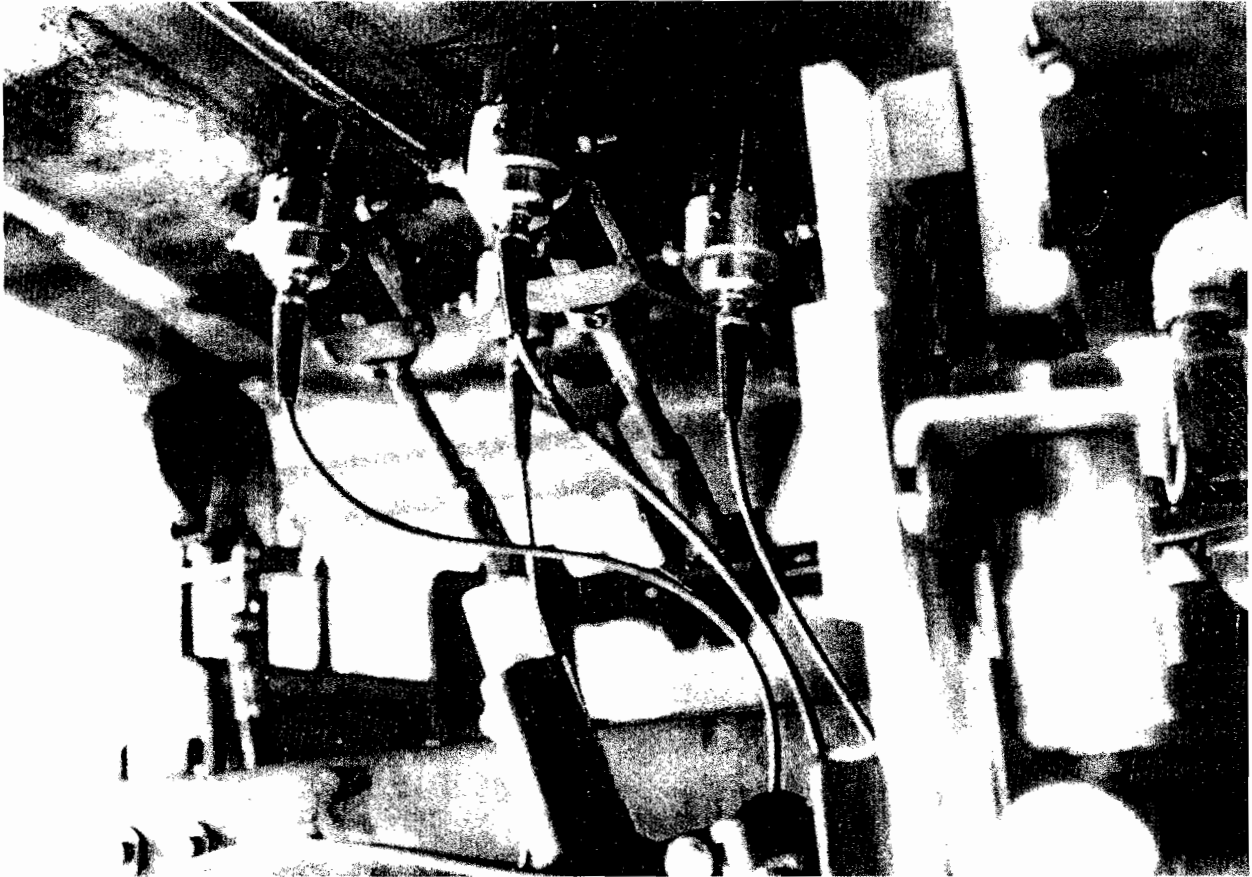
The importance of equally distributing fill material along each side of the penstock increases as the penstock becomes larger or more flexible. Sluicing of backfill and backfill settling by flooding also have been used in several installations. For large-diameter (16 feet or larger), thin-wall pipe for low-pressure design, the use of stiffeners provides a well designed and economic solution for buried pipe. Careful placement of backfill is required because the backfill provides the backing against the penstock. Primary attention in backfilling must be given to the space between the trench bottom and the penstock.

Backfill that is 6 to 8 inches above the top of the pipe may contain coarser material, but should be free of organic material or other objectionable matter that could cause improper consolidation and potential settlement. Flooding may be used for this portion of the backfill, provided precautions are taken against penstock flotation (due to deliberate or accidental flooding of the trench).

References*

1. Structural Welding Code—Steel. AWS D1.1. American Welding Society, Miami, FL.

* The most current version of a standard, code, or specification should be used for reference.



13.1 General

13.1.1 Owner/Engineer

Since the amount and type of nondestructive testing required affects the basic design, testing requirements must be determined early during the design stage. Testing and inspection alone do not ensure the integrity of the penstock. They provide only additional assurance that the completed penstock does in fact conform to the engineer's design considerations.

13.1.2 Contractor

The supplier, fabricator, and installer are responsible for quality control programs that provide the degree of quality control/quality assurance required by the project specifications. All inspection and testing must be documented (see Section 15). Inspection by the owner or his designated representative does not relieve the fabricator or installer of the responsibility of conforming to the project specifications.

13.2 Shop/Field Inspection

13.2.1 General

All penstock components are subject to inspection at the place of fabrication and/or installation. The inspector must be allowed to view any or all of the operations and have access to all report forms and radiographs.

13.2.2 Submittals

Upon request, the following documentation must be submitted to the owner:

(1) Fabrication, welding, and inspection procedures

Prior to the start of welding, copies of the following must be submitted: proposed welding procedure specification (WPS), procedure qualification reports (PQR), welding sequences, repair procedures, and welding rod control. Documentation must include a weld map or table identifying each type of weld joint, the assigned welding procedure specifications, the parts being joined, the material thickness, and any requirements for preheat and postweld heat treatment.

Welding procedures must be qualified in accordance with Section IX of the ASME Code.

All procedures for weld and nondestructive examination (NDE) must be submitted prior to the start of work.

(2) Personnel qualification records

Welders and welding operators must be qualified in accordance with Section IX of the ASME Code. Requirements for qualification include welder/welding operator performance qualification (WPQ) tests, recorded on ASME Code forms QW 482-484,1 or their equivalents.

nondestructive examination (NDE) personnel must be qualified under ASNT SNT-TC-1A² (visual examination excepted). Qualification records of NDE personnel must be submitted to the inspector upon request.

All nondestructive testing technicians and operators must be qualified at NDT Level II as defined in ASNT SNT-TC-1A.²

All inspection work must be performed by certified welding inspectors (CWI) who are certified in accordance with AWS QC1³ provisions.

(3) Weld examination and inspection reports

These reports must include detailed records showing evidence of quality of welding. For each section of weld inspected, a report form is required. The report must identify the work and show the welder's identification, the area of inspection, the acceptance of the welds, and

the inspector's approval signature. Each report must be completed at the time of inspection. A complete set of forms must be supplied upon completion of the work. For radiographic examination, in addition to the report forms, a complete set of radiographs must be supplied.

13.2.3 In-Process Inspection

Prior to the start of manufacture, the owner's inspector or designated representative must review the testing requirements of the specification and the applicable codes and standards for tolerances.

13.2.3.1 In-Process Inspections of Fabrication

In-process inspections must include the following where applicable:

- (1) Review of the purchase order and specification document requirements for materials, fabrication, welding, examination, testing, marking, tagging, cleaning, painting, and preparation for shipment.
- (2) Review of the suppliers' quality assurance programs for conformance with the specification requirements, and discussions with the suppliers' personnel about the quality verification activities to be performed during the course of the assignment.
- (3) Review of the shop drawings and other vendor documents that require submittal prior to fabrication, including a determination that they have been approved by the engineer, are properly stamped, and released for fabrication.
- (4) Review of mill test reports for all materials used in the fabrication of the penstock sections to verify that the materials are in compliance with the specification and applicable ASTM standards for chemical composition, mechanical properties, and Charpy impact tests, if applicable.
- (5) Verification that the pressure and attachment material identification, heat number, and thickness correspond with the certified mill test reports provided for the job. In addition, the owner may request the certified mill test reports for other materials. Copies of certified mill test reports must be available for inclusion in the job file.
- (6) Verification that welding procedures and welders have been qualified and are approved in accordance with the requirements of the specification, drawings, and applicable codes and standards. This may require an in-process review of the welding procedure specifications (WPS) and supporting procedure qualification reports (PQR), for conformance with the specification, drawings, and applicable codes and standards, particularly if there were design or personnel changes after the original submissions.

Welders and welding operators must have current certifications and be able to produce acceptably sound welds with the processes, materials, and welding procedures to be used in production.

- (7) Verification that welding electrodes, filler materials, and fluxes are identifiable, comply with the applicable welding procedure specification, and are properly stored in a clean and dry environment. Baked-out electrodes are required for high-strength materials greater than 80-ksi tensile strength. Low-hydrogen electrodes must be kept in holding ovens if not in their sealed containers.
- (8) Check of all in-process welding for conformance to approved welding procedures, drawings, and applicable codes. This includes all welding parameters, preheat, and interpass temperature requirements.
- (9) Where welding repairs are required, verification that the contractor has an approved repair and inspection procedure to ensure satisfactory defect removal and welding of the affected area.
- (10) Check of plate edges.
- (11) Inspection of shell courses for concentricity and roundness, and verification that the fit-up gaps of longitudinal and circumferential joints are within the tolerances given by the specifications.
- (12) Inspection of the various parts of the penstock to ensure that all appurtenances have been provided and that the projections and orientations are in accordance with approved shop drawings.
- (13) The inspector's witnessing of nondestructive examination (NDE) of materials or welds (spot check) on a random basis or as specified by the purchase order, and review of the required radiographic film to ensure that there are no indications exceeding the limitations specified by the applicable codes and standards.
- (14) Verification by the inspector that any required preheat or postweld heat treatment has been performed in accordance with the approved procedures. The inspector must obtain copies of time-temperature charts of each stress-relieving operation for inclusion in the job file.
- (15) For elbows, measurement of each chord to ensure that the units are fabricated as specified and have the desired radii.
- (16) For wye branches, verification of branch angle(s), projection, and orientation.
- (17) When bolted sleeve-type couplings and mechanical expansion joints are specified, a review for compliance with the procurement documents of the mill test reports provided for the couplings and gasket materials, and the documentation of nondestructive examination of welding (if required). Couplings and gaskets must be checked for physical damage, and dimension checks made to verify compliance with the drawings. The longitudinal or spiral seam welds on both ends of the penstock sections must have been ground flush in accordance with the drawings or specifications.

13.2.3.2 Coating and Lining Inspection Requirements

When corrosion protection, internal lining, or external surface coating is specified in the purchase order, a careful inspection for strict compliance with the specification, referenced standards, and Section 10 of this manual is mandatory. Details of inspection procedures for coating materials and their application are given in Section 10.

13.2.4 Final Shop Inspection

- (1) Ensure that all required test reports, as-built drawings, and other documentation generated during the fabrication of the penstock are available and processed in accordance with the procurement documents.
- (2) Prior to shipment, perform final inspection on the penstock sections, and verify that all deficiencies have been corrected and that each section has been properly marked.
- (3) Verify the identification marks on each penstock section for conformance with specification and drawing requirements. Confirm that the top and bottom center lines of special sections have been marked and are clearly visible on both ends of each section.
- (4) Verify that each penstock section has been properly braced to prevent damage during transit and handling. Obtain assurance that the penstock sections will be correctly blocked for shipment and that coated penstock sections are loaded on padded bunks or saddles as required.

13.2.5 Final Field Inspection

The entire penstock, including all coatings and linings, must be inspected for damage that may have occurred during shipment and erection. All damage must be repaired.

13.2.5.1 Damage

Damaged areas, such as scratches, gouges, grooves, or dents as determined by the inspector, must be corrected as specified in the project specification.

13.2.5.2 Field Weld Inspection

All field welds must be inspected for conformance to the project specification and to the requirements of Section VIII, Division 1, of the ASME Code.

13.2.5.3 Field Inspection of Linings and Coatings

All coatings and linings, including those applied in the shop or field, must be examined for damage that may have occurred during shipment and erection according to the appropriate requirements of the specification, referenced standards, and Section 10 of this manual. All damage must be repaired.

Cathodic protection systems must be tested for electrical continuity. If an impressed current system is in place, the power source must be given appropriate performance tests. Documentation in the owner's file must detail operating procedures.

13.2.5.4 Bedding and Backfill

All bedding and backfill must be inspected for conformance to the project specification during placement and after completion.

13.2.5.5 Reports

Ensure that all required test reports, as-built drawings, and other documentation generated during fabrication and erection of the penstock are available and processed in accordance with the procurement documents and specification.

Copies of reports for field and shop welding, NDE, and repair procedures must be included in the documentation in the project file.

13.3 Nondestructive Examination

13.3.1 General

The type of nondestructive examination (NDE) for all butt joints and certain full-fillet lap joints in pressure retaining parts is established by the designer in conjunction with the weld joint reduction factors described in Section 3.5.1.

The design engineer must document the required nondestructive examinations for these weld joints in a Fabrication and Installation NDE Data Sheet. For weld joints that are not described in Section 3.5.1 or where supplemental NDE is required, minimum nondestructive examination requirements are listed in Table 13-1.

Table 13-1 Minimum Nondestructive Examination Requirements

	$F_y < 55 \text{ ksi}$	$55 \text{ ksi} \leq F_y \leq 75 \text{ ksi}$	$F_y > 75 \text{ ksi}$
CORNER JOINTS AND FILLET OF PAD TO SHELL (SEE FIGURE 4.6.1-2(A))	100% MT*	BLEND GRIND** AND 100% MT	BLEND GRIND AND 100% WET MT
CORNER JOINTS (SEE FIGURE 4.6.1-2 (B) AND (C))	100% MT*	BLEND GRIND AND 100% MT	BLEND GRIND AND 100% WET MT
FILLET WELD (SEE FIGURE 4.6.1-2 (D))	10% SPOT MT	100% MT	BLEND GRIND AND 100% WET MT
STUD WELD (SEE FIGURE 4.6.1-2 (E))	VISUAL	10% BEND TEST	BLEND GRIND AND 100% WET MT
BUTT JOINTS (SEE FIGURE 4.6.1-1 (A))	SECTION 3.5.1	BLEND GRIND AND 100% MT	BLEND GRIND AND 100% WET MT
FILLET-WELDED BUTT STRAPS (SEE FIGURE 4.6.1-1 (B))	SECTION 3.5.1	BLEND GRIND AND 100% MT	BLEND GRIND AND 100% WET MT
BACKED-UP BUTT JOINT (SEE FIGURE 4.6.1-1 (C))	SECTION 3.5.1	BLEND GRIND AND 100% MT	BLEND GRIND AND 100% WET MT
DOUBLE-FILLET LAP JOINT (SEE FIGURE 4.6.1-1 (D))	SECTION 3.5.1	BLEND GRIND AND 100% MT	BLEND GRIND AND 100% WET MT
REPAIRS AND MATERIAL/OVERLAY	100% MT*	BLEND GRIND AND 100% MT	BLEND GRIND AND 100% WET MT

* MT applies to all final weld surfaces only.

** Blend grind is defined as the elimination of surface irregularities and fairing of edges of the weld.

Nondestructive examination for acceptance of any material subject to hydrostatic pressure testing must be performed prior to the hydrostatic tests. Repaired defects must be retested by the same NDE methods used for the original tests. If the engineer determines that the specified nondestructive testing is not possible because of conditions encountered in the work, the engineer must choose another method of inspection. The engineer may require additional tests and examinations.

13.3.2 Areas Requiring Special Consideration

- (1) Where required by the project specification or the weld procedure, weld bevel preparation in wye branches and penstocks must be magnetic-particle inspected prior to welding. Laminations and other linear defects must be repaired in accordance with ASTM A 20,⁴ Paragraph 9.4.
- (2) C-girders used for reinforcement of wye branches are subjected to through-thickness loading where the shell plates are welded to them. If the plate has laminations in the loaded area, lamellar tearing can occur. The likelihood of this defect occurring increases with plate thickness. Therefore, C-girder plates exceeding 1 inch in thickness should be tested ultrasonically in the area within 6 inches of the shell-to-C-girder weld. Ultrasonic examination and evaluation must comply with ASTM A 435,⁵ with the additional requirement that the 6-inch band adjacent to the weld area must undergo 100% ultrasonic examination. Any defect that shows a loss of back reflection that cannot be contained within a 1-inch-diameter circle is unacceptable. Laminations must be repaired in accordance with ASTM A 20,⁴ Paragraph 9.4. Shell-to-bar type joints in wye branches also must be examined for laminations in a similar manner.

Exposed edges of C-girders must be inspected by magnetic particle procedures after postweld heat treatment.

- (3) After postweld heat treatment, if required, all weld surfaces must be examined by the magnetic particle procedure in accordance with Appendix 6 of the ASME Code, Section VIII. If, in the opinion of the inspector, major repairs are required, stress relieving again may be required after repairs.

13.3.3 Nondestructive Examination Methods

13.3.3.1 Radiographic Examination

All radiographic examination must be in accordance with ASTM E 94⁶ and E 142,⁷ or the ASME Code, Section V, Article 2. Double film must be used.

Film must be marked with the date, owner's specification number, contract number, piece or section number, and weld number. Procedures must include an identification system that ensures traceability between the radiographic film and the weld examined, as well as clear location of weld defects. Radiographs and interpretation reports must be submitted to the owner, upon request, for permanent retention.

Acceptance standards for welded joints examined by radiography must be in accordance with Paragraph UW 51 or UW 52 of the ASME Code, Section VIII, Division 1.

All defects disclosed by radiography and determined unacceptable in accordance with the job specification must be repaired and re-radiographed.

13.3.3.2 Ultrasonic Examination

Ultrasonic examination and acceptance standards for welds other than spiral welds must be in accordance with Paragraph UW 53 and Appendix 12 of the ASME Code, Section VIII, Division 1. For spiral-welded pipe used in penstocks, ultrasonic test procedures and acceptance standards must be in accordance with Section 9 of API Standard 5L.⁸

13.3.3.3 Magnetic Particle Examination

Magnetic particle examination techniques are described in the ASME Code, Section V, Article 7 and in ASTM E 709.⁹ Acceptance criteria must conform to the requirements of Appendix 6 of the ASME Code, Section VIII, Division 1.

13.3.3.4 Liquid Penetrant Examination

Liquid penetrant examination procedures must conform to the requirements of the ASME Code, Section V, Article 6. Acceptance criteria must conform to the requirements of Appendix 6 of the ASME Code, Section VIII, Division 1.

13.3.3.5 Visual Examination

Visual examination procedures must conform to the requirements of Article 9 of the ASME Code, Section V.

The following visual indications are unacceptable:

- (1) Cracks
- (2) Undercut on surface greater than 1/32 inch deep
- (3) Lack of fusion on the surface
- (4) Incomplete penetration
- (5) Convexity of fillet-weld surface greater than 10% of the longer leg plus 3/100 inch
- (6) Concavity in groove welds
- (7) Concavity in fillet welds greater than 1/16 inch

- (8) Fillet-weld size less than indicated or greater than 1.25 times the minimum indicated fillet leg length.

All disclosed defects that are determined unacceptable in accordance with the job specification must be repaired and reexamined.

13.4 Hydrostatic Testing

13.4.1 General

Hydrostatic testing is not a mandatory requirement for penstocks. The need for hydrostatic testing is determined by the engineer. The following should be considered in determining the need for hydrostatic testing:

- (1) Site location, automatic shut-off systems, and head
- (2) Risk to the public and damage to property in the event of a failure
- (3) Structural complexity: complicated weldments, such as wye branches may warrant testing to verify their fabrication. The engineer may consider 100% radiographic examination (RT) or ultrasonic examination (UT), and magnetic-particle examination (MT) in lieu of hydrostatic testing.
- (4) The extent and type of nondestructive examination performed in the shop and field.

13.4.1.1 Component Testing

Pipe for mitered bends may be hydrostatically tested prior to making mitered cuts. Girth welds in mitered bends that have not been hydrostatically tested must be tested for defects using 100% radiographic or ultrasonic examination.

Fittings and attachments, such as nozzles, ring girders, and anchor rings, must be welded to the shell prior to hydrostatic testing. By agreement, nozzles, manholes, ring girders, and anchor rings may be fabricated into previously hydrostatically tested pipe with all welds made after hydrotesting provided they are tested using the appropriate NDE procedure.

13.4.1.2 Types of Hydrostatic Testing

If it is determined that hydrostatic testing is necessary, the location and time of testing must be established in the project specification. Hydrostatic testing can be performed either in the fabrication shop or in the field (after the penstock has been completely installed) or by a combination of both. The following must be considered in deciding whether to perform shop testing or field testing.

- (1) Hydrostatic testing of straight pipe sections can be performed using the manufacturer's hydrostatic testing machine. This method is usually less expensive than using test heads.

- (2) Shop testing does not test the system in its final installation condition.
- (3) Shop hydrostatic testing is not mandatory for penstocks subjected to 100% radiographic or ultrasonic examination, or if there is to be field hydrostatic testing of the completed system.
- (4) Manufacturer's hydrostatic testing machines may not be able to test very large-diameter and high-pressure penstocks. In these cases, welded test heads can be used or hydrostatic testing can be performed in the field after installation.
- (5) In-place testing may be more expensive, and if rupture occurs, the section must be removed and replaced.
- (6) Testing in the field on a steeply sloped system may over-pressure the lower portions and under-pressure the upper portions. When testing after installation, the penstock may have to be tested in sections to ensure an adequate test pressure at all points without over-pressure at other points.

13.4.2 Shop Hydrostatic Testing

When specified, all pipe sections must be shop tested hydrostatically to a pressure as specified in Section 13.4.2.1. Test procedures must conform to those described in AWWA Standard C-200.¹⁰ Testing must be performed prior to application of the coating and lining materials. Sections with attachments must be tested in the horizontal and upright position; test pressure must be measured at the centerline of the section. If hydrostatic testing machines are used, the end seals must not produce any significant inward pressure on the ends of the section.

The shop-tested sections must be filled slowly with water, and the pressure increased slowly until the required test pressure is reached. The test pressure must be held for 5 minutes, released, and then reapplied immediately and held while the test section is observed. There must be no leakage. Any defects in the work must be repaired, and the section must be hydrostatically tested again.

The hydrostatic tester must be equipped with a calibrated recording gage to record the test pressure and the duration of time that pressure is applied to each length of pipe. Records or charts must be available for examination at the plant.

13.4.2.1 Test Pressure

When specified, all or a specified length of pipe section must be hydrostatically tested to a pressure 1.5 times the working pressure (defined as the normal operating condition, P_{N2} , in Section 3.2.6.1) or to a pressure that will produce a stress not exceeding 80% (AWWA Standard C-200¹⁰ specifies 75%) of the minimum yield strength of the steel, whichever is less, as determined by the formula $P = 2St/D$, where P = hydrostatic test pressure, S = 0.8 times the minimum yield strength of the steel, t = wall thickness, and D = specified mean diameter of the pipe.

13.4.3 Field Hydrostatic Testing

When required by engineering or the specifications, the completed penstock must be hydrostatically tested in the field. For buried penstocks, sleeve-type couplings must not be buried until completion of the hydrotest and/or watering-up. To facilitate construction or to avoid overpressuring, the penstock may have to be tested in sections, using bulkheads for isolation. For field tests, some loads imposed on the structure may not be the same as those during operating conditions. The installation must be checked, prior to testing, to determine that anchorages and bulkheads can withstand the selected test pressure. Vacuum valves or standpipes must be installed to prevent vacuum collapse in the event of failure of a portion of the penstock under test. During the filling of the penstock, watering-up procedures described in Section 14.1 must be followed.

The test pressure must be held long enough to ensure that the entire section under test can be inspected and any leakage detected and the leakage rate measured. The leakage rate is determined by metering the water required to maintain pressure. Depending on the nature of the installation, the time required to perform the hydrostatic test may vary considerably. However, the duration of testing must be sufficient to ensure the detection of any problems. This duration may range from 1 to 24 hours. Upon reaching test pressure, readings from pressure gages must be recorded at 10- to 30-minute intervals depending on the duration of the hydrotest. Continuous surveillance for leakage must be performed during the first 30 minutes, and the test section must be visually inspected for leakage at 1-hour intervals thereafter. The hydrotest must be performed during daylight hours when practical. If any portion of the test is performed during night hours, suitable lighting for inspection must be provided.

Hydrostatic tests are accepted on the basis of the leakage rates described in the project specification. If a break or unacceptable leakage occurs during any of the testing operations, the test must be terminated and the engineer notified.

Defects disclosed during the hydrostatic test must be repaired. The section must be hydrostatically retested.

13.4.3.1 Field Test Pressure

When specified, all or a specified length of pipe section must be hydrostatically tested from a minimum pressure of 1.1 times the working pressure (P_{N2} in Section 3.2.6.1) to a maximum pressure of 1.5 times the working pressure or to a pressure that will produce a stress not exceeding 80% (AWWA Standard C-200¹⁰ specifies 75%) of the minimum yield strength of the steel, whichever is less, as determined by the formula $P = 2St/D$, where P = hydrostatic test pressure, S = 0.8 times the minimum yield strength of the steel, t = wall thickness and D = specified mean diameter of the pipe.

13.4.4 Hydrostatic Test Inspection Verification

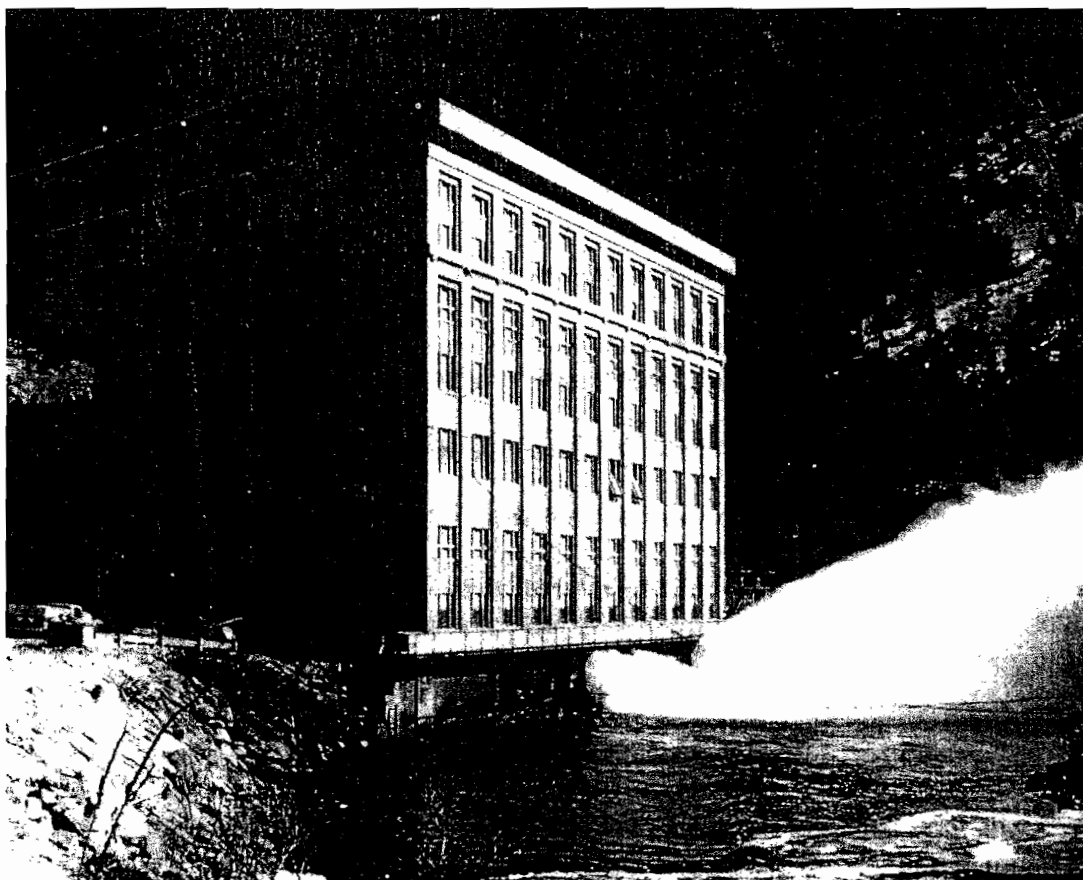
For both shop and field tests, the inspector must observe the testing and check the procedures, verifying the following:

- (1) The test medium, pressures, and duration of test are as specified for the penstock sections.
- (2) The calibrated pressure measuring devices provide for the recording of pertinent data, i.e., pressure, time, temperature, etc., as specified.
- (3) The gages used to monitor the pressure testing show evidence of current calibration traceable to the National Bureau of Standards.
- (4) All seams, nozzles, and manholes are inspected during the test for leaks, and findings are recorded.
- (5) The penstock section is drained, dried, and preserved (if specified) after completion of testing.

References*

1. Welder and Welding Procedure Qualification Records. ASME Code Forms, QW 482, 483, 484. ASME, New York, NY.
2. "Recommended Practice for Nondestructive Testing: Personnel Qualification and Certification". ASNT SNT-TC-1A. American Society of Nondestructive Testing, Columbus, OH.
3. Standard and Guide for Qualification and Certification of Welding. AWS QC1. American Welding Society, Miami, FL.
4. Standard Specification for General Requirements for Steel Plates for Pressure Vessels. ASTM A 20. ASTM, Philadelphia, PA.
5. Straight Beam Ultrasonic Examination of Steel Plates for Pressure Vessels. ASTM A 435. ASTM, Philadelphia, PA.
6. Recommended Practice for Radiographic Testing. ASTM E 94. ASTM, Philadelphia, PA.
7. Standard Method for Controlling Quality of Radiographic Testing. ASTM E 142. ASTM, Philadelphia, PA.
8. Standard for Line Pipe. API Standard 5L, American Petroleum Institute, Washington, DC.
9. Standard Practice for Magnetic Particle Examination. ASTM E 709. ASTM, Philadelphia, PA.
10. Steel Water Pipe 6 Inches and Larger. AWWA Standard C-200. AWWA, Denver, CO.

* The most current version of a standard, code, or specification should be used for reference.



14.1 Watering-Up

14.1.1 Initial Watering-Up

The initial watering-up of a penstock generally is the most critical watering-up operation because it is the first time that all the penstock components have been fully assembled and tested together under operational conditions. This initial watering-up must be performed under slow rates and controlled conditions. Where there have been long outages, with the penstock in a dewatered condition, watering-up also must be performed under slow rates and controlled conditions.

14.1.1.1 Start-Up Procedures

Prior to completion of the project, detailed project start-up procedures must be developed and approved. These procedures include, but are not limited to, powerhouse mechanical and electrical equipment testing (micro and macro), water conveyance system watering and

dewatering, and operational testing. Prior to watering-up, the emergency closure system must be completely operational, and an emergency contingency plan must be prepared. The scope of the work is dependent upon the scope of the project, i.e., new project or penstock replacement.

In addition, a detailed inspection procedure, the Civil Features Monitoring Program (CFMP), must be prepared and instituted. This program must provide a checklist of items to be monitored and inspected, inspection intervals, acceptance criteria, and criteria for unacceptable conditions.

14.1.1.2 Prior to Watering-Up

Following are examples of work to be completed prior to watering-up the system.

- (1) A baseline survey must be taken of all monuments to be used for short- and long-term monitoring, i.e., anchor blocks, tunnel portals, and penstock supports.
- (2) Baseline data must be taken using inclinometers, piezometers, extensometers, etc.
- (3) Emergency Closure Systems (ECS) must be complete, tested, and fully operational.
- (4) The interior of the penstock system must be inspected by a walk-down to ensure that all construction debris has been removed from the interior and that all personnel have vacated the interior.
- (5) The exterior of the system must be inspected by a walk-down to ensure that all manholes and valves have been closed and that air valves are open.
- (6) Clearances with the start-up engineer or powerhouse foreman must be signed off (verifying that the system is ready for watering-up).

14.1.1.3 During Watering-Up

Following are examples of work to be completed or performed during the initial watering-up of the system.

- (1) A low filling rate must be initiated to ensure full control of the system. An operator must be stationed at the intake during the entire watering-up process. Should a problem develop that would require rapid dewatering, a slow watering-up might prevent a potential disaster. A suggested maximum range for this initial filling rate for inclined penstocks and shafts is 20 to 50 vertical feet per hour, which usually allows adequate control for filling operations. For horizontal tunnel liners, this filling rate may be too high.
- (2) A system must be set up to provide communication between the inspection crew, the start-up engineer, and all operators.
- (3) The filling operation must be performed only during daylight hours. This is done for two reasons. The first is to provide for the safety of the personnel involved in the inspection

program. The second is to provide better visibility so that problem areas can be identified and acted upon quickly. Intermediate hold points should be considered to allow for a thorough inspection and evaluation. If practical, at least 4 to 6 hours of daylight should be available after completion of filling. During the night hours, intake head gates must be closed for safety reasons. The by-pass gates may be closed or left open.

- (4) After the penstock has been filled completely, it must be allowed to "soak" for 24 hours prior to any operational testing of the powerhouse equipment. A periodic inspection must be made during this period to identify any problems. Additional personnel must be on site or on call in the event that problems do develop. (Note: During the initial filling process, items such as sleeve couplings, valves, and other bolted connections may require re-torquing to stop leaks). Torquing must be done according to the manufacturer's recommendations (See Section 16).
- (5) For buried penstocks with sleeve-type couplings, bell holes should be left open for inspection of leaks and re-torquing of couplings.
- (6) For buried penstocks that have been fully pressurized, close the intake shutoff gate (ISG) or penstock shutoff valve (PSV) and monitor for any pressure drops to verify that there are no leaks. This may require water columns located at the ISG or PSV to monitor water loss.
- (7) After completion of all CFMP inspection procedures, the start-up engineer's start-up procedure must be signed off, indicating that the penstock is complete and ready for operational testing.

14.1.1.4 Inspection

Following are examples of inspection checks that must be made and logged.

- (1) Read floatwell water surface elevations.
- (2) Inspect sleeve coupling or expansion joints.
- (3) Inspect equipment, such as valves, manholes, and air/vacuum valves, for leaks.
- (4) Read inclinometer, piezometer, extensometer, and other monitoring systems.
- (5) Check for short-term settlement in foundations.
- (6) Check for cracking or distortion of the penstock support system.
- (7) Check for any leaks (exposed penstock) or wet soil, any increase in ground water, and any increase in weir or piezometer readings (for liner or buried penstock).
- (8) Compare movements or deflections to analysis.
- (9) Survey elevations.

14.1.2 Maximum Dewatering Rates

For inclined penstocks, the dewatering rate must not exceed 50% of the controlling air valve flow capacity or 100 vertical feet per hour, whichever is less.

For tunnels and horizontal tunnel liners, the maximum dewatering rate must not exceed from 10 to 60 vertical feet per hour. (Note: Depending on the rock formation (bulk modulus) and the liner design, lower rates may need to be considered.) If a differential pressure monitoring system is installed across the tunnel liner, these dewatering rates may be exceeded, provided an adequate factor of safety against buckling is maintained.

14.1.3 Subsequent Watering-Up

Subsequent penstock watering-up generally is not as critical as the initial watering-up operation. However, subsequent watering-up also must be performed under controlled conditions.

The following must be considered for subsequent watering-up operations.

- (1) The filling rate can be much higher than for the initial watering-up. A reasonable rate for a subsequent watering-up is 100 vertical feet per hour (penstock only). Again, the maximum dewatering rate must not exceed 50% of the controlling air valve flow capacities.
- (2) The initial and subsequent watering-up reports must be reviewed for the occurrence of any unusual events or circumstances.
- (3) A walk-down of the interior and exterior of the system must be conducted to ensure that all manholes have been closed, air valves/vacuums are operational and open, and drain valves are closed.
- (4) After the system has been fully pressurized but prior to being returned to operation, the entire system must be walked down, and any abnormal items conditions must be noted and corrected if necessary.
- (5) Clearance with the powerhouse foreman, start-up engineer, or other designated person must be signed off to certify that the system is ready to return to operation.

14.2 Operational Testing

14.2.1 General

As discussed in Section 9.1.1.1, one phase of the start-up procedure involves operational testing. This consists of performing load-rejection and load-acceptance testing of the turbine, pump, or pump/turbine. Physical testing must be conducted for all penstock operational conditions (normal condition) that are likely to occur. Pressures transmitted to the penstock must be compared against the penstock design criteria. For example, if the penstock is designed for a normal operational pressure (load rejection) of 10% over static pressure, then the

turbine/generator unit must be fully tested to ensure that the pressures in the penstock do not exceed 10% over static pressure.

14.2.1.1 Head Loss

If flow-measuring devices have been installed on or near the turbine/generator, penstock or system head loss can be measured directly; otherwise, conventional pressure gages can be used. The results then can be compared against the original design criteria.

Flow-measuring devices, methods, and systems in common use include:

(1) Acoustic flowmeters

Acoustic (or ultrasonic) flowmeters measure the water velocity by the interaction of an interrogating sound wave and the water. The most common type for penstock flows is based on the transit time method. This method is based on the principle that an acoustic pulse sent upstream travels slower than one sent downstream. By measuring the time difference between the upstream and downstream pulses, the average velocity of the water crossing the path of the pulse can be determined vectorially. There are several methods for using transit time ultrasonic flowmeters. These include: (1) permanently mounted "wetted" systems, in which the transducers are inserted into the pipe wall and come in contact with the water, (2) strap-on systems in which the transducers are temporarily mounted on the outside of the pipe, and (3) path systems for the acoustic wave (single and multi-path along the pipe diameter, or along chordal paths).

Under proper conditions, accuracies better than 1% can be obtained, particularly for multi-path systems. Under the proper conditions, typical single-path strap-on systems have claimed accuracies of between 1% and 3% (without calibrating in place against another flow standard). However, larger variations have been observed, depending on the installation. These systems have limitation in that: (1) they require a good flow profile for accurate measurements, typically at least 10 pipe diameters of undisturbed, straight pipe upstream of the meter, (2) the pipe inside diameter must be accurately determined, and (3) bubbles or other particles in the fluid can cause problems (due to reflection or scattering of the signal).

(2) Gibson method

This method, also called the pressure-time method, is based on Newton's second law of motion, in which the pressure forces acting on a mass of water are related to the change in momentum of that water as it is decelerated. For penstock flow measurements, the difference in pressure between two cross sections is measured as a downstream valve or gate is closed. By integrating the pressure difference over the time period, and knowing the appropriate pipe geometry data, the steady-state flow rate prior to shut-off can be determined. Modern instrumentation and computer numerical methods can be used to simplify the data collection and analysis. Accuracies of about 1% are generally assumed for the method. Limitations include the need for properly located pressure taps, the need

to shut the flow down to get the measurement (not an "on-line" method), and the somewhat complex data reduction required.

(3) Current meters

The current meter measures a point velocity using a propeller type meter whose rotational speed is proportional to the water velocity. A large number of current meters usually must be employed to obtain a good overall volumetric flow rate. Under proper conditions, accuracies of 1% can be obtained. Limitations include the need to mount a large number of current meters inside the penstock (if not an open canal situation) and the need for a good flow profile.

(4) Salt velocity method

The salt velocity method works on the principle that salt in solution (brine) increases the conductivity of water, and that the salt introduced at one point will travel as a "slug" for some distance down the penstock. The passage of this slug of brine past electrodes at two locations downstream of injection can be recorded, and the time the slug takes to travel the known distance between electrodes can be determined. The distance and time of travel are used to calculate the average flow velocity, which when multiplied by the penstock cross-sectional area, results in the volumetric flow rate. Under proper conditions, accuracies of 1% can be obtained. Limitations include the need to mount electrodes at two locations inside the penstock and a brine injection system at another location, and the need for accurate dimensional measurements between electrode stations.

(5) Dye dilution flow method

The dye dilution flow method directly measures the volumetric flow rate directly by utilizing the principle of conservation of mass of a tracer dye injected into the flowing fluid. Rhodamine WT dye typically is used as the tracer. A known concentration of dye is injected at a steady rate at one location, and a fluorometer is used to determine the mixed concentration at a second location sufficiently downstream to ensure complete mixing. The ratio of concentrations at the two locations and the measured injection rate are used to calculate the water flow rate. Under proper conditions, accuracies between 1% and 2% can be obtained. Limitations include the need for a sufficient distance between the injection and sampling locations (at least 100 penstock diameters is recommended) and the need for great care and accuracy in the making of dye standards for fluorometer calibrations.

(6) Pitot tubes

The Pitot tube measures point velocities by determining the difference between the total and static pressures at a point in the flow. A traverse must be done across the cross-section, typically along two diameters, to obtain a sufficient number of velocity points to calculate a good average. Twenty points along each diameter typically are used. The flow rate is then calculated by multiplying the average velocity by the cross-sectional area. A U-tube manometer typically is used to measure the differential pressure produced by the Pitot tube. Under proper conditions and with a calibrated Pitot tube, accuracies of about

2% can be obtained. Limitations include the difficulties in traversing across a large pipe diameter (Pitot tube support problems, vibrations, etc.), the need to obtain a good measurement of the penstock inside diameter, and the need for a good flow profile (at least 10 pipe diameters of undisturbed, straight pipe upstream of the Pitot tube).

(7) Winter-Kennedy method

The Winter-Kennedy method utilizes the differential pressure between two suitably located taps in the turbine spiral case to determine the flow rate. The flow rate is approximately proportional to the square root of the differential pressure. Winter-Kennedy flow measurements typically are used for index testing, where absolute flow measurements are not needed. These relative flow rates can be used to determine the shape of the turbine efficiency curve relative to some design point. For absolute flow determination, the constant of proportionality between flow and differential pressure, as well as the actual exponent for the differential pressure (it may not be an exact square root relationship) must be determined by calibration against another known flow method. The main limitation of this method is that accurate absolute flow measurements can be obtained only if a calibration is done using another known flow method.

(8) Thermodynamic method

The thermodynamic method directly measures turbine efficiency using the principles of conservation of energy. Turbine losses are manifested by a rise in the temperature of the water passing through the turbine. By appropriate temperature and pressure measurements at the inlet and outlet of the turbine, and by utilizing an energy balance analysis, turbine efficiency can be calculated. The water flow rate then can be calculated from the turbine efficiency by using separate measurements of net head and generator power output and assuming that the generator efficiency is known. Accuracies of better than 0.5% are claimed for the turbine efficiency. Flow rate accuracy will depend on the efficiency accuracy, as well as on the accuracy of the net head, generator power, and generator efficiency values. Limitations include the need to very accurately determine temperature differences between the turbine outlet and an inlet test chamber, and the relatively complex instrumentation system required.

(9) Weir method

The weir method utilizes a V-notch, trapezoidal, or rectangular sharp-crested plate interposed into a free surface flow. The measurement of the water head above the weir crest (measured upstream of the weir) can be used to calculate the flow rate. The weir generally is placed in the tailrace, downstream of the turbine. Under proper conditions, accuracies of about 2% can be obtained. Limitations include the need for a properly constructed and placed weir, and the need for smooth flow (without air bubbles or flow disturbances) in the approach canal.

(10) Magnetic flow meters

Magnetic flow meters utilize the principle of electromagnetic induction by a conductor moving through a magnetic field (Faraday's law). Water flowing through a magnetic field induces a voltage which is proportional to the velocity of the fluid. Single-point insertion types are available, as well as full pipe systems. Single-point insertion types must be traversed across the pipe diameter to obtain the overall flow rate (similar to a Pitot tube). For the insertion type, accuracies of about 1.5% can be obtained under proper conditions. Limitations (similar to those for the Pitot tube) include: difficulties in traversing across a large pipe diameter (support problems, vibrations, etc.), the need to obtain a good measurement of the penstock inside diameter, and the need for a good flow profile (at least 10 pipe diameters of undisturbed, straight pipe upstream of the meter). For the full pipe system, accuracies better than 1% can be obtained.

With the exception of the dye dilution method, these devices and methods require approximately 10 to 20 pipe diameters of undisturbed flow (with no change in diameter or bends). The selected device or method should be consistent with the required accuracy.

14.2.1.2 Operational Tests

Following are typical hydraulic tests performed during operational testing.

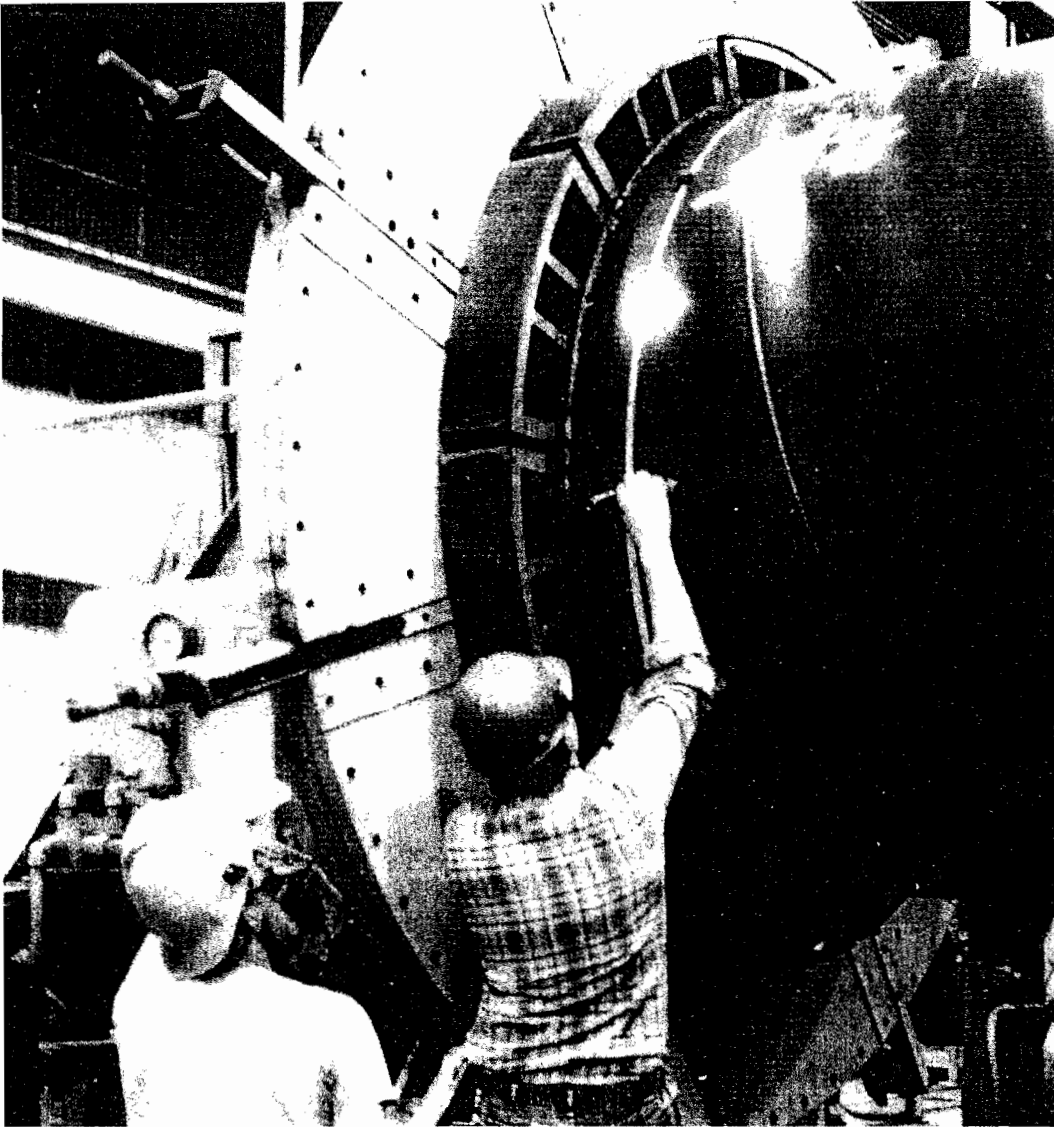
- (1) Speed—no load rejection (usually a critical pressure rise case for both Francis and Impulse machines)
- (2) 25% rated load rejection and load acceptance
- (3) 50% rated load rejection (usually a critical pressure rise case for a Francis machine) and load acceptance
- (4) 75% rated load rejection and load acceptance
- (5) 100% rated load rejection and load acceptance
- (6) 100% gate or needle rejection and load acceptance (usually a critical pressure drop case for an Impulse machine).

14.2.2 Instrumentation

To monitor the pressure rises generated during the testing described in Section 14.2.1.2, sufficient and accurate instrumentation must be installed. The monitored pressure data is used to ensure that the pressure rises and/or drops generated by unit operation are in compliance with the penstock design criteria. Acceptance criteria and allowable deviations must be established prior to the testing program. (Note: Pressure-rise and/or drop data must be reviewed and evaluated at the time of the test and not taken merely for the purpose of later evaluation or future reference.)

The following must be considered in using instrumentation equipment to monitor pressure.

- (1) Strain gages, if necessary, must be located at the most critical areas of the penstock system. See Section 13.4 for information on hydrostatic testing.
- (2) Bolt-on flow measuring devices can be used for unit flow and corresponding penstock headlosses.
- (3) Pressure transducers and analog/digital pressure charts should be used for measuring pressures and water surfaces (pressure gages do not provide sufficient speed response and accuracy). The type of pressure device should be compatible with the speed of the transient pressure wave. Analog pressure devices are adequate for surge-tank fluctuations and slow-moving valves. However, pressure transducers should be used for cases involving wicket gate closures, PRV operations, needle operations, etc. (Note: Long lengths of tubing should not be used for transient conditions because they can cause amplification or attenuation of data. They should be used only for steady-state conditions.)



15.1 Certified Mill Test Report

The penstock documentation package must contain a copy of the Certified Mill Test Report, which includes records for the following:

- (1) Steel plate, forgings, structural shapes, pipe, and flanges
- (2) Welding consumables

- (3) Connecting devices such as bolts and gaskets
- (4) Valves, expansion joints, and sleeve-type couplings
- (5) Other applicable items.

Suppliers of the above-listed items must provide records for each item for inclusion in the Certified Mill Test Report. Certified Mill Test Report records must be traceable to heat and lot numbers of material used, and must comply with the requirements outlined in the material specifications for the job.

15.2 Personnel Certification

15.2.1 Designers

Both in-house and outside contracted designers performing calculations on penstock systems, and personnel checking those calculations must have sufficient formal education, experience, and training to perform the calculations. For all outside contracted design, personnel qualifications should be left to the discretion of the design supplier, with the approval of the purchaser. For outside contracted design, current resumes of personnel working on the design team must be included for review by the purchaser, if required. All final design packages, including job and material specifications, however, must be signed off and certified by a registered professional engineer, registered in the state in which the project is located.

15.2.2 Welders and Welding Operators

All welders and welding operators performing welding on pressure or load carrying components, both in the shop and field, must be qualified by testing to the types of welding being performed on the penstock. Qualifications of welders and welding operators must be specified by the fabricator/installer to the satisfaction of the purchaser. For an example of qualification tests, see the ASME Code,¹ Section IX, QW-484. All welder qualification records must be maintained by the fabricator/installer of the penstock system, and turned over to the purchaser upon completion of the project.

15.2.3 Inspectors

All personnel performing nondestructive examination (NDE) on structural welds must be certified to Level 2 or 3 in the NDE method being applied. Personnel must work under the direct supervision of a welding inspector certified to AWS-QC1.² One method of NDE personnel qualification and certification currently used is ASNT SNT-TC-1A, "Recommended Practice for Nondestructive Testing: Personnel Qualification and Certification."³ NDE personnel must have sufficient experience and training, and must be certified by written and/or practical testing. Documentation of inspector and NDE personnel certification must be retained throughout fabrication and installation of the penstock system, and then turned over to the purchaser upon completion of the project.

15.3 Process Certification

15.3.1 Welding Procedure Specification (WPS)

Each fabricator and assembler must list the parameters applicable to weldments used in the construction of the penstock system. These parameters must be listed in a document known as the Welding Procedure Specification (WPS). The WPS must list the type of material to be joined, filler metals, the range of preheat and postweld heat treatments, the thickness range, and all other essential and nonessential variables required for each welding process. The WPS provides welding guidelines to permit welding in accordance with the project specifications, and must be readily available for reference by the welders and inspectors. Upon completion of the project, the applicable Welding Procedure Specifications must be retained in the permanent documentation files.

15.3.2 Procedure Qualification Record (PQR)

Each fabricator and installer must qualify the WPS by the welding of test coupons and the testing of specimens. The welding data and test results are recorded in the Procedure Qualification Record (PQR), which documents the essential variables of the specific welding process and the test results. The PQR must list what was used in qualifying the WPS and the test results, and also provide proof of the weldability of the variables described in the WPS. These documents must be certified by the fabricator or installer and must be available for examination by the inspector. Upon completion of the project, the applicable PQRs must be retained in the permanent documentation files.

15.3.3 Nondestructive Examination (NDE) Procedures

When NDE is required, the fabricator/installer must designate in writing suitable examination procedures, such as specified in the ASME Code, Section V. The fabricator/installer must ensure that the examination is performed by qualified personnel in accordance with the requirements of the design specifications. NDE procedures must comply with all applicable requirements to the satisfaction of the purchaser. NDE procedure documentation must be readily available to all NDE personnel for their reference and retained in the final documentation package upon completion of work.

15.4 Documentation Records

15.4.1 Coatings Documentation

The project technical specifications contain requirements for penstock coatings. Section 10 of this manual specifies requirements for penstock coatings. The following documents must be retained pending completion of the job:

- (1) List of materials

A list of coating materials used in the work, with specific products identified by manufacturer and catalog number (for each coating).

(2) Written coating procedures

Written procedures for storage, handling, surface preparation, environmental control, application, touch-up and repair, curing, and inspection of the coating system.

(3) Quality control inspection reports

Reports of quality control inspections performed in accordance with the specifications.

15.4.2 Shop and Installation Drawings

A copy of all shop fabrication and field installation record drawings must be included in the documentation records. These record drawings must include a designation of the material identification (heat and slab numbers) for each piece of material and the physical location of each item.

15.4.3 NDE Records

The documentation records must include checklists showing all nondestructive examinations along with traceability to each area examined and the identity of the individual making the examination.

15.4.4 Owner's Drawings

A copy of applicable owner's as-built drawings must be included in the documentation records.

15.4.5 Operations and Maintenance (O & M) Manuals

Fabricators, installers, suppliers, and/or constructors should provide the owner with O & M manuals for pertinent parts, components, and systems within their scope, such that the owner can operate and maintain the penstock system as originally designed.

15.4.6 Document Retrieval

All of the above documents must be logged and indexed by the owner to facilitate identification and retrieval.

15.5 Data Report

All completed steel penstocks must be documented by means of a Data Report, completed by the fabricator. The Data Report must include a description of the penstock, the material used, and the applicable design parameters, including design pressures, to the extent necessary to provide adequate identification. Each Data Report should include a Design Certification Statement, a Fabrication Certification Statement, and an Assembly Certification Statement. Figure 15-1 shows a sample Data Report.

DATA REPORT

Manufactured and certified by _____
(Name and Address)

Manufactured for _____
(Name and Address of Purchaser)

Location of installation _____
(Name and Address)

Type: _____
Horiz. or Sloping Manufacturer's Serial No. Drawing No. Year Built

Shell: _____
Mat'l Spec. No. Tensile Strength Nom. Thickness (in.) Min. Design Thickness (in.) O.D. ID (ft. & in.) Length (overall) (ft. & in.)

Seams: _____
(Long.) HT (Temp.) RT (Eff. %) Girth HT (Temp.) RT No. of Courses

Design Pressure: _____ at _____ Min. Pressure-test Temp _____ Hydro Test Pressure _____
(psi) (deg.F) (deg.F) (psi)

Nozzles, inspection and safety valve openings:

Purpose (inlet, outlet, drain, etc.)	Quantity	Dis. or Size	Type	How Attached	Material	Thickness	Reinforcement Material	Location

Supports: Ring Girders _____ Saddles _____ Legs _____ Other _____ Attached _____
(Yes/No) (Quantity) (Quantity) (Describe) (Where & How)

Remarks: _____

CERTIFICATION OF DESIGN

Design specification certified by _____ Prof. Eng. state _____ Reg. No. _____
Design report certified by _____ Prof. Eng. state _____ Reg. No. _____

CERTIFICATE OF SHOP COMPLIANCE

I, the undersigned, employed by _____
of _____ have inspected the component described in this data report on _____ and state that to the best of my knowledge and belief the fabricator has constructed this component in accordance with the Shop Fabrication Specification. By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this data report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____
(inspector)

CERTIFICATE OF FIELD ASSEMBLY COMPLIANCE

I, the undersigned, employed by _____
of _____ have inspected the component described in this data report on _____ and state that to the best of my knowledge and belief the assembler has constructed this component in accordance with the Field Construction Specification. By signing this certificate neither the inspector nor his employer makes any warranty, expressed or implied, concerning the component described in this data report. Furthermore, neither the inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ Signed _____
(inspector)

Figure 15-1 Sample Data Report

15.6 Nonconformance Report

During fabrication, installation, and start-up testing, any deviation from specifications and/or design drawings, including the specific final disposition of the deviation, must be documented in a Nonconformance Report. Nonconformance Reports must be retained in the final documentation package and turned over to the owner upon completion of the work.

References*

1. Manufacturer's Record of Welder or Welding Operator's Qualification Tests. ASME Boiler and Pressure Vessel Code, Section IX, QW-484. ASME, New York, NY.
2. Qualification and Certification of Welding Inspectors. AWS-QC1. American Welding Society, Miami, FL.
3. "Recommended Practice for Nondestructive Testing: Personnel Qualification and Certification". ASNT SNT-TC-1A. American Society of Nondestructive Testing, Columbus, OH.

* The most current version of a standard, code, or specification should be used for reference.



16.1 In-Service Inspection

A comprehensive, well-defined, well-documented in-service inspection program is one means of monitoring the condition of a steel penstock.

16.1.1 Items Inspected

Items that a comprehensive in-service inspection program must monitor include, but are not limited to:

- (1) Unusual movement

- (2) Excessive vibration
- (3) Leakage
- (4) Penstock age
- (5) Penstock shell condition (interior and exterior)
- (6) Welds
- (7) Bolts and rivets
- (8) Expansion joints and sleeve-type couplings
- (9) Air valves
- (10) Valves or other water control systems
- (11) Man holes and other penetrations
- (12) Anchor blocks and supports
- (13) Coatings (interior and exterior)
- (14) Instrumentation (as appropriate).

16.1.2 Penstock Shell Inspections

Penstock shell inspections must include, but are not limited to, the following.

- (1) The outer surface of the penstock, where exposed, must be examined for excessive leakage, corrosion, pitting, deterioration, and coating deficiencies. Inspection for leakage must be done while the penstock is watered.
- (2) Similarly, the inner surface of the penstock must be examined for excessive corrosion, pitting, and deterioration, as well as for coating deficiencies as applicable. One disadvantage of performing a visual interior inspection is that it is often dangerous because of limited access and ventilation problems. Also, the penstock must be dewatered. However, this type of inspection usually can be scheduled during normal plant outages.
- (3) Penstock shell thickness measurements must be taken and recorded for selected locations along the penstock. A history of these readings gives an indication of the expected yearly decrease in shell thickness. These readings can be taken easily on the outside of an exposed penstock without the need for dewatering. For a buried penstock, shell thickness readings can be taken from the inside during dewatered periods. These readings must be compared with the minimum acceptable plate thickness specified in the design to determine if any corrective action is needed.

- (4) A representative portion of all structural welding performed on the inside and outside of the penstock must be examined visually for signs of significant rusting, pitting, or other structural defects. If justified by visual inspection, structural weld integrity then must be evaluated by NDE methods.
- (5) The condition and integrity of selected bolted/riveted joints must be examined. These connections must be secure and free from excessive rust and deterioration.

16.1.3 Types of Inspection

Several types of inspection techniques are available to examine various aspects of penstocks in the field. Some of the more common inspection techniques are discussed below.

- (1) Visual inspection can be performed on exposed penstock surfaces and on the ground above buried penstocks when there is adequate access. Visual inspection is particularly effective for detecting surface defects and potential subsurface defects, which can be examined subsequently by other methods. Visual inspection must include, but is not limited to, the following.
 - (a) Inspect rivet heads on or around leaking circumferential and longitudinal straps.
 - (b) Inspect for rust blisters, which may indicate pinhole leaks from pitting.
 - (c) Inspect for linear indications, which may reveal cracks in the penstock shell.
 - (d) Inspect for dimpling of the shell around the horns of concrete saddles. Dimpling may indicate overstressing in these areas.
 - (e) Check that the horns of concrete saddles are not broken.
 - (f) Inspect the ground above a buried penstock to ensure that no trees or brush are established; their roots may disturb the pipe and backfill. Also, green growth, such as berry bushes, may indicate leaks in the penstock.
- (2) Ultrasonic devices can be used to determine shell thickness from the inside or outside surface of penstocks as well as to determine rivet integrity by picking up any laminations or other defects that may be present. Other NDE methods, such as MT, PT, and RT, can be used to detect potential problems on the steel plate surface and on older welded joints. Some possible applications of NDE methods include:
 - (a) Use of a B-scan system for a full interior surface profile while the penstock is dewatered, or for an exterior profile while the penstock is watered.
 - (b) Use of standard ultrasonic testing for localized spot readings as part of a grid profile.
 - (c) Use of MT or PT to follow up on surface indications that were detected by visual inspection.

- (d) Use of RT to evaluate interior pitting or plate/weld cracking and corrosion between steel shell and concrete saddles. The film must be placed between the steel and concrete, with the source on the other side of the penstock. The penstock must be dewatered; otherwise exposure time becomes excessive.
- (3) A relatively new technology in penstock inspection is the use of remote-controlled video equipment, or penstock-inspecting robots. These devices can perform inspections rapidly and safely inside slippery, steeply inclined penstocks. Internal paint, rust, and erosion conditions are monitored on an external CRT linked to a camera mounted on the inspection unit. The device can measure shell and paint thickness over the full 360-degree inner surface of the penstock.
- (4) Plate thickness can be measured accurately using drill borings or cut-out sections of the penstock. However, repair of the drill holes and cut-out sections requires subsequent material testing and analysis. For this reason, drill borings and cut-out sections rarely are used for shell thickness determination only.

16.1.4 Inspection Frequency

The frequency of in-service penstock inspections must be tailored to each individual hydroelectric facility, and take into account the following factors:

- (1) Accessibility for inspection
- (2) Consequences of a penstock failure
- (3) Frequency of penstock watering-up and dewatering
- (4) Climatological and environmental conditions
- (5) pH of the water flowing through the penstock
- (6) Amount of sediment in the water flowing through the penstock
- (7) Penstock age
- (8) Method of penstock manufacture, i.e., riveted, welded, mechanically coupled, etc.
- (9) Past history of penstock failure, if any.

Once these and any other pertinent factors have been addressed, the inspection frequency can be established. When practical, consideration should be given to visual observation of exposed penstocks through a monthly walkdown by operations personnel. If this is not practical because of excessive length, rough terrain, etc., then the walkdown must be performed at least once a year. The inner surface of exposed and buried penstocks must be visually examined at least every 2 to 3 years. Shell thickness readings must be taken at least every 5 to 7 years.

16.1.5 Multiple Facility Assessment

Owners of multiple, older hydroelectric facilities may need to develop a program to determine the priority of repair or replacement. The following must be considered when developing this type of penstock inspection program.

- (1) Development of a comprehensive database that includes such information as facility age, head, flow, penstock diameters, thicknesses, type of construction, strengths of steels, support types, coating types, transient operating pressures, wicket gate/needle timings, inspection dates, previous test dates, inspection record reference, records of previous problems and repairs, drawing and design references, and inclinometer data references.
- (2) Preparation of a complete and comprehensive list of all penstock drawings and calculations. The information must be easily accessible.
- (3) A vulnerability assessment in which the priority or sequence of inspections is based on:
(1) loss of property/environmental damage, (2) the risk to public safety, (3) penstock age, (4) the penstock maximum head, and (5) the energy value of the facility.
- (4) Testing that includes a determination of the pressures exerted on the penstock during operation of the turbine/generator. These tests must include a determination of the wet and dry timings of the wicket gates/needles for each penstock. In addition, load rejection tests must be performed to establish the penstock pressure rises and drops.
- (5) Development of inspection criteria (as discussed earlier in this section), and a systematic and comprehensive inspection of each penstock consistent with the prioritization schedule. Inspections must include interior inspections (lining condition and extent of corrosion) and exterior inspections (condition of the coatings, supports, and appurtenances). A geologic evaluation also must be performed. These inspections also may include mechanical testing of penstock, support, and foundation materials to obtain data for the data base and for evaluation of the facility.
- (6) Using the information in items 1 through 5 above, a complete structural (including appurtenances) and coating/lining assessment of each penstock can be performed. Improvements, modifications, or replacements that are required as a result of these evaluations must be budgeted for and made according to the prioritization schedule.

16.1.6 In-Service Inspection Records

To establish an accurate representation of the penstock condition at a given hydroelectric facility, the in-service inspection program must be well-documented and implemented by competent, well-trained personnel.

A log must be set up at the plant to record the date, type of inspection performed, and results of all inspections performed on penstocks. Inspection results should be forwarded to the engineering staff or other appropriate personnel for review and evaluation. These records must be retained for future reference. A documented chronology of inspections, results, evaluations,

and repairs will help identify the development of any adverse trends, and is essential for the proper maintenance of safe penstocks.

16.2 Prevention of Leakage

16.2.1 Prevention During Design and Construction

There is no way to guarantee that a penstock in service will not leak; however, there are steps that can be taken to minimize the frequency or amount of any leakage.

During the preparation of designs, calculations, and sketches, the designer must consider the possibility of leakage. Penstock appurtenances such as valves, expansion joint packing, bolted sleeve-type couplings, flanged connections, manholes, and air valves typically can experience in-service leakage, and must be given careful consideration by the designer.

The penstock fabricator and installer, both in the shop and field, must carefully install all penstock components in strict compliance with the design drawings and specifications. Particular care and attention must be given to the proper assembly of all connected parts such as bolted and welded connections. All valve and expansion joint packing material must be installed in accordance with manufacturers' recommendations.

Careful attention to the design and construction of penstock assemblies will reduce significantly the degree of any leakage that occurs when the penstock system is placed into service.

For more detailed information on new installations, see Section 14.1.1.

16.2.2 In-Service Leakage

16.2.2.1 General

Once a penstock is placed in service, all exposed portions must be inspected visually for leakage at least monthly, if practical (see Section 16.1.4). Preventative maintenance, such as bolt tightening, must be performed as needed to help prevent leakage. Usually this can be done at a minimal cost.

Whenever leakage is identified, it must be evaluated immediately by the appropriate personnel and repaired at the first practical opportunity (based on the severity of leakage).

One method in current use for the temporary repair of leaking penstocks is the pressurized injection of an expandable, resinous material to seal the opening at which the leakage occurs. This method is relatively inexpensive and usually does not require a plant shut-down during the repair process.

16.2.2.2 Mechanical Expansion Joints

Under normal operating conditions, mechanical expansion joints require very little maintenance. Packing chamber leakage may be caused by loosening of the bolts in the end ring or by drying out of the packing material located near the surfaces exposed to the sun. If this occurs, the bolts must be retightened using the recommended torque settings furnished by the manufacturer. If further tightening does not stop the leakage, it may be necessary to replace the packing in the packing chamber. The following procedure may be used:

- (1) After dewatering the penstock, loosen and remove all bolts.
- (2) Using a mechanical means, such as a come-a-long attached to the end-ring section at three places equidistant around the circumference, pull the end-ring section out of the packing chamber.
- (3) After the packing chamber area has been exposed, fish out the alternate rubber and lubricating rings. High-pressure jet streams are effective in removing gaskets on older couplings.
- (4) Install new packing rings according to the manufacturer's instructions, making sure to alternate the rubber with the flax rings, and successively alternate the splices 120 degrees for each material. There are usually five rubber rings and four flax rings. The rubber rings are always the end pieces that come into contact with the metal parts.
- (5) After the packing rings have been properly installed, push the end ring into place and follow the installation instructions to complete maintenance.

16.2.2.3 Bolted Sleeve-Type Couplings (BSTC)

With the penstock pressurized, axial movement or deflection may cause bolted sleeve-type couplings to be slightly displaced from their original position, and leakage may occur. The following leakage repair procedure is recommended.

- (1) Monitor the position of the end rings and sleeve in relation to the outside circumference gap between the coupling parts and the penstock shell.
- (2) Where gaps that exceed the spacing allowed during original installation are observed, mark these areas to properly identify the space and gap.
- (3) At areas where the gaps exceed the spacing allowed by the installation instructions, loosen the nuts a few turns to allow the gasket to creep.
- (4) Where minimum gaps are observed (even iron-to-iron) commence tightening in these areas to force the spacing to increase, thus diminishing spacing where it is excessive. The bolt torquing action will cause cold flow of the rubber gaskets to concentricize the gaps.

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- (5) After the gaps have been made uniform around the entire circumference, follow the manufacturer's recommended torquing procedure to again seat all the parts.

If leakage continues after following this procedure, it may be necessary to replace the BSTC gaskets after dewatering the penstock. This is accomplished by the following procedure:

- (1) Remove the bolts.
- (2) Separate the end rings from the sleeve and remove the gaskets by cutting at two or three places around the circumference. High-pressure jet streams are effective in removing gaskets from older BSTCs.
- (3) Slide one end ring completely away from the sleeve to allow the sleeve to be displaced from the original position, thereby exposing the pipe gap.
- (4) Using replacement gaskets from the manufacturer, insert them one at a time, feeding them through the pipe gap. The generally accepted practice is to stretch the gasket over the top of the sleeve to position it in the proper space between the end ring and sleeve on the one side. If gaskets need to be replaced and the pipe gap is not available to the maintenance personnel, use an unspliced or non-vulcanized gasket supplied by the manufacturer.
- (5) Take the proper measurements, cut the gasket on the bias with a sharp instrument, and bond the ends together to hold the gasket in shape. It is not necessary to bond the ends for the gasket to function properly (as the rubber expansion is radial), but positioning into the gasket flare is easier if the ends are bonded.
- (6) After all parts are properly positioned, follow the installation instructions furnished by the manufacturer.

The results of all visual inspections, evaluations, and repairs must be entered accurately into the plant maintenance log for future reference.

16.3 Maintenance of Air Valves

16.3.1 Scope

Air valves are required on penstocks to ensure proper operation and protection of the penstock during watering-up, dewatering, and normal operation, and therefore should be maintained properly.

16.3.2 Maintenance Operations

Each air valve and appurtenance feature must be checked for proper operation and for operational damage at least once each year. A maintenance procedure must be established that includes the following:

- (1) Inspection of vault or manhole structures for cleanliness and ease of access

- (2) Inspection of vent pipes to assure proper ventilation of air valve enclosures
- (3) Inspection of inside working parts of air valves (valve shafts, float stems, and lever mechanisms must not be struck with a hammer)
- (4) Repair or replacement of defective or nonworking parts, including bolts and gaskets
- (5) Flushing air valves after maintenance to ensure a proper seal and operation
- (6) Evaluation of protective coatings
- (7) Checking the operation of isolation valves and also checking that valves are left open after completion of maintenance
- (8) Protecting air valves from freezing (and discontinuation of the protection when not required)
- (9) Scheduling maintenance during normal outages if possible.

16.3.3 Maintenance Records

Air-valve maintenance records must be kept and should include the following items:

- (1) Date of inspection or maintenance
- (2) Location of each air valve assembly, including reference point measurements
- (3) List of items repaired in the vault or manhole structure
- (4) List of replaced or repaired air valve parts
- (5) Checklist of items observed
- (6) Environmental conditions, including groundwater elevation
- (7) Recommendations for additional repairs and improvements
- (8) Name of person completing the inspection/maintenance.

16.4 Elimination of Vibration

16.4.1 Introduction

As discussed in Section 1.6, when the amplitude of any penstock vibrations becomes excessive, fatigue and sudden failure of the penstock shell is a possibility since an excessive number of oscillations can occur in a relatively short period of time. In order to avoid this dangerous condition, the amplitude of the pressure pulsations generated in the hydraulic conduit at the

penstock's resonant frequency must be reduced significantly or the pressure pulsations eliminated. Alternatively, the natural frequencies at which the penstock vibrates can be changed to avoid resonance. The solution or combination of solutions adopted usually will be a matter of economics and practicality.

16.4.2 Control of Excitation Sources in the Conduit System

16.4.2.1 Low-Frequency Pressure Pulsations

The low-frequency pressure pulsations associated with vorticity and vortex shedding in the draft tube can have their frequency modified or their amplitude reduced by several means. These measures may be successful in certain situations but cannot be guaranteed to work in all cases. These include aeration of the turbine by natural aspiration of air at atmospheric pressure, compressed air, or water-air jet. Mechanical methods which meet with some limited success include introducing vertical fins in the draft tube, installing specially shaped runner cones in the stall zone below the runner, or installing coaxial-type diffuser cones.

16.4.2.2 High-Frequency Pressure Pulsations

The high-frequency pressure pulsations associated with turbine blade passing can be modified in frequency by introducing a new Francis turbine runner with fewer runner blades (i.e., fewer buckets). Alternatively, some success in reducing the amplitude of the pressure pulsations may be obtained by increasing the clearance between the distributor guide vanes (or the wicket gates) and the turbine runner by trimming material from the trailing edges of the guide vanes (or wicket gates) and a small amount of material from the leading edge of the turbine runner blades. It must be recognized that this may result in some loss of performance in the operation of the turbine and should be considered carefully.

16.4.3 Modifying the Penstock Frequency

When the source of pressure pulsations causing resonant vibration in the penstock is not economically or practically amenable to elimination or modification, then consideration can be given to changing the natural vibration frequency of the penstock. This can be accomplished either by introducing intermediate supports or by the installation of circumferential ring stiffeners at spaced intervals along the penstock span between supports.

The use of circumferential ring stiffeners is effective because they act to suppress the circumferential or breathing mode type of vibrations (see Figure 1-2) and significantly impede the longitudinal propagation of these radial deformations from one span to the next. A three-dimensional Finite Element dynamic analysis can be used to determine the most effective number and location of stiffener rings along a penstock span.

The stiffener rings can be arranged to be first bolted in place around the penstock shell and then subsequently welded directly to the penstock shell. The penstock should be dewatered and strip heaters applied to both sides of the stiffener ring to obtain the appropriate preheat temperature prior to welding.

16.4.4 Assessing the Need for Reducing or Eliminating Penstock Vibration

Excessive vibration amplitude of the penstock shell can result in fatigue and a sudden failure of the penstock shell and should not be allowed to continue without adopting corrective measures. A vibrometer can be used to measure the vibration frequency. The deformations associated with the vibration of the penstock can be monitored using strain gages to determine the incremental dynamic stress changes created between peak amplitudes of circumferential deformation in the radial direction. In addition, the maximum peak radial deformations should be located and measured for evaluation purposes.

It is recommended that vibration be considered excessive and remedial measures considered when the associated incremental dynamic stress exceeds 20% of the design stress or when the amplitude of the measured radial deformation exceeds $D/1000$, where D is the penstock internal diameter in inches.¹

16.5 Movement of Supports

This section addresses some of the more common problems associated with the movement of penstock supports. It concludes with some remedies for these problems.

16.5.1 Causes of Movement

Following are some of the common causes of penstock support movement.

(1) Undersized supports

If all loads were not considered during the initial design, it is possible for supports to move. If there is apparent movement in a support, the design must be reviewed to determine that the support is properly sized.

(2) Damaged supports

A support may be damaged (due to a variety of causes) such that the penstock is not being restrained by the full mass of the support.

(3) Improper sliding resistance coefficient

If the actual surface condition of the foundation that the concrete support is placed against differs from the condition assumed during initial design, the support may move.

(4) Unsound rock

If there was improper inspection during construction, rock that was assumed sound during initial design actually may be heavily fractured or unsound.

(5) Deteriorated foundation

When insufficient consideration is given to runoff or erosion, both rock and soil foundations can deteriorate, resulting in lower sliding resistance coefficients.

(6) Slope instability

Because of the steepness of many penstock installations, the slope on which the penstock supports are placed can move, causing the supports to move with the slope or an increased loading on the supports.

16.5.2 Repair and Replacement of Supports

Following are some remedies for the problem of penstock support movement.

(1) Support reinforcement

If the support is undersized, it may be supplemented by adding a system of grouted anchors and structural steel plates or additional concrete surrounding the existing support.

(2) Support replacement

If the support is undersized or damaged beyond repair, it may be necessary to replace it with a properly designed support system.

(3) Support repair

If the supporting foundation is in acceptable condition and the support is not severely damaged, a system of grouted rock anchors and structural steel plates or additional concrete surrounding the existing support may be used to repair the support.

(4) Foundation improvement

If the foundation is unstable or weathered, the foundation may need to be improved or replaced. Unsound rock can be consolidated using pressure grouting.

(5) Slope stability

Unstable slopes can be repaired using rock toes, a system of tie-back retainers, or in some instances, by removal and replacement using geotechnically reinforced soil slopes. It is recommended that geotechnical engineers be used for these types of problems.

16.6 Maintenance of Existing Steel Penstocks

This section addresses the requirements for proper evaluation of the condition of an existing steel penstock. Also discussed are some of the most common problems associated with existing steel penstocks and suggested remedies.

16.6.1 Risk Assessment

An estimate of the potential risks associated with penstock failure mechanisms must be developed. This type of assessment is site dependent and must include, but not be limited to, estimating potential failure modes, assessing the probability or likelihood of such an event, and assessing the potential for loss of life and damage to property associated with each identified failure mode. Penstocks that have been identified as having probable failure risks (especially where the potential for loss of life or damage to property exists) must be inspected on a more frequent basis. The penstock's age, operating and maintenance records, and proximity to a developed area provide a good initial indication of the apparent risk a penstock failure may present.

16.6.2 Evaluation of Penstock Condition

Key items involved in evaluating existing penstocks include:

- (1) Obtaining pertinent engineering documents including, but not limited to, drawings, material property or classification, age, type of fabrication, fabrication and/or construction records, and operating records (including any operational test data)
- (2) Inspecting the penstock to identify any defective areas and to determine if such defective areas are localized or global
- (3) Determining the cause of the defect
- (4) Determining if the defect is self-limiting or may tend to propagate further
- (5) Establishing the mechanical properties of the penstock material including, but not limited to, determining the stress vs. strain relationship, yield strength, ultimate strength, chemical composition, relative ductility, and fracture toughness. This information can be obtained by reviewing the original records if available, by performing destructive testing, and by performing a comparison with penstocks of similar material, construction, and operating records.

It may be impossible to obtain or accurately establish the design allowable stress level criteria, especially for older penstocks. However, such criteria must be conservatively estimated based on the available information.

- (6) Assessing the severity of the problem and any impact to the penstock's structural integrity and remaining service life
- (7) Evaluating nontechnical requirements or concerns that may pose additional constraints. Some common requirements include special operational and/or maintenance requirements,

accessibility, proximity to other structures or features (especially underground structures, cables, or pipes), and annual hydroplant outage times and durations. This information should be provided by the facilities hydro-operations department.

- (8) Developing alternative schemes at a conceptual level, including an examination of the pros and cons associated with each
- (9) Establishing a time frame for implementing any repair or replacement schemes.

16.6.3 Personnel Qualifications

Personnel involved in inspecting penstocks and in evaluating repair and replacement options must be highly qualified. Qualifications include:

- (1) Education in the various engineering disciplines including, but not limited to, material sciences, engineering mechanics, pressure vessel analysis and design (including structural and transient), piping and valve system design, and corrosion engineering. For evaluation of complex problems, a knowledge of welding and fracture mechanics is also essential.
- (2) Familiarity and experience with the various hydro-industry accepted design guidelines, codes, and trade publications (e.g., ASCE, ASME, USBR, AWWA, API, *Waterpower*, and *Hydro Review*).
- (3) Training in applicable federal and state safety regulations. Proper training and certification is essential because most evaluations involve inspection of the penstock interior, which requires working on steep inclinations and in confined spaces where hazardous gases may be present.

16.6.4 Types of Defects

16.6.4.1 Decrease in Shell Thickness

A reduction in wall thickness usually is attributed to uniform corrosion, localized corrosion or pits, cavitation at sharp boundary edges, or scour and abrasion via hydraulic transport of sediment. Corrosion and pitting can occur on both the exterior or interior surfaces. For uncoated penstocks, mill scale can cause anode/cathode reactions.

To determine the extent of material loss, wall thickness measurements must be obtained using the methods given in Section 13. From this data, the associated stress level can be determined and compared against the maximum allowable stress. Depending upon the extent and the effect of the lost wall thickness, the following restoration options are available:

- (1) Relining or recoating

Usually coatings and linings are intended to prevent wall thinning and/or pitting and not to act as structural replacement systems. Therefore, this method should be considered only when the

current stress levels are within an acceptable limit. Coating and lining information is discussed in greater detail in Section 10.

Coating or lining concerns include material life (which varies according to material product and method of application); defects associated with lack of complete adhesion or holidays; proper removal, handling, and disposal of the original coating or lining; and environmental application regulations.

(2) Repairs

Common methods of repairing steel penstocks include the installation of:

(a) Patches or wedding bands

Depending on the chemical composition and weldability of the penstock material, patches or "wedding bands" (circumferential banding) can be welded integrally to the penstock shell. (Note: Welding to cast sections requires special precautions and may not be applicable.) Patches can be installed over the affected area to provide additional reinforcement, or the affected area can be cut out and replaced. Whichever method is selected, all corners or sharp edges must be rounded to reduce the associated stress concentrations, and all welding must be performed with the penstock dewatered to reduce the possibility of thermal crack development in the welds' heat affected zone (HAZ) or the steel plate. All welding must be performed by certified welders as specified in this manual. In addition, all surfaces must be coated/lined to prevent further corrosion.

(b) Boiler plugs or redwood dowels

For localized areas, particularly in low head areas, where localized through-wall "pin holes" have developed, boiler plugs or redwood dowels can be used. However, this method is only a temporary measure to be used until a more permanent remedy can be implemented.

(c) Concrete encasement

Full concrete encasement is used to transfer all internal pressure loads to the concrete so that the steel shell functions only as a waterproof membrane. Concrete must be placed with the penstock completely dewatered and with circumferential reinforcement to prevent the internal penstock pressure from cracking the concrete.

(3) Replacement

Replacement sections must be designed and manufactured in accordance with Sections 2 and 11, respectively.

16.6.4.2 Changes in Steel Mechanical Properties

Aside from design-related deficiencies, changes in steel mechanical properties can be attributed primarily to: (1) fatigue-type loading caused by excessive vibration (see Section 1.6 for a discussion of vibration prevention and Section 16.4 for a discussion of vibration elimination), (2) in-service operating temperature range relative to the material's ductile-to-brittle transition zone, (3) excessive heat imparted to the steel, which usually occurs during the welding process and stress relieving, and (4) an abnormally high transient pressure rise that exceeds the yield point of the material or joints.

16.6.4.3 Decrease in Joint Efficiency

The causes of in-service reductions in joint efficiencies can be characterized by the type of joint or connection. The most common types of connections and associated problems are:

(1) Forge-welded connections

Problems with forge-welded joints include variable joint efficiency and accelerated corrosion at the joint (see Section 16.9). To verify the efficiency of a forge-welded joint, "coupons" must be removed and tested. From this data, an acceptable joint efficiency can be determined and used in ascertaining if repair or replacement schemes are warranted. If the tested joint efficiencies are determined to be too low, a fracture mechanics evaluation can be performed to demonstrate that certain flaw sizes and orientation are acceptable for the operating service loads. If the design steel specification is unknown, a metallurgical evaluation must be performed to determine the material's chemical composition and ascertain if weld repairs may be appropriate.

If the stresses in the joint are deemed unacceptably high, replacement may be the preferred alternative. However, some repair options exist. The most common joint repair options include:

(a) Weld repair of the joint

This approach is highly dependent on the chemical and mechanical properties of the steel and may require a postweld heat treatment to stress relieve the weld zone (especially where plate thickness exceeds 1½ inches and carbon content is too high).

(b) Installation of structural banding

Structural banding may be installed. However structural banding may be difficult to install and may not provide sufficient long-term reliability.

(2) Riveted connections

Design joint efficiencies for this type of connection usually are in the range of 46% to 95% (see Section 16.8 for methods of determining riveted joint efficiencies). Reductions in the design joint efficiency, or conversely, an increased stress distribution across the joint can be caused by improper installation of the rivet or corrosion of the rivet shanks. Scour or abrasion of a

rivet head is indicative of a loss of plate thickness (see Section 16.6.3.1). If the efficiency of the joint is determined to be unacceptable, common repair options include:

(a) Replacing rivets with bolts

Obtaining full plate contact of both the bolt and nut by applying sufficient torque to the bolts is of primary importance. If a cluster of rivets is to be replaced, the rivets must be replaced one at a time to prevent the joint from springing open.

b) Welding both sides of the lap joint

Welding both sides of the lap joint tends to increase the joint efficiency, assuming that the rivet strength is the critical element. However, it also introduces additional residual stresses associated with the welding process and may result in leakage around the rivets, which can lead to corrosion problems. Should this method be employed, all welding must be performed when the penstock is dewatered to minimize the thermal impact associated with welding on the cold plate, and the repair must be monitored during re-pressurization to verify that there is no leakage around any rivet heads. If leakage occurs, caulking around the interior of the rivet head or installing a flexible interior lining system may be required.

3) Welded connections

Typically, joint efficiency is established during the original design phase (see Section 3.5). The welded joint usually can be repaired by air-arc gouging or grinding out the defect and re-welding. Depending on the steel plate thickness, preheat or postweld heat treatment may be required. In addition, weld repairs on pipe sections encased in concrete may not be effective due to the difficulty in performing the necessary preheat and postweld heat treatment.

16.6.4.4 Cracks

Cracks develop at areas of high localized stress caused by inherent material or weld defects or by excessive loading. Once formed, certain types of cracks tend to propagate due to in-service conditions (i.e., when subjected to vibration, especially). Factors such as chemical composition, material properties, fracture toughness, crack geometry, and applied loading (and frequency) determine whether or not a crack will tend to propagate. To determine if repairs are required or even possible, both a fracture mechanics and a metallurgical evaluation must be performed. A fracture mechanics and metallurgical evaluation indicates the maximum crack geometry that will tend to be self limiting (e.g., cracks will not propagate further providing the mechanism responsible for initiating the crack can be arrested). In addition, this type of evaluation determines if repairs are feasible and, if so, indicates the type of repair procedure to be implemented.

Crack repairs typically consist of grinding or air-arcing out the crack and implementing an appropriate weld procedure. Depending upon the plate thickness and chemical composition, preheat or postweld heat treatment may be required. One cautionary note in performing weld

repairs is that, depending upon the chemical composition of the steel, the heat associated with the welding process may tend to propagate the crack further. Therefore, weld repair of cracks must be performed only under the careful supervision of a qualified metallurgist or welding engineer. In addition, if the penstock section in which cracks have been identified is to be left in service (whether or not it has been repaired), periodic inspections and non-destructive testing must be performed to confirm that no new cracks have developed and that unrepaired cracks are not propagating.

16.6.4.5 Casting Defects

Old steel castings typically are associated with penstock elbow, tee, transitions, and wye branch sections. Casting defects range from superficial lineation and blemishes to structural defects such as cracks associated with shrinkage during cooling near discontinuities, inclusions of foreign material in the casting, and poor chemical or crystal grain structure.

Repair of casting defects is dependent on the chemical composition, weldability, and geometry of the casting, whether the casting is encased in concrete or exposed, and the type of defect. Most repairs consist of air-arc gouging out or mechanically grinding out the defective area and may not require performing a weld repair. For shallow cracks or defects it may be sufficient to grind out the defect and feather back all sharp edges to minimize stress concentrations. For larger cracks or defects, weld repairs with preheat and potential postweld heat treatments may be required. Successful repair of casting defects requires special precautions and must be performed only under the careful supervision of a qualified metallurgist or welding engineer.

It should be noted that not all types of casting defects require repairs or can even be repaired. Small cracks and defects may be acceptable without repair, pending a review using fracture mechanics to determine the maximum acceptable crack size (length and depth) that is considered self limiting. Shrinkage-type defects generally are not considered repairable, but may be acceptable depending on defect size, relative location, material toughness, and relative state of stress. In addition, depending upon the weld procedure, weld repairs on casting encased in concrete may not be effective due to the difficulty in obtaining the necessary preweld and postweld heat treatment.

16.6.4.6 Ovalization/Out-of-Roundness

Thin-walled penstocks are most susceptible to losing their shape and becoming out of round. However, penstocks with an acceptable wall thickness also can lose their shape. Following are some of the most common causes of penstock ovalization.

- (1) When the normal internal pressures are low and the diameter-to-thickness ratios are high, the penstock typically will not maintain its circular shape. For low-head sections, the rounding effect associated with pressurizing the conduit may not be sufficient to offset the weight of the fluid acting outward and tending to flatten out the pipe. This problem may be of particular concern for penstocks on concrete saddles because of high stress concentration on the saddle horn (see Section 4.3.6.1).

- (2) Improper installation of buried or partially buried penstocks can cause the penstock to lose its shape. Typically, either improper compaction or the application of excessive surcharge loads can cause the penstock to lose its shape. Proper compaction from penstock invert to springline is essential to proper installation. Over-compaction at the springline can deflect the penstock sides inwardly, while under-compaction can cause the sides to splay outward. Exceeding the design surcharge or external pressure design loading (e.g., under road crossings) can also result in ovalization of the penstock.
- (3) Penstock sections that have not been designed for external loads and that are to be backfilled in soil or encased in concrete can become ovalized. Penstock sections that are buried deeply or are to be subjected to large surcharge loads must be designed for external loads with the penstock empty (typically the more severe loading).

16.6.4.7 Voids in Backfill or Concrete-Encased Penstock Sections

For buried or concrete-encased penstock sections, voids may be present in the backfill or concrete.

Voids in backfill are typically caused by ground water erosion of the backfill material near the invert of the penstock. Prolonged erosion of the backfill can undermine the penstock foundation, leading to differential settlement and potential failure. This type of defect usually can be detected by striking the penstock shell with a hammer at multiple locations and listening for a hollow sound. Once a void has been detected, the best method for repair is to excavate the backfill and then recompact backfill material into the void. Installation of a drainage system utilizing a geotextile fabric also helps to minimize soil erosion at such locations. Also, the water source should be located and a system developed to channel the water away from the penstock backfill.

Voids in concrete are caused by poor consolidation of the fresh concrete during concrete placement or by the trapping of excess water in the concrete "bleed water" near the penstock invert. Typically these types of voids are localized and relatively small. However, voids large enough to cause damage to the penstock can occur. The detection of voids in concrete is similar to the detection of voids in backfill. The repair of unacceptable voids usually consists of drilling through the penstock shell and filling the void with grout by pumping. A "patch" or plug is then installed on the drilled-out area of the penstock shell.

16.6.4.8 Localized Buckling

Localized circumferential buckling is an indication of longitudinal overstressing of the penstock. This defect is most common with thermal changes that affect a section of penstock between two fixed points, such as anchor blocks, and excessive external water pressures or dewatered tunnel liners. The penstock is most susceptible to thermal problems when the penstock has been dewatered and the ambient temperatures exceed the penstock's normal operating temperatures. Therefore, dewatering of the penstock for prolonged periods must be avoided during the warmest seasons of the year. For repair, the damaged penstock section must be removed and an expansion joint installed. If an expansion joint already is installed in the section, it must be inspected closely to determine that it is functioning properly.

16.6.5 Types of Problem Loading

16.6.5.1 Excessive Vibration

Vibration-related problems can surface at any time during the life of a penstock. The most typical problem associated with vibration is fatigue.

Methods of preventing or reducing vibration usually consist of changing (usually increasing) the penstock stiffness (i.e., natural frequency). In some cases, vibration-detecting sensors can be installed to monitor the penstock. The sensors trip the unit off-line when vibration levels become unacceptable. Methods for reducing vibration are discussed in greater detail in Sections 1.6 and 16.4.

16.6.5.2 Abnormal Transient Pressures

Load rejection and load acceptance conditions can produce unacceptably large positive and negative pressure waves along the entire length of the penstock.

Large positive pressure rises can produce unacceptably high tension stresses in the penstock shell. Remedies, aside from replacement with pipe of higher pressure capacity, include installing a bypass valve that opens when the turbine wicket gates close, installing rupture discs that open when the pressure rise exceeds the structural capacity of the disc (lower than penstock design pressure), changing the turbine gate/needle closure timing, installing jet deflectors on Pelton-type turbines, and eliminating or significantly reducing the time lag associated with mechanical linkage systems. All options should be explored based on effectiveness and economics.

Large negative pressure drops occur when the elevation of the hydraulic grade line drops below the penstock elevation. Water-column separation possibly then can occur. This condition can produce a negative internal pressure and very large re-combining pressures that could collapse or rupture the penstock. The previously described remedies can be used if the negative pressure drop is attributed to load rejection. However, if the negative pressure drop occurs during the "load on" condition, the needle valve timing can be increased.

16.7 Recoating/Relining

Surface preparation and keying into the existing coating/lining material is necessary to ensure that the recoating or relining provides performance capabilities equivalent to the original material.

All existing coating or lining materials surrounding the area to be recoated or relined must be checked for adhesion to the penstock. Loose coating or lining materials must be removed back to the point that the material exhibits sound adhesion to the penstock. Adhesion of adjacent or remaining materials must be evaluated by reference to the manufacturer's recommendations or by using a knife test (see AWWA C203-86²). Edges of adhering material adjacent to the area of the recoat/reline must be feathered to provide a smooth, even surface.

Surface preparation of the area to be repaired must be in accordance with the agreement of the coating/lining manufacturer and the owner. In addition, the area to be repaired must be cleaned with solvent (flash point > 100°F) wipes to remove oil and grease. Only clean rags and a solvent that does not leave a residue or have detrimental health effects should be used. Following solvent cleaning, the repaired area and coated or lined areas adjacent to the repaired area must be abrasive- or water-blasted to remove all loose materials, rust, and other foreign residue (cement-coated or lined penstocks must not be hydroblasted). Water for the hydroblasting must be clean and free of contaminants that may leave deposits.

The coating/lining materials must be the same as or compatible with the existing materials on the penstock. The material selection must be agreed upon with the owner prior to application. The applied materials must conform to the specified thickness.

All applied coating/lining materials applied to sand- or water-blasted substrates must be checked for adhesion either by reference to the manufacturer's recommendations or by tests using the adhesion test described in AWWA C203-86.²

All repaired areas must be electrically inspected for holidays in accordance with the original specification or by a method agreed upon by the owner. All holidays must be repaired prior to placing the penstock back in service.

16.8 Riveted Penstock Joints

16.8.1 Background

Although current penstock design and fabrication practices no longer use riveted joints, many penstocks still in service are either fully riveted or a combination of welded and riveted.

It is important to know more about the original design and construction of these older penstocks because many of them are connected to powerhouses that are being upgraded with additional powerhouse units through repowering programs. Also, with the increase in age of these older penstocks, many are being repaired or replaced as a result of safety evaluation programs or normal maintenance programs.

16.8.2 Types of Joints

The two types of riveted joint details that have been used for steel penstock fabrication and erection are:

- (1) Shop- or field-riveted longitudinal and field circumferential joints
- (2) Shop-welded longitudinal joints and field-riveted circumferential joints.

16.8.3 Composite Joint Design

The combination of shop welding and riveting was very common during the 1950s. These types of joints did not change until improved welding techniques were developed that made high-quality field welding feasible. Prior to the 1950s and dating back into the late 1800s, most joints were fully riveted. One exception was the use of forge-welded and banded steel penstocks during the 1910s and 1920s (see Section 16.9 and 16.10).

Usually all shop welded joints, either longitudinal or circumferential, were welded, radiographed and stress relieved in accordance with the ASME Code. In addition, it was quite common to pressure test each welded pipe section to 150% of the normal working pressure.

Prior to the 1950s, there was generally no field welding of the penstock shells, except in cases where access was a problem (tunnel liners made field riveting impractical).

It was not considered practical to pressure test riveted pipe in the shop because the strength could not be accurately computed and because of the difficulty of testing due to the lack of equipment.

16.8.4 Material Properties

Generally, two grades of steel were used for pressure shells: flange and firebox.

16.8.4.1 Chemical Composition

The chemical composition of the shell material usually conformed to the specifications given in Table 16-1.

Table 16-1 Shell Material Chemical Composition for ASTM A30

ELEMENT	FLANGE STEEL	FIREBOX STEEL
CARBON	—	PLATES $\leq 3/4$ IN., 0.12% - 0.25% PLATES $> 3/4$ IN., 0.12% - 0.30%
MANGANESE	0.30% - 0.60%	0.30% - 0.50%
PHOSPHORUS (ACID)	< 0.05%	< 0.04%
PHOSPHORUS (BASE)	< 0.04%	< 0.035%
SULPHUR	< 0.05%	< 0.04%
COPPER	—	< 0.05%

The chemical composition of the rivet steel material conformed to the requirements given in Table 16-2.

Table 16-2 Rivet Steel Material Chemical Composition

ELEMENT	COMPOSITION
MANGANESE	0.30 - 0.50%
PHOSPHORUS	< 0.04%
SULPHUR	< 0.045%

16.8.4.2 Mechanical Properties

The mechanical properties of the shell steel used usually conformed to the requirements given in Table 16-3.

Table 16-3 Mechanical Properties of Shell Steels for ASTM A30

MECHANICAL PROPERTY	FLANGE STEEL	FIREBOX STEEL
TENSILE STRENGTH, psi	55,000-65,000	55,000-63,000
YIELD POINT, MIN., psi	0.5 x TENSILE STRENGTH	0.5 x TENSILE STRENGTH
ELONGATION IN 8-IN., MIN., %	1.5x10 ⁶ /TENSILE STRENGTH	1.5 x 10 ⁶ /TENSILE STRENGTH

16.8.4.3 Crushing Strength of Mild Steel

The resistance to crushing of mild steel was 95,000 psi (for the cross-sectional area).

16.8.4.4 Strength of Rivets in Shear

Table 16-4 gives stresses for the cross-sectional area of the rivet shanks, used for calculating joint strengths of circumferential and longitudinal joints.

Table 16-4 Stresses for Cross-Sectional Area of Rivet Shanks

RIVET TYPE AND STRESS CONDITION	ALLOWABLE STRESS (psi)	ULTIMATE STRENGTH (psi)
IRON/SINGLE SHEAR	—	38,000
IRON/DOUBLE SHEAR	—	76,000
STEEL/SINGLE SHEAR	11,000	44,000
STEEL/DOUBLE SHEAR	22,000	88,000
STEEL/BEARING	23,750	95,000
STEEL/TENSION	13,750	55,000

16.8.5 Design

16.8.5.1 Minimum Thickness of Plate

The minimum thickness of any plate under pressure was 1/4 inch, except that for shells 24 inches and less in diameter the minimum plate thickness was 3/16 inch.

16.8.5.2 Minimum Thickness of Butt Straps

The minimum thickness of butt straps for double strap joints is given in Table 16-5.

Table 16-5 Minimum Thickness of Butt Straps for Double Strap Joints

THICKNESS OF STEEL PLATE (IN.)	MIN. THICKNESS OF BUTT STRAPS (IN.)
3/16	3/16
1/4	1/4
9/32	1/4
5/16	1/4
11/32	1/4
3/8	5/16
13/32	5/16
7/16	3/8
15/32	3/8
1/2	7/16
17/32	7/16
9/16	7/16
5/8	1/2
3/4	1/2
7/8	5/8
1	3/4
1-1/8	3/4
1-1/4	7/8

For plate thicknesses exceeding 1¼ inches, the thickness of the butt straps could not be less than two-thirds the shell thickness.

16.8.5.3 Allowable Stress

The maximum allowable working stress (S) was determined by:

$$S = 0.0125F_y(e + 8), \text{ and } S \leq 0.4F_y \text{ or } 0.2F_t \quad (\text{Equation 16-1})$$

Where:

- S = maximum allowable unit working tension stress, psi
- F_y = elastic limit of material
- F_t = tensile strength of material
- e = elongation of material in 8 inches, in percent

For example, the allowable stress for 55,000 psi material (F_t) is 11,000 psi.

16.8.5.4 Joint Efficiencies

The efficiency of a joint is the ratio between the strength of the joint and the strength of the solid plate. In the case of a riveted joint, this was determined by calculating the breaking strength of a unit section of the joint, considering each possible mode of failure separately, and then dividing the lowest result by the breaking strength of the solid plate of a length equal to that of the section considered. In computing the joint efficiency, the hole diameters were assumed to be 1/16 inch larger for drilled holes and 3/16 inch larger for punched holes than the nominal rivet diameter.

16.8.5.5 Longitudinal Joints

On longitudinal joints, the distance from the center of rivet holes to the edges of the plates, with the exception of rivet holes in the ends of butt straps, was not less than 1.5 times and not more than 1.75 times the diameter of the rivet holes. This distance was measured from the center of the rivet holes to the caulking edge of the plate before caulking.

The longitudinal joints of a shell that did not exceed 36 inches in diameter and were of lap-riveted construction had a maximum allowable working pressure of 100 psi. The plate was rolled and the two ends lapped over each other and then riveted (see Figure 16-1). Generally these joints had very low joint efficiencies that varied from 46% to 95% and therefore were used primarily for low-head locations.

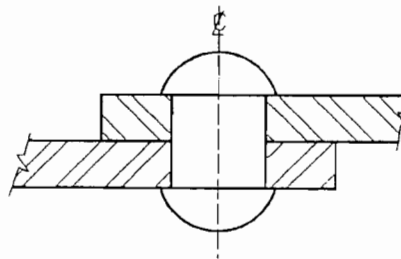


Figure 16-1 Riveted Lap Joint

The longitudinal joints of a shell that exceeded 36 inches in diameter were to be of butt and double-strap construction. The butt joint had outer and/or inner cover plates that were riveted together (see Figure 16-2). These joints had a much higher joint efficiency (77% to 95%) and were used for higher head applications.

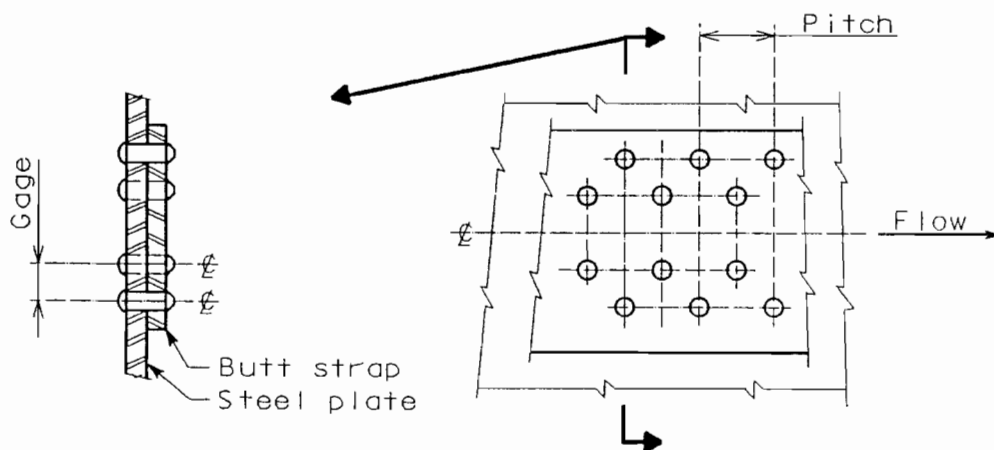


Figure 16-2 Butt-Strap Riveted Joint

16.8.5.6 Circumferential Joints

The strength of circumferential joints of penstocks was at least 50% the strength of the longitudinal joints of the same structure.

16.8.5.7 Butt Straps

Butt straps and the ends of shell plates forming the longitudinal joints were rolled or formed by pressure, not blows, to the proper curvature.

16.8.5.8 Rivets

(1) Centerline distances

The distance between the center lines of any two adjacent rows of rivets, or the "back pitch" measured at right angles to the direction of the joint, was to be at least twice the diameter of the rivets and also had to meet the following requirements:

- (a) Where each rivet in the inner row came midway between two rivets in the outer row, the sum of the two diagonal sections of the plate between the inner rivet and the two outer rivets was to be at least 20% greater than the section of the plate between the two rivets in the outer row.
- (b) Where two rivets in the inner row came between two rivets in the outer row, the sum of the two diagonal sections of the plate between the two inner rivets and the two rivets in the outer row was to be at least 20% greater than the difference in the section of plate between the two rivets in the outer row and the two rivets in the inner row.
- (c) The maximum spacing of holes along caulked edges was governed by the formula $P = 2.5t + d$ (not to exceed $1\frac{1}{2}$ inches), where P = pitch, t = plate thickness, and d = diameter of the rivet hole.

(2) Edge distance

On longitudinal joints, the distance from the centers of rivet holes to the edges of the plates, with the exception of rivet holes in the ends of butt straps, was not to be less than 1.5 times the diameter of the rivet holes.

(3) Rivet holes

Rivet holes were drilled full size with plates, butt straps, and heads bolted in position or were punched not to exceed $\frac{1}{4}$ inch less than full diameter for plates over $\frac{5}{16}$ inch in thickness, and $\frac{1}{8}$ inch less than full diameter for plates not exceeding $\frac{5}{16}$ inch in thickness, and then drilled or reamed to full diameter with plates, butt straps, and heads bolted in position. After drilling rivet holes, the plates and butt straps were separated and the burrs removed.

(4) Rivet length

Rivets were of sufficient length to completely fill the rivet holes and form heads at least equal in strength to the bodies of the rivets.

16.8.5.9 Manholes

(1) Size

An elliptical manhole opening was not less than 11 inches by 15 inches or 10 inches by 16 inches in size. A circular manhole opening was not less than 15 inches in diameter.

(2) Reinforcing

When a manhole reinforcing ring was used, it was of steel or wrought iron at least as thick as the shell plate. When reinforcing rings were required, they were to have the proper curvature. Shells over 48 inches in diameter were riveted to the shell with two rows of rivets, which could be pitched as shown in Figure 16-3. The strength of the rivets in shear on manhole frames and reinforcing rings was at least equal to the tensile strength of that part of the shell plate removed, on a line parallel to the longitudinal axis of the shell, through the center of the manhole or other opening.

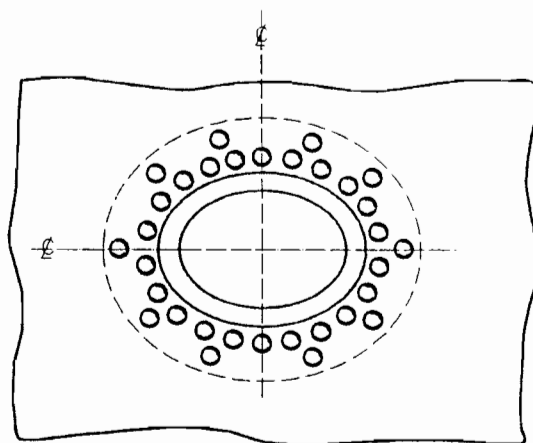


Figure 16-3 Manhole Rivet Connection

The following formulas determined the proportions of manhole frames and other reinforcing rings necessary to conform to the above specifications. It was assumed that the rings must have the same tensile strength as, and have a thickness equal to or greater than, the shell plate removed for the opening.

For a single-riveted ring:

$$W = \frac{l}{2} \left(\frac{t_1}{t} \right) + d \quad (\text{Equation 16-2})$$

For a double-riveted ring:

$$W = \frac{l}{2} \left(\frac{t_1}{t} \right) + 2d \quad (\text{Equation 16-3})$$

For two single-riveted rings:

$$W = \frac{l}{4} \left(\frac{t_1}{t} \right) + d \quad (\text{Equation 16-4})$$

For two double-riveted rings:

$$W = \frac{l}{4} \left(\frac{t_1}{t} \right) + 2d \quad (\text{Equation 16-5})$$

Where:

- W = least width of reinforcing ring, in.
- t_1 = thickness of shell plate, in.
- d = diameter of rivet when driven, in.
- l = length of opening in shell in direction parallel to axis of shell, in.
- t = thickness of reinforcing ring (not less than thickness of the shell plate), in.

The following formula was used to determine the number of rivets for a single or double reinforcing ring:

$$N = \frac{5.1Ta}{Sd^2} \quad (\text{Equation 16-6})$$

Where:

- N = number of rivets
- T = tensile strength of the ring, lb/in.² of section
- a = net section of one side of the ring or rings, in.²
- S = shearing strength of rivet, lb/in.² of section
- d = diameter of rivet when driven, in.

(3) Covers

Manhole cover plates were to be of wrought steel or steel castings.

(4) Bearing surface

The minimum width of a bearing surface for a gasket on a manhole opening was 1/2 inch. No gasket used on a manhole or handhole of any penstock was to have a thickness greater than 1/4 inch.

16.8.6 Fabrication

16.8.6.1 Rivet Application

Rivets were machine driven wherever possible, with sufficient pressure to fill the rivet holes, and allowed to cool and shrink under pressure.

16.8.6.2 Caulking

The caulking edges of plates, butt straps, and heads were beveled to an angle not sharper than 70 degrees to the plane of the plate. Every portion of the caulking edges of plates, butt straps, and heads was planed, milled, or chipped to a depth of not less than 1/8 inch. A round-nosed tool was used for caulking.

16.9 Forge-Welded Penstocks

16.9.1 Background

Although current penstock design and fabrication practices no longer use forge-welded joints, many penstocks of this type are still in service.

It is important to know more about the original design and construction of these older penstocks because many connect to powerhouses that are being upgraded with additional units through repowering programs. Also, with the increase in age of these older penstocks, many are being repaired or replaced as a result of safety evaluation programs or normal maintenance programs.

16.9.2 Material Properties

The steels used for forge-welded penstocks were the flange and firebox steels used for riveted penstocks (see Section 16.8) and open hearth steels described herein.

16.9.2.1 Chemical Composition of Open Hearth Steels

Following are specifications for the chemical composition of the open hearth steels used for forge-welded penstocks:

- (1) Carbon (plates 3/4 inch and less in thickness): Not to exceed 0.18%
- (2) Carbon (plates greater than 3/4 inch in thickness): Not to exceed 0.20%

- (3) Manganese: 0.40% - 0.60%
- (4) Phosphorus: Not to exceed 0.04%
- (5) Sulphur: Not to exceed 0.05%

Preferably, the steel for forge-welded plates was free of silicon, nickel, and chromium. Where these elements were present, the maximum content of any one element was not to exceed 0.05%.

16.9.2.2 Tensile Strength of Open Hearth Steels

The open hearth material used for forge-welded penstocks conformed to the following requirements for tensile properties (for mechanical properties of flange and firebox steels, see Section 16.8.4):

- (1) Tensile strength: 55,000 psi
- (2) Yield point, min.: 24,000 psi
- (3) Elongation in 8 inches, min., %: $1.5 \times 10^6 / (\text{tensile strength})$

16.9.3 Design

- (1) Minimum thickness

The minimum thickness for any plate was 1/4 inch, but in no case could the shell plate thickness be less than the diameter of the penstock divided by 200.

- (2) Allowable stress

The allowable stress for steels using the forge-welding process was 12,500 psi.

- (3) Joint efficiency

The efficiency of the joint when properly welded by the forge-welding process was taken as 85% of the minimum ultimate strength for flange and firebox steels and 95% for open hearth steels.

It should be noted that joint efficiency is directly related to the quality of manufacturing of the joint. Because the plates are heated to approximately 2,000°F, high-temperature oxides develop on the joint. If these oxides are not removed, adequate bonding does not occur during the pressure welding process. As a result, joint efficiency can drop considerably.

In reevaluating forge-welded penstocks, it is recommended that coupons be removed for destructive testing. These tests should include tensile tests in the joint and parent metal (for determination of yield strength, ultimate strength, and joint efficiency) and Charpy tests.

Alternatively, UT imaging inspection (A, B, and/or C-scan) and a fracture mechanics evaluation can be performed to determine the allowable joint strengths. Note that this material generally has poor fracture toughness.

16.9.4 Fabrication

16.9.4.1 Forge Welding

The edges to be welded together were lapped a distance at least equal to the thickness of the plate to be welded. All plates 1 inch and less in thickness were welded without scarfing. Plates exceeding 1 inch in thickness were scarfed, if desired by the manufacturer. The scarf started at least one-half the thickness of the plate from the side next to the weld. Figure 16-4 shows examples of joint details; details (A) through (C) show how various penetration systems are attached to the shell with forge welds; detail (D) shows the joint preparation before the welding process and also the welded joint.

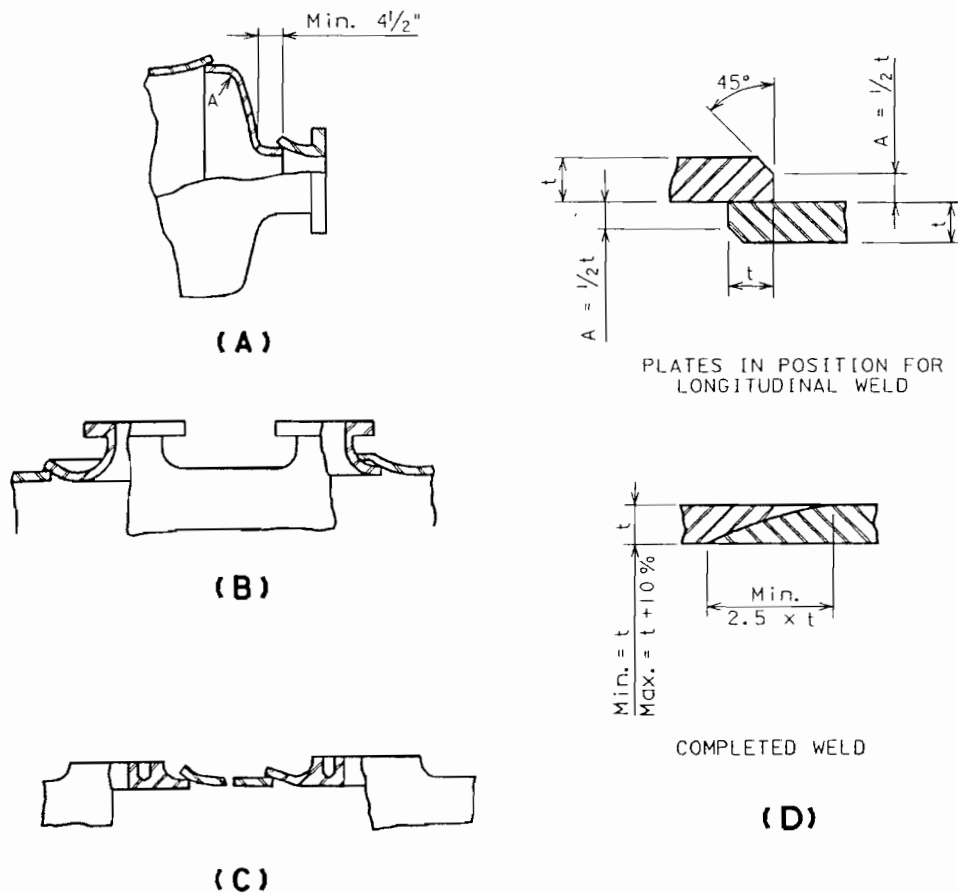


Figure 16-4 Forge-Welded Joint Details

16.9.4.2 Decarbonization

Because the forge-welding process requires heating the steels to approximately 2,000°F, decarbonization (loss of carbon) of the weld joint occurs. Because of this, the joints will corrode at a more accelerated rate than the base metal. If the interior lining and exterior coating do not provide adequate protection, severe wall corrosion can occur.

After the material was brought up to the proper welding temperature, it was placed between an anvil and a hammer, or between rolls, or mandrel and roll, or between mandrels. The plates were welded together by pressure (applied by the hammer, rolls, or mandrels that actually displaced the material during welding). The metal in and adjacent to the weld was not worked at what was termed the critical blue heat temperature of the steel (between 400°F and 800°F).

The minimum thickness (t_{min}) of the weld for all longitudinal and circumferential seams and special welds was the actual thickness of the shell. The maximum thickness (t_{max}) was 1.1 times the actual thickness of the shell. The engineer should pay particular attention to the actual thickness at these joints because many joints have shown to be thinner than the normal plate thickness.

The contact line of completed forge welds was equal to at least 2.5 times the thickness of the plate thickness.

16.9.4.3 Annealing

All longitudinal and circumferential welds were annealed by heating to the proper temperature to relieve strains and then were allowed to cool slowly. All longitudinal welds on the cylindrical pipes were heated in the area within 8 inches of the center of the weld or the entire shell, after which they were rolled to a commercially true cylindrical form. Any vessel distorted out of shape was reformed and then annealed or reformed at the proper annealing temperature. For a finished cylindrical shell, variations in diameter did not exceed 1% of the mean outside diameter when measured at any section. When a straight edge, two diameters long, was laid longitudinally along the outside of the shell, it had to be possible to set the straight edge such that no part of the edge exceeded a distance 1% of the mean outside diameter from the outer surface of the shell.

16.9.4.4 Hydrotest

All penstocks with seams or joints fabricated by the forge-welding process were hydrotested. They were tested to 50 psi over the working pressure when that pressure did not exceed 100 psi and to 1.5 times the working pressure for pressures over 100 psi. While under the hydrotest pressure, a thorough hammer or impact test was performed. It consisted of striking the shell on both sides of the welded seam with a sharp vibratory blow (hammer test) using a 2- to 6-pound hammer. The blows were struck 2 to 3 inches apart and within 2 to 3 inches of and on each side of the seam. The blows were as rapid as a man could conveniently strike a sharp, swinging blow and as hard as could be struck without indenting or distorting the metal of the shell. During this test the penstock was filled completely with water. This was the practice at the time of these designs and fabrication, but clearly is not current hydrotesting practice (see Section 13.4).

Welded seams or joints that did not pass this test (because of leaks, distortion, or other signs of distress) were not accepted until the defects were repaired and the test successfully passed. Defective sections of welded seam were cut out and rewelded, provided the shell integrity was not lowered. When in question, a coupon was cut out across the weld at points in question and subjected to microscopic or other examination.

16.10 Banded Steel Penstocks

16.10.1 Background

Although current penstock design and fabrication practices no longer use the banded steel pipe process, many penstocks of this type are still in service.

It is important to know more about the original design and construction of these older penstocks because many connect to powerhouses that are being upgraded with additional units through repowering programs. Also, with the increase in age of these older penstocks, many are being repaired or replaced as a result of safety evaluation programs or normal maintenance programs.

16.10.2 General

Banded steel penstock fabrication began around 1910 and continued for approximately 50 years. These penstocks were fabricated primarily in France, Germany, and Poland. They were used when the design pressures required plate thicknesses greater than approximately 1¼ inches. The lack of technology prohibited welding of plates exceeding this thickness.

There were two methods of fabricating banded steel penstocks. The first was to shrink the bands over the cylinders. The second was to install oversized rings on the pipe and expand the shell such that it pinched against the bands.

16.10.2.1 Band Shrinking

The banded pipe consisted of a cylindrical pipe manufactured in the same manner as for plain forge-welded pipes. The forge-welding process, using water gas to prevent oxidation of the metal, was used. The metal was reheated and circularly rolled. Any scale formed during this stage of the process was removed carefully using steel brushes. Areas over which the bands were to be shrunk were cleaned. The circumference was measured carefully to determine the proper size to which the bands were to be machined. The bands, which were seamless, were pressed and rolled, and finally machined to the correct size. Then they were heated sufficiently, without altering their mechanical properties, to allow them to fit. When they had cooled to the ambient temperature, they formed a single unit with the cylinder.

When the cylindrical part of the vessel was fabricated by this method, the end flanges were attached and the whole unit was stress relieved. Residual stresses due to welding, as well as the prestressing resulting from the hoop fitting process, were practically eliminated. The intervals for the bands were determined in accordance with the pressure for which the pipe was designed.

The amount of shrinkage was computed such that the bands and the cores were stressed to an equal factor of safety, thereby preventing the bands or the cores from being overstressed. As a result, the shell and the hoops were stressed such that the hoops were in tension and the shell in compression.

16.10.2.2 Expansion of Shell

The hoops, with a slightly larger inside diameter than the outside diameter of the pipe, were equally spaced. The pipe was placed between two end plates of a hydrotesting machine and the cylinder was expanded. It was pressurized until it reached its elastic limit, after which the cylinder expanded until it pressed against the hoops. Once the internal pressure was reduced to zero, the cylinder and hoops formed a single unit.

16.10.3 Advantages

Banded steel pipes had a number of advantages, as discussed below.

- (1) The weight, at the same factor of safety, was considerably lower than that of plain forge welded or riveted pipes of comparable pressure ratings. As a result, overall costs were less.
- (2) The factor of safety was accurately known because the bands (which represented approximately one-half of the material employed) had a 100% joint efficiency (being solid rolled).
- (3) Because the shell was fabricated much thinner than an equivalent plain forge-welded pipe, better notch toughness properties could be obtained for the banded pipe.
- (4) Should a failure occur (during an emergency or exceptional condition), any rupture would occur only at the weakest point, i.e., in the core between two bands. Thus, leakage would be limited to the length of pipe between two bands, ensuring the facility against a disaster. Banded pipes had a decided advantage when compared with seamless pipes; the combined use of materials and designs of different strengths resulting in banded pipes acting as an automatic relief in case of a failure.
- (5) Erection was simplified considerably resulting in decreased costs because the core pipes had practically one-half the thickness of a plain forge-welded pipe of comparable pressure ratings.

16.10.4 Material Properties

The material used in the manufacture of plain forge-welded and core pipe for banded steel pipes was a high-grade, open-hearth steel with the physical properties given in Table 16-6.

Table 16-6 Physical Properties of Material Used for Banded Steel Pipes

MATERIAL	YIELD STRENGTH (psi)	TENSILE STRENGTH (psi)	MINIMUM ELONGATION
PLATES FOR PLAIN-WELDED AND CORE PIPE	27,800	48,500 to 58,000	25%
	(51,000)*	(74,000)*	—
CAST STEEL FOR BENDS AND OTHER PARTS	35,000	64,000 to 85,000	18% - 23%
BANDS FOR REINFORCED PIPES	37,000	64,000 to 85,000	18%
	(113,000)*	(142,000)*	—
COMBINED SECTION (PIPE AND BANDS)	32,500	—	—

* After World War II, quenched and tempered high-strength alloy steel was used.

16.10.5 Design Stresses

The determination of the thickness was based upon the following criteria.

(1) Working stress in shell

Working stress at static pressure in the full section of a plain forge-welded pipe was not to exceed 10,000 psi (equal to the yield strength of 27,800 psi times the joint efficiency of 90% divided by 2.5).

(2) Maximum stress in shell

The maximum stress was not to exceed 18,800 psi. (Static head plus water hammer was not to exceed 75% of the yield strength of 27,800 psi times the joint efficiency of 90%.

(3) Working stress in bands

The working stress at static pressure in the steel bands was not to exceed 13,300 psi.

(4) Maximum stress in bands

The maximum stress at static pressure plus water hammer in the steel bands was not to exceed 25,000 psi.

(5) Working stress in combined section

The average combined stress used for design of the bands and the shell was 13,200 psi. At least 50% of the pressure was to be taken by the bands. The shell thickness and dimensions, spacing between bands, and shrinkage in the bands were selected to ensure that the longitudinal stresses in the shell (caused by beam action between bands) when combined with the circumferential tension did not result in extreme fiber stresses in excess of these values.

(6) Test pressure

The pipes usually were hydrotested to 175% to 200% of their working pressures.

(7) Factor of safety

Following is a comparison of the factor of safety (*FS*) of plain-welded steel pipe with that of pipe for pressures of static plus 25%.

(a) Forge-welded pipe:

$$FS = \frac{(48,500) (.90)}{10,000} = 4.36$$

(b) Banded pipe:

$$FS = \frac{48,500 + 64,500}{(2) (13,200)} = 4.30$$

16.10.6 Joints

Generally, for pipe shells up to 1 inch in thickness, single and double bump joints were used (see Figure 16-5). For shells with greater thicknesses, riveted band joints were provided (see Figure 16-6).

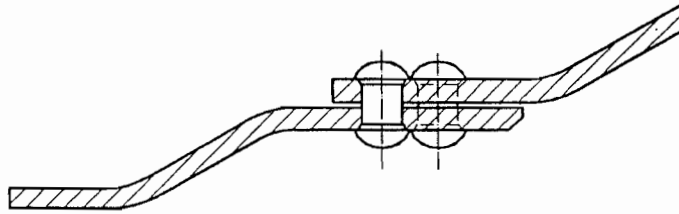


Figure 16-5 Double Bump Joint

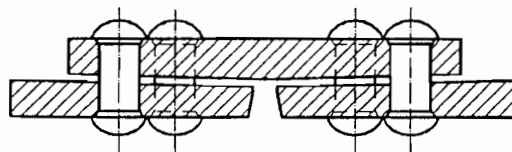


Figure 16-6 Riveted Band Joint

For banded pipe connection to specials, such as cast elbows, gasketed flanged joints were used (see Figures 16-7 and 16-8).

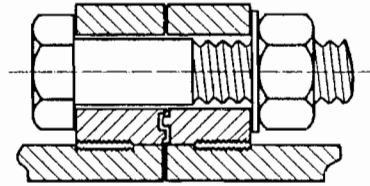


Figure 16-7 General Joint for Non-Encased Specials

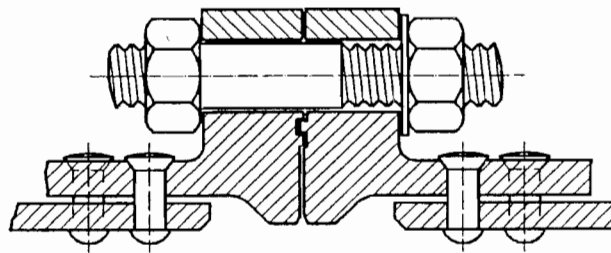


Figure 16-8 General Joint for Concrete-Encased Specials

16.10.7 Anchorage Rings

Anchorage rings usually were cast steel and manufactured in two parts for mounting on the bends (forge-welded or cast bends) or on adjacent pipes, thereby conveying the thrust to the penstock by means of rivets.

16.10.8 Expansion Joints

Expansion joints were provided immediately below each angle anchor block and where casing and sleeves were forge welded for low pressures. For higher pressures the casing was cast steel, and the sleeve was either welded or cast steel.

References*

1. Kito, F. "The Vibration of Penstocks." *Water Power* (Oct. 1959).
2. Field Bond Test for Primer and Enamel. AWWA Standard C203-86, Section 2.9. AWWA, Denver, CO (1986).

The following references are not cited in the text.

Bhave, S. K. et al. "Vibrations in Penstocks." *Water Power & Dam Construction* (Nov. 1987).

Denoor, G. and Billat, G. "High-Pressure Hooped Vessels." *Water Power* (June 1966).

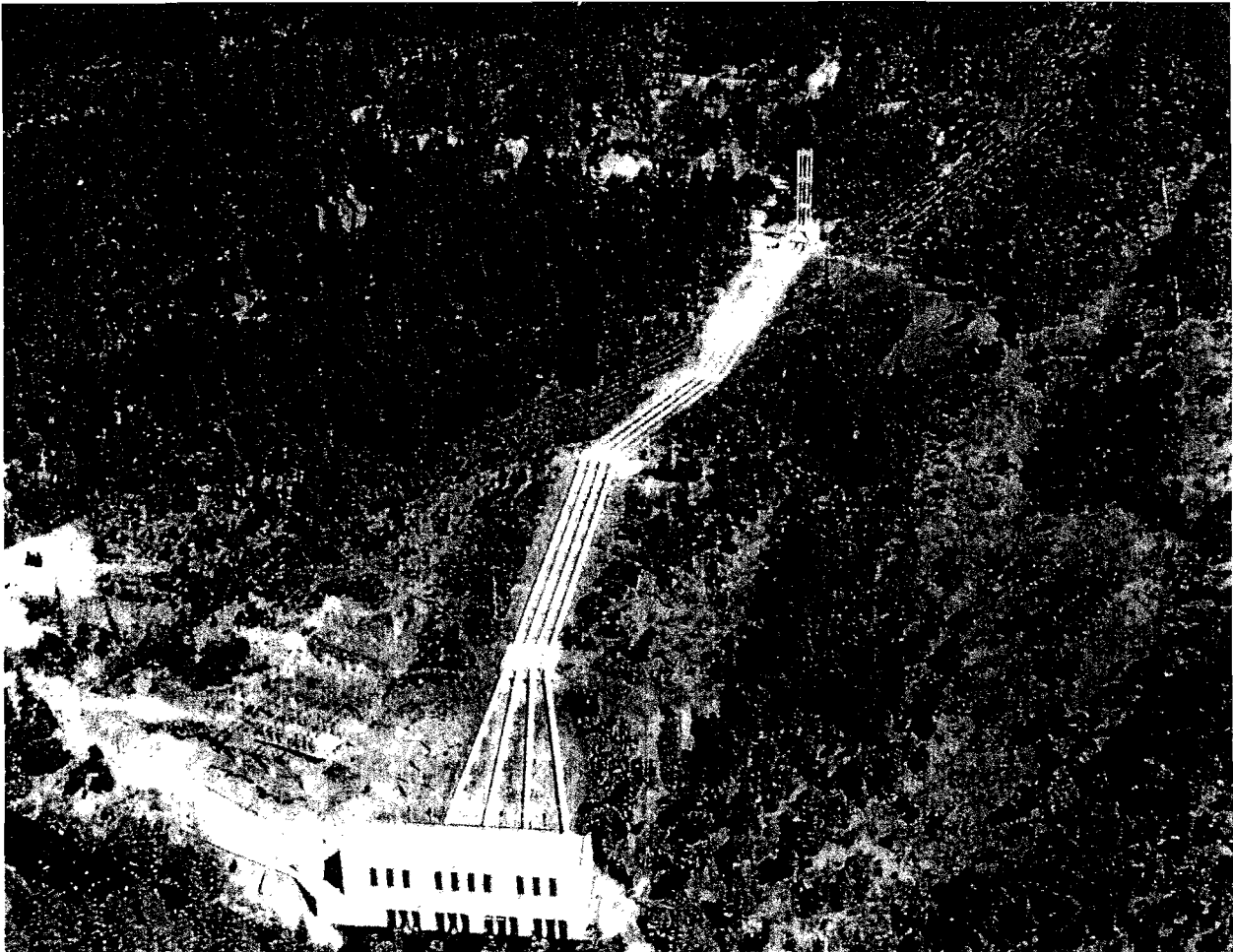
Experiences of the Bureau of Reclamation with Flow-Induced Vibrations. Hydraulics Branch, Laboratory Report No. Hyd.-538 (1964).

Fazalare, R.W. "Trends in Selecting and Procuring Hydro Turbines." *Water Power & Dam Construction* (Dec. 1986).

Mattioli, G. "Banded Pipes for Penstocks." *Water Power* (April 1962).

Shiraki, K., Nagata, O., and Homma, T. *The Penstock Vibration Characteristics—On the Vibrations of Simply Supported and Ring-Stiffened Cylindrical Shells Filled With Pressurized Water*. ASME Publication (June 1975).

* The most current version of a standard, code, or specification should be used for reference.



This section provides design examples for a hypothetical penstock project. The design calculations use the methods and formulas presented in this manual. It is assumed that the following references are readily available:

- (1) Roark, R. J. and Young, W.C. *Formulas for Stress and Strain*. McGraw-Hill Book Co., New York, NY.
- (2) "Welded Steel Penstocks." *Engineering Monograph No. 3*. US Bureau of Reclamation, Denver, CO.
- (3) Uniform Building Code. International Conference of Building Officials, Whittier, CA.

- (4) Specification for Structural Steel Buildings, Allowable Stress Design and Plastic Design, AISC, Chicago, IL.
- (5) "Welded Steel Penstocks and Tunnel Liners." *Steel Plate Engineering Data*—Vol. 4. American Iron and Steel Institute and Steel Plate Fabricators Association, Inc. (1984).

17.1 Design Criteria

17.1.1 Penstock Description

The hypothetical penstock extends from station 0 + 00 to station 13 + 00 and incorporates all three installation types (steel tunnel liner, buried pipeline, and exposed penstock) as follows:

- (1) Station 0 + 00 to station 3 + 00: Steel tunnel liner, including a transition
- (2) Station 3 + 00 to station 5 + 00: Buried penstock (under 5 feet of earth cover)
- (3) Station 5 + 00 to station 11 + 33.18: Exposed penstock (639.89 feet long)
- (4) Station 11 + 33.18 to station 11 + 64.51: Embedded in anchor block #1
- (5) Station 11 + 64.51 to station 12 + 33.34: Exposed penstock (91.45 feet long)
- (6) Station 12 + 33.34 to station 12 + 71.66: Embedded in anchor block #2
- (7) Station 12 + 71.66 to station 13 + 00: Exposed penstock (28.34 feet long)

There is a 10-foot-long transition section starting at station 0 + 00 which varies over its length from 15 feet square to 15 feet in diameter, after which the penstock diameter is a constant 15 feet up to the start of the bend at PI #3. Bend #3 is a reducing bend which varies from 15 feet to 12 feet in diameter. The remainder of the penstock to the end at station 13 + 00 is a constant 12 feet in diameter. All exposed sections are supported on ring girder type supports or saddles. Figure 17-1 shows the penstock profile used for the design examples.

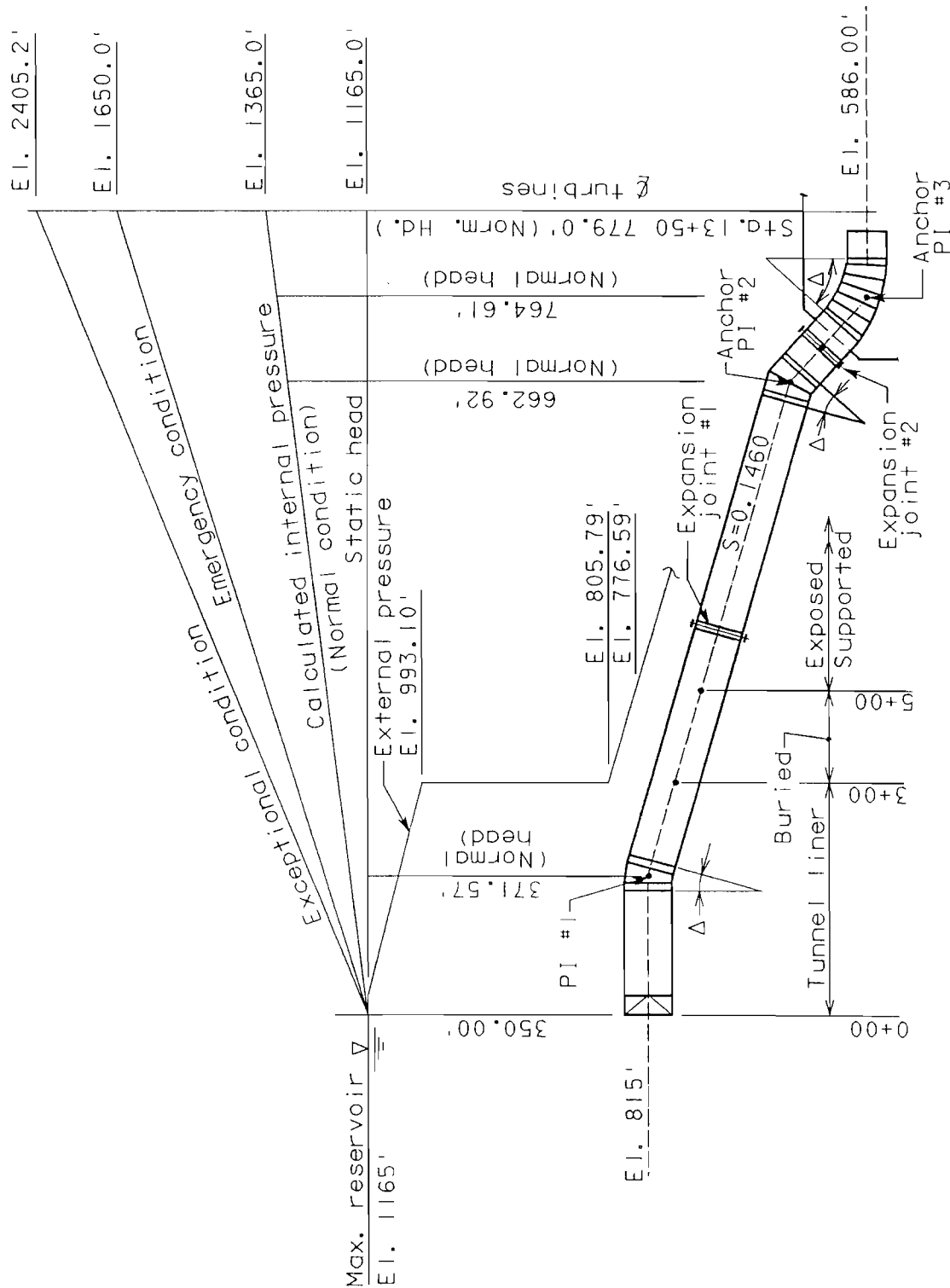


Figure 17-1 Penstock Profile Used for Example Problems

17 Penstock Design Examples

Table 17-1 gives penstock dimensions, elevations, and external pressures used in the design examples.

Table 17-1 Penstock Configuration and External Pressure

ITEM	STATION (FT)	ELEVATION (FT)	LENGTH (FT)	INSIDEDIAMETER (FT)	EXTERNAL PRESSURE (FT)
INLET	.00	815.00	.00	15.00	350.00
END TRANS.	10.00	815.00	10.00	15.00	350.00
PI #1	150.00	815.00	140.00	15.00	267.01
TUNNEL ADIT	300.00	793.10	151.59	15.00	200.00
TUNNEL ADIT	300.00	793.10	.00	15.00	15.00
END BACKFILL	500.00	763.90	202.12	15.00	15.00
EXPANSION JOINT 1	641.56	743.23	143.06	15.00	15.00
PI #2	1,150.00	669.00	513.83	15.00	15.00
EXPANSION JOINT 2	1,198.15	629.03	62.58	15.00	15.00
PI #3	1,250.00	586.00	67.38	15.00	15.00
PI #3	1,250.00	586.00	.00	12.00	15.00
END PENSTOCK	1,300.00	586.00	50.00	12.00	15.00
CENTER UNITS	1,350.00	586.00	50.00	12.00	—

Station and elevation distances are measured at the centerline of the penstock. The total penstock length to centerline of units equals 1,390.56 feet.

Table 17-2 gives the normal condition pressure gradients and required thickness. The normal condition pressure at centerline of units is 200 feet of head (34.54%); the normal condition pressure rise equals 0.1438 ft/ft (see Section 17.4).

Table 17-2 Normal Condition Pressure Gradients and Required Thickness

ITEM	INTERNAL PRESSURE (FT)			REQUIRED THICKNESS (IN.)
	STATIC HEAD	PRESSURE RISE HEAD	TOTAL HEAD	
INLET	350.00	.00	350.00	.585
END TRANS.	350.00	1.44	351.44	.587
PI #1	350.00	21.57	371.57	.621
TUNNEL ADIT	371.90	43.38	415.28	.694
TUNNEL ADIT	371.90	43.38	415.28	.694
END BACKFILL	401.10	72.45	473.55	.792
EXPANSION JOINT 1	421.77	93.02	514.79	.861
PI #2	496.00	166.93	662.93	1.108
EXPANSION JOINT 2	535.97	175.93	711.90	1.190
PI #3	579.00	185.62	764.62	1.278
PI #3	579.00	185.62	764.62	1.023
END PENSTOCK	579.00	192.81	771.81	1.032
CENTER UNITS	579.00	200.00	779.00	1.042

Table 17-3 gives the emergency condition pressure gradients and required thickness. The emergency condition pressure at centerline of units is 485 feet of head (83.77%); the emergency condition pressure rise equals 0.34877 ft/ft (see Section 17.4).

Table 17-3 Emergency Condition Pressure Gradients and Required Thickness

ITEM	INTERNAL PRESSURE (FT)			REQUIRED THICKNESS (IN.)
	STATIC HEAD	PRESSURE RISE HEAD	TOTAL HEAD	
INLET	350.00	.00	350.00	.390
END TRANS.	350.00	3.49	353.49	.394
PI #1	350.00	52.32	402.32	.448
TUNNEL ADIT	371.90	105.19	477.09	.532
TUNNEL ADIT	371.90	105.19	477.09	.532
END BACKFILL	401.10	175.68	576.78	.643
EXPANSION JOINT 1	421.77	225.58	647.35	.721
PI #2	496.00	404.80	900.80	1.004
EXPANSION JOINT 2	535.97	426.62	962.59	1.073
PI #3	579.00	450.12	1,029.12	1.147
PI #3	579.00	450.12	1,029.12	.918
END PENSTOCK	579.00	467.56	1,046.56	.933
CENTER UNITS	579.00	485.00	1,064.00	.949

Table 17-4 gives the exceptional condition pressure gradients and required thickness. The exceptional condition pressure at centerline of units is 1,240.20 feet of head (214.2%); the exceptional condition pressure rise equals 0.89185 ft/ft.

Table 17-4 Exceptional Condition Pressure Gradients and Required Thickness

ITEM	INTERNAL PRESSURE (FT)			REQUIRED THICKNESS (IN.)
	STATIC HEAD	PRESSURE RISE HEAD	TOTAL HEAD	
INLET	350.00	.00	350.00	.234
END TRANS.	350.00	8.92	358.92	.240
PI #1	350.00	133.78	483.78	.323
TUNNEL ADIT	371.90	268.98	640.88	.429
TUNNEL ADIT	371.90	268.98	640.88	.429
END BACKFILL	401.10	449.25	850.35	.569
EXPANSION JOINT 1	421.77	576.84	998.61	.668
PI #2	496.00	1,035.11	1,531.11	1.024
EXPANSION JOINT 2	535.97	1,090.92	1,626.89	1.088
PI #3	579.00	1,151.01	1,730.01	1.157
PI #3	579.00	1,151.01	1,730.01	.925
END PENSTOCK	579.00	1,195.61	1,774.61	.949
CENTER UNITS	579.00	1,240.20	1,819.20	.973

Table 17-5 gives the bend geometry at PI #1, PI #2, and PI #3.

17 Penstock Design Examples

Table 17-5 Bend Geometry

BEND NO.	CURVE RADIUS (FT)	TRUE ANGLE (DEGREES)	TAN LENGTH (FT)	MITERS (n)	LENGTH OF CURVE (FT)
PI #1	60.000	8.307	4.357	2	8.702
PI #2	60.000	31.386	16.857	4	32.919
PI #3	60.000	39.693	21.656	6	41.612

17.1.2 Materials

The following materials are used for the penstock design:

- (1) Steel penstock plate: SA-516 Grade 70 Normalized
- (2) Ring girder/supports: SA-36/SA-516 Grade 70 Normalized
- (3) Stiffeners: SA-516 Grade 70 Normalized

17.1.3 Design Parameters

Table 17-6 gives descriptions and values for parameters used in the sample penstock design.

Table 17-6 Data for Sample Design

ITEM	DESCRIPTION/VALUE
FOUNDATION MATERIAL	HARD AND BLOCKY ROCK
WELD JOINT REDUCTION FACTOR (E)	1.00
FLOW (Q)	2,600 cfs
GROUTING PRESSURE	30 psi
SEISMIC	UBC ZONE 3 FOR DESIGN BASIS CRITERIA (DBC) EARTHQUAKE
IMPORTANCE FACTOR	1.0
BACKFILL	WELL-GRADED, GRANULAR
EXPECTED DIFFERENTIAL SETTLEMENT OF A SINGLE SUPPORT	1/2 IN. MAXIMUM
RECOMMENDED SPACING OF RING GIRDER SUPPORTS	100 FT HORIZONTAL
RECOMMENDED SPACING OF SADDLE SUPPORTS	60 FT HORIZONTAL
TUNNEL LINER GAP/RADIUS RATIO	.0003
TEMPERATURE DIFFERENTIALS	100°F IN EXPOSED PENSTOCK SECTION; 40°F IN TUNNEL LINER SECTION
NORMAL CONDITION DATA	FOR DETAILED DATA, SEE TABLE 17-1 AND FIGURE 17-1.
MAXIMUM RESERVOIR ELEVATION	1,165 FT
SURGE + WATER-HAMMER PRESSURE	200 x $L_m/1,390$
INTERMITTENT CONDITION DATA	NOT SPECIFIED
EMERGENCY CONDITION DATA	REFER TO TABLE 17-1 AND FIGURE 17-1.
EXCEPTIONAL CONDITION DATA	REFER TO TABLE 17-1 AND FIGURE 17-1.

17.1.4 Welding and Nondestructive Testing

All longitudinal and circumferential seams in the penstock pressure boundary must be full-fusion butt welds. Welding must be in accordance with the requirements of Section 11.

All butt joints must be 100% radiographically examined. Joints other than butt joints that connect nonpressure attachments to pressure boundary material must be 100% magnetic-particle examined.

17 Penstock Design Examples

(6) Exceptional condition maximum pressure

$$\text{Station } 3 + 00 = 1165 - 793.1 + 268.97 = 640.87 \text{ (277.7 psi)}$$

or

$$\text{Station } 5 + 00 = 1165 - 763.9 + 449.23 = 850.33 \text{ (368.5 psi)}$$

17.1.7.2 Allowable Design Stress

For steel (ASTM A516-70), the basic allowable stress equals the lesser of:

$$2/3 \text{ yield} = (2/3) (38,000 \text{ psi}) = 25,333 \text{ psi}$$

or

$$1/3 \text{ tensile} = (1/3) (70,000 \text{ psi}) = 23,333 \text{ psi}$$

(where joint efficiency (E) = 100% for 100% radiographic inspection)

CONDITION	ALLOWABLE DESIGN STRESS INCREASE FACTOR (K)	ALLOWABLE DESIGN STRESS = (SI) (E) (K) = S _a
NORMAL OPERATING	1	23,333 psi
EMERGENCY	1.5	35,000 psi
EXCEPTIONAL	2.5	58,332 psi

17.1.7.3 Wall Thickness Calculations

(1) Handling thickness

The minimum penstock shell thickness for handling is calculated as follows:

$$\frac{D + 20}{400} = \frac{180 + 20}{400} = 0.50 \text{ in.}$$

or

$$\frac{D}{288} = \frac{180}{288} = 0.625 \text{ in.}$$

The larger value of 0.625 in. governs.

(2) Internal pressure thickness

The shell thickness for internal pressure is determined by:

$$t = \frac{PR}{S_a}$$

17.1.5 Concrete Anchor Blocks

Concrete anchor blocks must be designed in accordance with the requirements of Section 8.

17.1.6 Expansion Joints/Mechanical Bolted Couplings

Expansion joints and mechanical couplings must be designed to accommodate a longitudinal movement of 1 inch and a transverse movement of 1/2 inch in a span of 100 feet (horizontal) for ring-supported penstock sections and 60 feet (horizontal) for saddle-supported penstock sections.

17.1.7 Penstock Shell

17.1.7.1 Pressure Calculations

Following are preliminary pressure calculations:

$$(1) \text{ Normal condition pressure rise} = \frac{200 \text{ ft}}{1,390.60 \text{ ft}} = 0.1438 \text{ ft/ft}$$

$$\text{Station } 3 + 00 = (150 + 151.59) (.1438) = 43.37 \text{ ft}$$

$$\text{Station } 5 + 00 = (150 + 353.71) (.1438) = 72.43 \text{ ft}$$

- (2) Normal operating condition maximum pressure (equal to W. S. elevation – center-line elevation + pressure rise)

$$\text{Station } 3 + 00 = 1165.0 - 793.1 + 43.37 = 415.27 \text{ ft (180.0 psi)}$$

$$\text{Station } 5 + 00 = 1165.0 - 763.9 + 72.43 = 473.53 \text{ ft (205.2 psi)}$$

$$(3) \text{ Emergency condition pressure rise} = \frac{485 \text{ ft}}{1,390.60 \text{ ft}} = 0.34877 \text{ ft/ft}$$

$$\text{Station } 3 + 00 = (150 + 151.59) (.34877) = 105.19 \text{ ft}$$

$$\text{Station } 5 + 00 = (150 + 353.71) (.34877) = 175.68 \text{ ft}$$

- (4) Emergency condition maximum pressure

$$\text{Station } 3 + 00 = 1165 - 793.1 + 105.19 = 477.09 \text{ ft (206.7 psi)}$$

$$\text{Station } 5 + 00 = 1165 - 763.9 + 175.68 = 576.78 \text{ ft (249.9 psi)}$$

$$(5) \text{ Exceptional condition pressure rise} = \frac{1,240.2 \text{ ft}}{1,390.60 \text{ ft}} = 0.89185 \text{ ft/ft}$$

$$\text{Station } 3 + 00 = (150 + 151.59) (.89185) = 268.97 \text{ ft}$$

$$\text{Station } 5 + 00 = (150 + 353.71) (.89185) = 449.23 \text{ ft}$$

(6) Exceptional condition maximum pressure

$$\text{Station } 3 + 00 = 1165 - 793.1 + 268.97 = 640.87 \text{ (277.7 psi)}$$

or

$$\text{Station } 5 + 00 = 1165 - 763.9 + 449.23 = 850.33 \text{ (368.5 psi)}$$

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or

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(where joint efficiency (E) = 100% for 100% radiographic inspection)

CONDITION	ALLOWABLE DESIGN STRESS INCREASE FACTOR (K)	ALLOWABLE DESIGN STRESS = (SI) (E) (K) = S _a
NORMAL OPERATING	1	23,333 psi
EMERGENCY	1.5	35,000 psi
EXCEPTIONAL	2.5	58,332 psi

17.1.7.3 Wall Thickness Calculations

(1) Handling thickness

The minimum penstock shell thickness for handling is calculated as follows:

$$\frac{D + 20}{400} = \frac{180 + 20}{400} = 0.50 \text{ in.}$$

or

$$\frac{D}{288} = \frac{180}{288} = 0.625 \text{ in.}$$

The larger value of 0.625 in. governs.

(2) Internal pressure thickness

The shell thickness for internal pressure is determined by:

$$t = \frac{PR}{S_a}$$

Following are shell thickness calculations for stations 3 + 00 and 5 + 00:

(a) Normal operating condition

$$t = \frac{(180.2 \text{ psi})(90 \text{ in.})}{23,333 \text{ psi}} = .695 \text{ in.} \quad (\text{Station } 3 + 00)$$

$$t = \frac{(205.5 \text{ psi})(90 \text{ in.})}{23,333 \text{ psi}} = .793 \text{ in.} \quad (\text{Station } 5 + 00)$$

(b) Emergency condition

$$t = \frac{(207.1 \text{ psi})(90 \text{ in.})}{35,000 \text{ psi}} = .533 \text{ in.} \quad (\text{Station } 3 + 00)$$

$$t = \frac{(249.9 \text{ psi})(90 \text{ in.})}{35,000 \text{ psi}} = .643 \text{ in.} \quad (\text{Station } 5 + 00)$$

(c) Exceptional condition

$$t = \frac{(278.1 \text{ psi})(90 \text{ in.})}{(58,332 \text{ psi})} = .429 \text{ in.} \quad (\text{Station } 3 + 00)$$

$$t = \frac{(369.0 \text{ psi})(90 \text{ in.})}{58,332 \text{ psi}} = .569 \text{ in.} \quad (\text{Station } 5 + 00)$$

Therefore, the normal operating condition governs. Use $t = 0.695$ inch at station 3 + 00 and $t = 0.793$ inch at station 5 + 00.

17.2 Steel Tunnel Liner Design

The following calculations illustrate the methods for the design of tunnel liners subjected to external pressure. Tunnel liners initially are designed to withstand the internal pressure at each point along the penstock. The thickness determined for internal pressure then is checked for its capacity to accommodate the specified external pressure. The methods used for this check are given in Section 6. Tunnel liner thickness based on internal pressure may be adequate to accommodate the specified external pressure. If this is the case, no further analysis is required. If the liner thickness is inadequate, the following three alternate methods may be pursued.

(1) Method A

Increase the tunnel liner thickness to the point that it accommodates the specified external pressure.

17.1.5 Concrete Anchor Blocks

Concrete anchor blocks must be designed in accordance with the requirements of Section 8.

17.1.6 Expansion Joints/Mechanical Bolted Couplings

Expansion joints and mechanical couplings must be designed to accommodate a longitudinal movement of 1 inch and a transverse movement of 1/2 inch in a span of 100 feet (horizontal) for ring-supported penstock sections and 60 feet (horizontal) for saddle-supported penstock sections.

17.1.7 Penstock Shell

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Following are preliminary pressure calculations:

$$(1) \text{ Normal condition pressure rise} = \frac{200 \text{ ft}}{1,390.60 \text{ ft}} = 0.1438 \text{ ft/ft}$$

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- (2) Normal operating condition maximum pressure (equal to W. S. elevation – center-line elevation + pressure rise)

$$\text{Station } 3 + 00 = 1165.0 - 793.1 + 43.37 = 415.27 \text{ ft (180.0 psi)}$$

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$$(3) \text{ Emergency condition pressure rise} = \frac{485 \text{ ft}}{1,390.60 \text{ ft}} = 0.34877 \text{ ft/ft}$$

$$\text{Station } 3 + 00 = (150 + 151.59) (.34877) = 105.19 \text{ ft}$$

$$\text{Station } 5 + 00 = (150 + 353.71) (.34877) = 175.68 \text{ ft}$$

- (4) Emergency condition maximum pressure

$$\text{Station } 3 + 00 = 1165 - 793.1 + 105.19 = 477.09 \text{ ft (206.7 psi)}$$

$$\text{Station } 5 + 00 = 1165 - 763.9 + 175.68 = 576.78 \text{ ft (249.9 psi)}$$

$$(5) \text{ Exceptional condition pressure rise} = \frac{1,240.2 \text{ ft}}{1,390.60 \text{ ft}} = 0.89185 \text{ ft/ft}$$

$$\text{Station } 3 + 00 = (150 + 151.59) (.89185) = 268.97 \text{ ft}$$

$$\text{Station } 5 + 00 = (150 + 353.71) (.89185) = 449.23 \text{ ft}$$

(6) Exceptional condition maximum pressure

$$\text{Station } 3 + 00 = 1165 - 793.1 + 268.97 = 640.87 \text{ (277.7 psi)}$$

or

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For steel (ASTM A516-70), the basic allowable stress equals the lesser of:

$$2/3 \text{ yield} = (2/3) (38,000 \text{ psi}) = 25,333 \text{ psi}$$

or

$$1/3 \text{ tensile} = (1/3) (70,000 \text{ psi}) = 23,333 \text{ psi}$$

(where joint efficiency (E) = 100% for 100% radiographic inspection)

CONDITION	ALLOWABLE DESIGN STRESS INCREASE FACTOR (K)	ALLOWABLE DESIGN STRESS = (SI) (E) (K) = S _a
NORMAL OPERATING	1	23,333 psi
EMERGENCY	1.5	35,000 psi
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17.1.7.3 Wall Thickness Calculations

(1) Handling thickness

The minimum penstock shell thickness for handling is calculated as follows:

$$\frac{D + 20}{400} = \frac{180 + 20}{400} = 0.50 \text{ in.}$$

or

$$\frac{D}{288} = \frac{180}{288} = 0.625 \text{ in.}$$

The larger value of 0.625 in. governs.

(2) Internal pressure thickness

The shell thickness for internal pressure is determined by:

$$t = \frac{PR}{S_a}$$

Table 17-8 Allowable External Pressure (Jacobson and Amstutz)

(1) POINT	(2) STATION	(3) THICKNESS REQUIRED FOR INTERNAL PRESSURE(IN.)	(4) REQUIRED EXTERNAL PRESSURE (psi)	(5) ALLOWABLE EXTERNAL PRESSURE- JACOBSEN (psi)	(6) ALLOWABLE EXTERNAL PRESSURE- AMSTUTZ (psi)
1	0 + 10	.588	151.7	40.7	48.7
2	0 + 50	.598	144.4	42.0	50.4
3	1 + 00	.610	128.6	43.3	52.4
4	1 + 50	.622	115.7	45.3	54.5
5	2 + 00	.647	106.0	48.7	59.0
6	2 + 50	.671	96.4	52.7	63.5
7	3 + 00	.696	86.7	56.0	68.2

(1) Jacobsen formulas

The three equations that must be solved simultaneously for the three unknowns α , β , and p are:

$$r/t = \sqrt{\frac{[(9\pi^2/4 \beta^2) - 1] [\pi - \alpha + \beta (\sin \alpha / \sin \beta)^2]}{12 (\sin \alpha / \sin \beta)^3 [\alpha - (\pi \Delta/r) - \beta (\sin \alpha / \sin \beta) [1 + \tan^2 (\alpha - \beta)/4]]}}$$

$$p/E^* = \frac{(9/4) (\pi / \beta)^2 - 1}{12 (r/t)^3 (\sin \alpha / \sin \beta)^3}$$

$$\sigma_y/E^* = (t/2r) [1 - (\sin \beta / \sin \alpha)] + (pr \sin \alpha / E^* t \sin \beta) \left[1 + \frac{4\beta r \sin \alpha \tan (\alpha - \beta)}{\pi t \sin \beta} \right]$$

Where:

- α = one-half of the angle subtended to the center of the cylindrical shell by the buckled lobe
- β = one-half the angle subtended by the new mean radius through the half waves of the buckled lobe
- p = critical external buckling pressure, psi
- Δ/r = gap ratio, for gap between steel and concrete
- t = plate thickness, in.
- E^* = modified modulus of elasticity of steel liner = $E/(1 - \nu^2)$, where ν = Poisson's ratio for steel
- r = tunnel liner internal radius, in.

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(2) Amstutz formulas

$$\frac{\sigma_N - \sigma_v}{\sigma_F^* - \sigma_N} \left[\left(\frac{r}{i} \right) \sqrt{\frac{\sigma_N}{E^*}} \right]^3 \approx 1.73 \left(\frac{r}{e} \right) \left[1 - 0.225 \left(\frac{r}{e} \right) \frac{\sigma_F^* - \sigma_N}{E^*} \right]$$

$$P_{cr} \approx \left(\frac{F}{r} \right) \sigma_N \left(1 - 0.175 \left(\frac{r}{e} \right) \frac{\sigma_F^* - \sigma_N}{E^*} \right)$$

Where:

- $i = t\sqrt{12}$
- $e = t/2$
- $\sigma_v = -(k/r) E^*$
- $k/r =$ gap ratio, for gap ratio between steel and concrete
- $r =$ tunnel liner radius $= D/2$
- $D =$ tunnel liner diameter
- $t =$ plate thickness
- $E =$ modulus of elasticity
- $\sigma_F =$ yield strength
- $\sigma_N =$ circumferential axial stress in plate liner ring
- $\mu = 1.5 - 0.5 \left[\frac{1}{1 + 0.002 (E/\sigma_y)} \right]^2$
- $\sigma_F^* = \frac{\mu \sigma_F}{\sqrt{1 - \nu + \nu^2}}$
- $F =$ total cross-sectional area of ring between stiffeners $= t$
- $E^* = E/(1 - \nu^2)$
- $\nu =$ Poisson's ratio

Following are sample computer calculations for both the Jacobsen and Amstutz methods. These formulas may be readily solved using the computer codes presented in Reference 1. The calculations are for station 0 + 10.

(1) Jacobsen method calculations

α	β	p	$\Delta\alpha$	$\Delta\beta$	Δp
0.3519	0.3028	168.5	0.0519	0.0028	167.5376
0.4182	0.3935	97.5	0.0663	0.0907	-71.0171
0.4761	0.4538	60.7	0.0579	0.0603	-36.8583
0.4896	0.4647	61.2	0.0136	0.0109	0.5598

Data used in computations:

- $R = 90$ in.
- $\sigma_F = 38,000$
- Gap ratio $= .0003$
- $E = 30,000,000$

$$t = 0.588 \text{ in.}$$

$$FS = 1.5$$

For $t = 0.588 \text{ in.}$ and $R = 90 \text{ in.}$:

Critical buckling pressure = 61 psi
Allowable buckling pressure = 40.7 psi

(2) Amstutz method calculations

D/t	σ_N	P_α	ϵ
306.1224	12,138.5	73.0	10.2232

Data used in computations:

$$R = 90 \text{ in.}$$

$$\sigma_F = 38,000$$

$$\text{Gap ratio} = .0003$$

$$E = 30,000,000$$

$$t = 0.588 \text{ in.}$$

$$FS = 1.5$$

For $t = 0.588 \text{ in.}$ and $R = 90 \text{ in.}$:

Critical buckling pressure = 73 psi
Allowable buckling pressure = 48.7 psi

The results of both the Jacobsen and Amstutz calculations indicate that the liner thicknesses based on internal pressure cannot accommodate the specified external pressure. It is necessary to increase the thickness of the tunnel liner until the capacity of the liner matches the required external pressure. Table 17-9 indicates the required thickness.

Table 17-9 Thickness Required for External Pressure

(1) POINT	(2) STATION	(3) RECOMMENDED THICKNESS (IN.)	(4) REQUIRED EXTERNAL PRESSURE (psi)	(5) ALLOWABLE EXTERNAL PRESSURE—JACOBSEN (psi)	(6) ALLOWABLE EXTERNAL PRESSURE—AMSTUTZ (psi)
1	0 + 10	1.21	151.7	152.7	192.7
2	0 + 50	1.17	144.4	144	181.5
3	1 + 00	1.10	128.6	129.3	162.3
4	1 + 50	1.03	115.7	115.3	143.9
5	2 + 00	.99	106.0	107.4	133.8
6	2 + 50	.93	96.4	96	119.1
7	3 + 00	.88	86.7	86.7	107.3

As discussed in Section 6, the use of the Jacobsen formulas results in a lower allowable external pressure for a given thickness.

Each of the calculated allowable external pressures then is used to determine the stress level in the shell. These stress levels are based on the critical pressure, i.e., the allowable external pressure multiplied by the factor of safety. If the resulting stress is greater than 0.8 times the yield strength of the tunnel liner material, the value of E^* used in the Jacobsen or Amstutz formulas must be replaced by E_T^* using the formula:

$$E_T^* = E^* \left\{ 1 - \left[\frac{(\sigma_m - 0.8 \sigma_F)^2}{0.2 \sigma_F} \right] \right\}$$

Where:

$$\sigma_m = \frac{(P_{cr})(R)(FS)}{t}$$

The values given in Table 17-9 are less than 0.8 times yield strength. For example, at station 0 + 10:

$$\text{Stress} = \frac{(152.7)(1.5)(90)}{1.21} = 17,037 \text{ psi}$$

This value is less than 0.8 times the yield strength, σ_F .

17.2.2 Method B (Add Stiffeners)

Having established that the unstiffened tunnel liner thicknesses based on internal pressure are inadequate, the stiffener spacing needed to render the system acceptable can be determined. To do this, the designer must consider the following two types of buckling criteria: (1) rotary symmetric buckling, and (2) single lobe buckling.

17.2.2.1 Rotary Symmetric Buckling

The equations for rotary symmetric buckling are presented below.

(1) R. von Mises equation for rotary symmetric buckling:

$$P_{cr} = \frac{E(t/r)}{1 - \nu^2} \left[\frac{1 - \nu^2}{(n^2 - 1) \left(\frac{n^2 L^2}{\pi^2 r^2} + 1 \right)} \right] + \frac{E(t/r)^3}{12(1 - \nu^2)} \left(n^2 - 1 + \frac{2n^2 - 1 - \nu}{\frac{n^2 L^2}{\pi^2 r^2} - 1} \right)$$

Where:

- r = radius to neutral axis of shell in original formulation (for practical purposes, radius to outside of shell)
- L = length of tube between stiffeners, i.e., center-to-center spacing of stiffeners
- t = thickness of shell
- P_{cr} = collapsing pressure for FS = 1.0
- E = modulus of elasticity

- ν = Poisson's ratio
 n = number of lobes or waves in the complete circumference at collapse

(2) Donnell¹ equation for rotary symmetric buckling:

$$P_{cr} = \frac{EI_s}{R^3} \left[\frac{(n^2 + \lambda^2)^2}{n^2} \right] + \frac{Et}{R} \left[\frac{\lambda^4}{n^2 (n^2 + \lambda^2)^2} \right]$$

Where:

- E = Young's modulus
 I_s = shell bending stiffness = $\frac{t^3}{12(1 - \nu^2)}$
 ν = Poisson's ratio
 R = shell radius
 t = shell thickness
 λ = $\pi R/L$
 L = length of panel (stiffener spacing)
 n = number of circumferential lobes

To present a suitable comparison of methods, this sample calculation is based on a single stiffener cross section, thereby reducing the number of variables to the shell thickness and stiffener spacing. The stiffener size selected is 6 inches high by 7/8 inch thick. The stiffener spacing is the remaining variable.

Both the von Mises and Donnell formulas require that successive values of n be used to find the minimum value of P_{cr} . The value of n used in the von Mises formula may also be determined by formula. The values of n may not be the same for the two formulas.

Table 17-10 gives allowable external pressures based on a maximum stiffener spacing shown for each thickness.

Table 17-10 Allowable External Pressures for Stiffener Spacing

POINT	STATION	REQUIRED EXTERNAL PRESSURE (psi)	SHELL THICKNESS (IN.)	STIFFENER SPACING (IN.)	ALLOWABLE EXTERNAL PRESSURE— VON MISES (psi)	ALLOWABLE EXTERNAL PRESSURE— DONNELL (psi)
1	0 + 10	151.7	.588	48	152.6	147.3
2	0 + 50	144.4	.598	50	151.5	146.0
3	1 + 00	128.6	.610	58	132.4	128.9
4	1 + 50	115.7	.622	66	118.5	116.9
5	2 + 00	106.0	.647	76	110.5	109.5
6	2 + 50	96.4	.671	90	99.6	99.1
7	3 + 00	86.7	.696	108	89.2	89.1

17.2.2.2 Single Lobe Buckling

Shell thicknesses, stiffener size, spacing, and the other parameters previously described are used to determine the single lobe buckling capacity using the Jacobsen formulas shown below.

$$r/\sqrt{(12J/F)} = \left\{ \frac{[(9\pi^2/4\beta^2) - 1] [\pi - \alpha + \beta (\sin \alpha / \sin \beta)^2]}{12 (\sin \alpha / \sin \beta)^3 [\alpha - (\pi \Delta / r) - \beta (\sin \alpha / \sin \beta) [1 + \tan^2 (\alpha - \beta)/4]]} \right\}^{1/2}$$

$$(p/EF)\sqrt{(12J/F)} = \frac{[(9\pi^2/4\beta^2) - 1]}{(r^3 \sin^3 \alpha) / [(J/F)\sqrt{(12J/F)} \sin^3 \beta]}$$

$$\sigma_y/E = \frac{h \sqrt{(12J/F)}}{r \sqrt{(12J/F)}} \left(1 - \frac{\sin \beta}{\sin \alpha} \right) + \frac{p \sqrt{(12J/F)} r \sin \alpha}{EF \sqrt{(12J/F)} \sin \beta} \left[1 + \frac{8\beta h r \sin \alpha \tan (\alpha - \beta)}{\pi \sqrt{(12J/F)} \sqrt{(12J/F)} \sin \beta} \right]$$

Where:

- α = one-half of the angle subtended to the center of the cylindrical shell by the buckled lobe
- β = one-half the angle subtended by the new mean radius through the half waves of the buckled lobe
- p = critical external buckling pressure, psi
- Δ/r = gap ratio, for gap between steel and concrete
- r = tunnel liner radius to the centroid of the stiffened section used to calculate J
- E = modulus of elasticity of steel liner and stiffener material
- J = moment of inertia of the stiffener and $1.56\sqrt{rt}$ of the cylindrical shell
- F = area of the stiffener and the cylindrical shell
- h = distance from the centroid of the stiffened section and the extreme fiber of the stiffened section

These formulas may be readily solved using the computer codes presented by Moore.¹ The solutions are given in Table 17-11.

Table 17-11 Single Lobe Buckling Capacity

POINT	STATION	SHELL THICKNESS (IN.)	STIFFENER SPACING (IN.)	CALCULATED EXTERNAL ALLOWABLE PRESSURE (psi)	REQUIRED EXTERNAL PRESSURE (psi)	COMMENTS
1	0 + 10	.588	48	149.3	151.7	MARGINAL
2	0 + 50	.598	50	148.7	144.4	ACCEPTABLE
3	1 + 00	.610	58	140.7	128.6	ACCEPTABLE
4	1 + 50	.622	66	134.7	115.7	ACCEPTABLE
5	2 + 00	.647	76	130	106	ACCEPTABLE
6	2 + 50	.671	90	122	96.4	ACCEPTABLE
7	3 + 00	.696	108	114	86.7	ACCEPTABLE

A sample computer solution for station 0 + 10 is given below.

α	β	p	$\Delta\alpha$	$\Delta\beta$	Δp
0.7087	0.7169	1148.8	0.2087	0.2169	55,139.3300
1.0968	1.0969	-206.9	0.3881	0.3800	-65,073.5800
1.3132	1.2995	427.8	0.2164	0.2027	30,469.6900
1.5316	1.5294	302.4	0.2185	0.2299	-6,021.7750
1.7967	1.7932	207.6	0.2651	0.2638	-4,551.8570
1.8314	1.8286	223.9	0.0347	0.0353	782.4428
1.8288	1.8261	224.8	-0.0026	-0.0024	45.8607
1.8287	1.8260	224.8	-0.0001	-0.0001	1.3840
1.8287	1.8260	224.8	0.0000	0.0000	-0.0714

Data used in computations:

Shell thickness/stiffener size = 0.588/(6 × 7/8)

$J = 48.88$

$F = 33.474$

$H = 4.904$

$R = 91.684$

$\sigma_y = 38,000$

$E = 30,000,000$

Gap ratio = .0003

$L = 48$

$FS = 1.5$

For station 0 + 10:

P_{cr} = critical buckling pressure = 224 psi

P_{all} = allowable buckling pressure = 149.3 psi

In addition, a calculation of the average compression stress (σ_{ave}) is used to determine the validity of the value of E (for station 0 + 50) as follows:

$$\begin{aligned}\sigma_{ave} &= \frac{(P_{all}) (FS) (R) (L)}{(t_{shell}) (1.56\sqrt{rt}) + A_s} \\ &= \frac{(148.1) (1.5) (90) (50)}{(.598) [1.56 \sqrt{(90) (.598)}] + 6 (7/8)} \\ &= 28,549 \text{ (which is less than } 0.8 \times 38,000\text{)}\end{aligned}$$

If the value exceeds 0.8 times 38,000, the analysis must be rerun using the adjusted value for E given in Section 17.2.1.

17.2.3 Method C (Increase Tunnel Liner Thickness; Add External Stiffening)

In this method, an arbitrary thickness is selected for the entire length of stiffened tunnel liner, and the stiffener spacing is adjusted to accommodate the required external pressure at each station. Obviously, the thickness selected must be adequate to meet the internal pressure thickness requirements for all points. In this example, the thickness selected is equal to the maximum thickness required for internal pressure, $t = 0.696$ inch. After selecting the thickness, the steps used in Method B are followed.

The stiffener spacing that meets the required external pressure is determined using the rotary symmetric formulas of von Mises and Donnell. Table 17-12 gives results based on a tunnel liner thickness of 0.696 inch.

Table 17-12 Stiffener Spacing for Required External Pressure

POINT	STATION	SHELL THICKNESS (IN.)	STIFFENER SPACING (IN.)	REQUIRED EXTERNAL PRESSURE (psi)	CALCULATED ALLOWABLE EXTERNAL PRESSURE—VON MISES (psi)	CALCULATED ALLOWABLE EXTERNAL PRESSURE—DONNELL (psi)
1	0+10	.696	60	151.7	178.5	174.3
2	0+50	.696	68	144.4	153.0	150.1
3	1+00	.696	76	128.6	134.1	132.7
4	1+50	.696	84	115.7	119	117.9
5	2+00	.696	90	106	109.5	108.9
6	2+50	.696	100	96.4	98.2	98.2
7	3+00	.696	108	86.7	89.2	89.0

After determining the stiffener spacing for the $t = 0.696$ -inch design, the single lobe theory capacity is checked, using the Jacobsen formulas. These formulas may be readily solved using the computer codes presented by Moore.¹ These solutions are given in Table 17-13.

Table 17-13 Check of Single-Lobe Theory Capacity

POINT	STATION	SHELL THICKNESS (IN.)	STIFFENER SPACING (IN.)	CALCULATED EXTERNAL ALLOWABLE PRESSURE (psi)	REQUIRED EXTERNAL PRESSURE (psi)	COMMENTS
1	0+10	.696	60	153.3	151.7	ACCEPTABLE
2	0+50	.696	68	144.7	144.4	ACCEPTABLE
3	1+00	.696	76	137.3	128.6	ACCEPTABLE
4	1+50	.696	84	130.7	115.7	ACCEPTABLE
5	2+00	.696	90	126.0	106	ACCEPTABLE
6	2+50	.696	100	119.3	96.4	ACCEPTABLE
7	3+00	.696	108	114	86.7	ACCEPTABLE

Note that there is some excess capacity shown in Table 17-12 (the rotary symmetric results) for the first two points. The stiffener spacing was governed by the results of the single lobe theory analysis. The spacing of the stiffeners at these two points could be increased by increasing the height of the stiffener.

17.3 Buried Penstock

17.3.1 External Pressure Design

The following data is given for the external pressure design of the portion of the penstock that is buried.

ITEM	DESCRIPTION/VALUE
STATION	3 + 00
PENSTOCK CENTERLINE ELEVATION	793.10 FT
INTERNAL PRESSURE (NORMAL CONDITION)	415.28 FT
MINIMUM THICKNESS FOR INTERNAL PRESSURE	0.696 IN.
INSIDE DIAMETER (D)	15.00 FT
SOIL COVER (H)	5.00 FT
PENSTOCK LENGTH (21 RINGS @ 9.625 FT)	202.12 FT
EXTERNAL PRESSURE (VACUUM)	15.00 FT (6.5 psi)

17.3.1.1 Deflection of Buried Penstock

Deflection (Δx) is calculated as follows (from AWWA Manual M-11,² Equation 6-5).

$$\Delta x = D_l \left[\frac{K W r^3}{E I + 0.061 E' r^3} \right]$$

Where:

- x = horizontal deflection, in.
(allowable deflection = $.05D = .05 \times 180 \text{ in.} = 9.0 \text{ in.}$)
- D = penstock inside diameter, in. = 180 in.
- D_l = deflection lag factor = 1.1
- K = bedding constant = 0.1
- E' = modulus of soil reaction = 2,000 psi for coarse-grained soils with little or no fines/GW, GP, SW, SP (United Classification System) containing less than 12% fines. Bedding compaction moderate 80-95% Proctor, 40-70% relative density.
- W = load per unit of shell length ($W_c + W_L$)
- t = 0.694 in. (from Section 17.1.7.3 (2a))
- r = mean radius = 90.348 in.
- I = transverse moment of inertia per unit length of shell wall = $bt^3/12 = 1 \times .694^3/12 = 0.02786 \text{ in.}^4$
- E = modulus of elasticity = 29×10^6 psi for steel

Assume:

- W_c = $wHB_c/12$ = dead load (lb/in.) = $120 \text{ lb/ft}^3 \times 5.0 \text{ ft} \times 15.12 \text{ ft}/12 = 756 \text{ lb/in.}$
- H = cover over top of pipe = 5.0 ft
- B_c = pipe outer diameter = 181.390 in. = 15.116 ft
- w = unit weight of backfill = 120 lb/ft^3

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$$\begin{aligned}
 W_L &= \text{unit weight of live load} = \text{HS-20 Highway Loading (from Table 6-3 of} \\
 &\quad \text{AWWA Manual M-11}^2) = 250 \text{ lb/ft}^2 \times 15.12 \text{ ft/12} = 316 \text{ lb/in.} \\
 W &= 756 \text{ lb/in.} + 316 \text{ lb/in.} = 1,072 \text{ lb/in.}
 \end{aligned}$$

Then deflection equals:

$$1.1 \left(\frac{(.1) (1,072) (737,489)}{(29,000,000) (.02786) + (.061) (2,000) (737,489)} \right) = 0.96 \text{ in.}$$

Since the deflection is less than 9.0 inches, it is acceptable.

17.3.1.2 Buckling of Buried Penstock

The allowable buckling pressure (q_a) is calculated as follows from AWWA Manual M-11,² Equation 6-7. Note that this equation is valid under the following conditions:

- 1) Without internal vacuum: $2 \text{ ft} \leq h \leq 80 \text{ ft}$
- 2) With internal vacuum: $4 \text{ ft} \leq h \leq 80 \text{ ft}$

$$q_a = \left(\frac{1}{FS} \right) \left(32 R_w B' E' \frac{EI}{D^3} \right)^{1/2}$$

Where:

- q_a = allowable buckling pressure, psi
- h = height of ground surface above top of pipe, in.
- D = diameter of pipe, in.
- FS = design factor = 2.5 for $(h/D) \geq 2$ and 3.0 for $(h/D) < 2$
- h_w = height of water surface above top of pipe, in.
- R_w = water buoyancy factor = $1 - 0.33 (h_w/h)$, $0 \leq h_w \leq h$
- B' = empirical coefficient of elastic support (dimensionless)
- $B' = \frac{1}{1 + 4e^{(-0.065H)}}$
(where H = height of fill above pipe, ft)
- E' = modulus of soil reaction, psi (from Tables 6-1 and 6-2 of AWWA Manual M-11²)
- E = Young's modulus, psi
- I = moment of inertia of penstock wall per unit length, in.⁴
- EI = pipe wall stiffness, lb-in.²/in.
- t = wall thickness, in.

Assume:

- h_w = 0
- R_w = 1.0
- H = 7.5 ft
- B' = 0.257

$$\begin{aligned}
D &= 180 \text{ in.} \\
E &= 29 \times 10^6 \text{ psi} \\
E' &= 2,000 \text{ psi} \\
t &= 0.694 \text{ in.} \\
I &= t^3/12 = 0.02786 \text{ in.}^4 \\
EI &= 811,420 \text{ psi/in.} \\
FS &= 3.0 \\
t &= 0.694 \text{ in.}
\end{aligned}$$

Then:

$$q_a = \left(\frac{1}{3.0} \right) \left[(32) (1.0) (0.257) (2,000) \frac{(811,420)}{180^3} \right]^{1/2} = 15.9 \text{ psi}$$

The total external load on the penstock is calculated as follows (from AWWA Manual M-11,³ Equation 6-8). Note that the simultaneous application of live loads and internal vacuum is normally not considered.

$$\gamma_w h_w + R_w \frac{W_c}{D} + P_v \leq q_a$$

When live loads are considered, the following formula is used:

$$\gamma_w h_w + R_w \frac{W_c}{D} + \frac{W_L}{D} \leq q_a$$

Where:

$$\begin{aligned}
\gamma_w &= \text{specific weight of water} = 0.0361 \text{ lb/in.}^3 \\
h_w &= \text{height of water surface above pipe} = 0 \text{ in.} \\
P_v &= \text{internal vacuum pressure} = 6.50 \text{ psi} \\
W_c &= \text{soil dead load} = 756 \text{ lb/in.} \\
R_w &= \text{water buoyancy factor} = 1.0 \\
W_L &= \text{live load} = 174 \text{ lb/in.} \\
D &= \text{diameter of pipe, in.}
\end{aligned}$$

Then total load equals:

$$0.0361 (0) + \frac{(1)756}{180} + 6.50 = 10.7 \text{ psi}$$

and live load equals:

$$0.0361 (0) + \frac{(1)756}{180} + \frac{174}{180} = 5.1 \text{ psi}$$

Since 10.7 psi is less than q_a (16.0 psi), the equation is satisfied. Therefore, the 0.695-inch thickness is adequate for the external loading condition, including vacuum.

17.4 Exposed/Supported Penstock

This section provides sample design calculations for ring girders, saddles, stiffeners, bends, and transitions.

17.4.1 Ring Girder Design

Figure 17-2 illustrates a simplified ring girder.

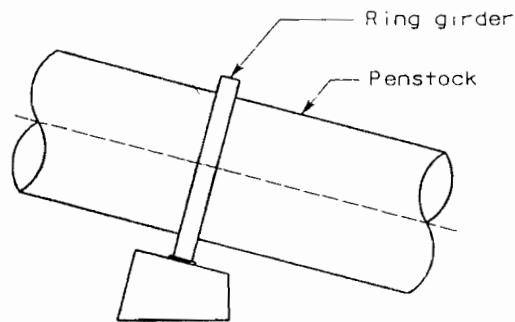


Figure 17-2 Ring Girder

The following calculations illustrate the methods for determining ring girder size and configuration. In addition to the given data in Section 17.1.1, the following data taken from the 1991 UBC is used in the calculations:

$R_w = 12$; S_1 soil profile; importance factor = 1.0; basic wind speed = 80 mph; Exposure C; shape factor $C_q = 0.8$

Ring girder size and configuration are determined for the following service load conditions:

- (1) Normal condition—service load combination 2, Table 3-1

$$P_{N2} = (663) (0.433) = 287 \text{ psi}$$

- (2) Intermittent condition—service load combination 3, Table 3-1

$$P_{N2} = 287 \text{ psi and EQ1} = \text{Design Basis Criteria (DBC) earthquake}$$

Note that snow and/or ice live loads have been omitted from the load calculations.

17.4.1.1 Preliminary Wall Thickness Calculations

First, it is necessary to determine normal condition shell thickness (t_n).

For $\sigma_n = 23,300$ psi:

$$t_n = \frac{P_n D}{2 \sigma_n E} = \frac{(287)(180)}{(2)(23,300)(1.0)} = 1.109 \text{ in.}$$

Where:

- σ_n = shell stress, normal conditions, psi
- t_n = shell thickness, normal conditions, in.
- P_n = design pressure, normal conditions, psi
- D = inside diameter, in.
- E = joint efficiency

17.4.1.2 Steel Pipe Properties and Stresses

Next, it is necessary to determine pipe properties, including section modulus, bending stress, longitudinal stresses, and axial stresses. Assume the penstock shell is thickened at the ring girder to 2 inches to handle all local stresses.

(1) Pipe properties

- (a) Penstock between ring girders: 180-inch ID x 1 1/8 inches

$$\text{Steel weight} = [(182.25)^2 - (180)^2] [(0.7854)(12)/1728] [490] = 2,178 \text{ lb/lin ft}$$

$$\text{Water weight} = (15)^2 (0.7854) (62.4) = 11,027 \text{ lb/lin ft}$$

$$\text{Total weight (w)} = 2,178 + 11,027 = 13,205 \text{ lb/lin ft}$$

The section modulus (S) of the pipe shell equals:

$$S = \pi r^2 t = \pi (90)^2 1.125 = 28,628 \text{ in.}^3$$

The cross-sectional area (A) of the pipe shell equals:

$$A = [(182.25)^2 - (180)^2] (0.7854) = 640 \text{ in.}^2$$

- (b) Penstock at ring girder: 180-inch ID x 2 inches

$$\text{Steel weight} = 3,891 \text{ lb/lin ft}$$

$$\text{Water weight} = 11,027 \text{ lb/lin ft}$$

$$\text{Total weight} = 3,891 + 11,027 = 14,918 \text{ lb/lin ft}$$

The section modulus (S) of the pipe shell equals:

$$S = \pi r^2 t = \pi (90)^2 (2) = 50,894 \text{ in.}^3$$

The cross-sectional area (A) of the pipe shell equals:

$$A = (184^2 - 180^2) (0.7854) = 1,143 \text{ in.}^2$$

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(2) Bending stress

Assume the maximum moment from penstock beam bending occurs at the ring girder supports (see Figure 17-3). The bending moment (M_l) is approximated by:

$$M_l = \frac{wl^2}{9}$$

Where:

- M_l = bending moment, in.-lb
- l = span, ft
- w = unit load, lb/ft

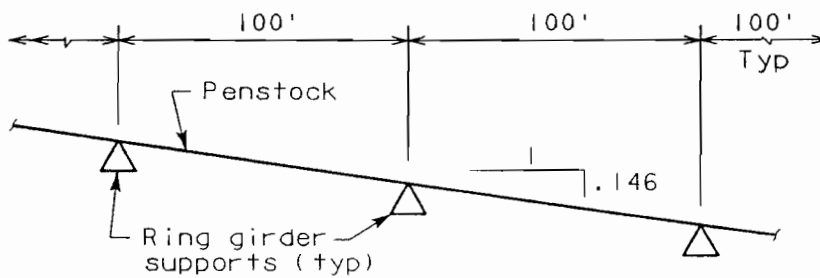


Figure 17-3 Penstock Support

The penstock slope equals: $\tan^{-1}(0.146) = 8.306^\circ$.

The bending moment (M_l) then equals:

$$M_l = \left(\frac{13,205}{\cos 8.306} \right) \frac{(100)^2 (12)}{9} = 177,933,290 \text{ in.-lb}$$

Bending stress (f_l) equals:

$$f_l = \frac{M_l}{S} = \frac{177,933,290 \text{ in.-lb}}{50,894 \text{ in.}^3} = 3,496 \text{ psi}$$

(3) Longitudinal stress due to gravity (f_l)

Refer to Figure 17-4.

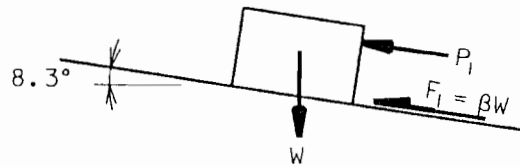


Figure 17-4 Free-body Diagram

Assume the coefficient of friction (β) = 0.05 and the total load (W) is for 500 feet of penstock (steel + water weight).

The sum of the forces (ΣF) parallel to the slope must equal zero:

$$\Sigma F = 0 = F_1 + P_1 - W \sin \alpha$$

Where:

- F_1 = frictional force resisting gravity, lb
- P_1 = force due to gravity, lb = $W \sin \alpha - F_1 = 0.144 W - 0.05 W = 0.0944 W$
- W = total load, lb
- α = penstock slope, degrees

Then the longitudinal stress due to gravity (f_1) equals:

Note that the minus sign (–) indicates compression stress and the plus sign (+) indicates tension stress.

$$f_1 = \frac{P_1}{A} = \frac{-629,873}{1,143} = -551 \text{ psi}$$

(4) Longitudinal stress due to frictional force at expansion joint (f_2)

Assume the frictional force at the expansion joint equals 500 pounds per diameter inch.

Then:

$$P_2 = 15 (12) \times 500 = -90,000 \text{ lb}$$

$$f_2 = \frac{P_2}{A} = \frac{-90,000}{1,143} = -79 \text{ psi}$$

(5) Longitudinal stress due to pressure at end of expansion joint (f_3)

First determine the normal pressure at the expansion joint.

The hydraulic grade line at the expansion joint equals:

$$\left(\frac{\Delta y}{\Delta x}\right)x + 1,165 = \left(\frac{1,365 - 1,165}{1,350}\right)642 + 1,165 = 1,260 \text{ ft}$$

Then:

$$P_n = (1,260 - 742) 0.433 = 225 \text{ psi}$$

$$P_3 = P_n A = 225 (640) = -144,000 \text{ lb}$$

And:

$$f_3 = \frac{P_3}{A} = \frac{-144,000}{1,143} = -126 \text{ psi}$$

(6) Total axial stress

The total axial stress equals:

$$f_1 + f_2 + f_3 = (-551) + (-79) + (-126) = -756 \text{ psi}$$

17.4.1.3 Calculations for Ring Girder Section Properties

Figure 17-5 shows the section of a ring girder. Following are values for A_i (cross-sectional area of ring segments) and y_i (distance from inside of shell to centroids of ring).

$$A_1 = 28 \text{ in.}^2, y_1 = 13 \text{ in.}$$

$$A_2 = A_3 = 10 \text{ in.}^2, y_2 = y_3 = 7 \text{ in.}$$

$$A_4 = 2 \times 34.94 \text{ in.} = 69.84 \text{ in.}^2, y_4 = 1 \text{ in.}$$

$$A_1 + A_2 + A_3 + A_4 = 117.84 \text{ in.}^2$$

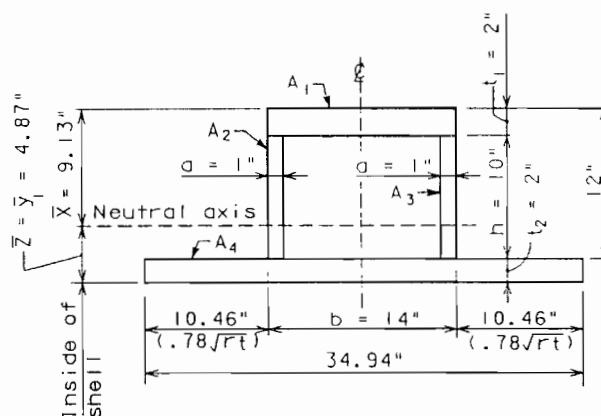


Figure 17-5 Section of Ring Girder

The distance ($\bar{y}_1$) from the inside of the shell to the neutral axis of the ring girder equals:

$$\bar{y}_1 = \bar{z} = \frac{\sum y_i A_i}{\sum A_i} = \frac{28(13) + (2)(10)(7) + (69.84)(1)}{177.84} = 4.87 \text{ in.}$$

The ring girder moment of inertia (I_R) equals:

$$I_R = \sum \frac{b_i h_i^3}{12} + \sum d_i^2 A_i$$

Where:

- $\bar{z}$ = $\bar{y}_1$ = distance from inside of shell to neutral axis of ring girder, in.
- $\bar{x}$ = distance from outside surface of ring girder to neutral axis of ring girder, in.
- b_i = width of ring segments, in.
- h_i = height of ring segments, in.
- d_i = distance from centroid of ring segments to neutral axis of ring girder, in.
- I_R = moment of inertia of ring girder, in.⁴
- A_i = cross-sectional area of ring segments, in.²

Then:

$$I_R = \frac{14(2)^3}{12} + \frac{(2)(1)(10)^3}{12} + \frac{34.94(2)^3}{12} + (4.87-1)^2 (69.84) + 2(7-4.87)^2 (10) + (13-4.87)^2 (28) = 3,188 \text{ in.}^4$$

The section modulus (S_{out}) of the outside of the ring girder equals:

$$S_{out} = \frac{I_R}{\bar{x}} = \frac{3,188}{14-4.87} = 349 \text{ in.}^3$$

The section modulus (S_{in}) of the inside of the ring girder equals:

$$S_{in} = \frac{I_R}{\bar{z}} = \frac{3,188}{4.87} = 655 \text{ in.}^3$$

17.4.1.4 Calculation for Local Shell Bending Stress

Refer to Figure 17-5. The local shell bending stress (f_b) is given by:

$$f_b = 1.82 \left(\frac{A_r - b t_2}{A_r + 1.56 t_2 \sqrt{r t_2}} \right) \left(\frac{P_n r}{t_2} \right)$$

Where:

- f_b = local shell bending stress, psi
- A_r = cross-sectional area of ring, in.²
- r = inside radius of penstock, in.

The cross-sectional area of the ring (A_r) equals:

$$A_r = 2ah + bt_1 + bt_2 = 2(1)(10) + 14(2) + 14(2) = 76 \text{ in.}^2$$

Then:

$$f_b = 1.82 \left(\frac{76 - 14(2)}{76 + 1.56(2)\sqrt{90(2)}} \right) \left(\frac{287(90)}{2} \right) = 9,573 \text{ psi}$$

17.4.1.5 Ring Stresses (Normal Conditions)

Refer to USBR Welded Steel Penstocks, *Engineering Monograph No. 3*,³ Figures 18 and 19.

Note that a complete analysis of ring girders would include a check of shear stresses; however, this has not been done in this design example.

(1) Ring constants and determination of ring girder thrust and moment.

Determine the ring constants B , q , and K (numerical coefficients used in Figures 18 and 19 of Reference 3).

$$B = \frac{r}{R} \left(1 - \frac{2K}{qr} \right) - \frac{x}{R}$$

Where:

- r = inside radius of penstock = 90 in.
- R = radius of ring girder centroid = 90 + 4.87 = 94.87 in.
- x = distance between centroid of ring girder and centroid of leg supporting ring, in.
- $q = \frac{1.285}{\sqrt{rt}} = \frac{1.285}{\sqrt{2(90)}} = 0.0958$

To determine x , find the center of gravity of the ring girder support leg (see Figure 17-6).

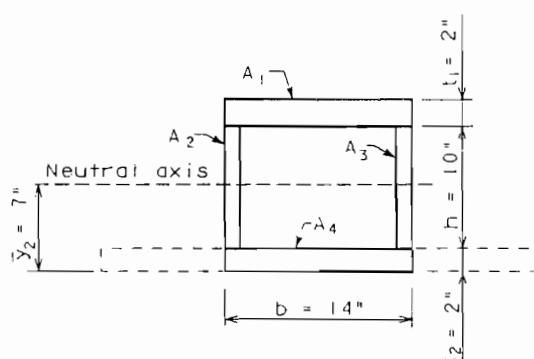


Figure 17-6 Section of Ring Girder Support Leg

Following are the values for A_1 , A_2 , A_3 , A_4 and y_1 , y_2 , y_3 , y_4 :

$$A_1 = A_4 = 28 \text{ in.}^2, \quad y_1 = 13 \text{ in.}, \quad y_4 = 1 \text{ in.}$$

$$A_2 = A_3 = 10 \text{ in.}^2, \quad y_2 = y_3 = 7 \text{ in.}$$

Since the leg is symmetric:

$$\bar{y}_2 = \frac{14}{2} = 7 \text{ in.}$$

$$A_1 + A_2 + A_3 + A_4 = 2(28 \text{ in.}^2) + 2(10 \text{ in.}^2) = 76 \text{ in.}^2$$

Then:

$$x = \bar{y}_2 - \bar{y}_1 = 7.00 - 4.87 = 2.13 \text{ in.}$$

$$K = \frac{r}{l} \left[\frac{\mu l^2}{12 r^2} + (1 - \mu^2) \left(1 - \frac{Q_s}{2Q} \right) + \frac{(2 + \mu)l}{4 q r^2} \right]$$

Where:

$$\mu = 0.3$$

$$l = 100 \times 12 = 1,200 \text{ in.}$$

$$Q_s = (100)(2,178) \left(\frac{1}{\cos 8.3} \right) = 220,109 \text{ lb}$$

$$Q = (100)(13,205) \left(\frac{1}{\cos 8.3} \right) = 1,334,500 \text{ lb}$$

$$\frac{Q_s}{2Q} = 0.0825$$

Then:

$$K = + \frac{90}{1,200} \left[\frac{(0.3)(1,200)^2}{12(90)^2} + (1 - 0.3^2)(1 - 0.0825) + \frac{(2 + 0.3)(1,200)}{4(0.0958)(90^2)} \right] = 0.46$$

$$B = \frac{90}{94.87} \left(1 - \frac{2(0.46)}{(0.0958)(90)} \right) - \frac{2.13}{94.87} = 0.825$$

The tension (N) due to internal pressure on the ring equals:

$$N = Pr \left[b + \frac{2(1 - \mu^2)}{q} \right] = (287)(90) \left[14 + \frac{2(1 - 0.3^2)}{0.0958} \right] = 852,336 \text{ lb}$$

The thrust (T) in the ring girder, exclusive of N , equals:

$$T = Q (K_1 + BK_2)$$

The bending moment (M) in the ring girder equals:

$$M = Q (RK_3 + xK_4)$$

(2) Ring thrusts and moments

Table 17-17 (from Figure 18 of "Welded Steel Penstocks—Design and Construction"³) gives K_1 through K_4 values.

Table 17-14 Vertical Load Constants

θ	K_1	K_2	K_3	K_4
0	-0.239	+0.318	+0.011	-0.068
90°	-0.250	0	0	+0.250
90°*	+0.250	0	0	-0.250
180	+0.239	-0.318	-0.011	+0.068

Following are ring thrust and moment calculations:

@ $\theta = 0^\circ$

$$T = 1,334,500 (-0.239) + (0.825) (0.318) = 31,160 \text{ lb}$$

$$M = 1,334,500 [(94.87) (0.011) + (2.13) (-0.068)] = 1,199,355 \text{ in.-lb}$$

@ $\theta = 180^\circ$

$$T = -31,160 \text{ lb}$$

$$M = -1,199,355 \text{ in.-lb}$$

@ $\theta = 90^\circ$

$$T = 1,334,500 [-0.25 + (0.825) (0)] = -333,625 \text{ lb}$$

$$M = 1,334,500 [(94.87) (0) + (2.13) (0.25)] = 710,621 \text{ in.-lb}$$

@ $\theta = 90^\circ*$

$$T = +333,625 \text{ lb}$$

$$M = -710,621 \text{ in.-lb}$$

(3) Ring stresses using ring girder properties calculated in Section 17.4.1.3

Following are calculations for ring stresses, where:

σ_{in} = ring inside bending stress, psi

σ_{out} = ring outside bending stress, psi

$$\begin{aligned}
 A &= 118 \text{ in.}^2 = A_1 + A_2 + A_3 + A_4 \\
 S_{in} &= 665 \text{ in.}^3 = \text{section modulus of inside of ring girder} \\
 S_{out} &= 349 \text{ in.}^3 = \text{section modulus of outside of ring girder}
 \end{aligned}$$

@ $\theta = 0^\circ$

$$\begin{aligned}
 \sigma_{in} &= \frac{T}{A} + \frac{M}{S_{in}} + \frac{N}{A} = \frac{31,160}{118} + \frac{1,199,355}{655} + \frac{852,336}{118} = +9,318 \text{ psi} \\
 \sigma_{out} &= \frac{T}{A} - \frac{M}{S_{out}} + \frac{N}{A} = \frac{31,160}{118} - \frac{1,199,355}{349} + \frac{852,336}{118} = +4,050 \text{ psi}
 \end{aligned}$$

@ $\theta = 180^\circ$

$$\begin{aligned}
 \sigma_{in} &= -264 + (-1,831) + 7,223 = +5,128 \text{ psi} \\
 \sigma_{out} &= -264 - (-3,436) + 7,223 = +10,395 \text{ psi}
 \end{aligned}$$

@ $\theta = 90^\circ$

$$\begin{aligned}
 \sigma_{in} &= \frac{-333,625}{118} + \frac{710,621}{655} + \frac{852,336}{118} = +5,480 \text{ psi} \\
 \sigma_{out} &= -2,827 - \frac{710,621}{349} + 7,223 = +2,359 \text{ psi}
 \end{aligned}$$

@ $\theta = 90^\circ$

$$\begin{aligned}
 \sigma_{in} &= +2,827 + (-1,085) + 7,223 = +8,965 \text{ psi} \\
 \sigma_{out} &= +2,827 - (-2,036) + 7,223 = +12,086 \text{ psi}
 \end{aligned}$$

17.4.1.6 Summary of Normal Condition Stresses

Table 17-15 summarizes the normal condition stresses.

Table 17-15 Normal Condition Stresses

(1) ANGLE, θ , FROM TOP OF RING GIRDER (DEGREES)	(2) RING GIRDER INSIDE STRESS, σ_x (psi)	(3) RING GIRDER OUTSIDE STRESS (psi)	(4) AXIAL STRESS, $f_1 + f_2 + f_3$ (psi)	(5) LOCAL SHELL BENDING STRESS (psi)	(6) LONGITUDINAL BENDING STRESS (psi)	(7) TOTAL LONGITUDINAL STRESS, σ_y , COL. 4 + 5 + 6 (psi)	(8) EFFECTIVE SHELL STRESS σ_e , σ_x = COL. 2, σ_y = COL. 7 (psi)
0	+9,318	+4,050	-756	+9,573	+3,496	+12,313	+11,122
90°	+5,480	+2,359	-756	+9,573	0	+8,817	+7,710
90°	+8,965	+12,086	-756	+9,573	0	+8,817	+8,892
180	+5,128	+10,395	-756	+9,573	-3,496	+5,321	+5,227

All stresses are less than 23,300 psi; therefore stresses for normal condition are acceptable.

17.4.1.7 Seismic Stresses

(1) Determine ring constants. Refer to Figure 17-7.

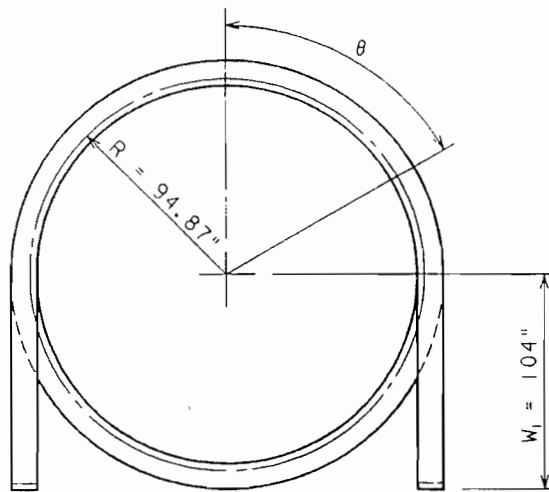


Figure 17-7 Ring Girder

For $-\pi/2 < \theta < \pi/2$:

$$B = \frac{r}{R} \left(1 - \frac{2K}{qr} \right) + \frac{\pi W_1}{4R}$$

Where:

- r = inside radius of penstock = 90 in.
- R = radius of ring girder centroid = 94.87 in.
- K = 0.46 (numerical coefficient from Section 17.4.1.5)
- q = 0.0958 (numerical coefficient from Section 17.4.1.5)
- W_1 = height of ring girder support leg = 104 in.

$$B = \frac{90}{94.87} \left(1 - \frac{(2)(0.46)}{(0.0958)(90)} \right) + \frac{\pi(104)}{(4)(94.87)} = 1.71$$

For $\pi/2 < \theta < 3\pi/2$:

$$B = \frac{r}{R} \left(1 - \frac{2K}{qr} \right) - \frac{\pi W_1}{4R}$$

$$B = \frac{90}{94.87} \left(1 - \frac{(2)(0.46)}{(0.0958)(90)} \right) - \frac{\pi(104)}{(4)(94.87)} = -0.013$$

(2) Apply Equation 34-1, Section 2334, of the 1991 UBC for seismic load:

$$V = \left(\frac{ZIC}{R_w} \right) W$$

Where:

$$\begin{aligned}
 V &= \text{total design base shear, lb} \\
 Z &= 0.3 \text{ for Seismic Zone 3 (from UBC Table No. 23-I)} \\
 I &= 1.0 \text{ (importance factor) (from UBC Table No. 23-L)} \\
 C &= \frac{1.25 S}{T^{2/3}} \text{ (where } C \leq 2.75 \text{ and } \frac{C}{R_w} \geq 0.5 \text{ (Section 2338(d), "Other Nonbuilding Structure," UBC))} \\
 S &= S_1 = 1.0 \text{ (a rock-like material) per UBC Table No. 23-J} \\
 T &= C_t(h_n)^{3/4} = \text{period of structure, seconds (Equation 12-3, Section 2312, UBC)} \\
 C_t &= 0.035 \text{ for steel moment resisting frames} \\
 h_n &= \text{height of structure, ft} \\
 R_w &= 12 \text{ for Special Moment Resisting Space Frames (steel) per UBC Table No. 23-O} \\
 T &= (0.035)(15)^{3/4} = 0.27 \text{ seconds} \\
 C &= \frac{(1.25)(1.0)}{(0.27)^{2/3}} = 2.99 \text{ (2.75 controls)} \\
 \frac{C}{R_w} &= \frac{2.75}{12} = 0.23 \text{ (0.5 controls)} \\
 W &= 1,334,500 \text{ lb (Q from section 17.4.2.5), combined weight of pipe shell and water in one span}
 \end{aligned}$$

Then:

$$V = [(0.3)(1.0)(0.5)] W = 0.15W = (0.15) 1,334,500 = 200,175 \text{ lb}$$

(3) Apply Equation 16-1, Section 2316, of the 1991 UBC for wind load:

$$p = (C_e)(C_q)(q_s)(I)$$

Where:

$$\begin{aligned}
 p &= \text{design wind pressure, lb} \\
 C_e &= 1.2 \text{ (0 to 20 ft height, exposure C) as given in UBC Table No. 23-G} \\
 C_q &= 0.8 \text{ (for round structures, tanks, chimneys) as given in UBC Table No. 23-H} \\
 q_s &= 17 \text{ psf (for 80 mph basic wind speed) as given in UBC Table No. 23-F} \\
 I &= 1.0 \text{ (importance factor) as given in UBC Table No. 23-L}
 \end{aligned}$$

Then:

$$p = (1.2)(0.8)(17)(1.0) = 16.3 \text{ psf}$$

And:

$$V = \left(\frac{100}{\cos 8.3^\circ} \right) (15)(16.3) = 24,708 \text{ lb}$$

The wind load is considerably less than the earthquake load; therefore, the earthquake load controls lateral force design of the ring girder.

17 Penstock Design Examples

(4) Determine ring girder moments, thrusts, and stresses for seismic loads.

Refer to Table 17-16 for earthquake load constants (from Figure 19 of "Welded Steel Penstocks—Design and Construction"³).

Table 17-16 Earthquake Load Constants

θ	K_1	K_2	K_3	K_4	B
0	0	0	0	0	1.71
90°	+0.0795	-0.250	+0.0795	-0.318	1.71
90°	+0.0795	+0.250	+0.0795	-0.318	-0.013
180	0	0	0	0	-0.013
270°	-0.0795	-0.250	-0.0795	+0.318	-0.013
270°	-0.0795	+0.250	-0.0795	+0.318	1.71

The ring girder bending moment (M) is given by:

$$M = n Q (RK_1 + W_1 K_2)$$

Where:

- n = 0.15 (seismic coefficient calculated from UBC Equation 34-1)
- Q = 1,334,500 lb (combined weight of pipe shell and water in one span)
- R = 94.87 in.
- W_1 = 104 in.
- K_1 and K_2 are determined from Table 17-16.

The ring girder thrust (T) is given by:

$$T = T_1 + \frac{(n)(Q - Q_s)}{\pi l} \left[b + \frac{2}{q} (1 - \mu^2) \right]$$

Where:

- $T_1 = (n)(Q)(K_3 + BK_4)$
- $K_3, K_4,$ and B are determined from Table 17-16.
- Q_s = 220,109 lb (weight of pipe shell in one span from Section 17.4.1.5)

Then:

$$T = T_1 + \frac{0.15 (1,334,500 - 220,109)}{\pi (100) (12)} \left[14 + \frac{2}{0.0958} (1 - 0.3)^2 \right] = T_1 + 1,463$$

@ $\theta = 0^\circ$ and 180° :

- $M = 0$
- $T_1 = 0$
- $T = 1,463$ lb
- $\sigma_{out} = \sigma_{in} = \frac{T}{A} \pm \frac{M}{S} = \frac{1,463}{118} \pm 0 = 12.4$ psi

@ $\theta = 90^\circ$:

$$M = (0.15)(1,334,500)[(94.87)(0.0795) + 104(-0.25)] = -3,694,797 \text{ in.-lb}$$

$$T_1 = (0.15)(1,334,500)[(0.0795) + (1.71)(-0.318)] = -92,937 \text{ lb}$$

$$T = -92,937 + 1,463 = -91,474 \text{ lb}$$

$$\sigma_{out} = \frac{T}{A} - \frac{M}{S_{out}} = \frac{-91,474}{118} - \frac{-3,694,797}{349} = +9,811 \text{ psi}$$

$$\sigma_{in} = \frac{T}{A} + \frac{M}{S_{in}} = \frac{-91,474}{118} + \frac{-3,694,797}{655} = -6,416 \text{ psi}$$

@ $\theta = 90^\circ$:

$$M = (0.15)(1,334,500)[(94.87)(0.0795) + 104(0.25)] = 6,714,303 \text{ in.-lb}$$

$$T_1 = (0.15)(1,334,500)[(0.0795) + (-0.013)(-0.318)] = 16,741 \text{ lb}$$

$$T = 16,741 + 1,463 = 18,204 \text{ lb}$$

$$\sigma_{out} = \frac{T}{A} - \frac{M}{S_{out}} = \frac{18,204}{118} - \frac{6,714,303}{349} = -19,084 \text{ psi}$$

$$\sigma_{in} = 154 + 10,405 = +10,788 \text{ psi}$$

@ $\theta = 270^\circ$:

$$M = (0.15)(1,334,500)[(94.87)(-0.0795) + 104(-0.25)] = -6,714,303 \text{ in.-lb}$$

$$T_1 = (0.15)(1,334,500)[(-0.0795) + (-0.013)(0.318)] = -16,741 \text{ lb}$$

$$T = -16,781 + 1,463 = -15,278 \text{ lb}$$

$$\sigma_{out} = \frac{T}{A} - \frac{M}{S_{out}} = \frac{-15,278}{118} - \frac{-6,714,303}{349} = +19,109 \text{ psi}$$

$$\sigma_{in} = -129 + (-10,251) = -10,380 \text{ psi}$$

@ $\theta = 270^\circ$:

$$M = (0.15)(1,334,500)[(94.87)(-0.0795) + 104(0.25)] = 3,694,797 \text{ in.-lb}$$

$$T_1 = (0.15)(1,334,500)[(-0.0795) + 1.71(0.318)] = 92,937 \text{ lb}$$

$$T = 92,937 + 1,463 = 94,400 \text{ lb}$$

$$\sigma_{out} = \frac{T}{A} - \frac{M}{S_{out}} = \frac{92,937}{118} - \frac{3,694,797}{349} = -9,787 \text{ psi}$$

$$\sigma_{in} = 800 + 5,640 = +6,44 \text{ psi}$$

(5) Add seismic stresses to normal vertical ring stresses

Table 17-17 shows the total inside and outside ring stresses resulting from the addition of seismic and normal vertical ring stresses.

At $\theta = 270^\circ$, the outside ring girder stress is +31,195 psi, which is .5% over the allowable of +31,033 psi.

(6) Check effective shell stresses (Hencky-von Mises)

The combined effective shell stress (σ_e) is given by:

$$\sigma_e = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2)^{1/2}$$

Table 17-17 Total Inside and Outside Ring Stresses

(1) θ (DEGREES)	VERTICAL		EARTHQUAKE		TOTAL	
	(2) INSIDE (psi)	(3) OUTSIDE (psi)	(4) INSIDE (psi)	(5) OUTSIDE (psi)	(6) INSIDE (psi)	(7) OUTSIDE (psi)
0	+9,318	+4,050	+12	+12	+9,330	+4,062
90°	+5,480	+2,359	-6,416	+9,811	-934	+12,170
90°*	+8,965	+12,086	+10,405	-19,084	+19,370	-6,998
180	+5,128	+10,345	+12	+12	+5,140	+10,407
270°	+8,965	+12,086	-10,380	+19,109	-1,415	+31,195
270°*	+5,480	+2,359	+6,441	-9,787	+11,921	-7,428

The values for σ_x are from column 6 of Table 17-17. The values for σ_y are from column 7 of Table 17-15. Table 17-18 gives the combined effective shell stresses.

Table 17-18 Combined Effective Shell Stresses

(1) ANGLE (DEGREES)	(2) RING GIRDER INSIDE STRESS σ_x (psi)	(3) TOTAL LONGITUDINAL INSIDE STRESS σ_y (psi)	(4) EFFECTIVE STRESS σ_e (psi)
0	+9,330	+12,313	+11,125
90°	-934	+8,817	+9,319
90°*	+19,370	+8,817	+16,797
180	+5,140	+5,321	+5,233
270°	-1,415	+8,817	+9,603
270°*	+11,921	+8,817	+10,712

The effective shell stress (column 4) is less than 31,033 psi and is therefore acceptable under seismic (intermittent) conditions.

Note that penstocks supported by ring girders typically should be investigated for half-full conditions as indicated in Table 3-1; however, for penstocks on steep slopes, similar to this example, a half-full condition cannot be obtained over the span length. Therefore, the half-full condition was not considered for this problem.

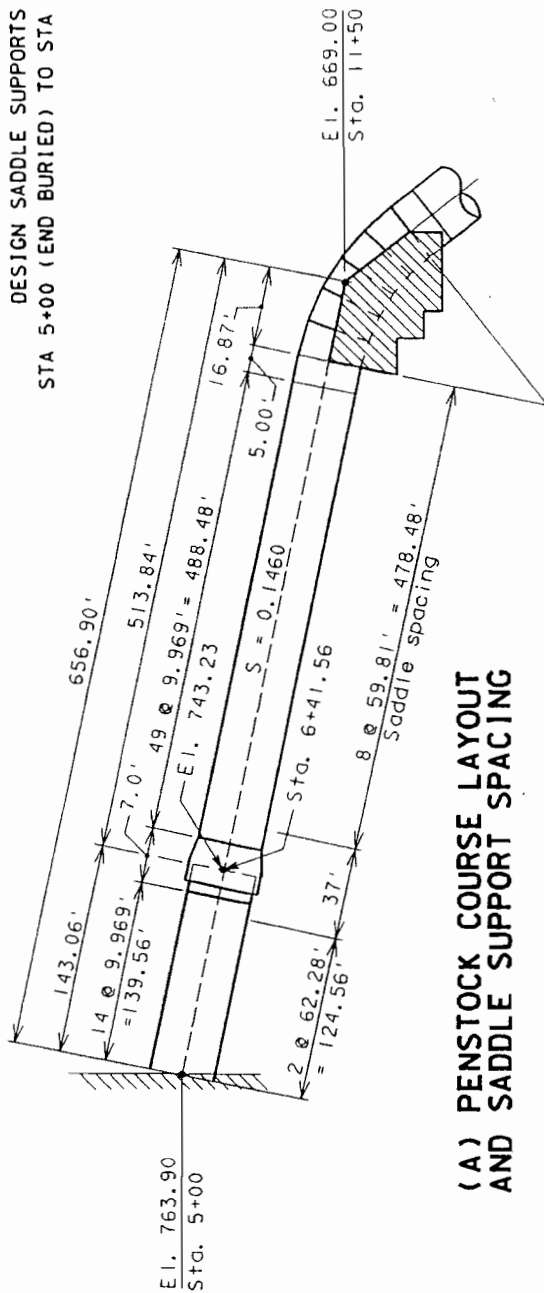
17.4.2 Saddle Design

This section gives calculations for the design of saddle supports for the exposed portion of the penstock extending from station 5 + 00 to station 11 + 50.

17.4.2.1 Preliminary

See Figure 17-8 for the penstock layout, support layout, and determination of reactions, shears, and bending moments from station 5 + 00 to the expansion joint (by the moment distribution method). It is assumed that EI is constant and there is no shear transfer across the expansion joint.

DESIGN SADDLE SUPPORTS BETWEEN
STA 5+00 (END BURIED) TO STA 11+50 (PI = 2)



(A) PENSTOCK COURSE LAYOUT AND SADDLE SUPPORT SPACING

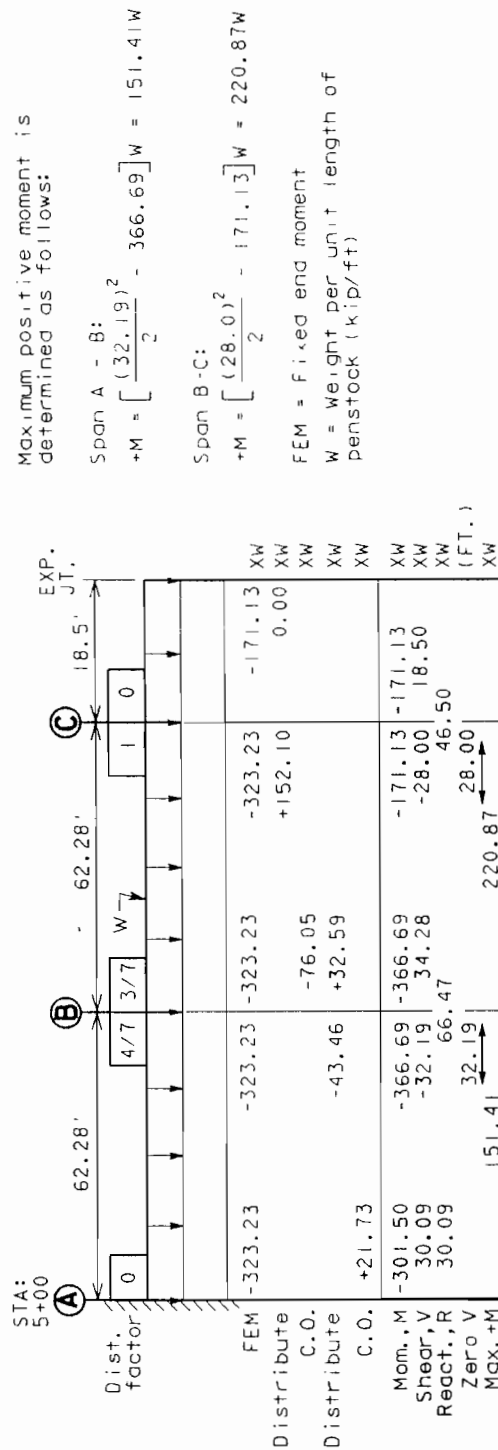


Figure 17-8 Saddle Support Spacing and Penstock Course Layout

The total weight (in k/ft) of the shell and water is:

$$\text{Weight of 1-in. thick shell} = 15(\pi) 40.8/1000 = 1.92 \text{ k/ft}$$

$$\text{Weight of water} = 15^2 (\pi/4) 62.4/1000 = 11.03 \text{ k/ft}$$

$$\text{Total weight} = 12.95 \text{ k/ft}$$

Multiply the total weight (in k/ft) by $\cos 8.3^\circ$ (0.9895) to determine the uniform load (w) normal to the penstock:

$$w = (12.95) (0.9895) = 12.8 \text{ k/ft}$$

The maximum bending moments, reactions, and shears are then calculated as follows:

$$\text{Maximum + moment} = (12.8) (220.87) = 2827.14 \text{ ft-kips}$$

$$\text{Maximum - moment} = (12.8) (-366.69) = -4693.63 \text{ ft-kips}$$

$$\text{Maximum reaction} = (12.8) (66.47) = 850.82 \text{ kips}$$

$$\text{Maximum shear} = (12.8) (34.28) = 438.78 \text{ kips}$$

Figure 17-9 gives saddle details.

17.4.2.2 Seismic Loads

Determine the seismic loads for Seismic Zone No. 3.

Base shear (V) is given by Equation 12-1, Section 2312, of the UBC as:

$$V = \frac{ZIC}{R_w} W$$

Where:

$$Z = 0.3 \text{ (seismic zone factor from UBC Table No. 23-J)}$$

$$I = 1.0 \text{ (importance factor from UBC Table No. 23-L)}$$

$$R_w = 4 \text{ (numerical coefficient from UBC Table No. 23-Q)}$$

$$C = 1.25 S/T^{2/3} \leq 2.75$$

$$C/R_w = 0.688 \text{ (with } C=2.75)$$

$$W = \text{Total seismic dead load}$$

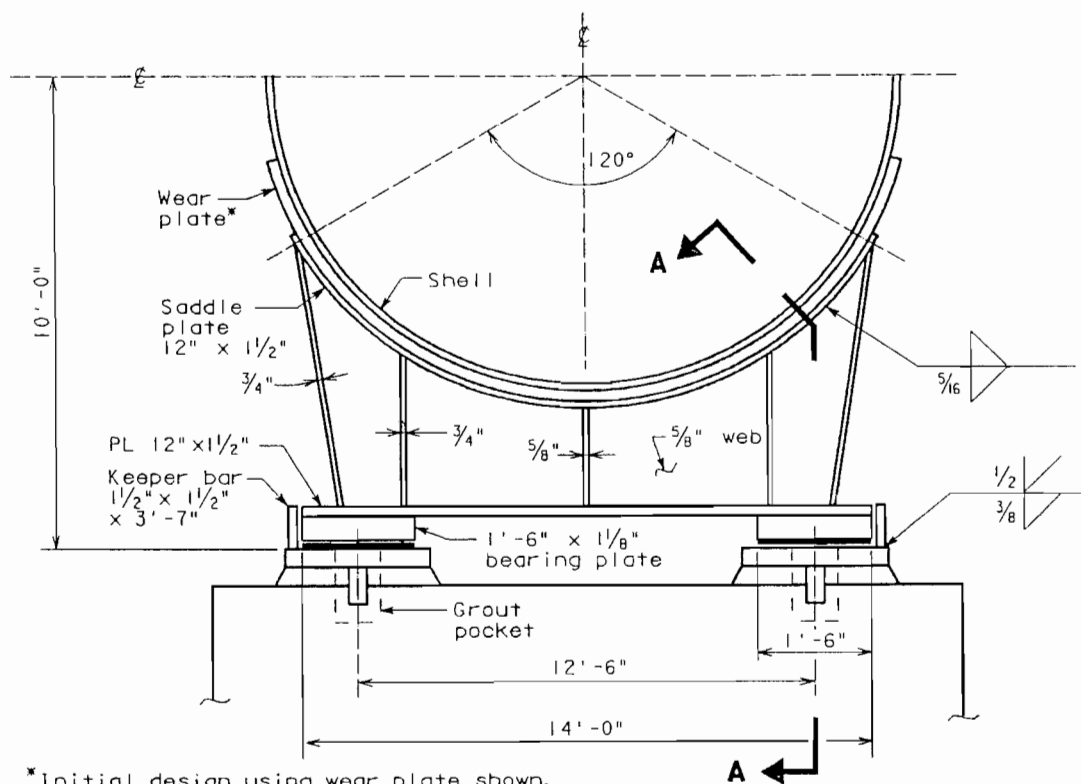
Design Basis Criteria (DBC) earthquake:

$$V = 0.3 (1.0) (0.688)W = 0.206W \text{ (see Section 1.8.1.1)}$$

$$\text{Vertical seismic} = 2/3 \text{ horizontal} = 0.137W \text{ (see Section 1.8.1.1)}$$

Dam Safety Criteria (DSC) earthquake:

$$2.0 \text{ DBC (see Section 1.8.1.2)}$$



*Initial design using wear plate shown. Final design replaced wear plate with a thickened shell plus two stiffener rings as shown in Figure 17-10.

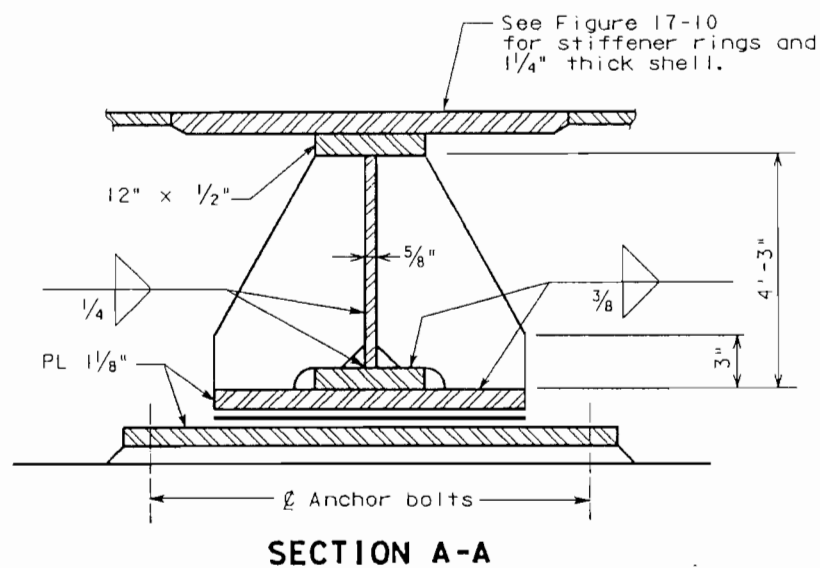


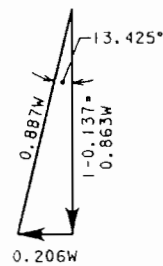
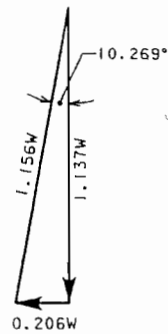
Figure 17-9 Saddle Details

17.4.2.3 Check Stability of Saddle

Refer to the discussion on saddle stability in Section 4.3.5.

- (1) Check DBC with vertical earthquake acting upward, then downward, for $\theta = 120$ degrees.

Vectorially add the vertical component to the horizontal component, as follows:

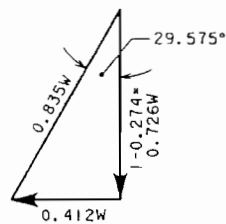
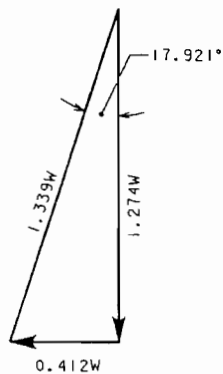


$$\tan^{-1}(0.206/1.137) = 10.269^\circ$$

$$\tan^{-1}(0.206/0.863) = 13.425^\circ \leq 0.125\theta = 15^\circ$$

which satisfies Equation 4.3.5-1 for DBC.

- (2) Check DSC as follows:



$$\tan^{-1}(0.412/1.274) = 17.921^\circ$$

$$\tan^{-1}(0.412/0.726) = 29.575^\circ \leq 0.25\theta = 30^\circ$$

which satisfies Equation 4.3.5-2 for DSC.

Therefore, 120-degree saddles are acceptable.

- (3) Check minimum bearing stress on supports (refer to Figure 4-5).

Stability considerations require that the minimum bearing on the right support in Figure 4-5 satisfies the expression:

$$\frac{W}{e} - \frac{Hh}{d} - L \left(h - R + \frac{b}{2} \right) \left(\frac{1}{c} \right) \geq 0$$

Assume that the total longitudinal load (L) equals zero; the equation then becomes:

$$\frac{W}{e} - \frac{Hh}{d} \geq 0$$

DBC:

$$\frac{0.863}{3 - \left[\frac{(0.206)(10.00)}{14.00} \right]} = +0.141$$

which is greater than zero, and therefore satisfies the equation.

DSC:

$$\frac{0.726}{2 - \left[\frac{(0.412)(10.00)}{14.00} \right]} = +0.069$$

which is greater than zero, and therefore satisfies the equation.

Check DBC with $L = 0.1W$ and $c = 3.25$ ft:

$$0.141 - 0.1 \left(\frac{10.00 - 7.50 + 7.5}{4} \right) \left(\frac{1}{3.25} \right) = 0.006$$

which is greater than zero, and therefore satisfies the equation.

- (4) Check maximum bearing stress for a 18 x 39-inch bearing plate (702.00 square inches).

Dead load plus water load bearing stress equals:

$$\frac{850.82}{(702.00)(2)} = 606 \text{ psi}$$

DBC:

The maximum reaction at one leg equals:

$$\left(\frac{1.137}{2}\right)W + \left[\frac{(0.206)(10.00)}{12.5}\right]W = 0.733W$$

Since $W = 850.82$ kips, the bearing stress equals:

$$\frac{(0.733)(850.82)}{702.00} = 889 \text{ psi}$$

DSC:

The maximum reaction at one leg support equals:

$$\left(\frac{1.274}{2}\right)W + \left[\frac{(0.412)(10.00)}{12.5}\right]W = 0.967W$$

The bearing stress equals:

$$\frac{(0.967)(850.82)}{702.00} = 1,170 \text{ psi}$$

The bearing stresses are based on the concrete bearing allowables and one-third increase for DBC and two for DSC.

17.4.2.4 Design Second Support in from Expansion Joint No. 1

Refer to the data in Table 17-1.

The thickness required for internal pressure equals:

$$\left(\frac{62.28}{143.06}\right)(0.861 - 0.792) + 0.792 = 0.822 \text{ in. (use } 7/8 \text{ in.)}$$

The head (normal condition) equals:

$$\left(\frac{62.28}{143.06}\right)(514.79 - 473.55) + 473.55 = 491.50 \text{ ft}$$

17.4.2.5 Check Shell Stresses

Refer to the discussion in Section 4.3.6.1.

(1) Section modulus over saddle

Refer to Figure 4-6 (A).

The shell mid-surface radius (R) equals:

$$R = 90 + .625 = 90.625 \text{ in. (where } t = 1.25 \text{ in.)}$$

The effective half angle (Δ) equals:

$$\frac{5}{12(\theta)} + 30^\circ = \frac{5}{(12)(120^\circ)} + 30^\circ = 80^\circ (1.3963 \text{ radians})$$

The distance (a) from the neutral axis to the top extreme fiber equals:

$$R \left[\left(\frac{\sin \Delta}{\Delta} \right) - \cos \Delta \right] = 90.625 \left[\left(\frac{\sin 80^\circ}{1.3963} \right) - \cos 80^\circ \right] = 48.183 \text{ in.}$$

The distance (b) from the neutral axis to the bottom extreme fiber equals:

$$R \left(1 - \frac{\sin \Delta}{\Delta} \right) = 26.706 \text{ in.}$$

Moment of inertia (I) equals:

$$R^3 t \left(\Delta + \sin \Delta \cos \Delta - \frac{2 \sin^2 \Delta}{\Delta} \right) = 165,670 \text{ in.}^4$$

Maximum negative moment equals:

$$(-4,693.63 \text{ ft-k}) (12 \text{ in./ft}) = -56,324 \text{ in.-kips}$$

Bending stress tension equals:

$$(56,324 \text{ in.-k}) \left(\frac{48.183 \text{ in.}}{165,670 \text{ in.}^4} \right) = 16.381 \text{ ksi (which is less than 23.33 ksi)}$$

Bending stress compression equals:

$$(56,324 \text{ in.-k}) \left(\frac{26.706 \text{ in.}}{165,670 \text{ in.}^4} \right) = -9.079 \text{ ksi}$$

17 Penstock Design Examples

The t/R value equals $1.25/90.625 = 0.013793$, which is greater than the maximum permitted by API to apply the $1.8 \times 10^6 (t/R)$ formula. Therefore, use the allowable compressive stress specified in the ASME Code, Section VIII, Division 1, UG-23(b):

$$A = \frac{0.125}{(R/t)} = 0.001724$$

From Figure 5–UCS-28.2 of the ASME Code, the allowable compression stress (B) equals 14.5 ksi and, therefore, the $1\frac{1}{4}$ -inch thickness is acceptable.

It should be noted that the above calculations are based on $t = 1.25$ inches. For a trial thickness of 0.875 inch, the tension over the saddle would be equal to $(1.25/0.875) \times 16.381$ ksi = 23.401 ksi, and the compression would be equal to $(1.25/0.875) (-9.079) = -12.970$ ksi (allowable = 13.0 ksi). Thus, both stresses are very close to the allowable. As shown below, it is necessary to increase the shell thickness to 1.25 inches and add full-circle stiffener rings to carry earthquake forces.

The bending moment (M) at midspan equals 2,827.14 ft-kips (as calculated in Section 17.4.3.1).

The section modulus equals:

$$\pi R^2 t = \pi (90.44^2) 0.875 = 22,484 \text{ in.}^3$$

Bending stress compression then equals:

$$(2,827.14) \left(\frac{12}{22,484} \right) = 1.509 \text{ ksi}$$

Since t/R equals $0.875/90.44 = 0.00967$, which is greater than the 0.00667 value specified in API 620, use the allowable compressive stress specified in the ASME Code, Section VIII, Division 1, UG-23(b):

$$A = \frac{0.125}{(R/t)} = \frac{0.125}{(90.44/0.875)} = 0.001209$$

The allowable compression stress (B) equals 13.0 ksi. The calculated compression stress of 1.509 ksi is well below the allowable and is, therefore, acceptable.

(2) Tangential shear stress over saddle

Refer to Figure 4-6 (B).

$$\alpha = 171^\circ - \left(\frac{19}{40} \right) \theta = 114^\circ (1.9897 \text{ radians})$$

$$K_2 = \frac{\sin \alpha}{(\pi - \alpha) + \sin \alpha \cos \alpha} = 1.1707$$

The shear stress (τ) equals $VK_2/(Rt)$, where V equals the shear at the support (438.78 kips).

Then:

$$\tau = \frac{(1.1707)(438.78)}{(90.625)(1.25)} = 4.535 \text{ ksi}$$

The allowable shear equals $0.85K$ or 18.66 ksi. 4.535 ksi is well below the allowable shear and is, therefore, acceptable.

(3) Circumferential compression directly over saddle invert

Refer to Figure 4-6 (C).

Assume the required wear plate dimensions are 18 x 3/4 inches ($g = 18$ inches, $t_2 = 3/4$ inch, $W = 850.82$ kips).

Then, compressive stress equals:

$$\frac{WK_5}{[1.56(Rt)^{0.5} + g]t + gt_2} = 15.628 \text{ ksi}$$

Where:

$$K_5 = \frac{1 + \cos \alpha}{(\pi - \alpha) + \sin \alpha \cos \alpha} = 0.7603$$

The allowable stress by Zick⁴ equals 1/2 yield or 19.0 ksi, and therefore 15.628 ksi is acceptable.

(4) Circumferential stress over horn of saddle

Refer to Figure 4-7.

Assume that the required wear plate dimensions are 18 x 3/4 inches ($g = 18$ inches, $t_2 = 3/4$ inch, $W = 850.82$ kips). Shell thickness is given as 0.875 inch.

Use the equations for moment and thrust given in Figure 4-7.

For $\theta = 120^\circ$, $\beta = 120^\circ$:

$$\text{Moment} = K_3RW = 0.05285(90.44)850.82 = 4,067 \text{ in.-kips}$$

$$\text{Thrust} = K_4W = 0.34047(850.82) = 290.0 \text{ kips}$$

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The length of shell resisting axial load is equal to:

$$1.56\sqrt{Rt} + g$$

Then the load-resisting area equals:

$$(1.56\sqrt{90.44 \times 0.875} + 18)(0.875) + (18.00)(0.75) = 41.393 \text{ in.}^2$$

The length of shell that can be used for resisting circumferential bending is equal to the lesser of the span length or eight times the radius (i.e., the lesser of $62.28 \times 12 = 747.36$ inches or $8 \times 90.44 = 723.52$ inches). Therefore, use 723.52 inches in calculating the section modulus.

Section modulus (Z) equals:

$$\frac{723.52 (0.75^2 + .875^2)}{6} = 160.15 \text{ in.}^3$$

Note that the square of the thicknesses is summed; this is a conservative assumption if the wear plate is welded to the shell.

The total stress equals:

$$\left(\frac{290.0}{41.393} \right) + \left(\frac{4,067}{160.15} \right) = 32.41 \text{ ksi}$$

The allowable stress by Zick⁴ equals:

$$(1.5)(23.33) = 35.00 \text{ ksi}$$

Therefore, the total stress is acceptable.

17.4.2.6 Check for Seismic DBC

Refer to Section 17.4.2.3.

$$\theta = 120^\circ$$

$$\tan^{-1} = (0.206 / 1.137) = 10.269^\circ$$

$$\tan^{-1} = (0.206 / 0.863) = 13.425^\circ$$

The effective saddle angle for DBC with vertical seismic forces acting downward equals:

$$2(60^\circ - 10.269^\circ) = 99.462^\circ$$

(Note that $\theta = 99.462^\circ$ for applying the formulas in Figure 4-6.)

The effective load (W) equals:

$$850.82 (1.137^2 + 0.206^2)^{1/2} = 983.13 \text{ kips}$$

- (1) Bending stresses over saddle (use $t = 1.25$ inches)

Bending moment increases by a factor of:

$$\frac{983.13}{850.82} = 1.1555$$

Top stress equals:

$$\frac{(56,324) (1.1555)}{(0.24231) (90.625) (1.25)} = 26.16 \text{ ksi}$$

Note that section modulus = coefficient $\times R^2 \times t$. The coefficient is calculated from the formulas in Figure 4-6(A) as Z_a with $\theta = 99.462^\circ$. That is, coefficient = $I/(a R^2 t)$.

Allowable top stress equals:

$$(23.33) (1.33) = 31.10 \text{ ksi}$$

Bottom stress equals:

$$\frac{(56,324) (1.1555)}{(.44681) (90.625) (1.25)} = -14.19 \text{ ksi}$$

Allowable bottom stress equals:

$$(14.50) (1.333) = -9.33 \text{ ksi}$$

Therefore, the bending stress is acceptable with a 1.25-inch shell thickness and no wear plate. By inspection, the midspan bending stress poses no problem.

- (2) Tangential shear stress over saddle

Refer to Figure 4-6 (B).

Shear stress (τ) equals:

$$\frac{K_2 (\text{shear})}{Rt}$$

Where:

$$\begin{aligned} \text{shear} &= (1.1555) (438.78 \text{ kips}) = 507.01 \text{ kips} \\ K_2 &= 1.5999 \end{aligned}$$

Then:

$$\tau = \frac{(1.5999)(507.01)}{(90.625)(1.25)} = 7.161 \text{ ksi}$$

The allowable shear equals:

$$0.8SK = 0.8(23.33)(1.333) = 24.88 \text{ ksi}$$

The tangential shear stress of 7.161 ksi is considerably less than the allowable shear, and is therefore acceptable.

(3) Circumferential compression over saddle invert

Refer to the equations given in Figure 4-6(C).

Compressive stress equals:

$$\frac{WK_5}{(1.56\sqrt{Rt} + g)t} = \frac{(983.13)(.85505)}{(16.604 + 12.00)1.25} = 23.51 \text{ ksi}$$

(Note that K_5 is calculated with $\theta = 99.462^\circ$.)

The allowable stress by Zick⁴ equals the product of 19.0 ksi (1/2 yield) times 1.333, or 25.33 ksi. Since 23.51 ksi is less than the allowable stress, the compressive stress is acceptable.

(4) Circumferential stress over horn of saddle

Refer to the equations given in Figure 4-7 for moment and thrust coefficients K_3 and K_4 .

$$\theta = 99.462^\circ; \beta = 130.27^\circ; \text{shell thickness} = 1.25 \text{ in.}$$

Moment equals:

$$K_3RW = (0.072196)(90.625)(983.13) = 6,432 \text{ in.-kips}$$

Thrust equals:

$$K_4W = (0.35546)(983.13) = 349.5 \text{ kips}$$

The length of shell resisting axial load is equal to:

$$1.56\sqrt{Rt} + g = 1.56\sqrt{90.625 \times 1.25} + 12.00 = 28.60 \text{ in.}$$

Then the load-resisting area equals:

$$1.25 (1.56 \sqrt{90.625 \times 1.25 + 12}) = 35.76 \text{ in.}^2$$

The length of shell that can be used for resisting circumferential bending equals the lesser of the span length or 8 times the radius (R).

$$\text{Span length} = (62.28) (12) = 747.36 \text{ in.}$$

$$8R = (8) (90.625) = 725.00 \text{ in.}$$

Therefore, use 725.00 inches for calculating the section modulus. The section modulus (Z) equals:

$$\frac{(725.00) (1.252)}{6} = 188.80 \text{ in.}^3$$

Then:

$$\text{Total stress} = \frac{349.5}{35.75} + \frac{6,432}{188.80} = 43.84 \text{ ksi}$$

The allowable stress by Zick⁴ equals:

$$(1.5) (23.33) (1.333) = 46.66 \text{ ksi}$$

Therefore, the total stress is acceptable.

17.4.2.7 Check for Seismic DSC

Refer to Section 17.4.2.3.

$$q = 120^\circ$$

$$\tan^{-1} (0.412/1.274) = 17.921^\circ$$

$$\tan^{-1} (0.412/0.726) = 29.575^\circ$$

The effective saddle angle for vertical seismic forces acting downward equals:

$$2 (60^\circ - 17.921^\circ) = 84.158^\circ$$

The effective load (W) equals:

$$850.82 \sqrt{1.274^2 + 0.412^2} = 1,139.2 \text{ kips}$$

(1) Bending stresses over saddle

Bending moments increase by a factor of:

$$\frac{1,139.2}{850.82} = 1.3389$$

Top stress equals:

$$\frac{(56,324) (1.3389)}{(0.18502) (90.625^2) (1.25)} = 39.702 \text{ ksi}$$

Allowable top stress equals:

$$(23.33) (2.0) = 46.67 \text{ ksi}$$

Bottom stress equals:

$$\frac{(56,324) (1.3389)}{(0.34611) (90.625^2) (1.25)} = -21.22 \text{ ksi}$$

Allowable bottom stress equals:

$$(14.50) (1.5) = -21.75 \text{ ksi}$$

Therefore, the bending stress is acceptable with a 1.25-inch shell thickness with no wear plate. Note that the stress increase factor (K) is limited to 1.5 because of buckling considerations. By inspection, the midspan bending stress poses no problem.

(2) Tangential shear stress over saddle

Refer to Figure 4-6 (B).

Shear stress (τ) equals:

$$\frac{K_2 (\text{shear})}{R_t}$$

Where:

$$\text{shear} = (1.3389) (438.78 \text{ kips}) = 587.48 \text{ kips}$$

Then:

$$\tau = \frac{(2.0981) (587.48)}{(90.625) (1.25)} = 10.88 \text{ ksi}$$

The allowable shear stress equals:

$$(0.8) (2.5) (S) = 46.66 \text{ ksi}$$

The tangential shear stress of 10.88 ksi is considerably less than the allowable shear, and is therefore acceptable.

(3) Circumferential compression over saddle invert

Refer to the equations given in Figure 4-6 (C).

Compressive stress equals:

$$\frac{WK_5}{(1.56 \sqrt{Rt+g}) t} = \frac{(1,139.2) (0.95559)}{(16.604 + 12.00) 1.25} = 30.45 \text{ ksi}$$

The allowable stress by Zick⁴ equals the product of 1/2 yield (19.0 ksi) times 2.5, or 47.5 ksi. Therefore, the compressive stress of 30.45 ksi is acceptable.

(4) Circumferential stress over horn of saddle

Refer to the equations given in Figure 4-7 for moment and thrust coefficients K_3 and K_4 .

$$\theta = 84.158^\circ; \beta = 137.921^\circ; \text{ Shell thickness} = 1.25 \text{ in.}$$

Moment equals:

$$K_3RW = (0.089537) (90.625) (1,139.2) = 9,244 \text{ in.-kips}$$

Thrust equals:

$$K_4W = (0.3593) (1,139.2) = 409.31 \text{ kips}$$

The length of shell resisting direct/axial load is:

$$1.56\sqrt{Rt} + g = 28.60 \text{ in.}$$

Then the load-resisting area equals:

$$(1.56 \sqrt{90.625 \times 1.25} + 12.00) 1.25 = 35.76 \text{ in.}^2$$

The length of shell that can be used for resisting circumferential bending equals the lesser of the span length or 8 times the radius (R).

$$\text{Span length} = (62.28) (12) = 747.36 \text{ in.}$$

$$8R = (8) (90.625) = 725.00 \text{ in.}$$

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Therefore, use 725.00 inches for calculating the section modulus. The section modulus (Z) equals:

$$\frac{725.00 (1.25^2)}{6} = 188.80 \text{ in.}^3$$

Then:

$$\text{Total stress} = \frac{409.31}{35.76} + \frac{9,244}{188.80} = 60.41 \text{ ksi}$$

The allowable stress by Zick⁴ equals:

$$(1.5) (23.33) (1.5) = 52.5 \text{ ksi}$$

The total stress is greater than the allowable stress. Therefore, the use of stiffener rings adjacent to the saddle is recommended.

Table 17-19 gives ring stress coefficients and other pertinent data used in the analysis and design of the support rings.

Table 17-19 Ring Stress Coefficients and Pressure Head

	M/RW	t/W	HEAD (FT)	CONDITION
NO SEISMIC	0.05808	0.2656	514.79	NORMAL
DBC	0.07094	0.2656	647.35	EMERGENCY
DSC	0.07844	0.2656	998.61	EXCEPTIONAL

Maximum moment and corresponding thrust occur at $\phi = 94^\circ$. The maximum moment produces tension on the outer fiber; the corresponding thrust is compressive at that point. Moment and thrust coefficients (M/RW and t/W) are from the equations in Figure 4-8.

Following are the section properties for the cross section shown in Figure 17-10 (two 10 x 1¼-inch rings spaced 13.50 inches clear and two 6 x 1¼-inch flanges).

$$\text{Ring area only} = 40.000 \text{ in.}^2$$

$$\text{Total area} = 80.755 \text{ in.}^2$$

$$\text{Moment of inertia} = 1,280.87 \text{ in.}^4$$

$$\text{Section modulus } (Z_{out}) = 177.605 \text{ in.}^3$$

$$\text{Section modulus } (Z_{in}) = 317.194 \text{ in.}^3$$

$$Q/I = 0.054/\text{in. at shell (one of pair)}$$

$$= 0.033/\text{in. at flanges (one flange)}$$

$$1.56\sqrt{Rt} + b = 32.604 \text{ in.}$$

$$\text{Radius to center of gravity} = 94.04 \text{ in.}$$

$$\text{Pressure stress} = 2.6 (\text{head} \times \text{diameter} \times \text{length}) / \text{Total area}$$

$$= \frac{2.6 (\text{Head}) (15.00) (32.604)}{(80.755) (1,000)} = 0.015746 (\text{Head})$$

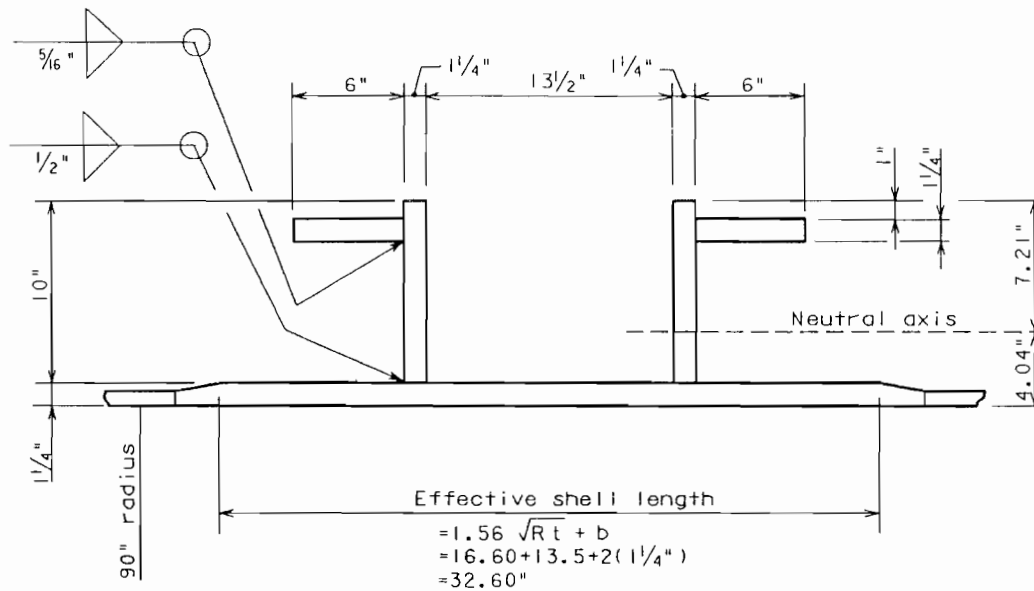


Figure 17-10 Ring Cross Section at Saddle Supports

Table 17-20 gives bending and axial stresses and allowables using information from Table 17-19.

Table 17-20 Bending and Axial Stresses and Allowables

	W (KIPS)	M (IN.-KIPS)	T (KIPS)	OUTSIDE BENDING + AXIAL STRESS (ksi)	PRESSURE STRESS (ksi)	TOTAL STRESS (ksi)	INSIDE BENDING + AXIAL STRESS (ksi)	ALLOWABLE (ksi)
NO SEISMIC	850.82	4,705	226	23.69	8.106	31.80	-17.63	23.33 x 1.5 = 35.00 (PL STRESS CATEGORY)
DBC	983.13	6,559	261	33.70	10.193	43.89	-23.91	35.00 x 1.5 = 52.50 (PL STRESS CATEGORY, K=1.5)
DSC	1,139.20	8,403	303	43.56	15.724	59.29	-30.24	35.00 x 2.5 = 87.50 (PL STRESS CATEGORY, K=2.5)

17.4.2.8 Check Rim Bending and Longitudinal Stresses

Rim bending equals:

$$1.82 \left(\frac{pR}{t} \right) \left[\frac{A_r}{(1.56\sqrt{Rt} + c)t + A_r} \right] = (1.82) \left[(223) \frac{90}{1.25} \right] \left[\frac{40.00}{(32.604)(1.25) + 40.0} \right] = 14.47 \text{ ksi}$$

Beam bending equals:

$$(4,693.63 \text{ ft-k}) (12/90.625^2 \pi 1.25) = \pm 1.746 \text{ ksi}$$

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Axial stress due to pressure and metal load parallel to the penstock centerline equals -1.00 ksi (assumed; it is low because of the expansion joint).

Membrane stress intensity (S_I) equals $8.106 - (-1.746 - 1.0) = 10.852$ ksi, which is less than the P_L allowable of 35.00 ksi.

Equivalent stress (S_e) equals:

$$\sqrt{8.106^2 - (8.106)(-1.746 - 1.0) + (-2.746)^2} = 9.773 \text{ ksi}$$

which is less than the P_L allowable of 35.00 ksi.

Stress intensity (SI), including rim bending, is equal to ring bending plus axial plus ring pressure stress combined with stresses in longitudinal direction, rim bending plus beam bending plus the axial stress.

$$\frac{4705}{317.1} - \frac{226}{80.76} + 8.106 - (-14.47 - 2.746) = 14.84 - 2.80 + 8.106 + 17.216 = 37.362 \text{ ksi}$$

The equivalent stress method gives:

$$[(14.84 - 2.80 + 8.106)^2 - (14.84 - 2.80 + 8.106)(-17.216) + (-17.216)^2]^{1/2} = 32.39 \text{ ksi}$$

The allowable is $3 S_m$ or 70 ksi.

Note that internal pressure, a compression radial stress on the inside surface of the penstock, does not enter into the stress intensity computations because internal pressure is less than beam bending, which is also a compressive stress.

17.4.2.9 Welds

(1) Flange-to-web

$$VQ/I = (851)(0.30)(0.033) = 8.42 \text{ k/in.}$$

$$\text{Weld size} = \frac{8.42}{(21)(2)(0.707)} = 0.284 \text{ k/in. (fillet weld) (Use } 5/16 \text{ in.)}$$

(2) Web-to-shell

$$VQ/I = (851)(0.30)(0.054) = 13.79 \text{ k/in.}$$

$$\text{Weld size} = \frac{13.79}{(21)(2)(0.707)} = 0.464 \text{ in. (Use } 1/2 \text{ in.)}$$

17.4.2.10 Saddle Design

(1) Preliminary

Use the equations given in Figure 4-9 and the following:

$$\begin{aligned} W &= 851 \text{ kips} \\ \theta &= 120^\circ \\ R &= 91.25 \text{ in.} \\ A &= (R^2/2) (\theta - \sin \theta) = 5,114 \text{ in.}^2 \\ \gamma &= 851/5,114 = 0.1664 \text{ k/in.}^2 \\ q &= R\gamma (1 - \cos \theta/2) = 7.592 \text{ k/in.} \\ \text{Saddle material is A516, Gr. 70.} \\ \text{Thickness of web} &> 7.592/23.33 = 0.325 \text{ in. (Use } 1/2 \text{ in.)} \end{aligned}$$

At $\phi = 0^\circ$ (invert of saddle):

$$\begin{aligned} e &= 91.25 (1 - \cos 60^\circ) = 45.63 \text{ in.} \\ A_1 &= (91.25^2/2) (1 - \cos 60^\circ) (\sin 60^\circ - 0) = 1,803 \text{ in.}^2 \\ A_2 &= (91.25^2/2) [(60 - 0)\pi/180 - \sin (60 - 0)] = 754.3 \text{ in.}^2 \\ U &= (0.1664) (1,803 + 754.3) = 425.5 \text{ kips (Note that } U = 1/2 W) \\ V &= (91.25^2/2)(0.1664) (\cos 0^\circ - \cos 60^\circ) = 173.2 \text{ kips (tension)} \\ M &= 91.25 [(173.2) (1) - (425.5) (0)] = 15,804 \text{ in.-kip} \end{aligned}$$

Then:

$$\frac{M}{U} = \frac{15,804}{425.5} = 37.14 \text{ in.}$$

U acts downward 37.14 inches to the right of the center. V acts at the invert and is a tension force.

See Figure 17-11 for the force diagram.

The vertical reaction at the bearing plates acts at a distance of:

$$1/2 (12 \text{ ft} - 6 \text{ in.}) = 75.0 \text{ in.}$$

Therefore, the resultant moment equals:

$$-15,804 + (425.5) (75) = 16,109 \text{ in.-kip}$$

The saddle cross section (see Figure 17-12) at the invert is made up of a web plate (25 x 1/2 inches) and two flanges (12 x 1 1/4 inches).

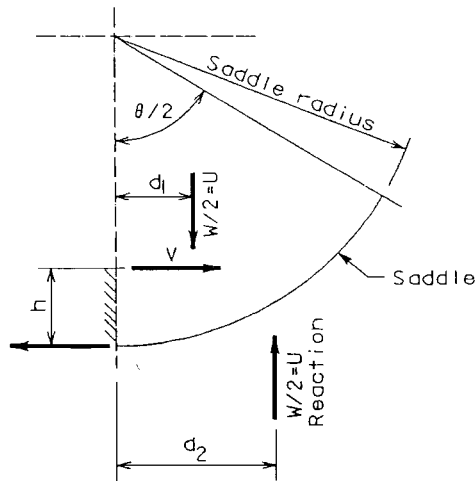


Figure 17-11 Force Diagram

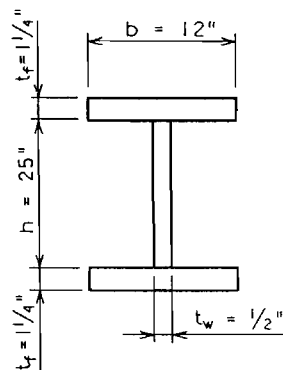


Figure 17-12 Saddle Cross Section

(2) Flanges

Section properties:

$$\text{Area (A)} = (2) (12) (1.25) + (25) (0.5) = 42.50 \text{ in.}^2$$

$$\text{Moment of inertia (I)} = (253) \left(\frac{1}{12} \right) (.5) + \frac{(26.252) (12 \times 1.25)}{2} + (2) (1.253) \left(\frac{15}{12} \right)$$

$$\text{Moment of inertia (I)} = 5,823.9 \text{ in.}^4$$

$$\text{Section modulus (Z)} = 5,823.9 / 13.75 = 423.6 \text{ in.}^3$$

Resolve forces and moments to the center of gravity of the section:

$$-16,109 + (173.2)(13.5) = -13,771 \text{ in.-kip}$$

This puts compression in the top flange.

Stresses at invert:

$$\text{Top flange stress} = 173.2/42.5 - 13,771/423.6 = 4.08 - 32.51 = -28.43 \text{ ksi}$$

$$\text{Bottom flange stress} = 173.2/42.5 + 13,771/423.5 = 4.08 + 32.51 = 36.59 \text{ ksi}$$

Referring to the AISC Specification for Structural Steel Buildings,⁵ the allowable bending for compact sections equals:

$$0.66 F_y = (0.66)(38) = 25.08 \text{ ksi}$$

Therefore, the saddle is overstressed.

Try using flanges with the following dimensions:

$$\text{Top flange} = 12 \text{ in.} \times 1\frac{1}{2} \text{ in.}$$

$$\text{Bottom flange} = 16 \text{ in.} \times 1\frac{1}{2} \text{ in.}$$

$$\text{Area (A)} = (12 + 16)(1.5) + (24.5)(0.5) = 54.25 \text{ in.}^2$$

$$\bar{x} = \frac{(18 \times 26.75) + (24 \times 0.75) + (12.25 \times 13.75)}{54.25} = 12.312 \text{ in.}$$

$$\text{Top arm} = 27.00 - \bar{x} = 27.00 - 12.312 = 14.688 \text{ in.}$$

New section properties:

$$\begin{aligned} \text{Moment of inertia (I)} &= (24.5^3)(1/12)(0.5) + (12.25 \times 1.438^2)(18) + \\ &\quad (11.562^2)(24) + \frac{(1.5^3)(12 + 16)}{12} = 7,351.1 \text{ in.}^4 \end{aligned}$$

$$\text{Section modulus, top (Z}_t\text{)} = 7,351.1/14.688 = 500.48$$

$$\text{Section modulus, bottom (Z}_b\text{)} = 7,351.1/12.312 = 597.07$$

Stresses at invert:

$$\text{Top flange stress} = 173.2/54.25 - 13,771/500.48 = -24.32 \text{ ksi}$$

$$\text{Bottom flange stress} = 173.2/54.25 + 13,771/597.07 = 26.26 \text{ ksi}$$

The bottom flange is slightly overstressed, but the stress is considered acceptable for purposes of this example.

(3) Web plate

Check shear in the web between flanges (refer to the AISC Specification for Structural Steel Buildings,⁵ F4, page 5-49).

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Compute the depth of the web over the center of bearing plate, 75 inches from the centerline.

$$\text{Radius} = 90 + 1.25 + 1.5 = 92.75 \text{ in.}$$

$$\text{Base to penstock centerline} = 92.75 + 24.5 \text{ (web at invert)} = 117.25 \text{ in.}$$

$$h = 117.25 - (92.75^2 - 75^2)^{1/2} = 62.68 \text{ in.}$$

$$h/t_w = 62.68/0.5 = 125.37$$

$$\text{Panel } a/h = \frac{(2)(75)}{62.68} = 2.4$$

$$380/\sqrt{F_y} = 61.64 < h/t_w$$

$$\text{Shear stress } (f_v) = \frac{425.5}{(62.68)(0.5)} = 13.58 \text{ ksi}$$

Allowable shear stress (F_v) equals:

$$F_v = (F_y/2.89) C_v$$

$$C_v = \frac{45,000k_v}{F_y(h/t_w)^2} \text{ when } C_v < 0.8$$

$$C_v = 190 \frac{\sqrt{k_v F_y}}{h/t_w} \text{ when } C_v > 0.8$$

$$C_v = \frac{(45,000)(6.03/38)}{(125.37)^2} = 0.454 \text{ (less than 0.8)}$$

$$k_v = \frac{4.00 + 5.34}{(a/h)^2} \text{ when } a/h < 1$$

$$k_v = \frac{5.34 + 4.00}{(a/h)^2} = \frac{5.34 + 4.00}{(150/62.68)^2} = 6.03 \text{ when } a/h > 1$$

$$F_v = \left(\frac{38}{2.89} \right) 0.454 = 5.97 \text{ ksi}$$

Try adding a stiffener at midspan and a 5/8-inch web:

$$\text{Shear stress } (F_v) = \frac{425.5}{(62.68)(0.625)} = 10.86 \text{ ksi}$$

$$h/t_w = 62.68/0.625 = 100.29$$

$$\text{Panel } a/h = \frac{1 \times 75}{62.68} = 1.20$$

$$C_v = 190 \frac{\sqrt{k_v F_y}}{h/t_w} \text{ when } C_v > 0.8$$

$$k_v = 5.34 + 4.00(1.20)^2 = 11.1 \text{ (when } a/h = 1.20 > 1.0)$$

$$C_v = 190 \frac{(11.1/38)^{1/2}}{100.29} = 1.0239 > 0.8$$

$$F_v = \left(\frac{38}{2.89} \right) 1.0239 = 13.46 \text{ ksi}$$

Use a 5/8-inch web plate and 5/8-inch intermediate stiffener at midspan.

(4) Bearing plate

Refer to Figure 17-13 and Section 17.4.2.3 for maximum bearing.

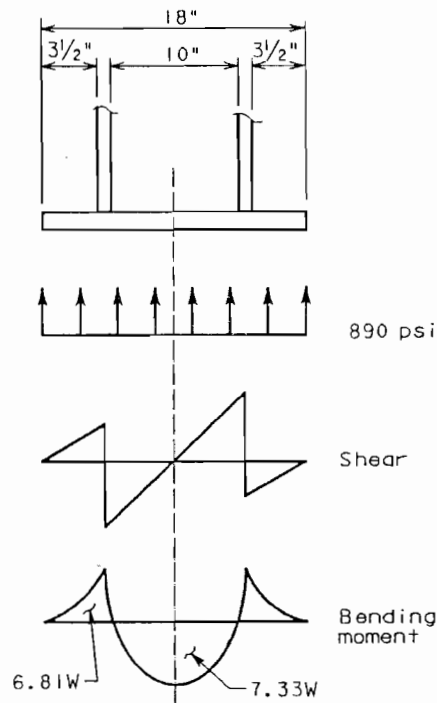


Figure 17-13 Bearing Plate Diagrams

Maximum DBC bearing pressure = 890 psi (let $w = 890$ psi)

Assuming 3/4-inch bearing stiffeners, the spans are 3.625, 10.75, and 3.625 inches (total of 18 inches)

Cantilever span moment = $(1/2) wl^2 = 5.85$ in.-k

Shear = 3.226 kips

At center span:

Moment = $(1/8) wl^2 - 5.85 = 7.01$ in.-kip

Shear = 4.78 kip

Refer to the AISC Specification for Structural Steel Buildings,⁵ F2, page 5-48 for allowable bending stress in a flat plate across thickness.

Allowable bending stress = $(0.75) (38) (1.33) = 37.91$ ksi

Required thickness = $\sqrt{\frac{6M}{S}} = \sqrt{\frac{6(7.01)}{37.91}} = 1.053$ in. (use $1\frac{1}{8}$ in. bearing plates)

Bearing stiffeners (try 3/4-inch stiffeners)

$$\begin{aligned}f_b &= 9 \times 0.89 / 0.75 = 10.68 \text{ ksi} \\b/t &= (39.0 - 0.625) / [(2) (0.75)] = 25.58 \\h/t &= 62.68 / 0.75 = 83.6\end{aligned}$$

Refer to the AISC Specification for Structural Steel Buildings,⁵ B5, page 5-98 for compression for a b/t ratio of 25.58.

$$K_c = \frac{4.05}{(h/t)^{0.46}} = 0.529$$

$$\sqrt{\frac{F_y}{K_c}} = 8.476$$

$$\frac{195}{\sqrt{F_y/K_c}} = 23.01 < b/t = 25.58$$

$$Q_s = \frac{26,200 K_c}{F_y (b/t)^2} = 0.5574$$

The allowable compression in the bearing stiffener equals:

$$0.60 F_y Q_s = 12.71 \text{ ksi}$$

Since the seismic DBC is being checked, the allowable could be increased by 1.33.

17.4.2.11 Saddle Welds

The allowable fillet weld shear equals:

$$0.3 \text{ (tensile strength of weld metal)} = 21.0 \text{ ksi.}$$

Tension or compression partial penetration welds; same allowable as base material.

First bearing stiffener:

The length of the first bearing stiffener (69.62 inches from the center) equals:

$$117.25 - \sqrt{92.75^2 - 69.62^2} = 55.97 \text{ in. less 1 in. for clipping of corners, or 54.97 in.}$$

As calculated previously, the maximum q load at the invert acting through the flange equals 7.592 k/in.

Bearing stiffener-to-bearing plate weld:

The required partial penetration weld size for the bearing stiffener to bearing plate equals:

$$\frac{(9)(0.89)}{23.3} = 0.344 \text{ in.}$$

(Weld is partial penetration type; therefore, allowable is 23.3 ksi.)

The required fillet-weld size equals:

$$\frac{(9)(0.89)}{(21)(0.707)} = 0.540 \text{ in.}$$

Use two 3/8-inch fillet welds.

Bearing stiffener-to-web weld:

The required size for the bearing stiffener-to-web weld equals:

$$\frac{(9)(0.89)(19.19/54.97)}{(21)(0.707)} = 0.188 \text{ in.}$$

Use two 1/4-inch fillet welds.

Bottom flange-to-web weld:

Shear = 425.5 kips

The depth at the first bearing stiffener center-to-center of flanges equals:

$$55.97 + 1.5 = 57.47 \text{ in.}$$

Therefore, the VQ/I shear equals:

$$\frac{425.5}{57.47} = 7.40 \text{ kips/in.}$$

The double-fillet weld size equals:

$$\frac{7.4}{(21)(0.707)(2)} = 0.249 \text{ in.}$$

Use two 1/4-inch fillet welds.

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Top flange-to-web weld:

Vectorially sum the q and VQ/I shears:

$$\sqrt{7.40^2 + 7.592^2} = 10.60 \text{ k/in.}$$

Weld size equals:

$$\frac{10.60}{(21)(2)(0.707)} = 0.357 \text{ in.}$$

This size can be reduced by using a q value calculated at the stiffener equal to:

$$q = \frac{(0.433)(851)}{91} = 4.05 \text{ kip/in.}$$

Then the shear load to be resisted equals:

$$\sqrt{7.40^2 + 4.05^2} = 8.44 \text{ k/in.}$$

And the weld size required to resist this shear load equals:

$$\frac{8.44}{(2)(21)(0.707)} = 0.284 \text{ in.}$$

Use two 5/16-inch fillet welds.

Design keeper bars (at each side of bearing plates):

Effective length = 39 in.

Lateral load DBC = $(851)(0.206) = 175.3$ kips (see Section 17.4.2.3(1))

Keeper bar thickness, min. = $\frac{175.3/39}{(0.4)(38)} = 0.296$ in.

Use a keeper bar for each leg, $1\frac{1}{2}$ in. $\times$ $1\frac{1}{2}$ in. $\times$ 3 ft-8 in. long. Attach each bar to the base plate with a $\frac{1}{2}$ -inch groove weld on the inside and a $\frac{3}{8}$ -inch fillet weld on the outside.

Note: The bars are made longer than the bearing plate to provide for temperature movements of the penstock, i.e., movement between anchor points for full range of temperature.

Other detail design and checks for DSC are left to the designer (see Figure 17-9).

17.4.3 Stiffeners for External Pressure (Exposed Penstock)

Refer to Section 17.4.2 for support spacing and ring girder sizes. Ring girders at support points are end stiffeners for buckling considerations and should be designed in accordance with the requirements of Section 4.4.3.

Given data:

$$\begin{aligned}\text{External pressure} &= 6.5 \text{ psi (15 ft)} \\ L &= 62.28 \text{ ft} \times 12 = 747.36 \text{ in.} \\ D_o &= 15 \text{ ft} + 1.623 \text{ in.} = 181.62 \text{ in.} \\ t \text{ (min.)} &= 0.813 \text{ in. (downstream of backfilled area)}\end{aligned}$$

Refer to the ASME Code, Section VIII, Division 1:

From Figure 5–UCS-28.2:

$$P_a = \frac{2AE}{3(D_o/t)}$$

$$E = 29.0 \times 10^6 \text{ psi}$$

From Figure 5–UGO-28.0:

$$A = 0.000091$$

Then:

$$P_a = \frac{2(0.000091)(29.0 \times 10^6)}{3(181.62/0.813)} = 7.875 \text{ psi}$$

From UG-29, the required moment of inertia (I_s') using the composite stiffener plus shell in calculating moment of inertia equals:

$$I_s' = \frac{D_o^2 L_s [t + (A_s/L_s)] A}{10.9}$$

Where:

$$\begin{aligned}A_s &= 80.75 \text{ in.}^2 \\ L_s &= L = 747.36 \text{ (in this case)} \\ A &= 2B/E = \frac{2(961.3)}{29.0(10^6)} = 0.0000663 \\ P &= 6.5 \text{ psi} \\ t &= 0.813 \text{ in.} \\ B &= \frac{3}{4} \left(\frac{PD_o}{t + A_s/L_s} \right) = \frac{3}{4} \left[\frac{(6.5)(181.62)}{(0.813) + (80.75/747.36)} \right] = 961.3 \text{ psi}\end{aligned}$$

Then:

$$I_s' = \frac{(181.62^2)(747.36)(0.813 + 80.75/747.36)(.0000663)}{10.9} = 138.11 \text{ in.}^4$$

$$I_{\text{provided}} = 1,280 \text{ in.}^4$$

Assuming an external pressure of 12 psi, then intermediate stiffeners between support ring girders would be required.

End stiffener parameters (note change from those above because of change in pressure):

$$B = 1,775.0 \text{ psi}$$

$$A = 0.0001224$$

$$I_s' = 255 \text{ in.}^4$$

$$I_{\text{provided}} = 1,280 \text{ in.}^4$$

Design intermediate stiffeners placed at midspan.

$$L = 747.36/2 = 373.68 \text{ in.}$$

$$LD_o = 2.0575$$

From Figure 5-UGO-28.0:

$$A = 0.00019$$

$$P_a = 2AE/[3(D_o/t)] = \frac{.00019}{.000091} \times 7.875 = 16.44 \text{ psi (greater than 12.0 psi)}$$

Therefore, use one intermediate stiffener at midspan between supports.

The moment of inertia (I) of intermediate stiffener rings should meet the following formula (see Section 4.4.3):

$$I \geq \frac{pLD_o^3}{77,300,000(N^2 - 1)}$$

Where:

$$N^2 = \frac{0.663}{\left(\frac{H}{D_o}\right)\left(\frac{t}{D_o}\right)^{1/2}} \leq 100$$

$$H = 747.36 \text{ in.}$$

$$N^2 = \frac{0.663}{4.115 \left(\frac{1}{223.4} \right)^{1/2}} = 2.41$$

$$\text{Minimum } N^2 = 4$$

Therefore:

$$I \geq \frac{(12.0)(373.68)(181.62^3)}{77,300,000(4-1)} = 115.84 \text{ in.}^4$$

The participating portion of the shell is equal to:

$$1.56\sqrt{Rt} = 1.56\sqrt{90.4 \times 0.813} = 13.37 \text{ in.}$$

Try a 10 × 0.75-inch plate stiffener. Refer to Figure 17-14.

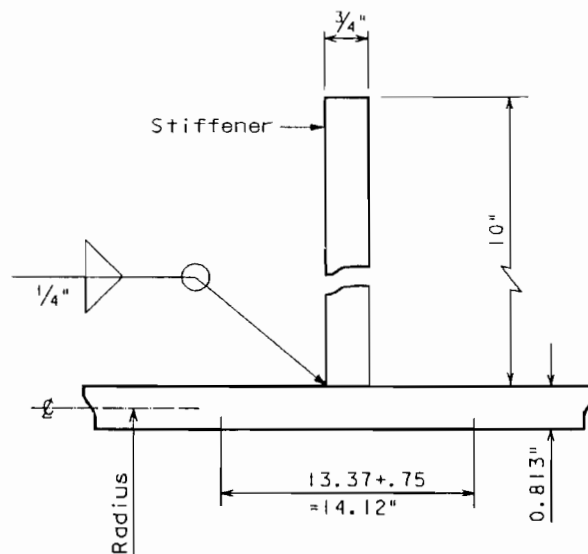


Figure 17-14 Plate Stiffener

$$\text{Area} = 18.98 \text{ in.}^2 \text{ (stiffener and shell)}$$

$$I_o = 195.1 \text{ in.}^4$$

The required I of 115.84 in.⁴ is less than the I provided. Therefore, use a 10 × 0.75-inch intermediate stiffener. Attach the stiffener to the shell with continuous 1/4-inch fillet welds on both sides.

17.4.4 Bend Design

This section gives bend design calculations for bend #3 (station 12 + 50, el. 586.0). The bend is a 15 × 12-foot reducing bend with the central angle equal to 39.6927 degrees). This section includes calculations for the layout of the reducing bend and the shell wall thickness. For the calculations, normal pressure = (764.61) (0.433) = 331 psi.

17.4.4.1 Layout of Reducing Bend

Refer to Figure 17-15 of this manual and Figure 22 of "Welded Steel Penstocks and Tunnel Liners, Steel Plate Engineering Data—Volume 4," published by AISI/SPFA (1984).⁶

Reducing bend dimensions:

Radius (R) of bend = $4D = 4(15) = 60$ ft

Tangent length (TL) = $(R)(\tan \Delta / 2) = 60 \tan(39.6927/2) = 21.6562$ ft = 259.87 in.

$n = 2$ (number of deflections) = 6

Inside diameter of large end (D_l) = 180 in. = 15 ft-0 in.

Inside diameter of small end (D_n) = 144 in. = 12 ft-0 in.

Central angle (Δ) = 39.6927°

Reducing bend formulas:

$$\rho = \frac{\Delta}{n} = \frac{39.6927^\circ}{6} = 6.6155^\circ$$

$$\sin \theta = \frac{D_l - D_n}{2(n-2)R \tan \rho} = \frac{15 - 12}{2(6-2)60 \tan 6.6155^\circ} = 0.0539$$

$$\theta = \sin^{-1} 0.0539 = 3.0892^\circ$$

$$\cos \theta = 0.9985$$

$$\text{Inside radius of large end } (r_l) = \frac{D_l}{2} = 180/2 = 90.0000 \text{ in.}$$

$$\text{Inside radius of small end } (r_n) = \frac{D_n}{2} = 144/2 = 72.0000 \text{ in.}$$

$$\begin{aligned} \text{Inside radius at any division } (r_x) &= r_l - (x-1)R \tan \rho \sin \theta \\ &= 90 - (x-1)(60)(12)(0.1160)(0.0539) \\ &= 90 - 4.5000(x-1) \end{aligned}$$

(x is the number of divisions from the point of curvature (PC) to the point under consideration.)

Inside diameter at any division (D_x) equals:

$$\begin{aligned} &\frac{D_l - 2(x-1)R \tan \rho \sin \theta}{\cos \theta} \\ &= \frac{180 - 2(x-1)(60)(12)(0.1160)(0.0539)}{0.9985} = 180.2704 - 9.0000(x-1) \end{aligned}$$

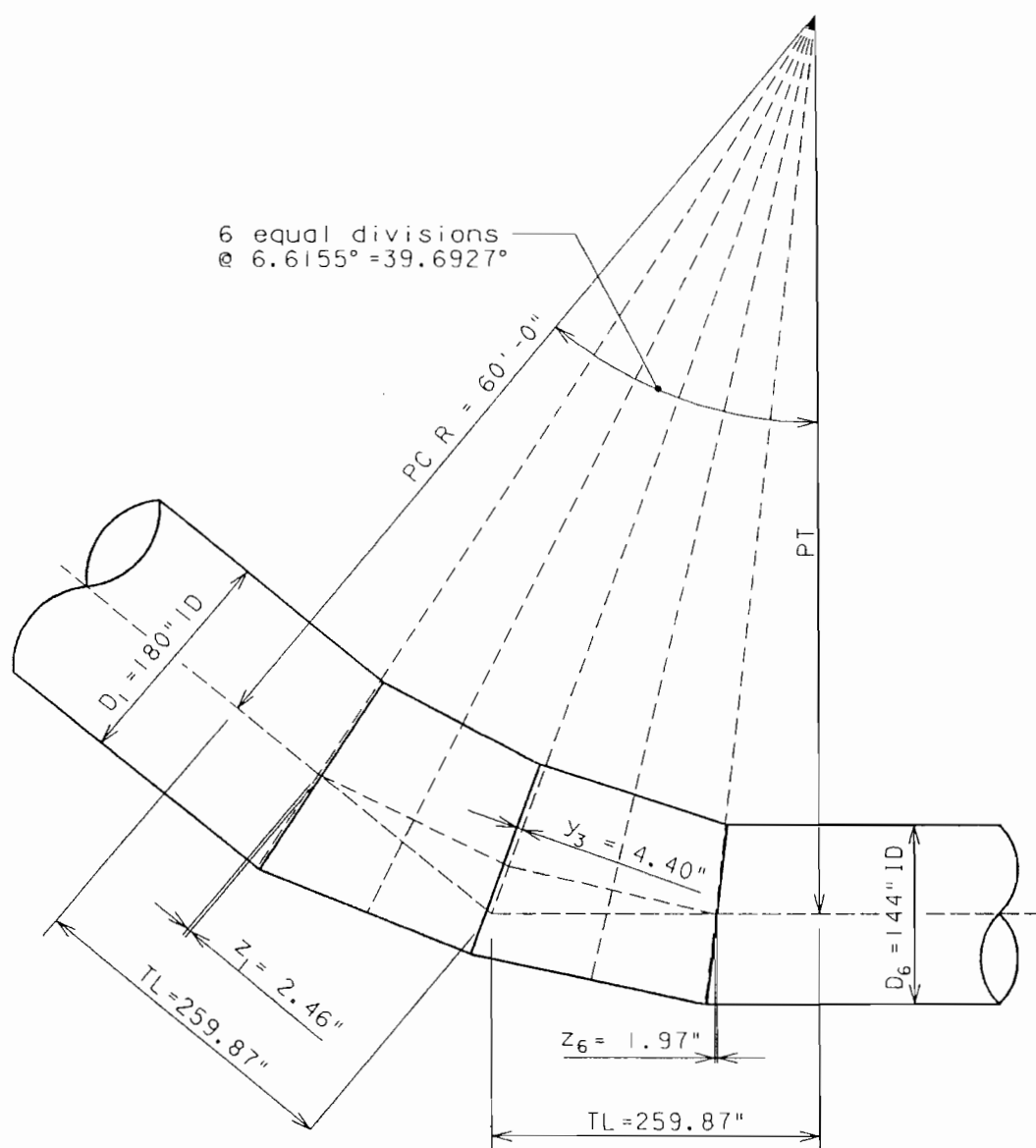


Figure 17-15 Reducing Bend

Inside diameters for the divisions at D_2 and D_4 equal:

$$D_2 = 180.27 - 9.0000 (2 - 1) = 171.26 \text{ in.}$$

$$D_4 = 180.27 - 9.0000 (4 - 1) = 153.23 \text{ in.}$$

$$\tan \phi = \frac{\sin 2p}{\cos 2p + \cos \theta} = \frac{0.2289}{0.9735 + 0.9985} = 0.1161$$

$$\phi = \tan^{-1} 0.1161 = 6.6203^\circ$$

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$$z_1 = \frac{r_1 \sin \theta}{\cos 2\rho + \cos \theta} = \frac{(90)(0.0539)}{0.9735 + 0.9985} = 2.4599 \text{ in.}$$

$$z_6 = \frac{r_6 \sin \theta}{\cos 2\rho + \cos \theta} = \frac{(72)(0.0539)}{0.9735 + 0.9985} = 1.9680 \text{ in.}$$

$$y_x = \frac{r_x \sin \theta}{\cos \rho} = \frac{r_x (0.0539)}{0.9933} = 0.0543 r_x$$

$$y_1 = 0.0543 (r_1) = 0.0543 (90) = 4.8870 \text{ in.}$$

$$y_3 = 0.0543 (r_3) = 0.0543 [90 - 4.5000 (3 - 1)] = 4.3983 \text{ in.}$$

$$y_5 = 0.0543 (r_5) = 0.0543 [90 - 4.5000 (5 - 1)] = 3.906 \text{ in.}$$

17.4.4.2 Shell Wall Thickness of Reducing Bend

The required elbow wall thickness (t) is given by:

$$t = \frac{PD}{Sf} \left[\frac{D}{3} \tan \left(\frac{\theta}{2} \right) + \frac{S}{2} \right]$$

Refer to Section 4.5.3 for a discussion of stress analysis.

Design pressure, normal conditions (P_n) = 331 psi

Shell stress, normal conditions (f_n) = 23,300 psi (for ASTM A516, Grade 70)

Segment deflection angle (θ) = 13.23°

Segment length inside of bend (S) equals:

$$R - \frac{D}{2} \tan \rho = 720 - \frac{180}{2} \tan 6.615^\circ = 146.12 \text{ in. (approximate formula)}$$

Outside diameter of elbow (D) = 183 in. (approx.)

The shell thickness under normal conditions (t_n) equals:

$$(t_n) = \frac{(33)(183)}{(146)(23,300)} \left[\frac{183}{3} \tan \left(\frac{13.23}{2} \right) + \frac{146}{2} \right] = 1.43 \text{ in.}$$

17.4.5 Transition Design

The transition is square to circular (15 × 15-foot square to 15 feet in diameter) located at stations 0 + 00 to 0 + 10 (see Section 17.1 and Table 17-1).

The external design pressure equals 1,165 – 815 = 350 feet (152 psi). Use a stiffened design.

The thickness required to contain internal pressure equals 0.588 inch.

17.4.5.1 Shell Thickness and Stiffener Spacing

Solve for ν/L using the following moment and stress equations:

$$\text{Moment } (M) = \frac{pL^2}{9}$$

$$\text{Stress } (S) = \frac{6M}{t^2}$$

Where:

L = spacing between stiffeners on surface of transition

Pressure (p) = 152 psi

Allowable stress (S) = 23,300 psi $\times$ 1.5 = 35,000 psi

Shape factor for bending (X) = 1.5 (see Section 3.4.7)

t = Thickness

Then, substituting and reducing the above moment and stress equations yields the following:

$$\nu/L = \sqrt{\left(\frac{2p}{3S}\right)} = \sqrt{\left(\frac{2 \times 152}{3 \times 35,000}\right)} = 0.0538$$

Then:

For $t = 11/16$ in., $L = 12.8$ in.

For $t = 3/4$ in., $L = 13.9$ in.

For $t = 5/8$ in., $L = 11.6$ in.

Therefore, use a 5/8-inch shell and 12-inch stiffener spacing.

Studs on 12-inch centers could be considered, but the size required is large—approximately $(12^2 \times 152)/10,000 = 2.19$ square inches (1.67-inch diameter). Therefore, use continuous tee-ring frames.

17.4.5.2 Concrete Stress Values and Allowables

It is assumed that the specified minimum concrete strength at 28 days (f'_c) equals 4,000 psi. Then the allowable bearing (f_b) equals 0.375, $f'_c = 1,500$ psi. The allowable diagonal shear (γ) equals $1.1 (f'_c)^{1/2} = 70$ psi (from UBC, Section 2626 (d)) and $\gamma = 1.7 (f'_c)^{1/2} = 108$ psi for bond for plain bars or studs.

The ultimate capacity diagonal tension (f_t) equals $4(f'_c)^{1/2} = 253$ psi. Applying a load factor of 1.7 and reduction factor (ϕ) of 0.65, the allowable diagonal tension equals $253 \times 0.65/1.7 = 97$ psi.

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Determine width of flange of tee as follows:

$$\text{Width of flange of tee} = \frac{\text{Span} \times \text{Pressure}}{\text{Allowable bearing}} = \frac{12 \text{ in.} \times 152 \text{ psi}}{1,500 \text{ psi}} = 1.22 \text{ in. (use 3 in.)}$$

$$\text{Bearing load/in.} = 12 \text{ in.} \times 152 \text{ psi} = 1,824 \text{ lb/in. of flange}$$

Determine flange thickness (t) by the formula:

$$t = \sqrt{\left(\frac{6M}{S(1.5)} \right)}$$

Where:

$$\text{Bending moment } (M) = (1/8) (1,824) (3.0 - 0.375)^2 = 1,571 \text{ in-lb/in.}$$

Then:

$$t = \sqrt{\left(\frac{(6) (1,571)}{(23,300) (1.5)} \right)} = 0.52 \text{ in. (use } 5/8 \text{ in.)}$$

$$\text{Tension through web } (\sigma) = 1,824/0.375 = 4,860 \text{ psi (using } 3/8\text{-inch web)}$$

Fillet weld allowable:

$$0.3 \times 60 = 18 \text{ ksi} \times 0.707 = 12.7 \text{ kip/1-in. fillet weld (AISC Specification for the Design, Fabrication and Erection of Structural Steel for Buildings}^7)$$

$$\text{Required weld size} = 1,824/12,700 = 0.144 \text{ in.}$$

Required stitch weld per foot of stiffener:

$$\frac{0.144 \times 12 \text{ in./ft}}{.25 \text{ in.}} = 6.9 \text{ in. (or 7 in.)}$$

Use 1/4-inch stitch welds 4 inches long at pitch each side of 12 inches.

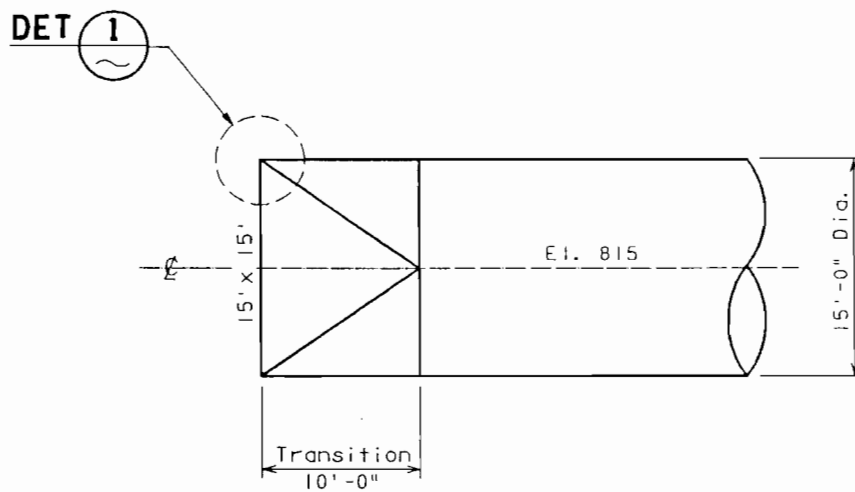
Required stiffener depth:

Use a diagonal tension allowable of 97 psi and determine the depth (d)

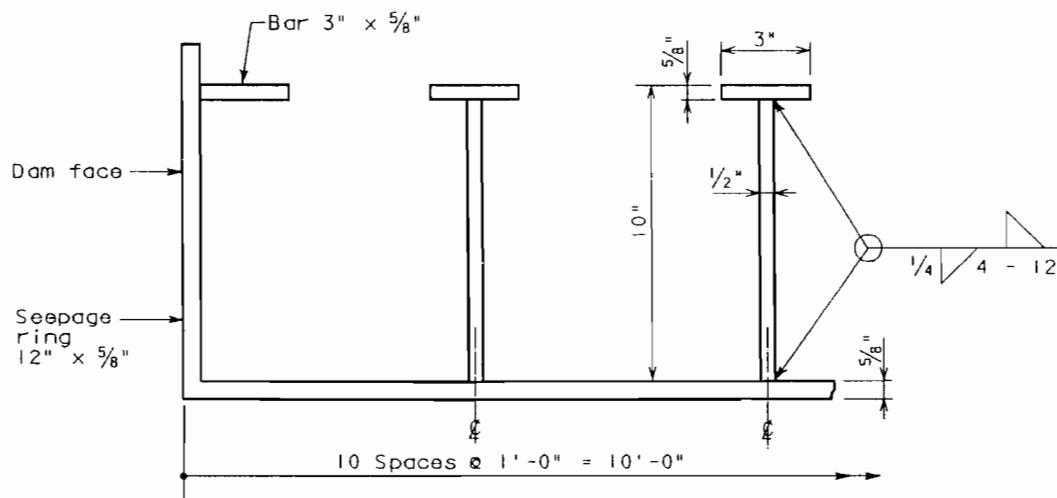
$$(2d) (1.414) (97) (0.707) = 1,824 \text{ lb}$$

$$d = 1824/194 = 9.4 \text{ in. (use 10 in.)}$$

Refer to Figure 17-16 for transition details.



ELEVATION



DETAIL ①

Figure 17-16 Transition Details

17.5 Expansion Joints/Sleeve Couplings

17.5.1 Mechanical Expansion Joint Design

(1) Wall thickness

First determine sleeve thickness (t) using the revised Barlow hoop stress formula:

$$S = pD/2t$$

or by:

$$t = pD/(2S - p)$$

Where:

- p = working pressure = (517.25 ft) (.4335) = 224.22 psi
- D = outside pipe diameter = 180 in.
- S = (allowable) for ASTM A516 Gr. 70 equals the lesser of 2/3 of yield or 1/3 of ultimate, or 23,333 psi

Then:

$$t = \frac{224.22 \times 180}{46,667 - 224.22} = .87 \text{ in. (use } 7/8 \text{ in.)}$$

(2) Number of bolts required

Determine minimum number of bolts required:

$$\text{Pipe OD} = 180 + 1.069 + 1.069 = 182.14 \text{ in.}$$

$$\text{Minimum number of bolts} = \text{Pipe OD} \times .400 = 182.14 \times .400 = 72.9 \text{ (use 74 bolts)}$$

Note that the number of bolts required may vary substantially depending on manufacturer or gasket material used. Bolts are ASTM A307 Gr. A, 13/16-inch diameter, with a bolt force of 7,680 pounds.

(3) Gasket pressure

Determine gasket pressure:

$$\text{Gasket thickness} = 1 \text{ in.}$$

$$\text{Gasket area/bolt} = \frac{(182.14 + 1) \pi}{74} = 7.78 \text{ in.}^2$$

$$\text{Gasket pressure} = 7,680/7.78 = 987 \text{ psi}$$

$$\text{Gasket pressure/working pressure} = 987/224.22 = 4.4 \times \text{working pressure (acceptable)}$$

A gasket pressure equal to 3.5 times the working pressure is acceptable. Note that gasket pressure may vary substantially depending on the manufacturer or gasket material used.

17.5.2 Bolted Sleeve-Type Coupling Design

(1) Wall thickness

First determine wall thickness (t) using the revised Barlow hoop stress formula:

$$S = pD/2t$$

or by:

$$t = pD/(2S - p)$$

Where:

- p = working pressure = (771.8 ft) (.4335) = 335 psi
- D = outside pipe diameter = 146.1 in.
- S = (allowable) for ASTM A516 Gr. 70 equals the lesser of 2/3 of yield or 1/3 of ultimate, or 23,333 psi

Then:

$$t = \frac{335 \times 146.1}{46,667 - 335} = 1.05 \text{ (use } 1\frac{1}{16} \text{ in.)}$$

(2) Number of bolts required

Determine minimum number of bolts required:

$$\text{Minimum number of bolts} = \text{Pipe OD} \times .400 = 146.1 \times .4 = 58.44 \text{ (use 60 bolts)}$$

Note that the number of bolts required may vary substantially depending on manufacturer or gasket material used. Bolts are ASTM A307 Gr. A, 13/16-inch diameter, with a bolt force of 7,680 pounds.

(3) Gasket pressure

Determine gasket pressure:

$$\text{Gasket thickness} = .625 \text{ in.}$$

$$\text{Gasket area/bolt} = \frac{(146.1 + .625) (.625) \pi}{60} = 4.8 \text{ in.}^2$$

$$\text{Gasket pressure} = 7,680/4.8 = 1,600 \text{ psi}$$

$$\text{Gasket pressure/working pressure} = 1,600/335 = 4.78 \times \text{working pressure (acceptable)}$$

A gasket pressure equal to 3.5 times working pressure is acceptable. Note that gasket pressure may vary substantially depending on manufacturer or gasket material used.

17.6 Anchor Block Design

The anchor block is at bend No. 2 of the exposed/supported penstock described in Section 17.4.2. Figure 17-17 shows the penstock reach anchored at bend No. 2 and the detail of the penstock bend.

17.6.1 Given Data

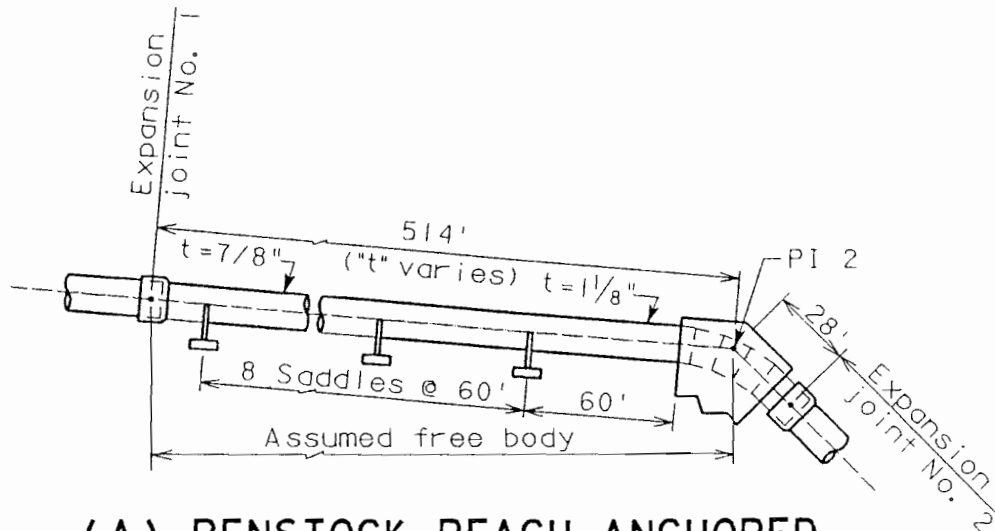
The following data is derived from Figures 17-8 and 17-17 or as given in Section 17.1.1.

Slope upstream of anchor (S_1) = -0.146 (-8.31°)
Slope downstream of anchor (S_2) = -0.830 (-39.69°)
Bend No. 2 angle of intersection = 31.38°
Design flow (Q) = $2,600 \text{ ft}^3/\text{sec}$.
Inside penstock diameter (D) = 15.00 ft
Inside penstock area (A) = 176.71 ft^2
Penstock thickness varies, $t = 7/8 \text{ in. to } 1\frac{1}{8} \text{ in.}$
Dead load of water = $11,027 \text{ lb/ft}$
Design flow velocity (v) = 14.71 ft/sec .
Inside area under upper expansion joint, using outside diameter (A_{xu}) = 180.17 ft^2
Inside area under lower expansion joint (A_{xl}) = 181.16 ft^2
Friction coefficient for bearings under saddles (f) = 0.12
Friction at expansion joint per inch of penstock diameter (EJF) = 500.00 lb

17.6.2 Design Method

Following is the generalized design method for the anchor at Bend No. 2. Loads acting are grouped by the system given in Section 8.2.2 and, in particular, Figures 8-2 and 8-4.

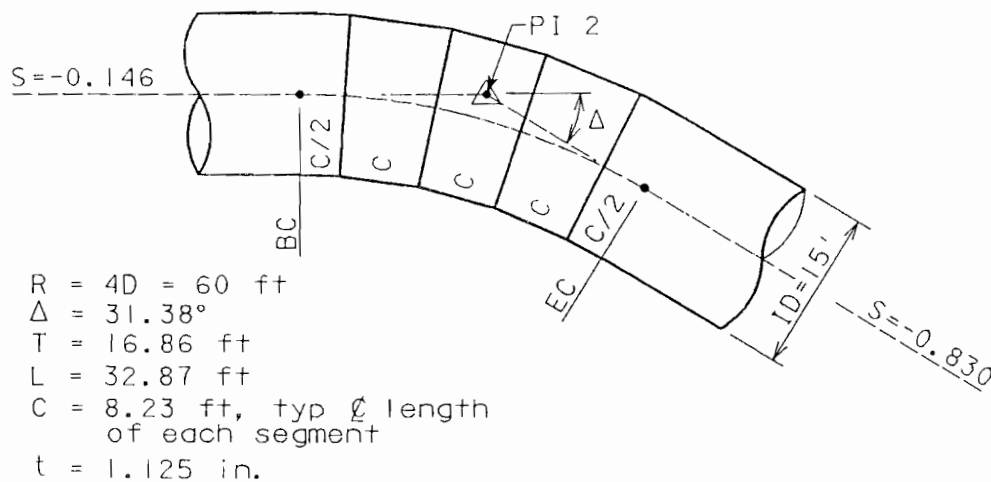
- (1) Assume a design for the upper part of the anchor block. Apply penstock bend loads entirely to this part of the anchor (see Figure 17-18).
- (2) Apply loading from groups "A" and "B" to the assumed upper part; consolidate all loads into two prime resultants and one moment about PI.
- (3) Add additional mass of concrete (or other systems) under the assumed upper part until an acceptable design is achieved.
- (4) Check the stability of the total anchor under seismic loadings, for other upsurge conditions, and for hydro-testing during construction.



(A) PENSTOCK REACH ANCHORED AT BEND NO. 2

NOTE:

Dimensions simplified for
this example.



(B) DETAIL OF PENSTOCK BEND

Figure 17-17 Anchored Penstock at Bend No. 2

This example considers a penstock on saddle supports and joined by expansion joints. Analysis for another anchor or for a sleeve-coupled penstock would be similar. Assumed details for the top portion of the concrete encasement are typical.

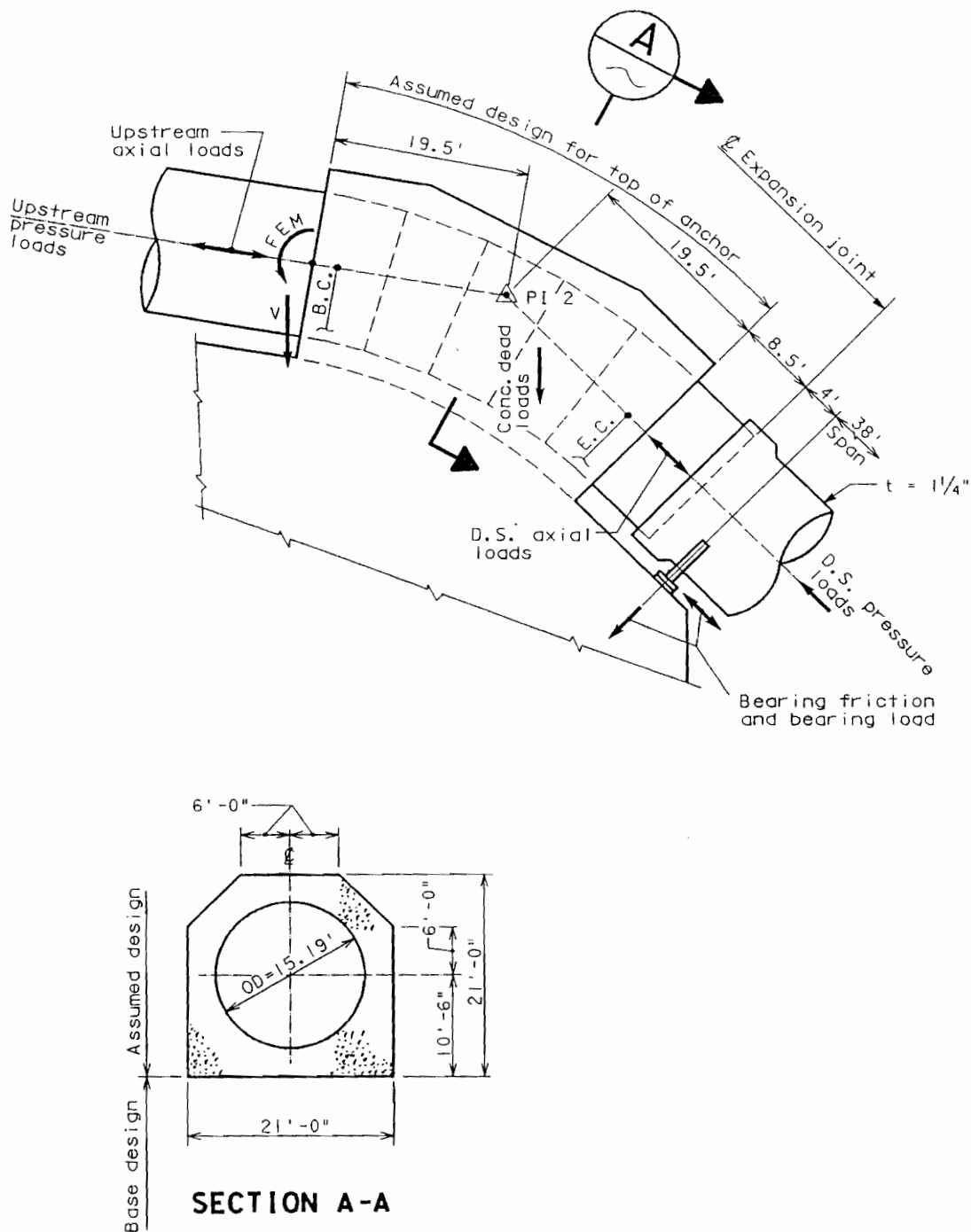


Figure 17-18 Assumed Anchor Design

The design for the base block depends on geotechnical conditions at each site and is beyond the scope of this example.

17.6.3 Group A Components of Design Loading

Group A is not affected by variables related to moving water. Friction loads are applied in the most adverse direction. All axial loads meet at PI.

- (1) Axial loads upstream of anchor (see Figures 17-8 and 17-18)

$$L = 514 - 19.5 = 494.5 \text{ ft}$$

Total estimated weight of upstream steel penstock (W_s) = 1,064 k.

Total weight of water (W_w) = 5,453 k.

(Typically, compute both values for later use in seismic loading calculations.)

Total $W_s + W_w = 6,517 \text{ k}$.

- (a) Axial load

$$L_{ax}: W_{sa} = (W_s + W_w) (\sin 8.31^\circ) = 942 \text{ k.}$$

- (b) Total friction loads on all saddle bearings

$$SFL_{ax} = (f) (W_{sa} + W_{wn}) = (0.12) (1,064 + 5,453) (\cos 8.31^\circ) = 774 \text{ k.}$$

- (c) Friction of upper expansion joint

$$EJL_{ax} = (500 \text{ lbs/in.}) (15 \text{ ft}) (12/1000) = 90 \text{ k.}$$

- (d) Total upstream axial loads

$$\text{Total} = 942 + 774 + 90 = +1,806 \text{ k.}$$

- (2) Gravity loads from embedded penstock barrel in top portion of anchor block (see Figure 17-19)

A single bevel is used at PI to shorten these calculations.

- (a) Upstream leg

$$L = 19.5 \text{ ft.}$$

Dead load (DL) of steel = 44 k.

Dead load (DL) of water = 215 k.

Total usDL = 259 k.

Upstream axial = UAX = (usDL) ($\sin 8.31^\circ$) = +37 k.

Upstream normal = UN = (usDL) ($\cos 8.31^\circ$) = 256 k.

UN horizontal = -37 k.

UN vertical = -253 k.

Moment of UN about PI = 256 (9.75) = -2,496 ft-k.

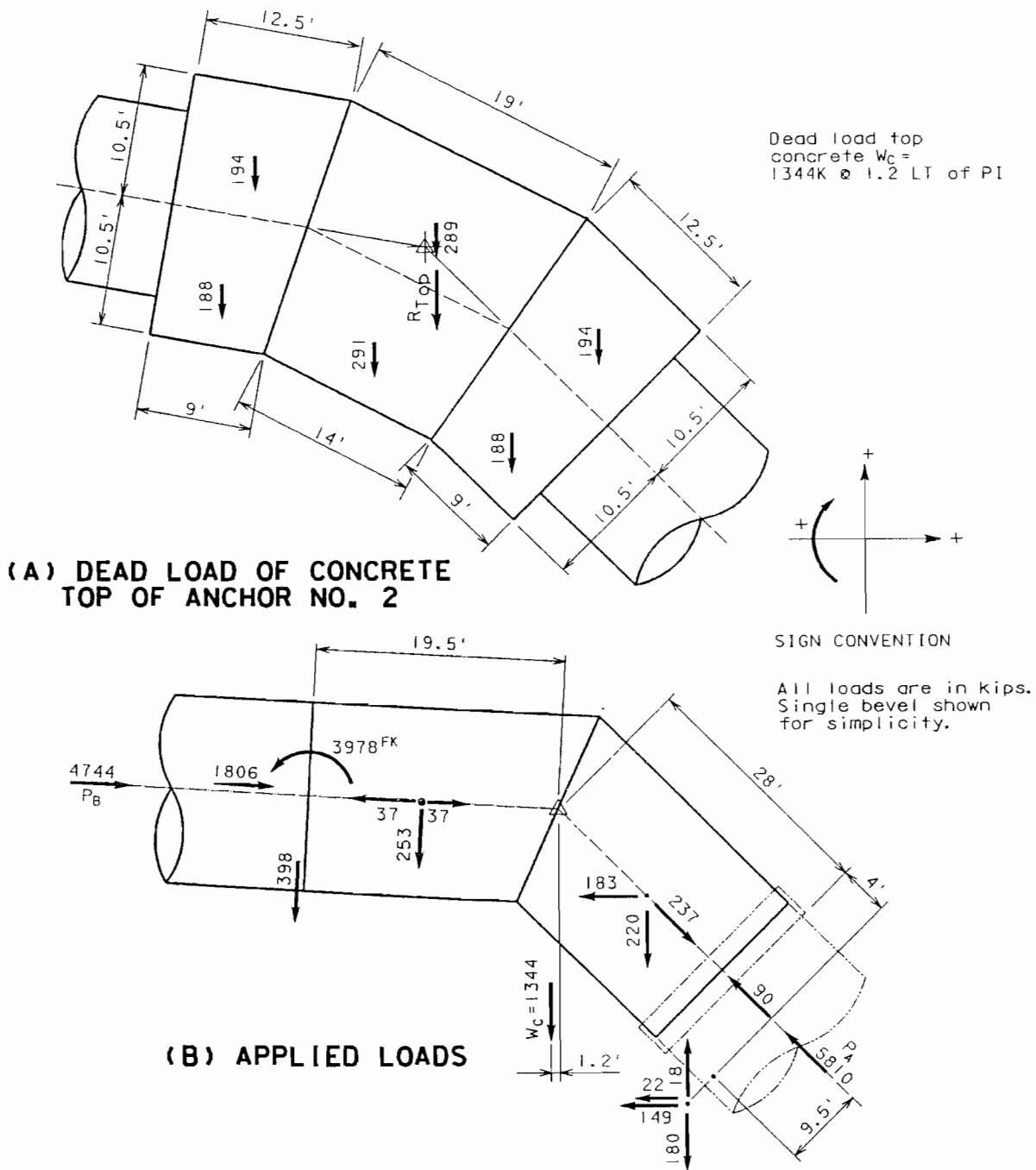


Figure 17-19 Anchor Loads

(b) Downstream leg

$$L = 28 \text{ ft.}$$

$$\text{Dead load (DL) of steel} = 63 \text{ k.}$$

$$\text{Dead load (DL) of water} = 309 \text{ k.}$$

$$\text{Total dsDL} = 372 \text{ k.}$$

$$\text{Downstream axial} = \text{DAX} = (\text{dsDL}) (\sin 39.69^\circ) = +237 \text{ k.}$$

$$\text{Downstream normal} = \text{DN} = (\text{dsDL}) (\cos 39.69^\circ) = 286 \text{ k.}$$

$$\text{DN horizontal} = -183 \text{ k.}$$

$$\text{DN vertical} = -220 \text{ k.}$$

$$\text{Moment of DN about PI} = 286 (14) = +4,004 \text{ ft-k.}$$

(3) Fixed end loads at upstream limit of concrete.

$$(a) \text{ Shear dead load for 60-foot span: steel} = 134 \text{ k.; water} = 661 \text{ k.}$$

$$V = 1/2 DL = -398 \text{ k.}$$

$$\text{Moment} = (398) (19.5) = -7,761 \text{ ft-k.}$$

(b) Fixed end moment equals:

$$\frac{w l^2}{12} = \frac{(13.26) (60)^2}{12} = -3,978 \text{ ft-k.}$$

(4) Other gravity loads affecting anchor

(a) Bearing load from downstream saddle

$$t = 1\frac{1}{4} \text{ in.}$$

$$L = 38 \text{ ft}$$

$$\text{Dead load end reaction of shell, saddle, and expansion joint} = 56 \text{ k.}$$

$$\text{Dead load reaction of water} = 248 \text{ k.}$$

$$\text{Total dead load (DL)} = 304 \text{ k.}$$

$$\text{Normal load} = (\text{DL})(\cos 39.69^\circ) = 234 \text{ k. (All axial loads, this reach, resisted at Bend 3)}$$

$$\text{NL horizontal} = -149 \text{ k.}$$

$$\text{NL vertical} = -180 \text{ k.}$$

$$\text{Moment about PI} = (234) (28 + 4 \text{ ft}) = +7,488 \text{ ft-k.}$$

(b) Bearing friction, downstream saddle

$$\text{SFL} = (0.12)(234) = 28 \text{ k.}$$

$$\text{Horizontal} = -22 \text{ k.}$$

$$\text{Vertical} = +18 \text{ k.}$$

$$\text{Moment about PI} = (28) (9.50) = +266 \text{ ft-k.}$$

(c) Friction of downstream expansion joint

$$EJF_{ax} = -90 \text{ k.}$$

(5) Static pressure vectors acting at PI

Static pressure is mobilized at joints nearest the anchor. No correction is made for flow friction. No correction is made for elevation because the water component is included in other axial loads.

(a) At upper expansion joint (see Table 17-1)

$$P_B = (H) (A_{x1}) (0.0624) = (422) (180.17) (0.0624) = +4,744 \text{ k. (H = ft of water)}$$

(b) At downstream expansion joint (see Figure 17-17)

$$P_A = (H) (A_{x1}) (0.0624) = (514) (181.16) (0.0624) = -5,810 \text{ k.}$$

(6) Dead load of assumed concrete encasement; designer's option

Detailed calculations are not given (see Figures 17-18 and 17-19).

$$\begin{aligned} \text{Dead load of total encasement: @ 1.2 ft Lt. PI } W_c &= -1,344 \text{ k.} \\ \text{Total moment} &= -1,692 \text{ ft-k.} \end{aligned}$$

(7) Summation of loads in Group A

(a) Group A forces and vectors

Table 17-21 gives a listing of Group A forces and vectors.

(b) Resolution of Group A axials into two components

$$\begin{aligned} \text{Horizontal: } &+(6,587) (\cos 8.31^\circ) - (5,663) (\cos 39.69^\circ) = +2,160 \text{ k. (right)} \\ \text{Vertical: } &-(6,587) (\sin 8.31^\circ) + (5,663) (\sin 39.69^\circ) = +2,665 \text{ k. (up)} \end{aligned}$$

(c) Total Group A loads

$$\begin{aligned} \text{Horizontal: } &-391 + 2,160 = +1,769 \text{ k.} \\ \text{Vertical: } &-2,297 + 2,665 = +368 \text{ k.} \end{aligned}$$

Table 17-21 Group A and B Combined Forces and Vectors

LOAD COMPONENTS IN SEC. 17.6.3	HORIZONTALS (K)	VERTICALS (K)	UPSTREAM AXIALS (K)	DOWNSTREAM AXIALS (K)	MOMENTS ABOUT PI (FT-K)
(1)	—	—	+1,806.	—	—
(2)a	- 37.	-253.	+ 37.	—	-2,496.
(2)b	-183.	-220.	—	+ 237.	+4,004.
(3)a	—	-398.	—	—	-7,761.
(3)b	—	—	—	—	-3,978.
(4)a	-149.	-180.	—	—	+7,488.
(4)b	- 22.	+ 18.	—	—	+ 266.
(4)c	—	—	—	- 90.	—
(5)	—	—	+4,744.	-5,810.	—
SUBTOTAL	-391.	-1,033.	+6,587.	-5,663.	-2,477.
* (7)b	+2,160.	+2,665.	*	*	*
TOTAL	+1,769.	+1,632.	—	—	-2,477.
(6)	—	-1,344.	—	—	-1,612.
TOTAL LOADING GROUP A	+1,769.	+ 288.	—	—	-4,089.
SEC. 17.6.4 LOADING GROUP B	+ 436.	+ 979.	—	—	—
TOTAL DESIGN LOADING	+2,205.	+1,267.	—	—	-4,089.

* See (b), above

17.6.4 Group B Loads (Upsurge Pressures and Momentum)

Group B includes all axial loads intersecting at PI.

(1) Upsurge pressure vectors

Compute variable upsurge P_U at PI 2.

$H = 173$ ft at PI (assumed). See Table 17-1.

$$P_U = (173) (176.71) (0.0624) = 1,908 \text{ k.}$$

(2) Momentum loads at PI 2

$$\text{Momentum vector} = (Q) (62.4/32.2) (v) (1/1000) = (2600) (62.4/32.2) (14.71) = 74 \text{ k.}$$

(3) Combined loading at PI 2

$$\text{Horizontal: } (P_U + \text{momentum}) (+\cos 8.31^\circ - \cos 39.69^\circ) = (P_U + 74) (0.2200) = +436 \text{ k.}$$

$$\text{Vertical: } (P_U + \text{momentum}) (-\sin 8.31^\circ + \sin 39.69^\circ) = (P_U + 74) (0.494105) = +979 \text{ k.}$$

These loads are combined with Group A in Table 17-21 and Figure 17-20.

(4) Resultant of loading

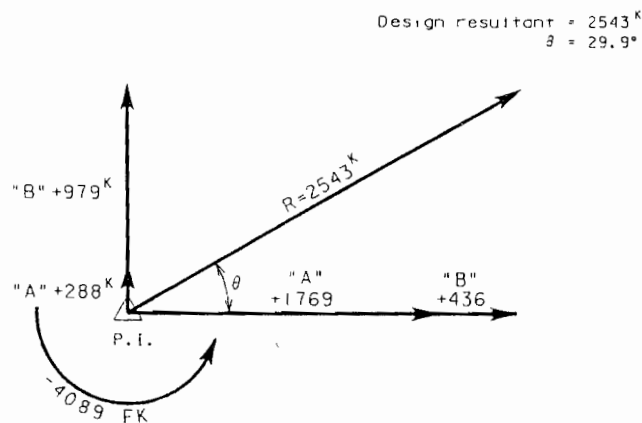


Figure 17-20 Resultant of Loading

17.6.5 Group C Loads

Seismic loads are considered below. Wind and snow loading are not factors at this location. Axial pressure loads during hydro-testing depend on the locations of the bulkheads because anchorages are affected only if unrestrained joints are included within the pressurized reach. Specifications for field testing normally should not control anchorage design except in unusual cases.

17.6.6 Group D Loads

Group D loads are all axial loads. These are not used for design but are applied with Group A loads, and the extreme total loads are checked for compliance with safety factors.

Reference to Table 17-1, et seq, shows two exceptional loads over the design upsurge, or Group B loading. The ratios of these with respect to B Loads are:

Emergency, $405/167 = 2.4$

Exceptional, $1035/167 = 6.2$

For evaluation, these ratios are applied to P_u of the B loading and the extreme effects computed.

17.6.7 General Comments

The relatively flat angle (30 degrees) for the design resultant indicates the main anchorage should rely more on a tie back or restraining system for equilibrium than for dead load friction.

Design loads allow estimation of the relative size of a buried, solid-base block. A vertical load of 1,267 kips (with FS = 1.5) requires approximately $(1,267)(1.5)/(0.15)(27) = 469$ cubic yards of

concrete or other dead load force. Horizontally, lateral earth resistance without deformation must provide a $(2,205)(1.5) = 3,307$ k. force.

These values suggest that alternatives to a dead-weight base should be considered. If founded in sound rock, battered, pretensioned anchor tendons are more efficient. In less sound material, drilled anchor caissons or anchor piles are available to provide load resistance by engaging a larger volume of foundation material.

For earthquake design, seismic loads are added to static design or Group A loads. These must be resisted entirely by the remaining lower part of the anchor.

17.6.8 Reinforcement of Concrete Encasement

In encasement, an upside-down saddle block transfers all loads from PI into the very stiff base section. Reinforcement design requires approximating loading conditions for analysis. This also assumes that concrete is placed while the penstock is under internal pressure and that shell deformation does not affect the block.

(1) Calculate load acting on top of saddle

Table 17-22 shows calculations for the load acting on top of the saddle at PI based on loads in Table 17-21 and Figure 17-19.

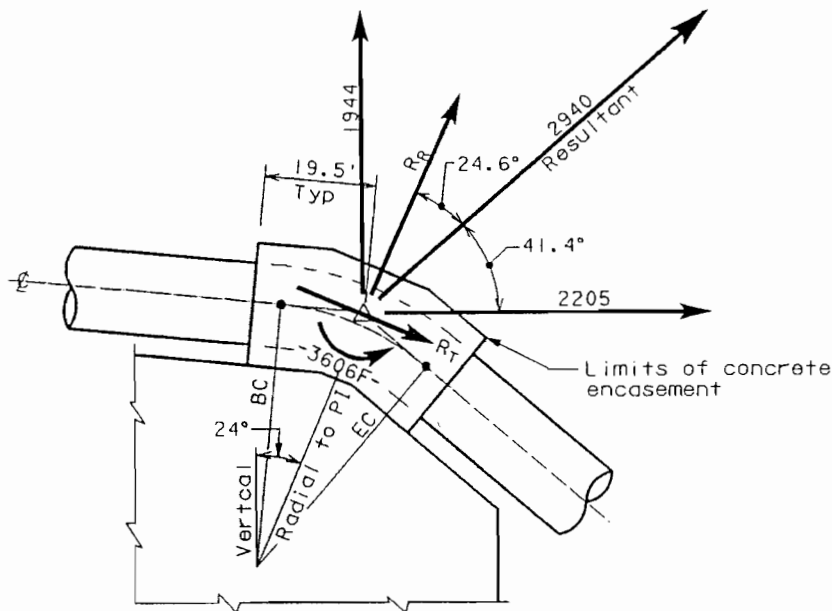
Table 17-22 Calculations for Load Acting On Top of Saddle

	HORIZONTAL (K)	VERTICAL (K)	MOMENTS ABOUT PI (FT-K)
AXIAL-GROUP A	+1,769	+288	-4,089
DL, TOP (DEDUCT)	—	+677	+483
TOTAL W/O P_u	+1,769	+965	-3,606
UPSURGE, P_u	+436	+979	—
TOTAL LOAD ON SADDLE	+2,205	+1,944	-3,606

The resultant of the horizontal and vertical loads equals 2,940 k. at 41.4 degrees above the horizontal. The resultant and other calculations are shown in Figure 17-21. Resolve the skewed load into one radial load through PI and one normal to the first. The radial load converts to a uniform load, and the normal load is resisted by shear in the block.

Encasement details and dimensions are taken from Figures 17-17 and 17-20. The encasement length equals the sum of center line arc length and two tangential extensions. Each of the latter is 19.5 less the curve tangent distance = 2.64 feet. Thus, total length = 38.15 feet. The minimum width of encasement of each side is 1/2 of 21 feet less OD of penstock = 2.90 feet.

This design requires that the skewed resultant be resisted by two force components developed within the encasement. The radial component is developed by four rows of reinforcing bars, two on each side of the penstock. The tangential component is developed by the continuity of the concrete and yet not exceed design stress parameters.

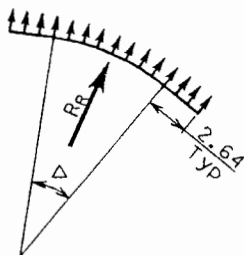


COMPONENTS OF RESULTANT:

$$R_R = \text{Radial} = 2940^K (\cos 24.6^\circ) = 2673^K$$

$$R_T = \text{Tangential} = 2940^K (\sin 24.6^\circ) = 1224^K$$

(A) DEVELOPING LOADING TO DESIGN TOP HALF OF PENSTOCK ENCASEMENT



CHECK REINFORCEMENT IN ENCASEMENT:

$$w = \text{Tension per foot of concrete}$$

$$R_R = 2wrsin \Delta + 2w(2.64)$$

$$2673^K = 2w(60)(\sin 31.38^\circ) + 2w(2.64)$$

$$w = 39.44^K/\text{ft}$$

Use reinforcement in two sides, 2 layers each.

$$f_s = 20 \text{ ksi}$$

$$A_s = \frac{39.44}{20} (1/4) = 0.49 \text{ in}^2/\text{ft}, \text{ each layer}$$

CHECK CONCRETE SHEARING RESISTANCE:

$$\text{Area} = 38.15 (2.90)(2) = 221.27 \text{ ft}^2$$

$$v = \frac{R_T}{A} = 1224^K/221.27 = 5.53^K/\text{ft}^2 = 38.4 \text{ psi}$$

$$= < 90 = \text{ok}$$

(B) ANALYSIS OF LOADING

Figure 17-21 Saddle Loads

The effect of the unbalanced moment may be analyzed by estimating the section modulus of the net encasement block on the center line elevation of the encasement. In this case, this results in a force system easily contained by appropriate increases in the steel reinforcing bar areas.

(2) Design other re-bar in block

A uniform load bears upward on the top half of the block. Tension occurs in the sides, and bending occurs at the sides and top. It is necessary to approximate a solution using loading coefficients or a similar analysis, unless a finite-element program is available. References 2 and 3 are also available, assuming that the load acts from inside to out.

17.6.9 Alternative Design Without Encasement

Full concrete encasement may not be desirable because of high seismic loads. An alternative design uses ring girders and large tie-anchors embedded into the base. For this design, upward and tangential loads are computed as above. Rings are designed for upside-down conditions by typical design methods. Typically, the penstock shell is bedded 120 degrees in concrete, and the embedded portion of the rings transfers shearing loads into the concrete base.

17.7 Riveted Penstock Joints

17.7.1 General

The following calculations do not apply to the design example outlined in Section 17.1. Calculations are presented only to provide examples of how riveted joints have been designed and to provide guidance for evaluating existing joints for verification.

The efficiency of riveted joints is defined as the ratio of the strength of a unit length of a riveted joint to the strength of the same unit length of the parent plate material. The efficiency of a joint is calculated by the general method illustrated in the following examples.

The following nomenclature is used in the sample calculations:

T_s	=	tensile strength stamped on plate, psi
t	=	thickness of plate, in.
b	=	thickness of butt strap, in.
P	=	pitch of rivets (on row having greatest pitch), in.
d	=	diameter of rivet after driving (equal to diameter of rivet hole), in.
a	=	cross-sectional area of rivet after driving, in. ²
s	=	shearing strength of rivet in single shear, psi
S	=	shearing strength of rivet in double shear, psi
c	=	crushing strength of mild steel, psi
n	=	number of rivets in single shear in a unit length of joint
N	=	number of rivets in double shear in a unit length of joint

17.7.2 Single-Riveted Lap Joint (Longitudinal or Circumferential)

The following formulas are used in determining joint efficiency.

Strength of solid plate (A):

$$A = (P) (t) (T_s)$$

Strength of plate between rivet holes (B):

$$B = (P - d) (t) (T_s)$$

Shearing strength of one rivet in single shear (C):

$$C = (n) (s) (a)$$

Crushing strength of plate in front of one rivet (D):

$$D = (d) (t) (c)$$

Divide by A the lesser of the values for B, C, or D. The quotient is the efficiency of a single-riveted lap joint, shown in Figure 17-22.

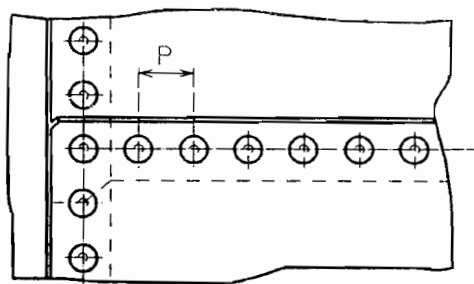


Figure 17-22 Single-Riveted Lap Joint (Longitudinal or Circumferential)

The following values are given:

T_s	=	55,000 psi
t	=	1/4 in. = 0.25 in.
P	=	1 5/8 in. = 1.625 in.
d	=	1/16 in. = 0.0625 in.
a	=	0.3712 in. ²
s	=	44,000 psi
c	=	95,000 psi

Substituting in the values gives:

$$\begin{aligned} A &= (1.625) (0.25) (55,000) = 22,343 \text{ psi} \\ B &= (1.625 - 0.6875) (0.25) (55,000) = 12,890 \text{ psi} \\ C &= (1) (44,000) (0.3712) = 16,332 \text{ psi} \\ D &= (0.6875) (0.25) (95,000) = 16,328 \text{ psi} \end{aligned}$$

Joint efficiency then equals:

$$\frac{B}{A} = \frac{12,870}{22,343} = 0.577$$

17.7.3 Double-Riveted Lap Joint (Longitudinal or Circumferential)

The following formulas are used in determining joint efficiency.

Strength of solid plate (*A*):

$$A = (P) (t) (T_s)$$

Strength of plate between rivet holes (*B*):

$$B = (P - d) (t) (T_s)$$

Shearing strength of two rivets in single shear (*C*):

$$C = (n) (s) (a)$$

Crushing strength of plate in front of two rivets (*D*):

$$D = (nd) (t) (c)$$

Divide by *A* the lesser of the values for *B*, *C*, or *D*. The quotient is the efficiency of a double-riveted lap joint, shown in Figure 17-23.

The following values are given:

$$\begin{aligned} T_s &= 55,000 \text{ psi} \\ t &= 5/16 \text{ in.} = 0.3125 \text{ in.} \\ P &= 2\frac{7}{8} \text{ in.} = 2.875 \text{ in.} \\ d &= 3/4 \text{ in.} = 0.75 \text{ in.} \\ a &= 0.4418 \text{ in.}^2 \\ s &= 44,000 \text{ psi} \\ c &= 95,000 \text{ psi} \end{aligned}$$

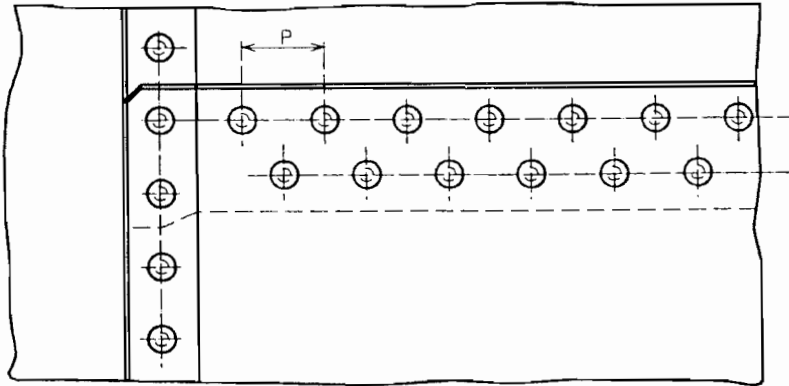


Figure 17-23 Double-Riveted Lap Joint (Longitudinal or Circumferential)

Substituting in the values gives:

$$\begin{aligned} A &= (2.875) (0.3125) (55,000) = 49,414 \text{ psi} \\ B &= (2.875 - 0.75) (0.3125) (55,000) = 36,523 \text{ psi} \\ C &= (2) (44,000) (0.4418) = 38,878 \text{ psi} \\ D &= (2) (0.75) (0.3125) (95,000) = 44,531 \text{ psi} \end{aligned}$$

Joint efficiency then equals:

$$\frac{B}{A} = \frac{36,523}{49,414} = 0.739$$

17.7.4 Double-Riveted Butt and Double Strap Joint

The following formulas are used in determining joint efficiency.

Strength of solid plate (A):

$$A = (P) (t) (T_s)$$

Strength of plate between rivet holes in the outer row (B):

$$B = (P - d) (t) (T_s)$$

Shearing strength of two rivets in double shear, plus the shearing strength of one rivet in single shear (C):

$$C = (N) (S) (a) + (n) (s) (a)$$

Strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row (D):

$$D = (P - 2d) (t) (T_s) + (n) (s) (a)$$

Strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row (E):

$$E = (P - 2d) (t) (T_s) + (d) (b) (c)$$

Crushing strength of plate in front of two rivets, plus the crushing strength of butt strap in front of one rivet (F):

$$F = (N) (d) (t) (c) + (n) (s) (a)$$

Crushing strength of plate in front of two rivets, plus the shearing strength of butt strap in front on one rivet (G):

$$G = (N) (d) (t) (c) + (n) (s) (a)$$

Strength of butt straps between rivet holes in the inner row (H):

$$H = (P - 2d) (2b) (T_s)$$

(This method of failure is not possible for thicknesses of butt straps required by this manual and the computation need only be made where thin butt straps have been used. For this reason, this method of failure will not be considered for other joints.)

Divide by A the lesser of the values for B , C , D , E , F , G , or H . The quotient is the efficiency of a butt and double strap joint, double-riveted, shown in Figure 17-24.

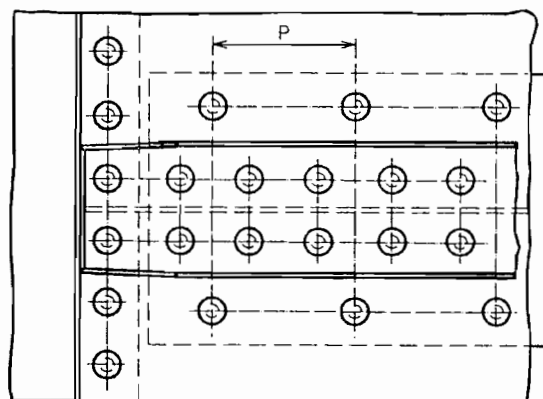


Figure 17-24 Double-Riveted Butt and Double Strap Joint

17 Penstock Design Examples

The following values are given:

$$\begin{aligned}T_s &= 55,000 \text{ psi} \\t &= 3/8 \text{ in.} = 0.375 \text{ in.} \\b &= 5/16 \text{ in.} = 0.3125 \text{ in.} \\P &= 4\frac{7}{8} \text{ in.} = 4.875 \text{ in.} \\d &= 7/8 \text{ in.} = 0.875 \text{ in.} \\a &= 0.6013 \text{ in.}^2 \\s &= 44,000 \text{ psi} \\S &= 88,000 \text{ psi} \\c &= 95,000 \text{ psi}\end{aligned}$$

Number of rivets in single shear in a unit length of joint = 1

Number of rivets in double shear in a unit length of joint = 2

Substituting in the values gives:

$$\begin{aligned}A &= (4.875) (0.375) (55,000) = 100,547 \text{ psi} \\B &= (4.875 - 0.875) (0.375) (55,000) = 82,500 \text{ psi} \\C &= (2) (88,000) (0.6013) + (1) (44,000) (0.6013) = 132,286 \text{ psi} \\D &= (4.875 - 2 \times 0.875) (0.375) (55,000) + (1) (44,000) (0.6013) = 90,910 \text{ psi} \\E &= (4.875 - 2 \times 0.875) (0.375) (55,000) + (0.875) (0.3125) (95,000) = 90,429 \text{ psi} \\F &= (2) (0.875) (0.375) (95,000) + (0.875) (0.3125) (95,000) = 88,320 \text{ psi} \\G &= (2) (0.875) (0.375) (95,000) + (1) (44,000) (0.6013) = 88,800 \text{ psi}\end{aligned}$$

Joint efficiency then equals:

$$\frac{B}{A} = \frac{82,500}{100,547} = 0.820$$

17.7.5 Triple-Riveted Butt and Double Strap Joint

The following formulas are used in determining joint efficiency:

Strength of solid plate (A):

$$A = (P) (t) (T_s)$$

Strength of plate between rivet holes in the outer row (B):

$$B = (P - d) (t) (T_s)$$

Shearing strength of four rivets in double shear, plus the shearing strength of one rivet in single shear (C):

$$C = (N) (S) (a) + (n) (s) (a)$$

Strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row (*D*):

$$D = (P - 2d) (t) (T_s) + (n) (s) (a)$$

Strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row (*E*):

$$E = (P - 2d) (t) (T_s) + (d) (b) (c)$$

Crushing strength of plate in front of four rivets, plus the crushing strength of butt strap in front of one rivet (*F*):

$$F = (N) (d) (t) (c) + (n) (d) (b) (c)$$

Crushing strength of plate in front of four rivets, plus the shearing strength of one rivet in single shear (*G*):

$$G = (N) (d) (t) (c) + (n) (s) (a)$$

Divide by *A* the lesser of the values for *B*, *C*, *D*, *E*, *F* or *G*. The quotient is the efficiency of a butt and double strap joint, triple-riveted, shown in Figure 17-25.

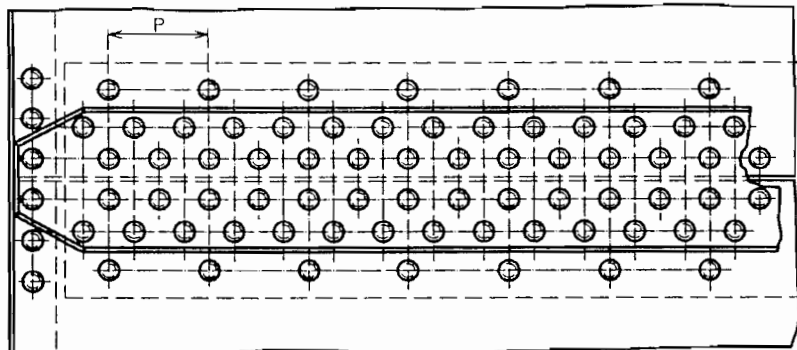


Figure 17-25 Triple-Riveted Butt and Double Strap Joint

The following values are given:

T_s	=	55,000 psi
t	=	3/8 in. = 0.375 in.
b	=	5/16 in. = 0.3125 in.
P	=	6½ in. = 6.5 in.
d	=	13/16 in. = 0.8125 in.
a	=	0.5185 in. ²
s	=	44,000 psi
S	=	88,000 psi
c	=	95,000 psi

17 Penstock Design Examples

Number of rivets in single shear in a unit length of joint = 1

Number of rivets in double shear in a unit length of joint = 4

Substituting in the values gives:

$$A = (6.5) (0.375) (55,000) = 134,062 \text{ psi}$$

$$B = (6.5 - 0.8125) (0.375) (55,000) = 117,304 \text{ psi}$$

$$C = (4) (88,000) (0.5185) + (1) (44,000) (0.5185) = 205,326 \text{ psi}$$

$$D = (6.5 - 2 \times 0.8125) (0.375) (55,000) + (1) (44,000) (0.5185) = 123,360 \text{ psi}$$

$$E = (6.5 - 2 \times 0.8125) (0.375) (55,000) + (0.8125) (0.3125) (95,000) = 124,667 \text{ psi}$$

$$F = (4) (0.8125) (0.375) (95,000) + (1) (0.8125) (0.3125) (95,000) = 139,902 \text{ psi}$$

$$G = (4) (0.8125) (0.375) (95,000) + (1) (44,000) (0.5185) = 138,595 \text{ psi}$$

Joint efficiency then equals:

$$\frac{B}{A} = \frac{117,304}{134,062} = 0.875$$

References*

1. Moore, E.T. "Designing Steel Tunnel Liners for Hydro Plants." *Hydro Review* (Oct.1990).
2. Steel Pipe—*A Guide for Design and Installation*. AWWA Manual M11. American Water Works Association, Denver, CO.
3. "Welded Steel Penstocks—Design and Construction." Engineering Monograph No. 3. U.S. Bureau of Reclamation, Denver, CO (1986).
4. Zick, L.P. "Stresses in Large Horizontal Cylindrical Pressure Vessels on Two Saddle Supports." *Welding Journal* (Welding Research Supplement). 30:435-S (Sept. 1951).
5. Specification for Structural Steel Buildings—Allowable Stress Design and Plastic Design, June 1, 1989. American Institute of Steel Construction, Inc., Chicago, IL.
6. "Welded Steel Penstocks and Tunnel Liners, Steel Plate Engineering Data—Volume 4." American Iron and Steel Institute and Steel Plate Fabricators Association, Inc. (1984).
7. Specification for the Design, Fabrication and Erection of Structural Steel for Buildings (Nov. 1, 1978). American Institute of Steel Construction, Inc., Chicago, IL.

* The most current version of a standard, code, or specification should be used for reference.

MEASUREMENT CONVERSIONS

ACCELERATION

Unit	ft/s ²	m/s ²
1 Foot per second squared (ft/s ²)	1	0.3048
1 Meter per second squared (m/s ²)	3.2808	1

AREA

Unit	ft ²	m ²	ha	Acre	mi ²
1 Square foot (ft ²)	1	0.0929	9.2903×10 ⁻⁶	2.2956×10 ⁻⁵	3.5870×10 ⁻⁸
1 Square meter (m ²)	10.7639	1	1×10 ⁻⁴	2.4711×10 ⁻⁵	3.8610×10 ⁻⁷
1 Hectare (ha)	1.0764×10 ⁵	10,000	1	2.4711	3.8610×10 ⁻³
1 Acre	43,560	4046.85	0.4047	1	1.5625×10 ⁻³
1 Square mile (mi ²)	2.7878×10 ⁷	2.5900×10 ⁶	259	640	1

ENERGY

Unit	J	ft-lb	Btu	kcal	hph	kWh
1 Joule (J)	1	0.7376	9.481×10 ⁻⁴	2.389×10 ⁻⁴	3.725×10 ⁻⁷	2.778×10 ⁻⁷
1 Foot-Pound (ft-lb)	1.356	1	1.285×10 ⁻³	3.239×10 ⁻⁴	5.051×10 ⁻⁷	3.766×10 ⁻⁷
1 British thermal unit (Btu)	1,055	777.9	1	0.252	3.929×10 ⁻⁴	2.930×10 ⁻⁴
1 Kilocalorie (kcal)	4,086	3,087	3.968	1	1.559×10 ⁻³	1.163×10 ⁻³
1 Horsepower-hour (hph)	2.685×10 ⁶	1.980×10 ⁶	2,545	641.4	1	0.7457
1 Kilowatt-hour (kWh)	3.6×10 ⁶	2.655×10 ⁶	3,413	860.1	1.341	1

FORCE

Unit	dyn	N	lbf	kgf	kip
1 Dyne (dyn)	1	1.0×10 ⁻⁵	2.248×10 ⁻⁶	1.020×10 ⁻⁶	2.248×10 ⁻¹⁰
1 Newton (N)	100,000	1	0.2248	0.1020	2.248×10 ⁻⁴
1 Pound (lbf)	444,800	4.448	1	0.04536	0.001
1 Kilogram (kgf)	980,700	9.807	2.205	1	2.205×10 ⁻³
1 Kip	4.448×10 ⁹	4,448	1,000	453.5	1

MEASUREMENT CONVERSIONS — *Continued*

LENGTH

Unit	in.	ft	m	km	mi
1 Inch (in.)	1	0.0833	0.0254	2.540×10^{-5}	1.5782×10^{-5}
1 Foot (ft)	12	1	0.3048	3.048×10^{-4}	1.8939×10^{-4}
1 Meter (m)	39.3710	3.2808	1	0.001	6.2136×10^{-4}
1 Kilometer (km)	39,370	3,280.84	1,000	1	0.6212
1 Mile (mi)	63,360	5,280	1,609.36	1.6093	1

MASS

Unit	lb	kg	Metric slug	Slug	Metric ton	Long ton
1 Pound (lb)	1	0.4536	0.0462	0.0311	4.536×10^{-4}	446.4×10^{-4}
1 Kilogram (kg)	2.205	1	0.1020	0.0685	0.001	9.842×10^{-4}
1 Metric slug	21.62	9.807	1	0.6721	0.0098	0.0096
1 Slug	32.17	14.59	1.490	1	0.0146	0.0144
1 Metric ton	2,205	1,000	102.0	68.52	1	0.9842
1 Long ton	2,240	1,016	103.7	69.63	1.016	1

POWER (Rate of Energy Flow)

Unit	Btu/h	ft-lb/s	hp	kW
1 Btu/hour (Btu/h)	1	0.2161	3.929×10^{-4}	2.920×10^{-4}
1 Foot-pound/second (ft-lb/s)	4.628	1	1.818×10^{-3}	1.356×10^{-4}
1 Horsepower (hp)	2,545	550	1	0.7457
1 Kilowatt (kW)	3,413	737.6	1.341	1

1 Watt = 1 J/s.

1 kW is generated by 11.81 ft³/s of water falling 1 foot (at 100% efficiency) or by 0.102 m³/s falling 1 meter (at 100% efficiency).

PRESSURE

Unit	Pa	H ₂ O ft	Hg in.	lb/in. ²	atm
1 Pascal (Pa)	1	3.3456×10^{-4}	2.9533×10^{-4}	1.4504×10^{-4}	9.8692×10^{-6}
1 Foot of water @ 39.4°F (H ₂ O ft)	2,989	1	0.88275	0.43352	0.0295
1 Inch of Mercury (Hg in.)	3,386	1.13282	1	0.4911	0.03342
1 Pound per square inch (lb/in. ²)	6,894.757	2.30671	2.03625	1	0.068046
1 Atmosphere (atm)	101,325	33.89945	29.92471	14.69595	1

1 Pa = 1 N/m² = 10 dyne/cm².

MEASUREMENT CONVERSIONS — *Continued*

VELOCITY

Unit	ft/d	km/h	ft/s	mi/h	m/s
1 Foot per day (ft/d)	1	1.27×10^{-5}	1.157×10^{-5}	7.891×10^{-6}	3.528×10^{-6}
1 Kilometer per hour (km/h)	78.740	1	0.9113	0.6214	0.2778
1 Foot per second (ft/s)	86.400	1.097	1	0.6818	0.3048
1 Mile per hour (mi/h)	126.700	1.609	1.467	1	0.447
1 Meter per second (m/s)	283,500	3.600	3.281	2.237	1

VOLUME

Unit	L	gal	ft ³	m ³	acre-ft
1 Liter	1	0.264	0.035	0.001	8.11×10^{-7}
1 U.S. gallon (gal)	3.785	1	0.134	0.00379	3.07×10^{-6}
1 Cubic foot (ft ³)	28.317	7.48	1	0.02832	2.30×10^{-5}
1 Cubic meter (m ³)	1000	264	35.315	1	8.11×10^{-4}
1 Acre-ft (acre-ft)	1,233,500	325,851	43,560	1,233.48	1

1 U.S. gallon = 231 in.³ = 0.83 Imperial gallons.

1 L = 1,000 cm³ = 1.05 quarts = 1,000 grams of water.

1 Barrel = 42 U.S. gallons.

1 ft³ of water = 62.4 lb.

SI PREFIXES AND SYMBOLS

Multiplication factor	Prefix	Symbol
1,000,000,000,000,000,000 = 10 ¹⁸	exa	E
1,000,000,000,000,000 = 10 ¹⁵	peta	P
1,000,000,000,000 = 10 ¹²	tera	T
1,000,000,000 = 10 ⁹	giga	G
1,000,000 = 10 ⁶	mega	M
1,000 = 10 ³	kilo	k
100 = 10 ²	hecto	h
10 = 10 ¹	deka	da
0.1 = 10 ⁻¹	deci	d
0.01 = 10 ⁻²	centi	c
0.001 = 10 ⁻³	milli	m
0.000,001 = 10 ⁻⁶	micro	μ
0.000,000,001 = 10 ⁻⁹	nano	n
0.000,000,000,001 = 10 ⁻¹²	pico	p
0.000,000,000,000,001 = 10 ⁻¹⁵	femto	f
0.000,000,000,000,000,001 = 10 ⁻¹⁸	atto	a

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