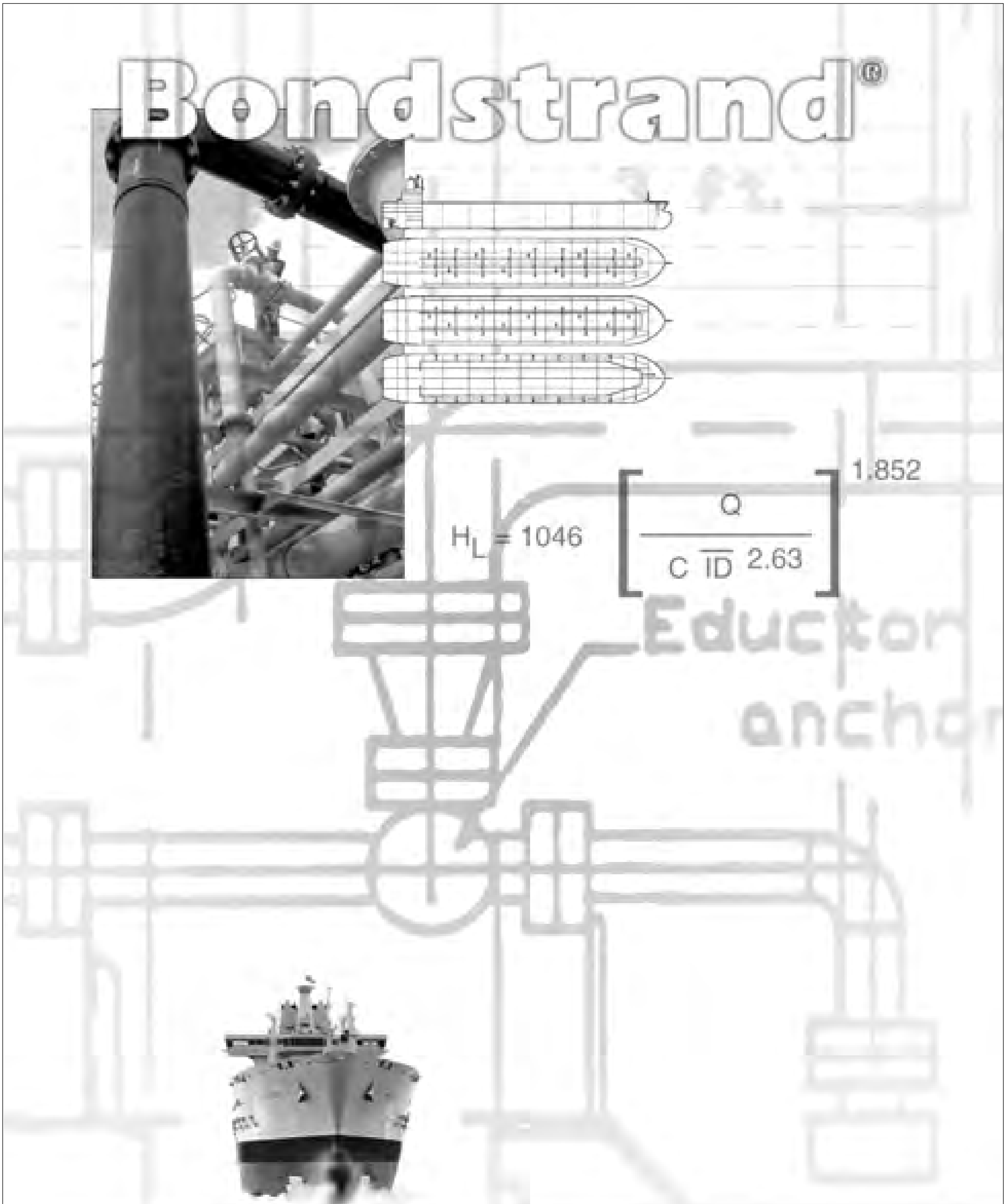


NOV Fiber Glass Systems

Bondstrand® Design Manual

for Marine Piping Systems

Table of Contents

1.0 Introduction

1.1 GENERAL

Historically, offshore exploration, production platforms and ship owners have had to face the grim reality of replacing most metal piping two or three times during the average life of a vessel or platform. This has meant, of course, that piping systems end up costing several times that of the original investment since replacement is more expensive than new installation. When you add the labor costs, the downtime and the inconvenience of keeping conventional steel or alloy piping systems in safe operating condition, the long-term advantages of fiberglass piping become very obvious.

1.2 PRODUCT RANGE AND SERIES

Bondstrand® provides four distinct series of filament-wound pipe and fittings using continuous glass filaments and thermosetting resins for marine and naval applications:

Series 2000M
A lined epoxy pipe and fittings system for applications which include ballast lines, fresh and saltwater piping, sanitary sewage, raw water loop systems and fire protection mains where corrosion resistance and light weight are of paramount importance.

Series 2000M-FP
A lined epoxy system covered with a reinforced intumescent coating suitable for dry service in a jet fire.

Series 2000USN
An epoxy system meeting the requirements of MIL-P-24608B (SH) for nonvital piping systems on combatant and non-combatant vessels. Available in sizes from 1 to 12 inches (25 to 300mm).

Series 5000M
A lined vinylester pipe and fittings system in 2 inch diameter (50mm) for seawater chlorination.

Series 7000M
An epoxy pipe and fittings system with anti-static capabilities designed for white petroleum products and applications passing through hazardous areas. Properly grounded Series 7000M prevents the accumulation on the exterior of the pipe of dangerous levels of static electricity produced by flow of fluids inside the pipe or by air flow over the exterior of the pipe. This is accomplished by NOV FGS patented method of incorporating electrically conductive elements into the wall structure of pipe and fittings during manufacture.

PSX™•L3
A polysiloxane-modified phenolic system for use in normally wet fire protection mains - also suitable for confined spaces and living quarters due to low smoke and toxicity properties. Also available in a conductive version.

PSX™•JF
A polysiloxane-modified phenolic system for use in deluge piping (normally dry). PSX™•JF has an exterior jacket which allows the pipe to function even after 5 minutes dry exposure to a jet fire (follow by 15 minutes with flowing water). Also available in a conductive version.

1.3 STANDARDS AND SPECIFICATIONS

Bondstrand® marine pipe and fittings are designed and manufactured in accordance with the following standards and specifications:

MIL-P-24608A (SH)
U.S. Navy standards for fiberglass piping systems onboard combatant and noncombatant ships.

ASTM (F1173)
U.S. standards for fiberglass piping systems onboard merchant vessels, offshore production and explorations units.

1.4 CLASSIFICATION SOCIETY APPROVALS

NOV FGS works closely with agencies worldwide to widen the scope of approved shipboard applications for fiberglass pipe systems. Certificates of approval and letters of guidance from the following agency concerning the use of Bondstrand piping on shipboard systems are currently available from NOV FGS. Others are pending.

American Bureau of Shipping
Biro Klasifikasi Indonesia
Bureau Veritas
Canadian Coast Guard, Ship Safety Branch
Det Norske Veritas
Dutch Scheepvaartinspectie
DDR-Schiffs-Revision UND-Klassifikation
Germanisher Lloyd
Korean Register of Shipping
Lloyd's Register of Shipping
Nippon Kaiji Kyokai
Polski Rejestr Statkow
Registro Italiano Navale
Register of Shipping
The Marine Board of Queensland
United States Coast Guard
USSR Register of Shipping

1.5 USES AND APPLICATIONS

Series 2000M
Approved for use in air cooling circulating water; auxiliary equipment cooling; ballast/segregated ballast; brine; drainage/sanitary service/sewage; educator systems; electrical conduit; exhaust piping; fire protection mains (IMO L3) fresh water/service (nonvital); inert gas effluent; main engine cooling; potable water; steam condensate; sounding tubes/vent lines; and tank cleaning (saltwater system); submersible pump column piping; raw water loop systems and drilling mud pumping systems.

Series 2000M-FP
Designed for use where pipe is vulnerable to mechanical abuse or impact or for dry deluge service.

Series 5000M
Approved for use in seawater chlorination.

Series 7000M
Approved for use in ballast (adjacent to tanks); C.O.W. (crude oil washing); deck hot air drying (cargo tanks); petroleum cargo lines; portable discharge lines; sounding tubes/vent cargo piping; stripping lines and all services listed for Series 2000M in hazardous locations.

PSX™•L3
Designed and approved for use in fire protection ring mains and for services in confined spaces of living quarters where flame spread, smoke density and toxicity are critical.

PSX™•JF
Designed and approved for dry deluge service where pipe may be subject to a directly impinging jet fire.

1.6 JOINING SYSTEMS

Bondstrand® marine and naval pipe systems offer the user a variety of joining methods for both new construction and for total or partial replacement of existing metallic pipe.

All Series:

1-to 16-inch ....Quick-Lock® straight/taper adhesive joint;

2-to 24-inch (2000M)......Van stone type flanges with movable flange rings for easy bolt alignment.

1-to 36-inch ....One-piece flanges in standard hubbed or hubless heavy-duty configuration.

2-to 36-inch ....Viking-Johnson or Dresser-type mechanical couplings.

1.7 FITTINGS AND FLANGE DRILLINGS

NOV FGS offers filament-wound fittings, adaptable for field assembly using adhesive, flanged, or rubber-gasketed mechanical joints. Tees, elbows, reducers and other fittings provide the needed complete piping capability.

Bondstrand marine and naval flanges are produced with the drillings listed below for easy connection to shipboard pipe systems currently in common use. Other drillings, as well as undrilled flanges, are available.

ANSI B16.5 Class 150 & 300;
ISO 2084 NP-10 & NP-16;
JIS B2211 5kg/cm^2;
JIS B2212 10kg/cm^2;
JIS B2213 16kg/cm^2;
U.S. Navy MIL-F-20042

1.8 CORROSION RESISTANCE

Bondstrand pipe and fittings are manufactured by a filament-winding process using highly corrosion-resistant resins. The pipe walls are strengthened and reinforced throughout with tough fiberglass and carbon fibers (Series 7000 only) creating a lightweight, strong, corrosion-resistant pipe that meets U.S. Coast Guard Class II and U.S. Navy MIL-P-24608A (SH) standards for offshore and most shipboard systems.

1.9 ECONOMY

Bondstrand offshore piping and Bondstrand marine and naval pipe systems have corrosion resistance surpassing copper-nickel and more exotic alloys, but with an installed cost less than carbon steel. Numerous shipyards have recorded their Bondstrand installation costs on new construction projects and report savings from 30 to 40 percent compared to traditional steel pipe.

2.0 Design for Expansion & Contraction

2.1 LENGTH CHANGE DUE TO THERMAL EXPANSION

Like other types of piping material, in an unrestrainted condition, Bondstrand fiberglass reinforced pipe changes its length with temperature. Tests show that the amount of expansion varies linearly with temperature, in other words, the coefficient of thermal expansion in Bondstrand pipe is constant, it equals to 0.00001 inch per inch per degree Fahrenheit (0.000018 millimeter- per millimeter per degree centigrade).

The amount of expansion can be calculated by the formula:

$$\Delta L = \alpha \, L \, \Delta T$$

where

ΔL = change in length (in. or mm),
α = coefficient of thermal expansion (in./in./°F or mm/mm/°C),
L = length of pipeline (in. or mm), and
ΔT = change in temperature (°F or °C).

Example: Find the amount of expansion in 100 feet (30.48 meter) of Series 2000M pipe due to a change of 90°F (50°C) in temperature:

a. English Units:

$$\Delta L = \alpha \, L \, \Delta T$$

where

$\alpha = 10 \times 10^{-6}$ in./in./°F
ΔT = 90°F
L = 100 ft. = 1200 in.
ΔL = (1200 in.) (10×10^{-6} in./in./°F) (90°F)
ΔL = 1.08 in.

b. Metric Units:

$$\Delta L = \alpha \, L \, \Delta T$$

where

$\alpha = 18 \times 10^{-6}$ mm/mm/°C
ΔT = 50°C
L = 30.48 m = 30480 mm
ΔL = (30480 mm) (18×10^{-6} mm/mm/°C) (50°C)
ΔL = 27.4 mm

Note that 27.4 mm is equal to 1.08 in. which is the calculated thermal expansion for the same length of pipe due to the same amount of temperature change.

In normal operating temperature range, the length change - temperature relationship can be represented by a straight line as illustrated in Figure 2-1 on the next page.

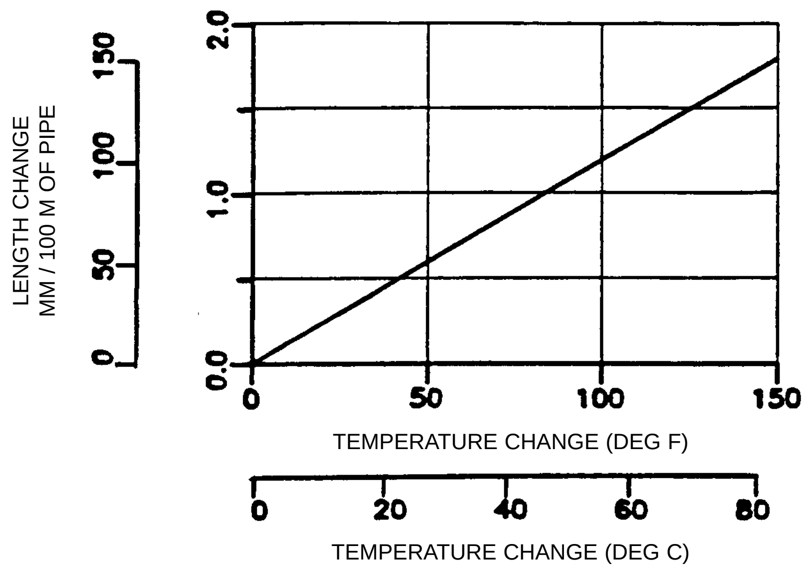


Fig. 2-1

2.2 LENGTH CHANGE DUE TO PRESSURE

2.2.1 Unrestrained System

Subjected to an internal pressure, a free Bondstrand pipeline will expand its length due to thrust force applied to the end of the pipeline. The amount of this change in the pipe length depends on the pipe wall thickness, diameter, Poisson's ratio and the effective modulus of elasticity in both axial and circumferential directions at operating temperature.

$$\triangle L = L \left[\frac{p\,\overline{ID}^2}{4t\,D_m\,E_l} - \nu_{lc}\,\frac{p\,\overline{ID}^2}{2t\,D_m\,E_c} \right]$$

The first term inside the bracket is the strain caused by pressure end thrust while the second term,

$$\nu_{lc}\,\frac{p\,\overline{ID}^2}{2t\,D_m\,E_c}$$

is the axial contraction due to an expansion in the circumferential direction, the Poisson's effect. The result is a net increase in length which can be calculated by the simplified formula:

$$\triangle L = L\,\frac{p\,\overline{ID}^2}{4t\,E_l\,D_m} \left[1 - 2\nu_{lc}\,\frac{E_l}{E_c} \right]$$

where L = length of pipe (in. or cm.),

p = internal pressure (psi or kg./cm^2),

ν_{lc} = Poisson's ratio for contraction in the longitudinal direction due to the strain in the circumferential direction.

E_c = circumferential modulus of elasticity (psi or kg./cm^2),

E_l = longitudinal modulus of elasticity (psi or kg./cm²),

D_m = mean diameter of pipe wall = $\overline{ID}$ + t,

$\overline{ID}$ = inside diameter of the pipe (in. or cm.), and

t = thickness of pipe wall (in. or cm.)

Example: Find the length change in 10 meters of Bondstrand Series 2000M, 8-inch pipe which is subjected to an internal pressure of 145 psi (10 bars) at 75° F (24°C).

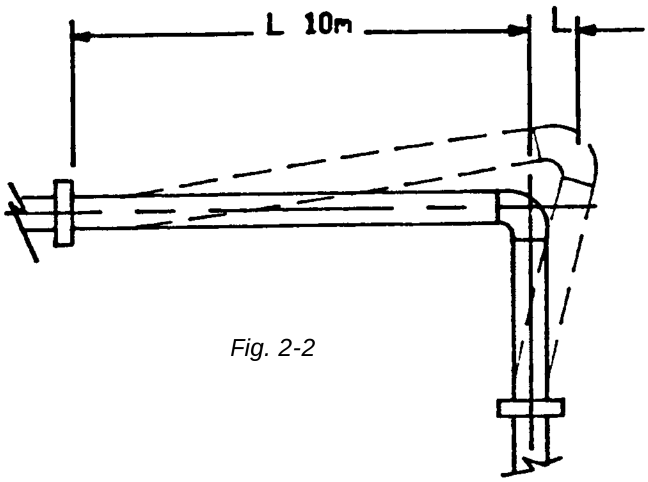


Fig. 2-2

a.English Units:

The physical properties of the pipe can be found from Bondstrand Series 2000M Product Data (FP194):

ν_{lc} = 0.56

E_c = 3,600,000 psi

E_l = 1,600,000 psi

ID = 8.22 in.

t = 0.241 in.

D_m = 8.46 in.

p = 145 psi

L = 394 in.

Note: Physical properties vary with temperature. See Bondstrand Series 2000M Product Data (FP194).

$$\triangle L = (394 \text{ in.}) \frac{145 \text{ psi } (8.22 \text{ in.})^2}{4\,(.241 \text{ in.})\,(8.46 \text{ in.})\,1{,}600{,}000 \text{ psi}} \left[1 - 2\,(.56)\,\frac{1{,}600{,}000 \text{ psi}}{3{,}600{,}000 \text{ psi}}\right]$$

$$\triangle L = 0.147 \text{ in.}$$

b. Metric Units:

$\nu_{lc} = 0.56$

$E_l = 113490 \text{ kg/cm}^2$

$D_m = 21.5 \text{ cm}$

$\overline{ID} = 20.9 \text{ cm}$

$t = 0.612 \text{ cm}$

$p = 10 \text{ bars} = 10.02 \text{ kg/cm}^2$

$L = 1000 \text{ cm}$

$$\triangle L = (1000 \text{ cm}) \frac{10.02 \text{ kg/cm}^2\,(20.9 \text{ cm})^2}{4\,(.612 \text{ cm})\,(21.5 \text{ cm})\,(113490 \text{ kg/cm}^2)} \left[1 - 2\,(.56)\,\frac{113490 \text{ kg/cm}^2}{253105 \text{ kg/cm}^2}\right]$$

$$\triangle L = 0.373 \text{ cm}$$

Table 2-I provides the calculated length increase for 100 feet (30.48 meters) of Bondstrand Series 2000M Pipe caused by 100 psi (7 kg/cm^2) internal pressure. The Table is valid through the temperature range of application. (The effect of temperature on length change due to pressure is small.)

Table 2-I

Size		Length Increase	
(in.)	(mm.)	(in.)	(mm)
2	50	0.2	5.0
3	80	0.3	7.8
4	100	0.3	7.6
6	150	0.4	10.2
36	900	0.4	10.2

Obtain length increase for other pressure by using a direct pressure ratio correction. For example, to find the length change caused by 150 psi pressure in a 6-inch pipe, multiply 0.4 inch by the pressure ratio 150/100 to obtain an amount of 0.6 inch length increase.

2.2.2 Restrained Systems

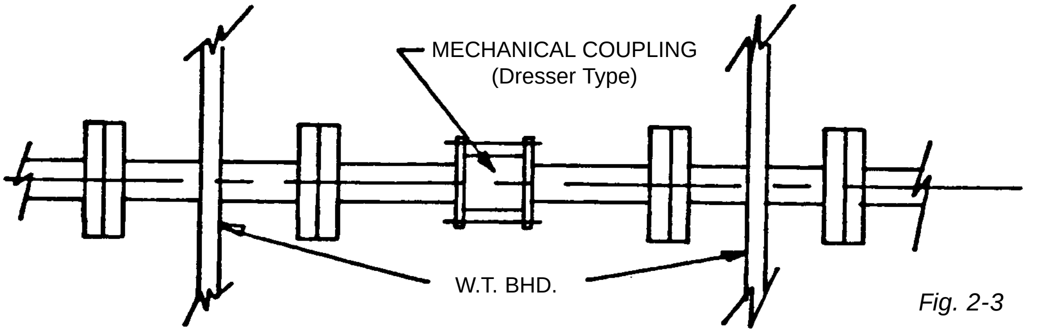


Fig. 2-3

In the piping system, shown in Figure 2-3, all longitudinal thrusts are eliminated by the use of fixed supports; therefore, the pipe is subjected only to load in the circumferential direction. Without the end thrust present, the first term in the equation is dropped and the length change becomes:

$$\triangle L = L \left[-\nu_{lc} \quad \frac{p\,\overline{ID}^2}{2t\,E_c\,D_m} \right]$$

where L = length of pipe (in. or cm),

p = internal pressure (psi or kg/cm^2),

ν_{lc} = Poisson's ratio

E_c = circumferential modulus of elasticity, (psi or kg/cm^2)

ID = inside diameter of the pipe (in. or cm),

t = thickness of pipe wall (in. or cm),

D_m = mean diameter of pipe wall = ID + t.

Example: Find the change in length in 12 meters (39.4 feet) of restrained Bondstrand Series 2000M, 8-inch diameter pipe operating at 10 bars (145 psi) internal pressure.

a. English Units:

ν_{lc} = .56

$\overline{p}$ = 145 psi

ID = 8.22 in.

t = 0.241 in.

D_m = 8.46 in.

E_c = 3,600,000 psi

L = 472 in.

$$\triangle L = (472 \text{ in.})(-.56) \frac{145 \text{ psi } (8.22 \text{ in.})^2}{2\ (.241 \text{ in.})\ (8.46 \text{ in. })\ 3{,}600{,}000 \text{ psi}}$$

$$L = -.175 \text{ in. or } .175 \text{ in. reduction in length}$$

b.Metric Units:

ν_{lc} = .56

p = 10.02 kg/cm^2

ID = 20.9 cm

$\overline{D}_m$ = 21.5 cm

t = 0.612 cm

E_c = 253105 kg/cm^2

L = 1200 cm

$$\triangle L = (1200 \pm \text{ cm})\ (-.56) \left[\frac{10.02 \text{ kg/cm}^2 (20.9 \text{ cm})^2}{2\ (0.612 \text{ cm})\ (21.5 \text{ cm})\ (253105 \text{ kg/cm})^2} \right]$$

$$\triangle L = -\ .442 \text{ cm or } .442 \text{ cm reduction in length}$$

As indicated by the formula and demonstrated by the example, in a restrained installation where a mechanical coupling is used, application of pressure will result in a contraction of the pipe. This shortening effect is found favorable in most applications where the designer can use the reduction in length to compensate for thermal expansion. Conversely, allowances should be made where operating temperature is significantly lower than the temperature at which the system is installed.

2.3 LENGTH CHANGE DUE TO DYNAMIC LOADING

Piping installed on board ship is often subjected to another type of load at the supports which results from sudden change of the support's relative location. This dynamic loading should be accounted for in the design. The degree of fluctuation in length between the two support points depends on the ship's structural characteristics, i.e., the ship size, the size of the dynamic load, etc. This type of movement in the piping system should be considered with other length changes previously discussed; however, calculation of expansion and contraction due to dynamic loading is beyond the intended scope of this manual.

2.3.1 Equipment Vibration

Under normal circumstances, Bondstrand pipe will safely absorb vibration from pumping if the pipe is protected against external abrasion at supports.

Vibration can be damaging when the generated frequency is at, or near, the natural resonance frequency of the pipeline. This frequency is a function of the support system, layout geometry, temperature, mass and pipe stiffness.

There are two principal ways to control excessive stress caused by vibration. Either install, observe during operation, and add supports or restraints as required; or add an elastometric expansion joint or other vibration absorber.

2.4 FLEXIBLE JOINTS, PIPE LOOPS, Z AND L TYPE BENDS

Bondstrand piping is often subjected to temperature change in operation, usually in the range of 50°F to 100°F (32°C to 82°C). Since a piping system operating at low stress level provides longer service life, it is good practice to reduce the amount of stress caused by thermal and/or pressure expansion. This can be accomplished by using one or more of the following:

A. Flexible Joints
   a.1 Mechanical coupling (Dresser-type), or
   a.2 Expansion joint.

B. Pipe Loops

C. Z type configurations or change of direction at bends.

2.5 DESIGN WITH FLEXIBLE JOINTS

Both Dresser-type couplings and expansion joints are recognized as standard devices to absorb thermal expansion. They are easy to use and commercially available.

2.5.1 Mechanical Couplings (Dresser-type)

These are primarily designed to be used as mechanical connection joints. The elastomeric seal offers some flexibility that will relieve thermal expansion in the pipe; however, this can only absorb a limited amount of axial movement, usually about 3/8 in. (10mm) per coupling. Thus, more than one coupling must be used if the expected movement is greater than 3/8 in. (10mm).

It should be noted here that fixed supports are always required in a mechanical system. In moderate temperature and pressure application, such as often found in ballast piping systems, the total expansion of a 40-foot Bondstrand pipe is within the coupling recommended limit. *For additional information on mechanical type couplings see Appendix A.*

2.5.2 Expansion Joints

Expansion joints are widely accepted as standard devices to relieve longitudinal thermal stress. Unlike the mechanical coupling, this joint offers a wider range of axial movement giving more flexibility in design. This is advantageous in long section of pipe such as in cargo piping which sometimes runs the entire length of the ship. An expansion joint is normally not needed in ballast piping system where short sections of pipe are anchored at bulkheads.

When an expansion joint is used in the pipeline to relieve longitudinal stress, it must be fairly flexible, such as a teflon bellows which is activated by the thrust of a low modulus material.

Support for expansion joints must be correctly designed and located to maintain controlled deflection. Besides adding weight, most of these joints act as partial structural hinges which afford only limited transfer of moment and shear. Where the expansion joint relies on elastomers of thermoplastics, the structural discontinuity or hinging effect at the joint changes with temperature.

When using an expansion joint in a pipeline carrying solids, consider the possibility that it could stiffen or fail to function due to sedimentation build up in the expansion joint. Failure of the expansion joint could cause excessive pipe deflection. Regular schedule maintenance and cleaning of the expansion joint is recommended to assure adequate function of the piping system.

2.6 DESIGN WITH PIPE LOOPS

Where space is not a primary concern, expansion loops are the preferred method for relieving the thermal stress between anchors in suspended piping systems since it can be easily fabricated using pipe and elbows at the job site.

Loops should be horizontal wherever possible to avoid entrapping air or sediment and facilitate drainage.

- For upward loops, air relief valves aid air removal and improve flow. In pressure systems, air removal for both testing and normal operation is required for safety.
- For downward loops, air pressure equalizing lines may be necessary to permit drainage.
- In both cases, special taps are necessary for complete drainage.

The size of the loop can be determined by using the "Elastic-Center Method." The concept is outlined as follows:

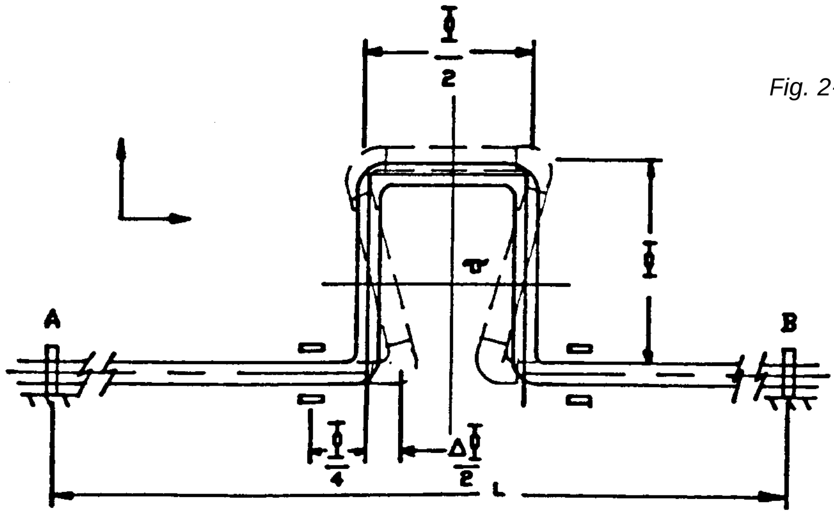


Fig. 2-4

Consider a properly guided expansion loop as shown in Figure 2-4. The centroid "0" of this structure is located at the center of the guides A and B, and the line of thrust will lie parallel to a line joining the guides. The only force that acts on this loop is in the x direction and can be found by the equation.

$$F_x = \frac{\Delta \bar{\Phi} EI}{I_x}$$

where $\Delta \bar{\Phi}$ = total linear expansion which will be absorbed by the loop,

F_x = force in the x direction,

E = modulus of elasticity of the pipe,

I = beam moment of inertia of the pipe, and

I_x = moment of inertia of the line about the x axis of the centroid.

$$\text{Since } I_x = \left[\frac{\bar{\Phi}}{4}\right]\left[\frac{\bar{\Phi}}{2}\right]^2 + \left[\frac{\bar{\Phi}}{2}\right]\left[\frac{\bar{\Phi}}{2}\right]^2 + \left[\frac{\bar{\Phi}}{4}\right] = \left[\frac{\bar{\Phi}}{2}\right]^2 \quad \frac{\bar{\Phi}^3}{4}$$

$$F_x = \frac{4\Delta\ell EI}{\ell^3}$$

Substituting $$M = F_x \left[\frac{\ell}{2}\right] \text{ and}$$

$$S_A = \frac{M\,D}{2\,I}$$

and arranging the required length ℓ in terms of other known values we obtain:

$$\ell = \left[\begin{array}{c}\ell\\ \hline \Delta \quad ED\\ SA\end{array}\right]^{1/2}$$

Where
M = bending moment, maximum at elbows,
SA = allowable stress,
D = outside diameter of pipe,
ℓ = required length of the expansion loop.

It should be noted here that similar result can be obtained using the Guided Cantilever Method of pipe flexibility calculation.

Where $$\frac{\Delta\ell}{2} = \frac{1}{4} \qquad \frac{F\ell^3}{EI} \qquad = \qquad \frac{M\ell^2}{4EI} \qquad = \qquad \frac{SA\ell^2}{2ED}$$

and again $$\ell = \left[\frac{\Delta\ell\ ED}{S_A}\right]^{1/2}$$

Calculation example: Determine the required expansion loop for 8-inch Bondstrand Series 2000M piping subjected to the following condition:

Operating temperature:	65°C (149°F)
Installation temperature:	20°C (68°F)
Total length of pipe between anchors:	100 meter (328 ft)

From Product Data Sheet for Bondstrand 2000M (FP194) we obtain at 150°F (66°C):

Allowable bending stress = $\frac{548 \text{ kg/cm}^2}{3}$ = 183 kg/cm^2 (2600 psi)

Thermal expansion coefficient	=	18×10^{-6} m/m/°C (10×10^{-6} in/in/°F)
Modulus of elasticity at 65°C	=	91,400 kg/cm2 (1,300,000 psi)
Pipe O.D.	=	22.1 cm (8.7 inch)

First determine the total thermal expansion for the entire length of the pipe section in question:

$$\begin{aligned}\Delta L &= \alpha\ L\ \Delta T\\ &= 18 \times 10^{-6}/°C\ (45°C)\ (100 \times 10^2)\ cm\\ &= 8.1\ cm\end{aligned}$$

Then

$$\ell = \left[\frac{\Delta \ell E D}{S_A} \right]^{1/2}$$

$$\ell = \left[\frac{8.1 \text{ cm } (91{,}400 \text{ kg/cm}^2)\ (22.1 \text{ cm})}{183 \text{ kg/cm}^2} \right]^{1/2} = 299 \text{ cm}$$

$$\ell = 2.99 \text{ meter}$$

Calculation of length ℓ can also be performed in English units:

$$\ell = \left[\frac{3.18 \text{ in } (1{,}300{,}000 \text{ psi})\ 8.7 \text{ in}}{2{,}600 \text{ psi}} \right]^{1/2} = 118 \text{ in}$$

$$\ell = 9 \text{ ft. - } 10 \text{ in.}$$

which is equivalent to 2.99 meters.

Table 2-II tabulates the length of loop in feet and meters required to absorb expansion.

TABLE 2-II: REQUIRED LENGTH FOR EXPANSION LOOP

NPS	Length Change in Inches																							
	1/2		1		1-1/2		2		2-1/2		3		3-1/2		4		4-1/2		5		5-1/2		6	
	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m
2	2	.6	3	.9	4	1.2	4	1.2	5	1.5	5	1.5	6	1.8	6	1.8	6	1.8	7	2.1	7	2.1	7	2.1
3	3	.9	4	1.2	5	1.5	5	1.5	6	1.8	6	1.8	7	2.1	7	2.1	8	2.4	8	2.4	9	2.7	9	2.7
4	3	.9	4	1.2	6	1.8	6	1.8	7	2.1	7	2.1	8	2.4	8	2.4	9	2.7	9	2.7	10	3.0	10	3.0
6	4	1.2	5	1.5	7	2.1	7	2.1	8	2.4	9	2.7	10	3.0	10	3.0	11	3.4	11	3.4	12	3.7	12	3.7
8	4	1.2	6	1.8	8	2.4	8	2.4	9	2.7	10	3.0	11	3.4	12	3.7	12	3.7	13	4.0	14	4.3	14	4.3
10	5	1.5	6	1.8	9	2.7	9	2.7	10	3.0	11	3.4	12	3.7	13	4.0	14	4.3	15	4.6	15	4.6	16	4.9
12	5	1.5	7	2.1	10	3.0	10	3.0	11	3.4	12	3.7	13	4.0	14	4.3	15	4.6	16	4.9	16	4.9	17	5.2
14	5	1.5	7	2.1	10	3.0	10	3.0	12	3.7	13	4.0	14	4.3	15	4.6	16	4.9	17	5.2	17	5.2	18	5.4
16	6	1.8	8	2.4	11	3.4	11	3.4	13	4.0	14	4.3	15	4.6	16	4.9	17	5.2	18	5.4	18	5.4	19	5.8
18	6	1.8	8	2.4	12	3.7	12	3.7	13	4.0	15	4.6	16	4.9	17	5.2	18	5.4	19	5.8	19	5.8	21	6.4
20	6	1.8	9	2.7	13	4.0	13	4.0	14	4.3	15	4.6	17	5.2	18	5.4	19	5.8	20	6.0	20	6.0	22	6.7
24	7	2.1	10	3.0	14	4.3	14	4.3	15	4.6	17	5.2	18	5.4	19	5.8	21	6.4	22	6.7	22	6.7	24	7.3
28	7	2.1	10	3.0	15	4.6	15	4.6	17	5.2	18	5.4	20	6.0	21	6.4	22	6.7	23	7.0	23	7.0	26	7.9
30	8	2.4	11	3.4	15	4.6	15	4.6	17	5.2	19	5.8	20	6.0	22	6.7	23	7.0	24	7.3	24	7.3	17	8.2
36	8	2.4	12	3.7	17	5.2	17	5.2	19	5.8	21	6.4	22	6.7	24	7.3	25	7.6	27	8.2	27	8.2	29	8.8

2.7 DESIGN USING Z LOOPS AND L BENDS

Similarly the Z-loop and L-bends can be analyzed by the same guide cantilever method.

$$\Delta_L = \frac{F_x L^3}{4EI} = \frac{M L^2}{4EI} = \frac{S_A L^2}{2ED}$$

$$L = \left[\frac{2\Delta L \;\; ED}{S_A}\right]^{1/2}$$

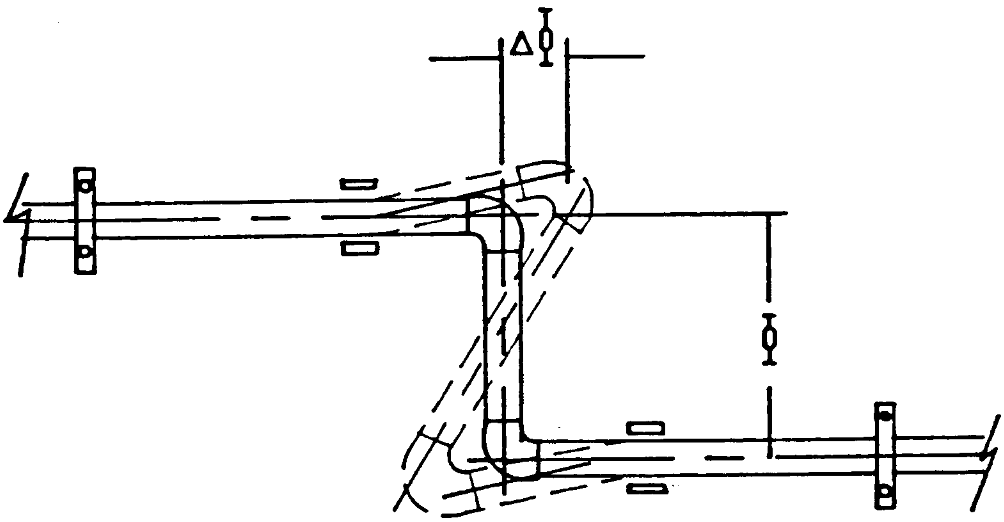


Fig. 2-5

Note: In special cases where the pipe is insulated, longer length is needed to compensate for the stiffer loop members.

The required length Φ in this case should be adjusted by a factor

$$(EI_{\text{insulated pipe}}/EI_{\text{bare pipe}})^{1/2}$$

which was derived as follows:

$$\Delta\Phi_{bp} = \frac{M\Phi^2_{bp}}{2EI_{bp}} \qquad \Phi_{bp} = \left[\frac{2\Delta\Phi_{bp}\, EI_{bp}}{M}\right]^{1/2}$$

$$\Delta\Phi_{ip} = \frac{M\Phi^2_{ip}}{2EI_{ip}} \qquad \Phi_{ip} = \left[\frac{2\Delta\Phi_{ip}\, EI_{ip}}{M}\right]^{1/2}$$

For the same application condition:

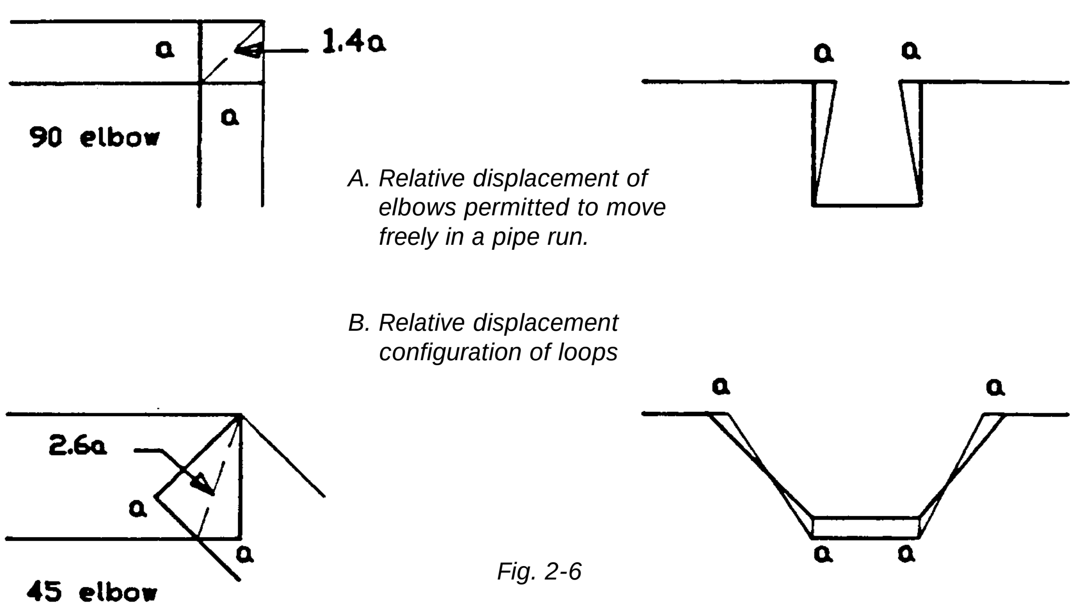
$$\Delta\Phi_{bp} = \Delta\Phi_{ip}$$

$$\Phi_{ip} = \Phi_{bp} \left[EI_{ip}/EI_{bp}\right]^{1/2}$$

Loops using 90° elbows change length better than those using 45° elbows. Unlike a 90° turn, a 45° turn carries a thrust component through the turn which can add axial stress to the usual bending stress in the pipe and fittings. Alignment and deflection are also directly affected by the angular displacement at 45° turns and demand special attention for support design and location.

A 45° elbow at a free turn with the same increment of length change in each leg will be displaced 86 percent more than a 90° elbow. The relative displacement in the plane of a loop is also more of a problem. Figure 2-6 illustrates the geometry involved.

Comparison of Displacement in 90° vs. 45° elbows caused by a Unit Length Change:



A. Relative displacement of elbows permitted to move freely in a pipe run.

B. Relative displacement configuration of loops

Fig. 2-6

Table 2-III tabulates the length of loop or bend in feet and meters required to absorb expansion.

TABLE 2-III: REQUIRED LENGTH FOR Z TYPE LOOP AND L BEND

NPS	Length Change in Inches																							
	1/2		1		1-1/2		2		2-1/2		3		3-1/2		4		4-1/2		5		5-1/2		6	
	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m	ft	m
2	3	.9	4	1.3	6	1.9	6	1.9	7	2.1	7	2.3	8	2.5	9	2.6	9.2	2.8	9.7	2.9	10	3.1	10.6	3.2
3	4	1.1	5	1.6	6	2.0	7	2.3	8	2.5	9	2.8	10	3.0	10	3.2	11	3.4	12	3.6	12	3.7	13.0	3.9
4	4	1.3	6	1.8	6	2.2	8	2.6	9	2.9	10	3.1	11	3.4	12	3.6	13	3.8	12	4.0	14	4.2	15.0	4.4
6	5	1.6	7	2.2	7	2.7	10	3.1	11	3.5	12	3.8	13	4.1	14	4.2	15	4.7	16	4.9	17	5.2	18.0	5.4
8	6	1.8	8	2.4	9	3.1	12	3.5	13	3.8	14	4.3	15	4.7	16	5.0	17	5.8	18	5.6	19	5.9	20.0	6.1
10	6	2.0	9	2.8	10	3.4	13	4.0	15	4.4	16	4.8	17	5.2	18	5.6	19	5.9	21	6.3	22	6.6	22.0	6.9
12	7	2.2	10	3.0	12	3.6	14	4.3	16	4.8	17	5.3	19	5.7	20	6.1	21	6.5	22	6.8	23	7.1	24.0	7.5
14	7	2.3	10	3.2	13	3.9	15	4.5	17	5.1	18	5.5	20	6.0	21	6.4	22	6.8	23	7.1	25	7.5	26.0	7.8
16	8	2.4	11	3.4	14	4.2	16	4.8	18	5.4	19	5.9	21	6.4	22	6.8	24	7.2	25	7.6	26	8.0	27.0	8.4
18	8	2.6	12	3.6	15	4.4	17	5.1	19	5.7	21	6.3	22	6.8	24	7.2	25	7.7	27	8.1	28	8.5	29.0	8.9
20	9	2.7	13	3.8	15	4.7	18	5.4	20	6.0	22	6.6	23	7.1	25	7.6	27	8.1	28	8.5	29	9.0	31.0	9.4
24	10	3.0	14	4.0	17	5.1	19	5.9	22	6.6	24	7.2	26	7.8	27	8.4	29	8.9	31	9.4	32	9.8	34.0	10.2
28	10	3.2	15	4.5	18	5.5	21	6.4	23	7.1	26	7.8	28	8.5	30	9.1	31	9.6	33	10.1	35	10.6	36.0	11.1
30	11	3.3	15	4.6	19	5.7	22	6.6	24	7.4	27	8.1	29	8.7	31	9.4	33	9.9	34	10.5	36	11.0	38.0	11.5
36	12	3.6	17	5.1	21	6.3	24	7.2	27	8.0	29	8.9	31	9.6	34	10.2	36	10.9	38	11.5	39	12.0	42.0	12.5

3.0 Design for Thrust (Restrained Systems)

3.1 GENERAL PRINCIPLES

Occasionally, the layout of a system makes it impossible to allow the pipe to move freely, as for example, a ballast line running thwart-ships between longitudinal bulkheads. Or it may be necessary to anchor certain runs of an otherwise free system. In a fully restrained pipe (anchored against movement at both ends), the designer must deal with thrust rather than length change. Both temperature and pressure produce thrust which must be resisted at turns, branches, reducers and ends. Knowing the magnitude of this thrust enables the designer to select satisfactory anchors and check the axial stress in pipe and shear stress in joints. Remember that axial thrust on anchors is normally independent of anchor spacing.

Caution: In restrained systems, pipe fittings can be damaged by faulty anchorage or by untimely release of anchors. Damage to fittings in service can be caused by bending or slipping of an improperly designed or installed anchor. Also, length changes due to creep are induced by high pressures or temperatures while pipe is in service. When anchors must later be released, especially in long pipe runs, temporary anchors may be required to avoid excessive displacement and overstress of fittings.

3.2 THRUST IN AN ANCHORED SYSTEM

Both temperature and pressure produce thrust, which is normally independent of anchor spacing. In practice, the largest compressive thrust is normally developed on the first positive temperature cycle. Subsequently, the pipe develops both compressive and tensile loads as it is subjected to temperature and pressure cycles. Neither compressive nor tensile loads, however, are expected to exceed the thrust on the first cycle unless the ranges of the temperature and pressure change.

3.3 THRUST DUE TO TEMPERATURE

In a fully restrained Bondstrand pipe, length changes induced by temperature change are resisted at the anchors and converted to thrust. The thrust developed depends on thermal coefficient of expansion, the cross-sectional area, and the modulus of elasticity.

3.4 THRUST DUE TO PRESSURE

Thrust due to internal pressure in a suspended but restrained system is theoretically more complicated. This is because in straight, restrained pipelines with all joints adhesive bonded or flanged, the Poisson effect produces considerable tension in the pipe wall.

As internal pressure is applied, the pipe expands circumferentially and at the same time contracts longitudinally. This tensile force is important because it acts to reduce the hydrostatic thrust on anchors. In lines with elbows, closed valves, reducers or closed ends, the internal pressure works on the cross-sectional area of the ends. This thrust tends to be about twice as great as the effect of pressure on the pipe wall.

The concurrent effects of pressure and temperature must be combined for design of anchors. Similarly, on multiple pipe runs, thrusts developed in all runs must be added for the total effect on anchors.

3.5 FORMULAS FOR CALCULATING THRUST IN RESTRAINED PIPELINES

3.5.1 Thrust Due To Temperature Change In An Anchored Line

The thrust due to temperature change in a system fully restrained against length change is calculated by:

$$P = \alpha \Delta T A E_l$$

where P = thrust (lbf or kg),

α = coefficient of thermal expansion (in./in./°F or m/m/°C),

ΔT = change in temperature (°F or °C),

E_l = longitudinal modulus of elasticity at lower temperature (psi or kg/cm²),

A = average cross-sectional area of the pipe wall (in.² or cm²), See Table 4-IV.

For example:

$\alpha = 10 \times 10^{-6}$ in./in./°F

$\Delta T = 150$°F

$A = 4.23$ in² for 6 inch pipe

$E_l = 1.6 \times 10^6$ psi

then $P = (10 \times 10^{-6})(150)(4.23)(1.6 \times 10^6) = 10{,}150$ lbf. or from Table 3-1

$P = 6{,}770 \times 1.5 = 10{,}150$ lbf.

3.5.2 Thrust Due To Pressure In An Anchored System

In a fully restrained system, calculate the thrust between anchors induced by internal pressure using:

$$P = \frac{\pi p D_m \overline{ID}}{2} \; \frac{E_l}{E_c} \; (-\nu_{lc})$$

where P = internal pressure (psi or kg/cm²),

$\overline{ID}$ = internal diameter (in. or cm),

E_l = longitudinal modulus of elasticity (psi or kg/cm²),

E_c = circumferential modulus of elasticity (psi or kg/cm²), and

ν_{lc} = Poisson's ratio.

Note: *Use elastic properties at lowest operating temperature to calculate maximum expected thrust.*

For example, assume that

$$\overline{ID} = 6.26 \text{ in.,}$$

$$D_m = 6.44 \text{ in.,}$$

$$P = 100 \text{ psi.}$$

$$E_l = 1.6 \times 10^6 \text{ psi,}$$

$$E_c = 3.6 \times 10^6 \text{ psi, and}$$

$$\nu_{lc} = 0.56$$

then $$P = \frac{3.14\,(100)\,(6.44)\,(6.26)}{2} \; \frac{(1.6)}{(3.6)} \; (0.56) = 1{,}580 \text{ lbf (tension)}$$

or read the value of 1,580 lbf from Table 3-II.

3.5.3 Thrust Due To Pressure On A Closed End

Where internal pressure on a closed end exerts thrust on supports, calculate thrust using:

$$P = \frac{\pi \overline{ID}^2}{4}\, p$$

where $\overline{ID}$ = inside diameter of the pipe (in. or cm).

Values are given in Table 3-III.

For example: If there is 100 psi in a 6-inch (6.26 $\overline{ID}$) pipe, thrust is

$$P = \frac{3.14\,(6.26)^2}{4} \times 100 = 3{,}080 \text{ lbf}$$

3.6 LONGITUDINAL STRESS IN PIPE AND SHEAR STRESS IN ADHESIVE

Stress in the pipe is given in each of the above cases by:

$$f = \frac{P}{A}$$

where f = longitudinal stress (psi or kg/cm^2).

In the last example for pressure on a closed end:

$$f = \frac{3{,}080}{4.23} = 728 \text{psi}$$

The allowable stress is one third of the longitudinal tensile strength at the appropriate temperature as given in the Bondstrand Product Data Sheet. For Series 2000M and Series 7000M pipe the allowable stress at 70°F is 8,500 psi/3.0 = 2830 psi (199 kg/cm2). For short-term effects such as those resulting from green sea loads, a higher allowable stress may be justified.

Shear stress in an adhesive bonded joint is:

$$\sigma = \frac{P}{\pi D_j L_b}$$

where σ = shear stress in adhesive (psi or kg/cm^2),

D_j = joint diamater (in. or cm), see Table 3-IV.

L_b = bond length (in. or cm), see Table 3-IV.

For example: In the case of 100 psi pressure on a closed end 6-inch pipe, as previously calculated:

$$P = 3{,}080 \text{ lbf}$$

$$\sigma = \frac{3{,}080}{3.14\,(6.54)\,2.25} = 67 \text{ psi}$$

The allowable shear stress for RP-34 adhesive (normally used with Series 2000M products) is 250 psi (17.6 kg/cm^2). The allowable shear stress for RP-60 adhesive (normally used with Series 7000M products) is 212 psi (14.4 kg/cm^2).

TABLE 3-I

THRUST IN AN ANCHORED PIPELINE DUE TO TEMPERATURE CHANGE

FOR BONDSTRAND PIPING

Nominal Pipe Size (in)	(mm)	For ΔT = 100°F (lbf)	For ΔT = 100°C (kg)
2	50	1,810	1,490
3	75	2,720	2,250
4	100	4,370	3,600
6	150	6,770	5,580
8	200	11,220	9,230
10	250	17,280	14,260
12	300	24,160	19,900
14	350	27,840	22,900
16	400	36,000	29,600
18	450	44,960	37,200
20	500	55,200	45,600
24	600	78,560	64,800
28	700	113,900	94,000
30	750	130,200	107,200
36	900	184,000	152,000

Note:
1. For temperature change other than 100°F or 100°C use linear ratio for thrust.
2. Calculations are based on elastic properties at room temperature.
3. Calculations are based on IPS dimensions for sizes 2 to 24 inch, MCI dimensions for 28 to 36 inch.

TABLE 3-II

THRUST FORCE DUE TO INTERNAL PRESSURE IN AN ANCHORED PIPELINE

FOR BONDSTRAND PIPING

Nominal Pipe Size (in)	Nominal Pipe Size (mm)	For Internal Pressure = 100 psi (lbf)	For Internal Pressure = 10kg/cm^2 (kg)
2	50	182	118
3	75	423	273
4	100	700	451
6	150	1,580	1,020
8	200	2,710	1,750
10	250	4,300	2,780
12	300	6,120	3,950
14	350	7,090	4,570
16	400	9,260	5,980
18	450	11,700	7,560
20	500	14,500	9,330
24	600	20,800	13,400
28	700	30,500	19,700
30	750	35,000	22,600
36	900	50,000	32,300

Note:
1. For temperature change other than 100 psi or 10 kg/cm^2, use linear ratio for tensile force.
2. Calculations are based on elastic properties at room temperature.
3. Calculations are based on IPS dimensions for sizes 2 to 24 inch, MCI dimensions for 28 to 36 inch.

TABLE 3-III

THRUST DUE TO PRESSURE ON A CLOSED END

FOR BONDSTRAND PIPING

Nominal Pipe Size (in)	Nominal Pipe Size (mm)	For Internal Pressure = 100 psi (lbf)	For Internal Pressure = 10kg/cm2 (kg)
2	50	343	221
3	75	814	525
4	100	1,350	869
6	150	3,080	1,900
8	200	5,310	3,420
10	250	8,410	5,430
12	300	12,000	7,730
14	350	13,900	8,950
16	400	18,100	11,700
18	450	22,900	14,800
20	500	28,300	18,300
24	600	40,800	26,300
28	700	59,700	38,500
30	750	68,400	44,200
36	900	97,900	63,200

Note:
1. For temperature change other than 100 psi or 10 kg/cm^2, use linear ratio for thrust.
2. Calculations are based on IPS dimensions for sizes 2 to 24 inch, MCI dimensions for 28 to 36 inch.

TABLE 3-IV

ADHESIVE BONDED JOINT DIMENSIONS

Nominal Pipe Size (in)	(mm)	Joint Diameter (in)	(mm)	Bond Length (in)	(mm)
2	50	2.33	5.92	1.81	4.60
3	75	2.45	8.76	1.81	4.60
4	100	4.43	11.25	1.81	4.60
6	150	6.54	16.62	2.25	5.72
8	200	8.54	21.70	2.50	6.35
10	250	10.68	27.13	2.75	6.99
12	300	12.68	32.22	3.00	7.62
14	350	13.93	35.33	3.50	8.89
16	400	15.91	40.41	4.00	10.16
18	450	17.93	455.3	4.13	10.48
20	500	19.93	506.1	4.38	11.11
24	600	23.93	607.7	5.00	12.70
28	700	28.97	735.8	6.00	15.24
30	750	31.20	787.9	6.50	16.51
36	900	37.12	942.9	8.00	20.32

Note: 1. Joint Diameters are based on IPS dimensions for sizes 2 to 24 inch, MCI dimensions for 28 to 36 inch.

2. Adhesive bonded joints are available for field joining of pipe and fittings in size range 2 to 16 inch. Only adhesive bonded flanges are available for field joints above 16 inch.

4.0 Support Location & Spacing

4.1 GENERAL

This section gives recommendations on placement of supports and maximum support spacing. These recommendations give minimum support requirements. Additional support may be needed where pipe is exposed to large external forces as for example, pipe on desk subject to green wave loading.

Techniques used in determining support requirements for Bondstrand are similar to those used for carbon steel piping systems; however, important differences exist between the two types of piping. Each requires its own unique design considerations. For example, Bondstrand averages 16 percent of the weight of schedule 40 steel, has a longitudinal modulus 14 times smaller, and a thermal coefficient of expansion 50 percent larger.

4.2 ABRASION PROTECTION

Bondstrand should be protected from external abrasion where it comes in contact with guides and support, particularly in areas of significant thermal expansion, in long runs of pipe on weather decks, or in passageways which would be affected by dynamic twisting of the ship's structure. Such protection is achieved through the use of hanger liners, rider bars or pads made of teflon or other acceptable material. Refer to Table 4-I for details.

TABLE 4-I

PIPE HANGER LINER, RIDER BAR, OR PAD MATERIAL FOR ABRASION PROTECTION

Material	Thickness (in.)	Thickness (mm)	Maximum Temperature (°F)	Maximum Temperature (°C)
Silicone Rubber R-765 Class 2, 600 Durometer or equal	1/8	3.2	450	232
Synthetic Rubber Sheet (Mil-R-6855, Class 2) 60 Durometer or equal	1/8	3.2	180	82
Flurocarbon Elastomer (Viton, Fluorel)	1/8	3.2	650	343
Teflon (Flurocarbon Resin)	1/4	6.0	750	399

4.3 SPANS ALLOWING AXIAL MOVEMENT

Supports that allow expansion and contraction of pipe should be located on straight runs of pipe where axial movement is not restricted by flanges or fittings. In general, supports may be located at positions convenient to nearby ships structures, provided maximum lengths of spans are not exceeded.

4.4 SPAN RECOMMENDATIONS

Recommended maximum spans for Bondstrand pipe at various operating temperatures are given in Table 4-II. These spans are intended for normal horizontal piping arrangements, i.e., those which have no fittings, valves, vertical runs, etc., but which may include flanges and nonuniform support spacings. The tabular values represent a compromise between continuous and single spans. When installed at the support spacings indicated in Table 4-II, the weight of the pipe full of water will produce a long-time deflection of about 1/2 inch, (12.7 mm), which is usually acceptable for appearance and adequate drainage. Fully continuous spans may be used with support spacings 20 percent greater for this same deflection; in simple spans, support spacings should be 20 percent less. For this purpose, continuous spans are defined as interior spans (not end spans), which are uniform in length and free from structural rotation at supports. Simple spans are supported only at the ends and are either hinged or free to rotate at the supports. In Table 4-II, recommendations for support spacings for mechanical joints assume simple spans and 20 ft. (6.1m) pipe length. For additional information regarding the special problems involved in support and anchoring of pipe with mechanical joints, see Appendix E.

4.4.1 Formula for Calculating Support Spacing for Uniformly Distributed Load

Suspended pipe is often required to carry loads other than its own weight and a fluid with a specific gravity of 1.0. Perhaps the most common external loading is thermal insulation, but the basic principle is the same for all loads which are uniformly distributed along the pipeline. The way to adjust for increased loads is to decrease the support spacing, and conversely, the way to adjust for decreased loads is to increase the support spacing. An example of the latter is a line filled with a gas instead of a liquid; and longer spans are indicated if deflection is the controlling factor.

For all such loading cases, support spacings for partially continuous spans with a permissible deflection of 0.5 inch are determined using:

$$L = 0.258 \left[\frac{(EI)}{w} \right]^{1/4}$$

TABLE 4-II

RECOMMENDED MAXIMUM SUPPORT SPACINGS FOR

PIPE AT 100°F (38°C) AND 150°F (66°C) OPERATING TEMPERATURES

(FLUID SPECIFIC GRAVITY = 1.0)

Nominal Diameter		Uniform Load				Partially Continuous Spans				For Mechanical Joints			
		Pipe		Fluid		@100°F(38°C)		@150°F(66°C)		@100°F(38°C)		@150°F(66°C)	
(in)	(mm)	lb/in.	kg/m	lb/in.	kg/m	ft	m	ft	m	ft	m	ft	m
2	50	0.07	1.2	0.13	2.2	11.8	3.6	11.2	3.4	9.5	2.9	9.0	2.7
3	80	0.10	1.9	0.29	5.3	13.6	4.1	12.8	3.9	10.9	3.3	10.3	3.1
4	100	0.17	3.0	0.48	8.7	15.4	4.7	14.6	4.4	12.4	3.8	11.7	3.6
6	150	0.26	1.6	1.1	20.	17.5	5.3	16.5	5.0	14.0	4.3	13.2	4.0
8	200	0.43	7.8	1.9	34.	20.0	6.1	18.9	5.8	16.0	4.9	15.1	4.6
10	250	0.68	12.	3.0	54.	22.4	6.8	21.2	6.5	18.0	5.5	17.[illegible]	5.2
12	300	0.95	17.	4.3	77.	24.4	7.4	23.0	7.0	19.5	6.0	18.5	5.6
14	350	1.1	20.	5.1	89.	25.	7.7	24.	7.2	20.	6.1	19.1	6.1
16	400	1.4	25.	6.8	117.	27.	8.2	25.	7.7	20.	6.1	20.	6.1
18	450	1.7	32.	8.3	148.	29.	8.7	27.	8.3	20.	6.1	20.	6.1
20	500	2.2	39.	10.2	182.	30.	9.2	28.	8.7	20.	6.1	20.	6.1
24	600	3.1	56.	14.7	263.	33.	10.	31.	9.5	20.	6.1	20.	6.1
28	700	4.2	75.	20.0	358.	33.	10.	33.	9.5	20.	6.1	20.	6.1
30	750	1.8	86.	23.0	411.	33.	10.	33.	10.	20.	6.1	20.	6.1
36	900	6.9	123.	33.1	591.	33.	10.	33.	10.	20.	6.1	20.	6.1

Note:

1. For 14- through 36-inch diameters, loads tabulated are for Iron Pipe Size and are 7 to 12 percent less than for Metric Cast Iron sizes. However, recommended spans are suitable for either.
2. Span recommendations apply to normal horizontal piping support arrangements and are calculated for a maximum long-time deflection of 1/2 inch to ensure good appearance and adequate drainage.
3. Includes Quick-Lock adhesive bonded joints and flanged joints.
4. Maximum spans for mechanically joined pipe are limited to one pipe length.
5. Modulus of elasticity for span calculations:

   E = 2,100,000 (psi)-6000 (psi/°F) x T (°F). See Table 4-III.

where L = support spacings, ft.

(EI) = beam stiffness (lb-in^2, from Table 4-III and 4-IV)

w = total uniformly distributed load (lb/in.).

In metric units:

$$L = 0.124 \left[\frac{(EI)}{w} \right]^{1/4}$$

where L = support spacings (m)

(EI) = beam stiffness (kg-cm^2) (from Table 4-III and 4-IV)

w = total uniformly distributed load (kg/m)

For example: Calculate the recommended support spacing for 6-inch Bondstrand Series 2000M pipe full of water at 150°F:

$$L = 0.258 \left[\frac{1{,}200{,}000 \times 19.0}{1.36} \right]^{1/4} \quad 16.5 \text{ ft.}$$

4.5 SUSPENDED SYSTEM RESTRAINED FROM MOVEMENT

Anchors may be used to restrict axial movement at certain locations (see Section 5 for anchor details). Such restriction is essential:

- Where space limitations restrict axial movement.
- To transmit axial loads through loops and expansion joints.
- To restrain excessive thrusts at turns, branches, reducers, and ends
- To support valves. This is done not only to support the weight of valves and to reduce thrust, but it also prevents excessive loads on pipe connections due to torque applied by operation of valves.

Refer to Section 3 for determining thrust in an anchored system.

TABLE 4-III

MODULUS OF ELASTICITY FOR CALCULATIONS OF SUPPORT SPACINGS

Temperature	E_1
100°F (38°C)	1,500,000 psi (105,500 kg/cm2)
150°F (66°C)	1,200,000 psi (84,400 kg/cm2)
200°F (93°C)	900,000 psi (63,300 kg/cm2)

In pipe runs anchored at both ends, a method of control must be devised in order to prevent excessive lateral deflection or buckling of pipe due to compressive load. Guides may be required in conjunction with expansion joints to control excessive deflection. Tables 4-V and 4-VI give recommendations on guide spacing versus temperature change for marine pipe with restrained ends.

4.6 EULER AND ROARK EQUATIONS

The Euler equation is first used to check the stability of the restrained line.

$$L = \pi \left[\frac{I}{\alpha \Delta T \ A} \right]^{1/2}$$

where L = unsupported length or guide spacing (in. or cm),

I = beam moment of inertia (in^4 or cm^4) see Table 4-IV,

α = coefficient of thermal expansion (in./in./°F or m/m/°C),

A = cross-sectional area (in^2 or cm^2) see Table 4-IV,

ΔT = change in temperature (°F or °C).

The equation gives maximum stable length of a pipe column when fixed ends are assumed.

In Tables 4-V and 4-VI this maximum length is reduced by 25 percent to allow for non-Euler behavior near the origin of the curve.

TABLE 4-IV

PIPE DIMENSIONS AND SECOND MOMENT OF AREAS (SERIES 2000M)

IRON PIPE SIZE (IPS)

Nominal Diameter in.	Nominal Diameter mm.	Diameter Inside ID in.	Diameter Inside ID mm.	Diameter Outside OD in.	Diameter Outside OD mm.	Total Wall: Minimum Thickness, T_t in.	Minimum Thickness, T_t mm.	Average Sectional Area, A in^2	Average Sectional Area, A cm^2	Beam Second Moment of Area, I in^4	Beam Second Moment of Area, I cm^4
2	50	2.09	53.1	2.37	60.3	.140	3.56	1.13	7.3	.59	25.
3	80	3.22	81.8	3.50	88.9	0.140	3.56	1.70	11.0	1.98	83.
4	100	4.14	105.2	4.50	114.3	0.180	4.57	2.73	17.6	5.50	229.
6	150	6.26	159.0	6.64	168.6	0.188	4.77	4.23	27.3	19.0	792.
8	200	8.22	208.8	8.70	221.0	0.241	6.12	7.01	45.2	55.7	2320.
10	250	10.35	262.9	10.95	278.0	0.298	7.57	10.8	69.8	139.	5770.
12	300	12.35	313.7	13.05	331.5	0.351	8.92	15.1	97.4	278.	11500.
14	350	13.29	337.6	14.04	356.6	0.376	9.55	17.4	112.	372.	15600.
16	400	15.19	385.8	16.04	407.5	0.428	10.87	22.5	145.	634.	26400.
18	450	17.08	433.8	18.04	458.1	0.478	12.15	28.1	182.	1010.	42000.
20	500	18.98	482.1	20.04	509.0	0.529	13.44	34.5	223.	1530.	63800.
24	600	22.78	578.9	24.05	610.9	0.631	16.04	49.1	317.	3170.	132000.

METRIC IRON SIZE

28	700	27.57	700.4	29.09	739.0	0.760	19.30	71.2	460.	6780.	282000.
30	750	29.52	749.8	31.14	791.0	0.812	20.62	81.4	525.	8900.	370000.
36	900	35.31	896.9	37.25	946.1	0.967	24.57	115.	745.	18100.	755000.

Notes:

1. Outside diameters approximate those for iron pipe size, ISO International Standard 559 - 1977 and for cast iron pipes, ISO Recommendation R13-1965 as follows:

Diameter	2	3	4	5	6	7	8
2000M	60.3	88.9	114.3	168.6	221.0	278.0	331.5
ISO 559	60.3	88.9	114.3	168.3	219.1	273	323.9
ISO R13	--	98	118	170	222	274	326

2. Values are for composite moment of area of structural wall and liner cross-section in terms of the structural wall for Series 2000M. Beam second moment of area is also known as beam moment of Inertia.

Using the length developed by the Euler equation, the weight of and the physical properties at the operating temperature deflection of a horizontal pipe is calculated using the equation from Roark[1]:

$$y = \frac{-wL}{2KP} \left(\tan \frac{KL}{4} - \frac{KL}{4}\right)$$

where $$K = \left[P/(E_l) \right]^{1/2}$$

$$P = \frac{\pi^2 (E_l)}{L^2} = \alpha \triangle TAE$$

E_l = longitudinal modulus of elasticity (psi or kg/cm^2), see Table 4-III

w = uniform horizontal load (lb/in or kg/cm),

L = guide spacing (in. or cm).

If “y” is less than 0.5 inch (1.27cm), the “L” obtained using the Euler equation is the recommended guide spacing. If “y” is greater than .5 inch (1.27cm), choose a shorter length “L” and solve the Roark equation again for “y”. A final length recommendation is thus determined by trial and error when “y” closely approximates 0.5 inch (1.27cm).

4.7 SUPPORT OF PIPE RUNS CONTAINING EXPANSION .JOINTS

The modulus of elasticity for Bondstrand pipe is approximately 1/14th that of steel pipe. For this reason, the force due to expansion of Bondstrand pipe is not great enough to compress most varieties of expansion joints used in steel piping systems. Bondstrand requires elastomeric expansion joints.

The use of elastomeric expansion joints has somewhat limited marine applications. These joints have very limited resistance to external forces and, therefore, are not suitable for use in the bottom of tanks. However, it can be used for piping systems installed in the double bottoms were hydrostatic collapse pressure is not a requirement. During the installation careful consideration must be given to the proper support and guidance.

(1) R.J. Roark, Formulas for Stress and Strain, 3rd Edition, McGaw-Hill Book Co., New York, 1954.

TABLE 4-V

GUIDE SPACING VS. TEMPERATURE CHANGE FOR PIPE WITH RESTRAINED ENDS

Nom size		L in meters for Delta T in Deg C											
in	mm	10 °C	20 °C	30 °C	40 °C	50 °C	60 °C	70 °C	80 °C	90 °C	100 °C	110 °C	120 °C
2	50	3.2	2.3	1.9	1.6	1.4	1.3	1.2	1.1	1.1	1.0	1.0	0.9
3	80	4.8	3.4	2.8	2.4	2.2	2.0	1.8	1.7	1.6	1.5	1.5	1.4
4	100	6.3	4.5	3.7	3.2	2.8	2.6	2.4	2.2	2.1	2.0	1.9	1.8
6	150	9.5	6.7	5.5	4.7	4.2	3.9	3.6	3.3	3.2	3.0	2.9	2.7
8	200	12.6	8.9	7.3	6.3	5.6	5.1	4.8	4.4	4.2	4.0	3.8	3.6
10	250	16.0	11.3	9.2	8.0	7.2	6.5	6.0	5.7	5.3	5.1	4.8	4.6
12	300	19.1	13.5	11.0	9.6	8.6	7.8	7.2	6.8	6.4	6.1	5.8	5.5
14	350	20.6	14.6	11.9	10.3	9.2	8.4	7.8	7.3	6.9	6.5	6.2	6.0
16	400	23.7	16.7	13.7	11.8	10.6	9.7	8.9	8.4	7.9	7.5	7.1	6.8
18	450	26.7	18.9	15.4	13.4	12.0	10.9	10.1	9.5	8.9	8.5	8.1	7.7
20	500	29.7	21.0	17.1	14.9	13.3	12.1	11.2	10.5	9.9	9.4	9.0	8.6
24	600	35.8	25.3	20.7	17.9	16.0	14.6	13.5	12.7	11.9	11.3	10.8	10.3
28	700	43.5	30.8	25.1	21.8	19.5	17.8	16.5	15.4	14.5	13.8	13.1	12.6
30	750	46.6	33.0	26.9	23.3	20.9	19.0	17.6	16.5	15.5	14.7	14.1	13.5
36	900	56.0	39.6	32.3	28.0	25.0	22.8	21.2	19.8	18.7	17.7	16.9	16.2

Note: For horizontal pipe, values below the line may be taken from Table 4-II. For vertical pipe, use tabulated values as shown.

TABLE 4-VI

GUIDE SPACING VS. TEMPERATURE CHANGE FOR PIPE WITH RESTRAINED ENDS

Nom Size		L in ft. for Delta T Deg F																			
nn	mm	10 °F	20 °F	30 °F	40 °F	50 °F	60 °F	70 °F	80 °F	90 °F	100 °F	110 °F	120 °F	130 °F	140 °F	150 °F	160 °F	170 °F	180 °F	190 °F	200 °F
2	50	14	10	8	7	6	6	5	5	5	4	4	4	4	4	4	4	3	3	3	3
3	80	21	15	12	11	9	9	8	7	7	7	6	6	6	6	5	5	5	5	5	5
4	100	28	20	16	14	12	11	11	10	9	9	8	8	8	7	7	7	7	7	6	6
6	150	42	29	24	21	19	17	16	15	14	13	13	12	12	11	11	10	10	10	10	9
8	200	55	39	32	28	25	23	21	20	18	17	17	16	15	15	14	14	13	13	13	12
10	250	70	50	41	35	31	29	27	25	23	22	21	20	20	19	18	18	17	17	16	16
12	300	84	60	49	42	38	34	32	30	28	27	25	24	23	23	22	21	20	20	19	19
14	350	91	64	52	45	41	37	34	32	30	29	27	26	25	24	23	23	22	21	21	20
16	400	104	74	60	52	47	43	39	37	35	33	31	30	29	28	27	26	25	25	24	23
18	450	118	83	68	59	53	48	44	42	39	37	35	34	33	31	30	29	29	28	27	26
20	500	131	92	75	65	58	53	49	46	44	41	39	38	36	35	34	33	32	31	30	29
24	600	158	112	91	79	71	64	60	56	53	50	48	46	44	42	41	39	38	37	36	35
28	700	192	135	111	96	86	78	72	68	64	61	58	55	53	51	49	48	46	45	44	43
30	750	205	145	119	103	92	84	78	73	68	65	62	59	57	55	53	51	50	48	47	46
36	900	246	174	142	123	110	101	93	87	82	78	74	71	68	66	64	62	60	58	56	55

Note: For horizontal pipe, values below the line may be taken from Table 4-II. For vertical pipe, use tabulated values as shown.

There are also very distinct advantages to these expansion joints. They reduce vibration caused by equipment, are very compact and lightweight, and will compensate for axial movement.

When using an expansion joint to allow movement between anchors, the expansion joint should be placed as close as possible to one anchor or the other. The opposite side of the expansion joint should have a guide placed no further than five times the pipe's diameter from the expansion joint with a second guide positioned farther down the pipe. To determine the spacing for the second guide, find manufacturer's specifications on force required to compress the joint and refer to Figure 4-1 for recommended spacing.

The horizontal line at the top of each curve represents maximum support spacing for a totally unrestrained system. The lower end of the curve also becomes horizontal at the value for maximum guide spacing for a totally restrained system. This graph only shows values for pipes smaller than 12 inch diameter. In large diameters, the slightly increased guide spacing is not great enough to compensate for the added cost of the expansion joint.

The guide spacing for variable end thrust as produced by an expansion joint may be calculated as follows:

$$L = \pi \left[\frac{I}{\alpha \triangle TA} \right]^{1/2} = \pi \left[\frac{IE_l}{F} \right]^{1/2}$$

L = guide spacing (in. or cm.)

$F = \alpha \triangle TAE_l$ = force of compressing an expansion joint (lb or kg),

α = coefficient of thermal expansion (in/in/°F or m/m/°C).

E_l = longitudinal modules of elasticity at the highest operating temperature (psi or kg/cm^2), see Table 4-III

$\triangle T$ = change in temperature (°F or °C),

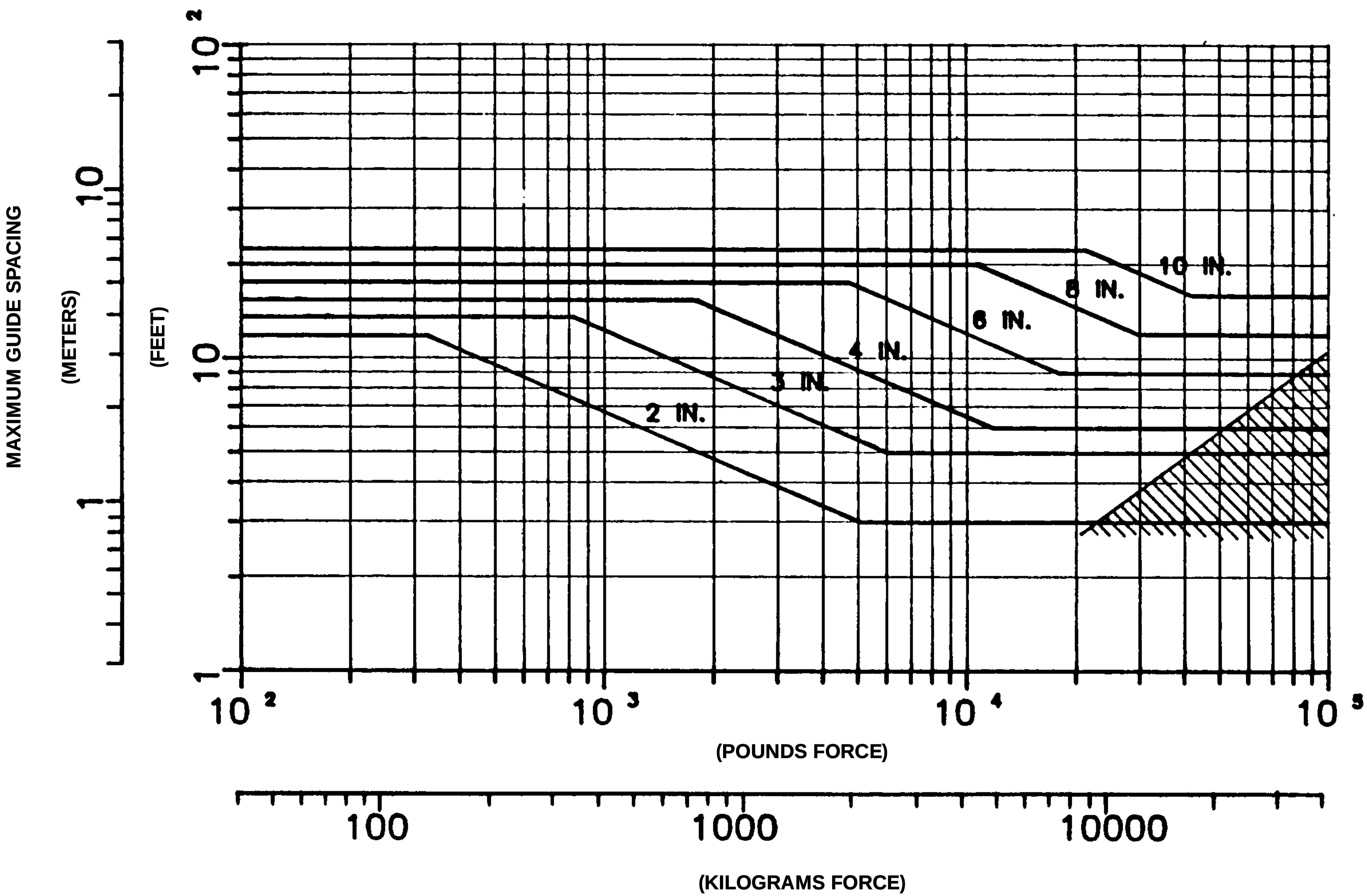
A = cross-sectional area (in^2 or cm^2), see Table 4-IV.

I = beam second moment of area (in^4 or cm^4), see Table 4-IV.

The values shown in Fig. 4-1 are calculated at 100°F (38°C) and reduced by 25 percent. Within the cross-hatched area, the pipe will crush prior to compression of the expansion joint based on a compressive allowable stress of 20,000 psi (1400 kg/cm^2).

FIGURE 4-1

AXIAL FORCE COMPRESSING AN EXPANSION JOINT VS. GUIDE SPACING



4.8 SUPPORTS FOR VERTICAL RUNS

Install a single support anywhere along the length of a vertical pipe run more than about ten feet (3mm) long. See Section 5 for suggested details. If the run is supported near its base, use loose collars as guides spaced as needed to insure proper stability.

Vertical runs less than ten feet (3mm) long may usually be supported as part of the horizontal piping. In either case, be sure the layout makes sufficient provision for horizontal and vertical movement at the top and bottom turns.

In vertical pipe runs, accommodate vertical length changes if possible by allowing free movement of fittings at either top or bottom or both. For each 1/8 inch (3mm) of anticipated vertical length change, provide 2 feet (62cm) of horizontal pipe between the elbow and the first support, but not less than 6 feet (1.9m) nor more than 20 feet (6.1m) of horizontal pipe. If the pipeline layout does not allow for accommodations of the maximum calculated length change, there are two possible resolutions:

- Anchor the vertical run near its base and use intermediate guides at the spacing shown in Tables 4-V or 4-VI, or
- Anchor the vertical run near its base and use intermediate Dresser-type couplings as required to accommodate the calculated expansion and contraction.

Treat columns more than 100 feet (30m) high (either hanging or standing) as special designs; support and provision for length change are important. The installer should be especially careful to avoid movement due to wind or support vibration while joints are curing.

4.9 CASE STUDY: VERTICAL RISER IN BALLAST TANK

A 210,000 DWT Tanker trades between Alaska and Panama. Segregated ballast tanks next to cargo tanks are served by 16 inch (400mm) Bondstrand Series 7000M pipe with RP-60 adhesive as shown in Figure 4-2. Maximum working pressure is 225 psi (15.5 bars). Maximum cargo temperature is 130ºF (54ºC). Minimum cargo temperature is 70ºF (21ºC). Minimum ballast water temperature in Alaska is 30ºF (-1ºC). Length of riser is 80 ft. (24.4m). Ambient temperature at time of pipe installation is 70ºF (21ºC). Maximum ambient temperature in Panama is 110ºF (43ºC).

4.9.1 What relative movement is expected between bottom of riser and bulkhead assuming no restraint on riser and no dresser-type couplings in the riser pipe?

Maximum relative movement due to temperature occurs when the steel bulkhead is at cargo temperature (1300F) and the fiberglass pipe is at minimum ballast water temperature (300F); i.e. at time of loading cargo in Alaska.

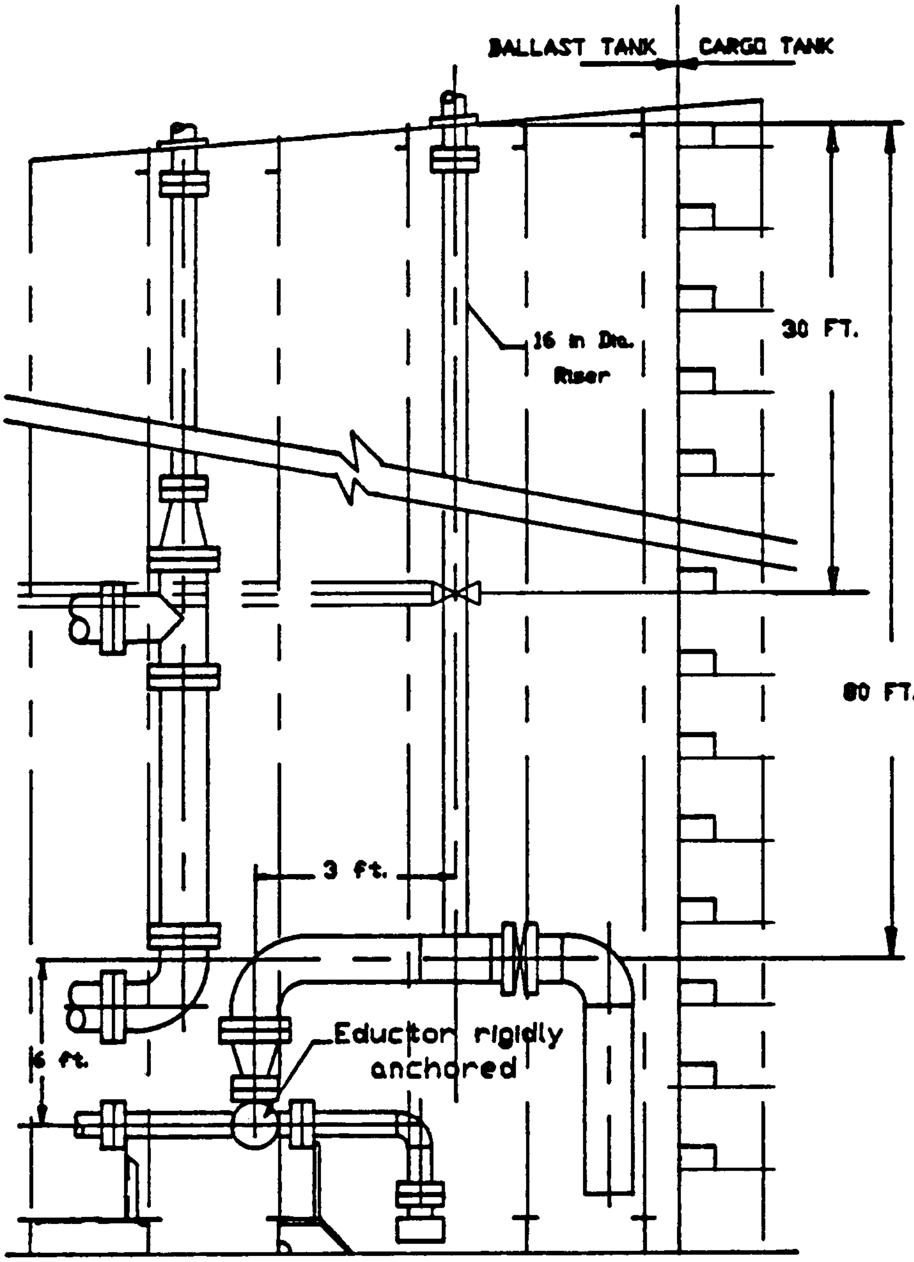
Expansion of bulkhead $= \alpha L \triangle T$
$= 6.38 \times 10^{-6} (80 \times 12) (130 - 70)$
$= 0.37$ inches

Contraction of pipe $= \alpha L \triangle T = 10 \times 10^{-6}(80 \times 12) (70 - 30)$
$= 0.38$ inches

Total relative movement due to temperature
$= 0.37 + 0.38 = 0.75$ inch

Note that pressure in the pipe under these conditions will cause the pipe to lengthen and reduce the relative movement between pipe and bulkhead.

Maximum relative movement due to pressure will occur at ambient temperature during ballasting in Panama.



VERTICAL RISER IN BALLAST TANK

FIGURE 4-2

$$\triangle L = (80 \times 12) \frac{225\ (15.19)^2}{4\ (.47)\ 1{,}6000{,}000\ (15.66)} \left[1\text{-}2\ (.56) \frac{1.6}{3.6} \right]$$

$$= 0.53 \text{ inches or see Table 2-I}$$

Thus the maximum expected relative movement is 0.75 inch as caused by temperature.

4.9.2 Does the pipeline layout below the riser allow enough flexibility to absorb the expected relative movement?

The eductor is rigidly anchored to prevent vibration; therefore, the riser support forms a Z loop. Interpolating from Table 2-III for a length change of 0.75 inch, the required leg length is 9.5 ft. Since the layout provides only 3 ft., there is insufficient flexibility to absorb movement.

Two solutions are possible:

A. Anchor the riser pipe near the bottom and provide guides as required to prevent buckling.

B. Insert Dresser-type couplings into the riser pipe to absorb the expected movement.

4.9.3 Solution A: Restrain the riser pipe

E_l at 30ºF = 2,100,000 — 6,000 (30) = 1,920,000 psi

Force on anchor, $P = E_l A \triangle L/L$
$= 1{,}920{,}000\ (22.5)\ 0.75/(80\text{x}12)$
$= 33{,}750$ lbf. due to temperature change

Note that pressure causes a reduction in anchor force due to temperature.

From Table 3-II, the force due to pressure alone is

$$P = 9260\ (225/100) = 20{,}840 \text{ lbf.}$$

Thus the anchor must be designed for 33,750 lbf.

The guide spacing should be established for a condition of empty ballast tank in Panama (110°F) and full cargo tank at 70°F. The pipe $\triangle T$ = 110-70=40°F. From Table 4-VI the guide spacing is 52 feet. Since the maximum unguided length is 30 ft., no additional guides would be required.

Check maximum tensile stress in pipe wall: In this case, assume hot cargo tank, cold ballast tank and maximum pressure occur simultaneously.

$$f = (33{,}750 + 20{,}840)/22.5$$
$$= 2{,}426 \text{ psi} < 2{,}830 \text{ psi allowable}$$

Check shear stress in RP—60 adhesive (See Table 3-IV):

$$a = (33{,}750 + 20{,}840)/[ir(15.91)(4.00)]$$
$$= 273 \text{ psi} > 212 \text{ psi allowable}$$

Solution A is not feasible due to shear stress in adhesive.

4.9.4 Solution B: Dresser-type couplings. Contraction in riser pipe due to pressure:

$$\triangle L = (80 \times 12) \left[(.56) \frac{225\,(15.9)^2}{2(.47)\ 3{,}600{,}000\ (15.19 + .47)} \right]$$

$$= 0.53 \text{ inches}$$

Thus the total contraction due to pressure and temperature is 0.75 + 0.53 = 1.28 inches. Each coupling allows 0.375 inch movement (See Appendix A) without gasket scuffing. However, considering the infrequent nature of the worse-case condition, two couplings should be sufficient. Light duty anchors will be required between couplings.

The riser bottom should be anchored against closed-end force. From Table 3-Ill, the force is:

$$P = 18{,}100\ (225/100) = 40{,}740 \text{ lbf.}$$

For anchor details see Section 5.

5.0 Anchor And Support Details

5.1 INTRODUCTION

Proper support of fiberglass piping systems is essential far the success of marine fiberglass installations. In dealing with installations of fiberglass pipe by shipyards, riding crews, arid owners throughout the world, the need for a Chapter dedicated to commonly used installation details has become evident.

The recommendations and details herein are based on sound engineering principles and experience in successful fiberglass piping installations. They are offered as alternatives and suggestions for evaluation, modification and implementation by a qualified Marine Engineer. Taking short cuts to save material or cost can cause grave consequences.

Notes:

1. Unless otherwise indicated, details are considered suitable for all approved piping systems.
2. Details are not intended to show orientation. Assemblies may be inverted or turned horizontal for attachment to ship's structure, bulkhead or deck. Good practice requires that support lengths in pipe runs provide the minimum dimensions needed for clearance of nuts and bolts.
3. Location, spacing and design of hangers and steel supports are to be determined by the shipyard, naval architect, or design agency. The necessary properties of fiberglass pipe are found in Chapters 2, 3 and 4.
4. Fiberglass piping systems on board ships are often designed to absorb movement and length changes at mechanical joints. To control deflections, the designer must allow for the weight and flexibility (hinge effect) introduced by mechanical couplings or expansion joints. See Appendix E.
5. Detailed dimensions are in inches and (mm) unless otherwise indicated.
6. Flange gaskets shall be 1/8 in. (3mm) thick, full face elastomeric gaskets with a Shore A Durometer hardness of 60 + 5. A Shore flurometer hardness of 50 or 60 is recommended for elastomeric pads.
7. Refer to ASTM F708 for additional details regarding standard practice for design and installation of rigid pipe hangers.

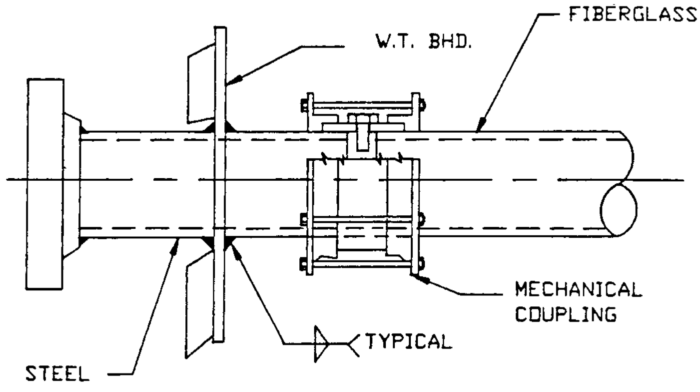
5.2 DETAILS

5.2.1 Water Tight Bulkhead Penetration, Flanged One End (Figure 5—1 On Following Page)

All water tight bulkheads and deck penetrations must be accomplished in steel and/or a non-ferrous metal capable of being welded water tight to the steel structure and must comply with classification societies rules. Fiberglass pipe can be attached to this penetration by a mechanical coupling (Dresser-type) between the metallic spool piece and fiberglass plain end. A step down coupling can also be used when the diameter of the metallic spool piece differs from the outside diameter of the fiberglass pipe.

Note: All spool pieces must be aligned with the longitudinal axis of the piping system within tolerance permitted by the mechanical coupling manufacturer regardless of the deck or bulkhead slope.

Fig. 5—1

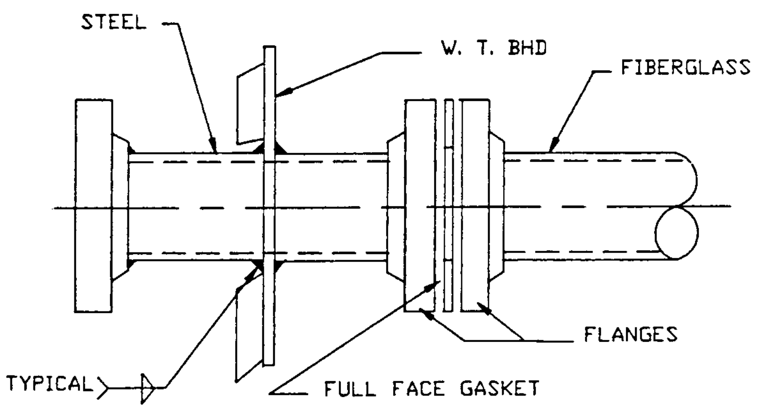


5.2.2 Water Tight Bulkhead Penetration, Flanged Both Ends (Figure 5—2)

The difference between this water tight spool piece and the previous one is the incorporation of flanges at both ends of the water tight bulkhead. This spool piece penetration is commonly used if a valve must be attached at the bulkhead penetration as required for design, safety reasons or classification society rules.

The alignment between the steel and fiberglass flanges must be within the tolerance discussed later in Paragraph 5.2.13 and shown by Figure 5—13. Special attention is required when valves are mounted on the flanges; lock washers shall be placed on the steel side (compressed by the nut) and flat washers on the fiberglass side (supported by the bolt).

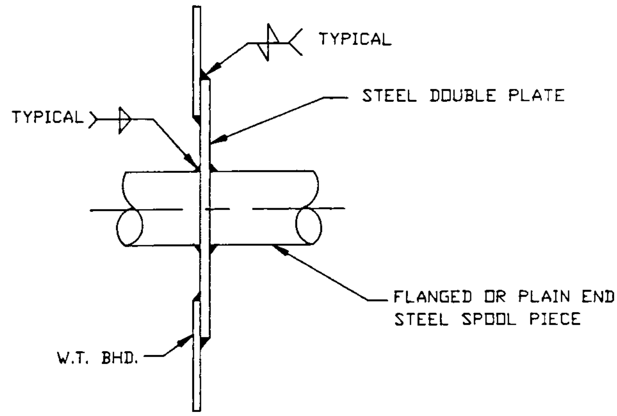
Fig. 5—2



5.2.3 Adjustable Water Tight Bulkhead Penetration, Flanged or Plain End. (Figure 5—3)

This particular spool piece connection allows tack welding at the bulkhead prior to final assembly so that the pipe is truly aligned, thus relieving fabrication stresses in the system. Two tanks can be aligned simultaneously with the use of this adjustable bulkhead penetration for proper alignment of the fiberglass pipe and fittings.

Fig. 5—3



5.2.4 Anchor Supports. (Figure 5—4)

This particular detail uses fiberglass saddle stock halfcollars to anchor the pipe and prevent longitudinal displacement along the axis. The gap between each 1800 saddle and the flat bar type clamp is 1/8 in. (3mm). These steel clamps are fabricated by the shipyard conforming to I.P.S. or M.C.I. outside diameters.

Notes: 1. The steel clamp should fit squarely against the angle bar support where the clamp will be bolted. Inserts, washers and spacers should not be used.

2. For thickness of the steel clamps refer to Note 3 under Paragraph 5.1.

5.2.5 Pipe Anchor Using 1800 Saddle Stock Full Collar (Figure 5—5 On Preceding Page)

This anchor support is accomplished in the same manner as Figure 5—4. It restricts the pipe from axial movement. The additional saddles will increase the area of contact between the saddle and the pipe to accommodate axial forces.

Calculations of thrust are discussed in Chapter 3. If the shear value of the adhesive to be used on a particular systems is exceeded (see Section 3.6), alternate types of anchors should be used; especially at fittings. See Figures 5—8 and 5—9 for examples.

Fig. 5—4

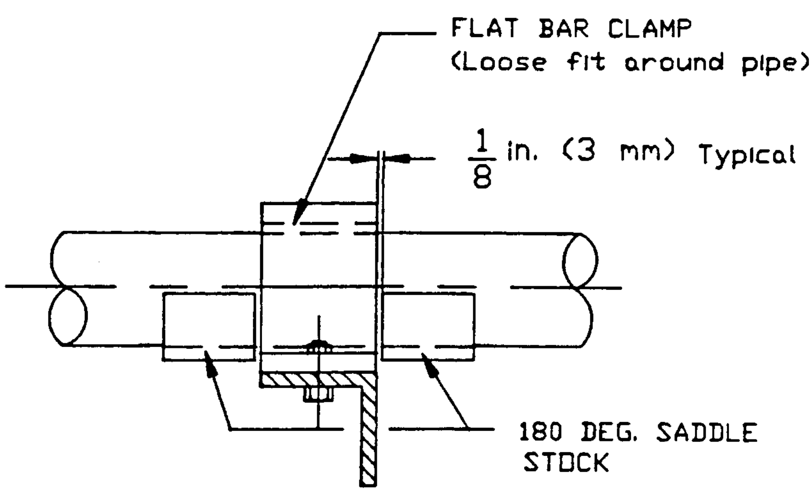
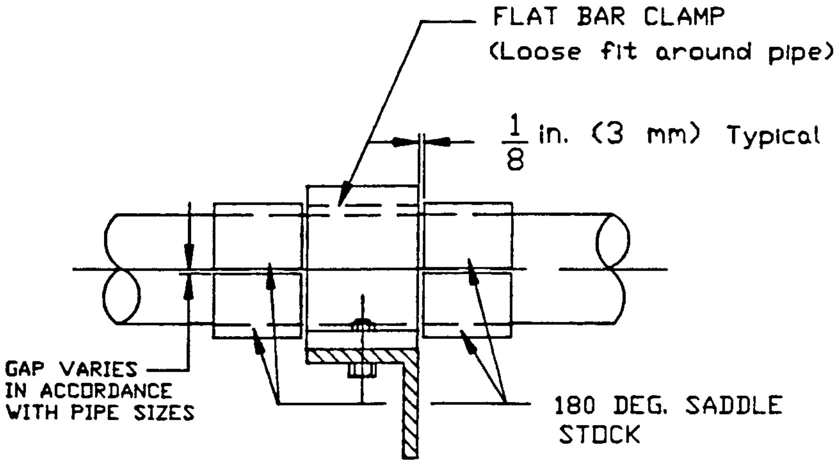


Fig. 5—5



5.2.6 Anchor Supports Using Full Metal Clamp (Figure 5—6)

The flat bar clamp is designed to restrain the pipe from axial movement. Saddle stock is installed on both sides of the steel clamp. In order to hold the pipe without damage see Table 5—1 below for recommended space between the bottom part of the clamp and upper part of the clamp.

For small pipe diameters 1—6 in. (25—150mm) it is useful to use a 1/4 thick (6mm) neoprene pad (Durometer A 50—60) compressed between the pipe and metal clamp. This will not prevent movement of the pipe in the axial direction. To prevent movement, the pipe must be properly anchored with saddle supports using half or full collars depending on the thrust imposed by the hydrostatic pressure or temperature change in the piping system.

Notes: 1. The steel clamp should fit squarely against the angle bar support where the clamp will be bolted. Inserts, washers and spacers should not be used.

2. For thickness of the steel clamps refer to Note 3 under Paragraph 5.1.

TABLE 5—I

NPS	Clearance At Bolts (Without Liner) (in)	(mm)	NPS	Clearance At Bolts (Without Liner) (in)	(mm)
1	1/8	3	20	5/8	16
1 1/2	1/8	3	22	5/8	16
2	1/8	3	24	5/8	16
3	1/4	6	26	5/8	16
4	1/4	6	28	5/8	16
6	3/8	10	30	5/8	16
8	3/8	10	32	5/8	16
10	1/2	12	34	5/8	16
12	1/2	12	36	5/8	16
14	5/8	16			
16	5/8	16			
18	5/8	16			

5.2.5 Pipe Anchor Using 180º Saddle Stock Full Collar (Figure 5—5)

This anchor support is accomplished in the same manner as Figure 5—4. It restricts the pipe from axial movement. The additional saddles will increase the area of contact between the saddle and the pipe to accommodate axial forces.

Calculations of thrust are discussed in Chapter 3. If the shear value of the adhesive to be used on a particular systems is exceeded (see Section 3.6), alternate types of anchors should be used; especially at fittings. See Figures 5—8 and 5—9 for examples.

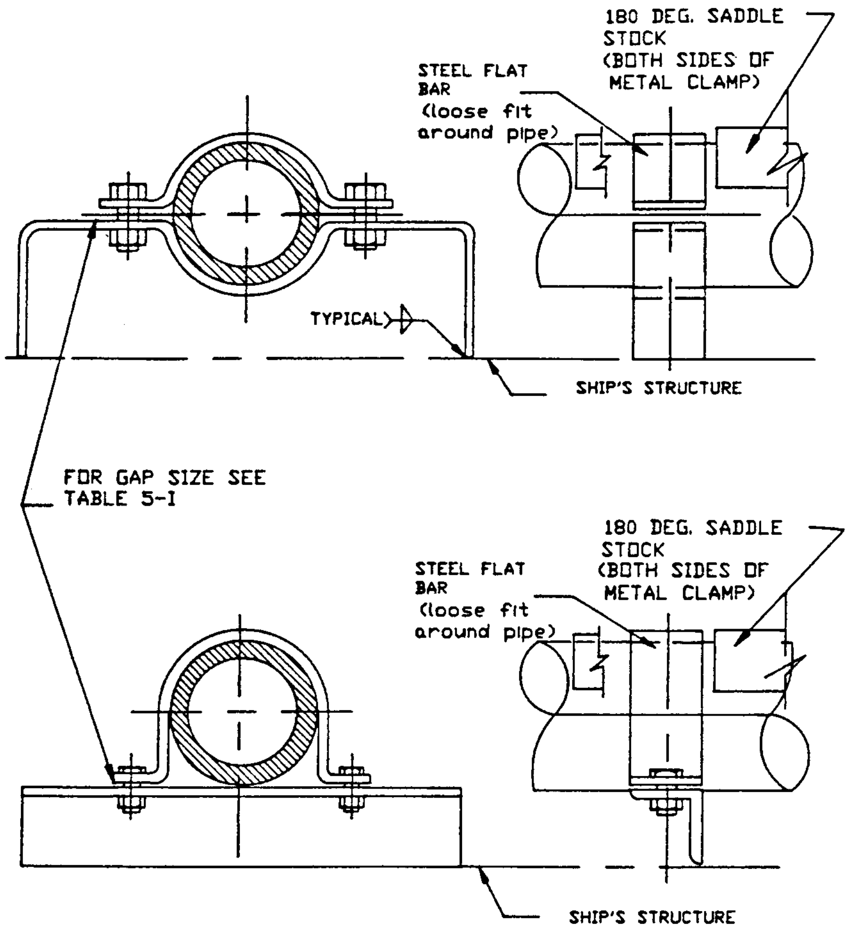
5.2.6 Anchor Supports Using Full Metal Clamp (Figure 5—6)

The flat bar clamp is designed to restrain the pipe from axial movement. Saddle stock is installed on both sides of the steel clamp. In order to hold the pipe without damage see Table 5—1 below for recommended space between the bottom part of the clamp and upper part of the clamp.

For small pipe diameters 1—6 in. (25—150mm) it is useful to use a 1/4 thick (6mm) neoprene pad (Durometer A 50—60) compressed between the pipe and metal clamp. This will not prevent movement of the pipe in the axial direction. To prevent movement, the pipe must be properly anchored with saddle supports using half or full collars depending on the thrust imposed by the hydrostatic pressure or temperature change in the piping system.

Fig. 5—6

Fig. 5—7



5.2.7 Anchor Supports Using Flat Bar Top Half and Steel Shape Bottom (Figure 5—7 Previous Page)

This type of anchor support is similar in purpose to that shown in Figure 5—6. Many shipyards prefer this type.

Caution: Dimensions of the steel clamp must provide for a loose fit around the fiberglass pipe when attached to the steel angle shape below. If the pipe is clamped against the flat steel surface on the bottom half, the force imposed at the tangential point of contact between the pipe and steel can damage the fiberglass pipe. (See Table 5—I). For diameters greater than 8 inches this problem is less severe due to increased thickness of the pipe wall. (See Chapter 4, Table 4—IV)

Note: The supports shown in Figs. 5—4, 5—5, 5—6 and 5—7 are designed to restrain axial movement of the pipe when they are fitted with 180 deg. saddles.

5.2.8 Thrust Support For 90º and 45º Elbows (Figure 5—8 on Following Page)

The thrust support plate of Figure 5—8 is used when the hydrostatic force or thrust in the piping system will exceed the shear strength of the adhesive bonded joint. It is recommended that this type of support be used in transferring the load from the joint directly into the body of the fitting. The fitting will absorb thrust imposed on the piping system. The support plate will be permanently attached to the standard foundation detail produced by the shipyard with addition of a torsional support plate bolted directly onto a flange of the elbow to prevent a torsional displacement of the fitting.

It is recommended that a .394 in. (10mm) thick neoprene pad with a Durometer A of 50-60 be installed between the thrust support plate and the outside of the elbow completely covering the inside curved surface which will contact the pipe. The neoprene pad should be fully compressed against the thrust plate. If the thrust plate support cannot be made into a smooth radius, an alternative method is to weld together straight plates (Lobster-Back configuration). In this case the neoprene pad must be sufficiently thick so that when the pad is compressed between the fitting and the Lobster-Back support, a full contact of the outside diameter of the pipe is accomplished with the compression of the neoprene pad. This assures that the forces will be transmitted directly to the steel thrust support plate and no slippage will occur by an improperly compressed neoprene pad.

Note: It is recommended that a mechanical coupling (Dresser-type only) be incorporated on either side of the fitting using thrust support plates to allow axial movement in the piping system and relieve part of the thrust imposed on the fitting. This practice has been used successfully in previous installations. See Note in Section 5.2.9.

5.2.9 Thrust Support Plate For Tees (Figure 5—9 On Page 5.8)

The thrust support plate of Figure 5—9 is used when the hydrostatic force or thrust in the piping system will exceed the shear strength of the adhesive bonded joint. It is recommended that this type of support be used in transferring the load from the joint directly into the body of the fitting. The fitting will absorb thrust imposed on the piping system. The thrust support plate for the tee is simpler in design than the previous thrust support for elbows. The construction is straight and simple without compound curvature and can be accomplished by rolling the plate to conform to the outside diameter of the tee.

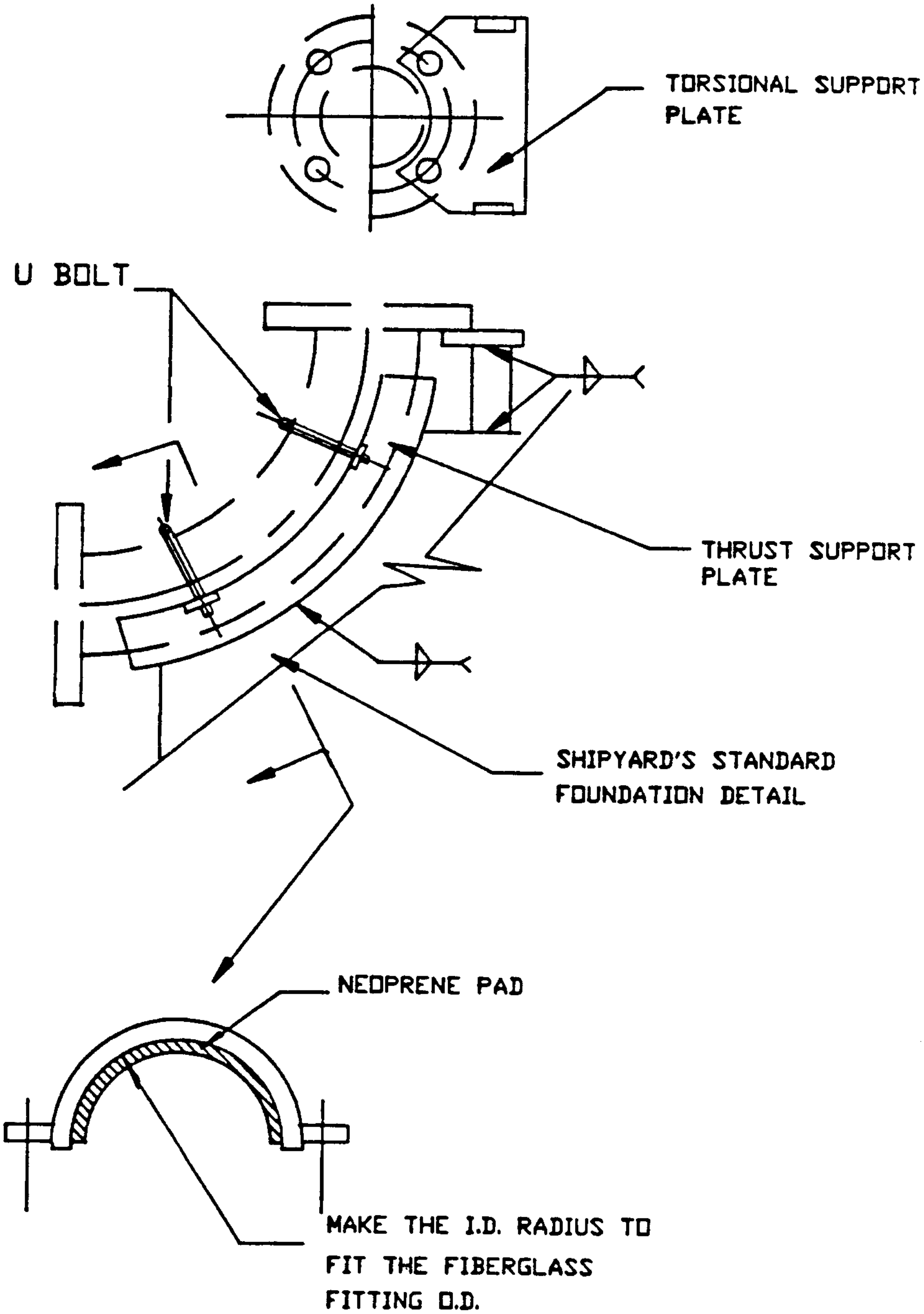
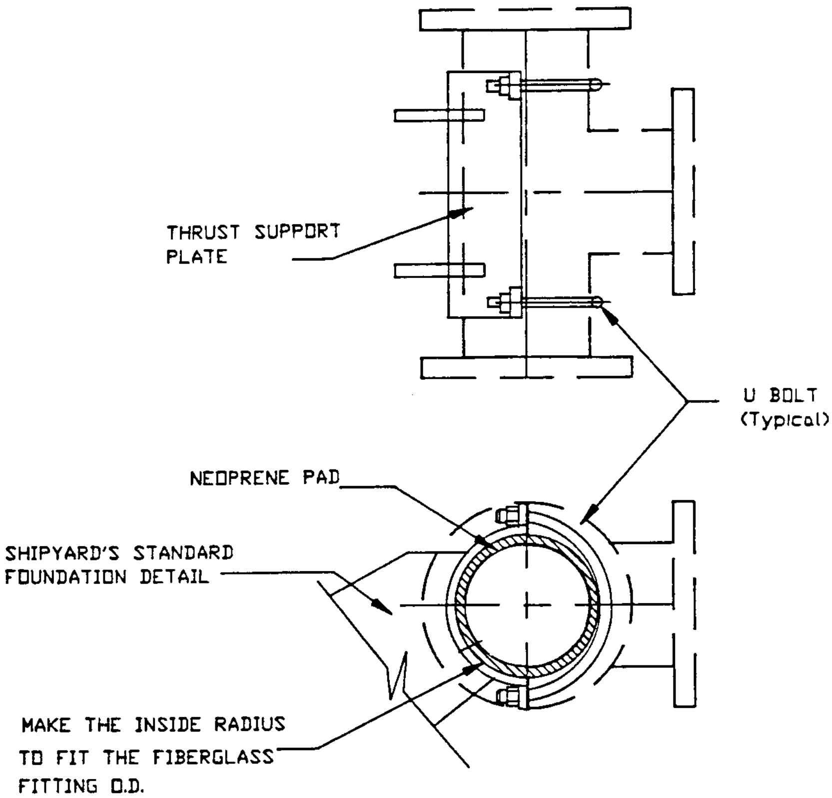

TORSIONAL SUPPORT PLATE
U BOLT
THRUST SUPPORT PLATE
SHIPYARD'S STANDARD FOUNDATION DETAIL
NEOPRENE PAD
MAKE THE I.D. RADIUS TO FIT THE FIBERGLASS FITTING O.D.

Fig. 5—8

Fig. 5—9



The accommodation of the neoprene pad will be the same as Figure 5—8 with the objective to transfer the thrust force of the piping system into the thrust support plate and not into the flange or bonded joints of the tee. Because of the geometrical configuration of the tee, a torsional plate will not be required. All the rest of the recommendations previously discussed in Figure 5—8 are also applicable to the tee support.

Note: It is advisable to coat the U bolts which hold the elbows and tees against the thrust support plates with Amercoat, urethane or similar coatings to protect against corrosion, and also cushion between the fittings and the U bolt. Another method used by some shipyards is to introduce a neoprene sleeve around the U bolts. This Note applies to all supports using U bolts.

5.2.10 Anchor Support Plate Bolted to a Flanged Fitting (Figure 5—10 On Following Page)

This anchor support is used for flange fittings when the hydrostatic forces imposed by the design of the piping system do not exceed the adhesive shear stress value. (See Section 3.6 of this manual.)

Figure 5—10 shows the plate pattern covering a minimum of four bolts (for all pipe sizes). Figure 5—10 shows a design used by shipyards to anchor large diameter elbows. See Note 3 on page 5.2.

5.2.11 Steel Supports for Large and Small Valves (Figure 5—11 On Page 5.10)

The steel supports shown in Figure 5—11 apply for various kinds of valves. Valves in sizes 4 in. and under are relatively light can normally be supported with a single support. Gate valves and similar large and heavy valves in sizes 6 in. and up require two supports to accommodate the weight and directly transmit it to the ship's structure. Valves such as globe or gate valves with reach rods extending to the above decks require double support.

See Table 5—II below for required number of bolts in support plates.

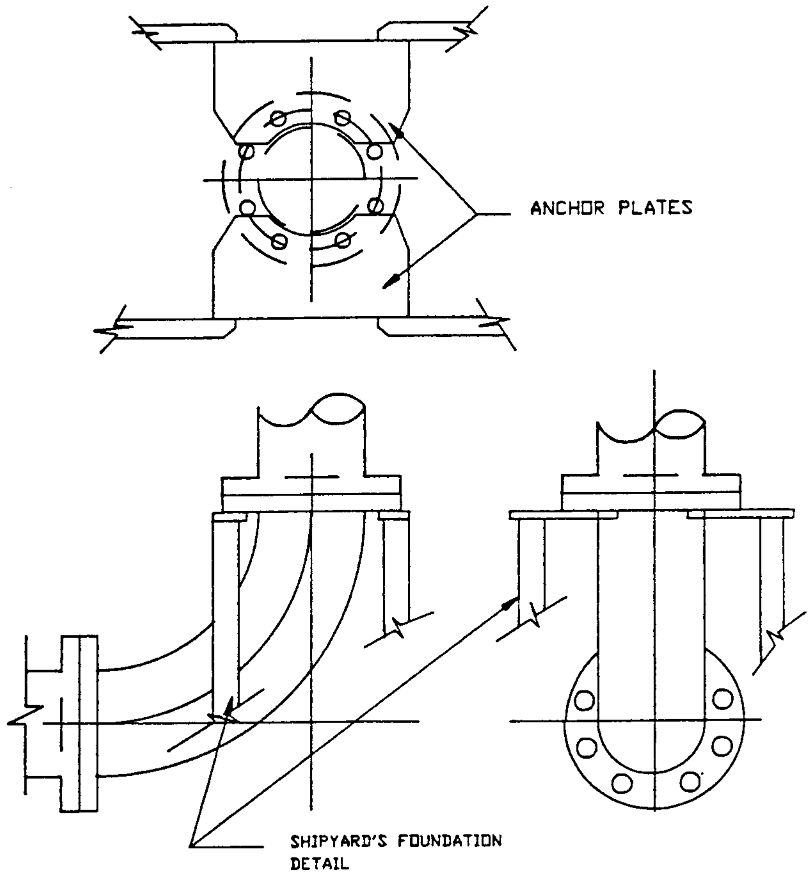


Fig. 5—10

Flanged plates must be properly designed to support the weight of valves and transmit it directly to the ship's structure. It is recommended that all steel components in a piping system be supported. This will prevent shifting the weight to the fiberglass piping system.

TABLE 5—II

Flange Size	Required Minimum Number Of Bolts Attached To Support Plate	Flange Size	Required Minimum Number Of Bolts Attached To Support Plate
1	2	20	8
1 1/2	2	22	8
2	2	24	10
3	4	26	10
4	4	28	10
6	4	30	12
8	4	32	12
10	6	34	12
12	6	36	12
14	6		
16	6		
18	8		

Note: Flanges should be two-hole oriented as a general practice in shipbuilding.

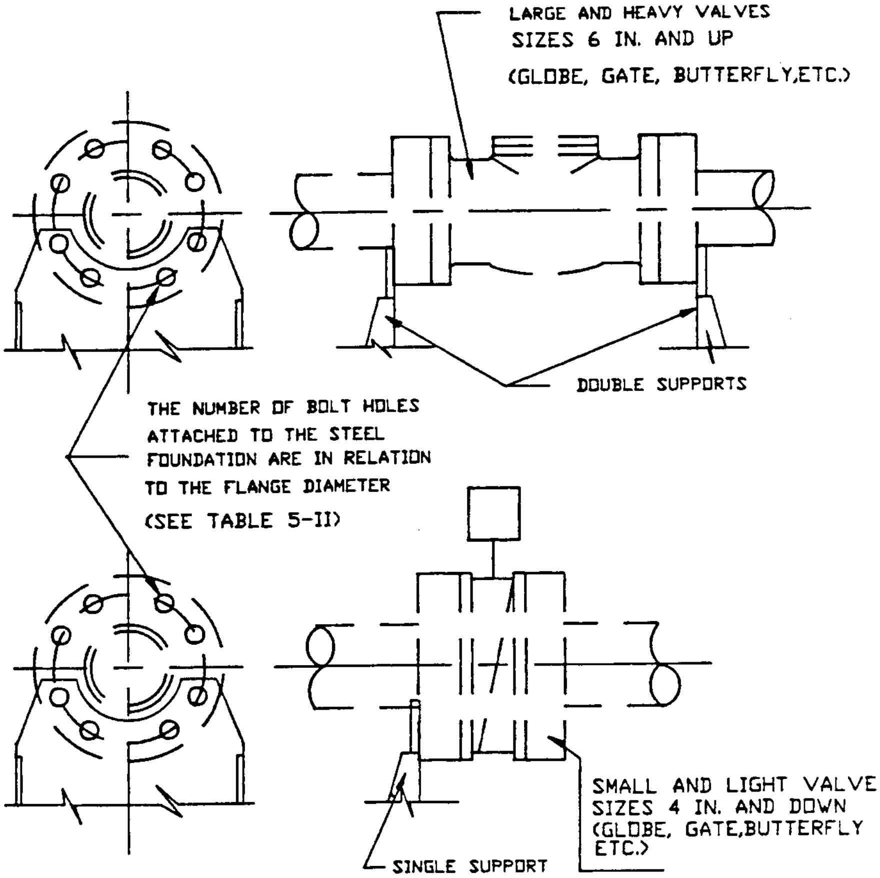


Fig. 5—11

5.2.12 Guidance Support for Fiberglass Pipe. Teflon Sliding Pad (Figure 5—12)

This simple design has been adopted almost universally for guides in ship construction. Teflon has self—lubricating properties which help to reduce friction between the surface of the pipe and the steel without inducing abrasion on the fiberglass component. Teflon also is inert to most chemicals and petroleum derivatives used in tank ships, white product, and chemical carriers. The minimum thickness of the teflon pad is recommended to be 1/5 inch (5mm). Teflon thickness should be increased proportionally to the largest size of the piping system i.e., 1/4 inch (6mm) for 20 inches and above. The teflon pad can be utilized (or installed) in different configurations, some shipyards feel that the teflon pad in conjunction with the holes for the U bolt will be sufficient. Others shipyards prefer to have an indentation on the teflon pad to prevent any sliding in the center between the two holes supporting the pad. The third anchor point will be in the center of the teflon pad and the metal bar as shown as an alternative on Figure 5—12. It is also recommended that the U bolts be coated with Amercoat, urethane or hot dip coating to prevent corrosion.

5.2.13 Maximum Flange Misalignment Allowance (Figure 5—13)

The Table in Figure 5—13 shows allowable misalignment for flanges from 1—16 inches diameter and from 18—36 inches diameter. It is recommended that these allowances not be exceeded in order to accomplish a proper seal between flanges without inducing unacceptable stresses.

Fig. 5—12

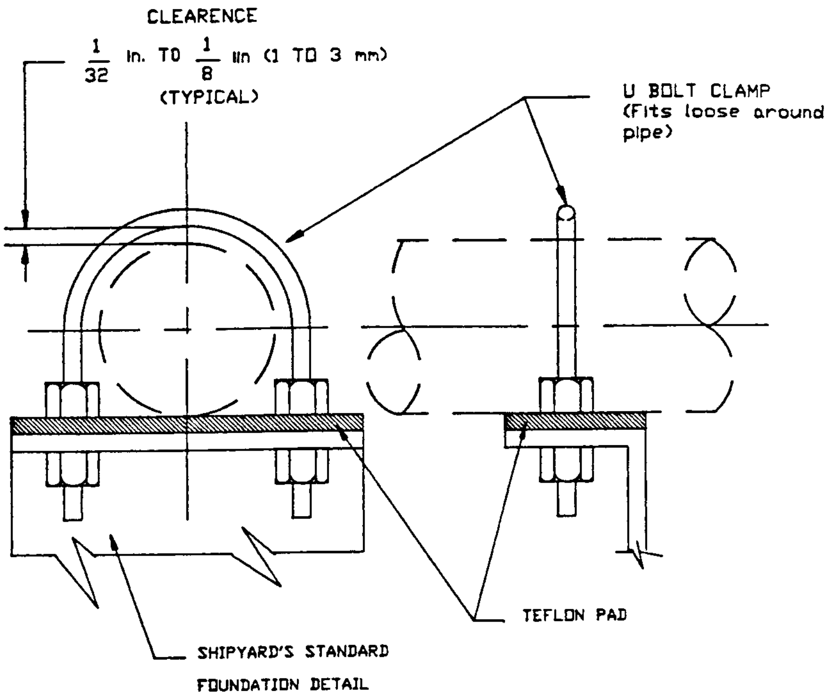
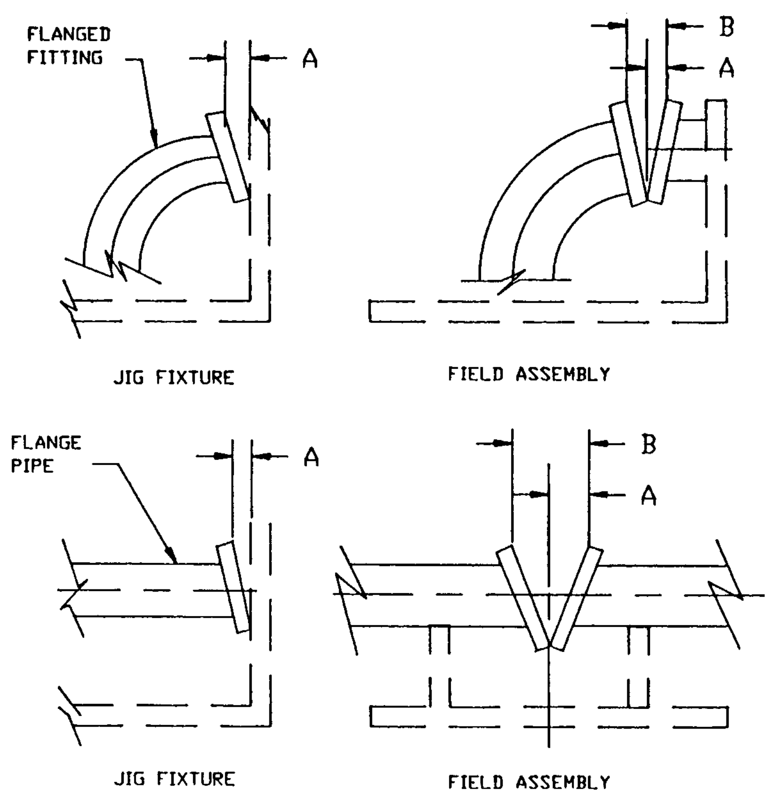


Fig. 5—13



FLANGE NPS RANGE	MAX. MISALIGNMENT ALLOWANCE	
	A in. (mm)	B in. (mm)
1 TO 16	1/16 (1.6)	1/8 (3.2)
18 TO 36	1/8 (3.2)	1/4 (6.4)

5.2.14 Pipe Misalignment Between Supports (Figure 5—14)

The Table in Figure 5—14 shows allowable misalignment for different sizes of pipe assuming 20 ft. (6m) between supports. Figure 5—14 also provides a formula to calculate the maximum misalignment between supports for other support spacings.

Note: When joints are made with mechanical couplings, see manufacturer's literature for permissible misalignment.

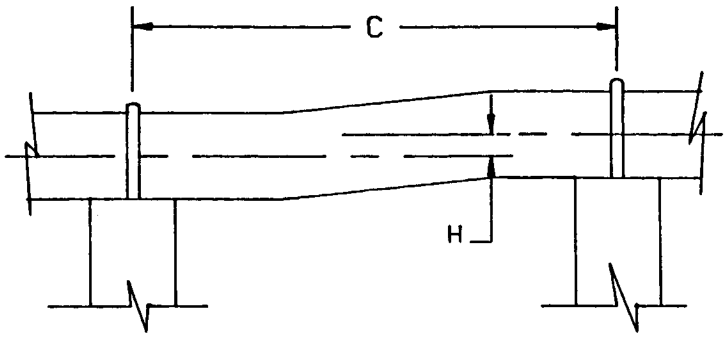


Fig. 5—14

$$H = H_{20} \times \frac{C^2}{400}$$

Where

H = Total allowable misalignment in (in.)

C = Support span in (ft.)

H_{20} = See Table

NPS RANGE	C= 20 ft. (6m) H in.	mm.
2-4	1	25.4
6-12	3/4	19.1
14 - 20	1/2	12.7
24 - 36	3/8	9.5

Notes: 1. For supports spans other than 20 feet the total misalignment can be calculated using the above formula

2. Misalignment applicable applicable to any direction parallel to axis

6.0 Internal and External Pressure Design

6.1 INTERNAL PRESSURE

$$P_i = \frac{2st}{(\overline{OD} - t)}$$

Where: P_i = rated internal pressure, psi or kg/cm2,

s = allowable hoop stress, 6000 psi. (422kg/cm2) for Series 2000M and 7000M Bondstrand pipe,

$\overline{OD}$ = minimum outside diameter (in. or cm) see Table 4—IV,

t = minimum reinforced wall thickness (in. or cm) = tt — ti,

tt = minimum total thickness (in. or cm) see Table 4—IV,

t_l = liner thickness, 0.020 in. (0.51 cm) for Series 2000M, zero for Series 7000M.

$(\overline{OD} - t)$ = $\overline{ID} + t + 2t_l$

$\overline{ID}$ = inside diameter (in. or cm).

To convert pressure in psi to bars, divide by 14.5. To convert pressure in kg/cm^2 to bars, divide by 1.02.

Based on the formula given above, the rated operating pressure for Series 2000M and Series 7000M pipe is tabulated in Table 6—I. This provides long—term performance in accordance with the cyclic Hydrostatic Design Basis (ASTM D2992, Method A) and provides a 4 to 1 safety factor on short—term hydrostatic performance as required by proposed ASTM Marine Piping Specifications.

Note: Fittings and/or mechanical couplings may reduce the system working pressure below that shown in Table 6—I. See Bondstrand Product Data Sheets FP168 and FP169 and coupling manufacturer's literature.

TABLE 6—I

Rated Internal Operating Pressure for Series 2000M and Series 7000M Pipe

Nominal Diameter		Rated Internal Operating Pressure at 2000F (930C)	
in.	mm	psi	bar
2	50	550	38
3	80	450	31
4	100	450	31
6	150	300	21
8	200	300	21
10	250	300	21
12	300	300	21
14	350	300	21
16	400	300	21
18	450	300	21
20	500	300	21
24	600	300	21
28	700	300	21
30	750	300	21
36	900	300	21

Note: Fittings and flanges have a lower pressure rating than the pipe.

6.2 EXTERNAL COLLAPSE PRESSURE.

$$P_c = \frac{2E_c\, t_a^{\,3}}{(1-\nu_c\nu_l)\, \overline{ID}^3}$$

Where P_c = external collapse pressure (psi or kg/cm²),

E_c = effective circumferential modulus of elasticity (psi or kg/cm²), see Table 6—II,

t_a = average reinforced wall thickness (in. or cm), .875 is used because the minimum thickness is 87.5% of nominal.

= $(tt / .875) — t_l$

t_t = minimum total thickness (in. or cm) see Table 4—IV,

t_l = liner thickness, 0.020 in. (0.51 cm) for Series 2000M, zero for Series 7000M,

$\overline{ID}$ = pipe inside diameter (in. or cm), see Table 4—IV,

ν_l = Poisson's ratio for contraction in the circumferential direction due to tensile stress in the longitudinal direction, see Table 6—II,

ν_c = Poisson's ratio for contraction in the longitudinal direction due to the tensile stress in the circumferential direction, see Table 6—II.

To convert external pressure in psi to bars, divide by 14.5. Atmospheric pressure at sea level is 14.7 psi. To convert kg/cm^2 to bars, divide by 1.02.

When installing pipe in the bottom of tanks, the pipe must resist the combined external fluid pressure and internal suction. It is assumed that a positive displacement pump can pull a maximum of 75 percent vacuum. The designer should also allow for a safety factor of 3 in accordance with proposed ASTM Specifications. Thus the allowable hydrostatic head, H in ft. is:

$$H = 2.31 \left[\frac{P_c}{3.0} — 11.0 \right]$$

Tabulated values of allowable hydrostatic head are shown in Table 6—III on page 6.6 for temperatures of 1000F(380C) and 2000F(930C). For example, calculate the collapse pressure and allowable hydrostatic head in English units for 12 inch Series 2000M pipe at 2000F:

$$\underline{ID} = 12.35 \text{ inch}$$
$$t_t = 0.351 \text{ inch}$$

$$t_l = 0.020 \text{ inch}$$

$$t_a = (.351/.875) — .020 = .381 \text{ inch}$$

$$P_c = \frac{2(2.20 \times 10^6).381^3}{[\,1 - .7\,(.41)]\ 12.35^3} = 181 \text{ psi}$$

$$H = 2.31 \left[\frac{181}{3.0} — 11.0 \right] = 114 \text{ ft.}$$

Or read the appropriate values from Table 6—III.

Table 6—II

Elastic Properties for Calculation of External Collapse Pressure for Series 2000M and 7000M Pipe

Temperature		E_c		μ_c	μ_l
ºF	ºC	psi	kg/cm^2		
70	21	3.15 x 106	2.21 x 10^5	0.56	0.37
100	38	3.06 x 106	2.15 x 10^5	0.57	0.38
150	66	2.90 x 106	2.04 x 10^5	0.60	0.39
200	93	2.20 x i06	1.55 x 10^5	0.70	0.41

Note: Ec is based on external collapse tests per ASTM D2924. Values of Poisson's ratio are based on tests per ASTM D1599

TABLE 6—III

External Collapse Pressure and Allowable Hydrostatlc Head
for Series 2000M and Series 7000M Pipe

Nom. Pipe Size		1000F(380C)				2000F(930c)			
		Collapse Pressure		Allowable Hydrostatic Head		Collapse Pressure		Allowable Hydrostatlc Head	
(in)	(mm)	(psi)	(Bars)	(ft)	(in)	(psi)	(Bars)	(ft)	(in)
2	50	2,331	160	1,770	540	1,855	565	1,403	427
3	80	637	43.9	465	142	507	35.0	365	111
4	100	703	48.5	516	157	559	38.6	405	123
6	150	234	16.1	155	47	186	12.8	118	36
8	200	231	15.9	153	47	184	12.7	116	35
10	250	231	15.9	153	47	184	12.7	116	35
12	300	228	15.7	150	46	181	12.5	114	35
14	350	228	15.7	150	46	181	12.5	114	35
16	400	228	15.7	150	46	181	12.5	114	35
18	450	227	15.6	149	45	181	12.5	114	35
20	500	227	15.6	149	45	181	12.5	114	35
24	600	226	15.5	149	45	180	12.4	114	35
28	700	226	15.5	149	45	180	12.4	114	35
30	750	226	15.5	149	45	180	12.4	114	35
36	900	225	15.5	148	45	179	12.3	112	34

7.0 Hydraulics

7.1 INTRODUCTION

When comparing Fiberglass and carbon steel piping systems it becomes evident that selection of Fiberglass pipe can result in significant savings due to favorable hydraulic properties.

7.2 HEAD LOSS

The frictional head loss in a pipe is a function of velocity, density, and viscosity of the fluid; and of the smoothness of the bore, and the length and diameter of the pipe. Therefore, the best means of minimizing this pressure drop in a particular piping service is to minimize the internal roughness of the pipe. This internal roughness causes movement of the fluid particles in the boundary layer adjacent to the pipe wall, which causes flow through the pipe to be impeded.

Fiberglass pipe has a smoother inner surface than new steel piping. There is an even more significant difference between the inner surface of Fiberglass and steel pipe after the pipes have been in service for a while. In most systems Fiberglass maintains its low head loss performance for life.

Fiberglass does not scale, rust, pit or corrode electrolytically or galvanically. It resists growth of bacterial algae, and fungi that could build up on the inner surface. Also, Fiberglass has high chemical and abrasion resistance. In marine applications, where pipelines are usually short, the major portion of the total pressure drop in a system occurs in the valves and fittings. It is customary to express the resistance of valves and fittings in terms of equivalent length of pipe, these are added to the actual length for purposes of pressure drop calculation for the total system.

7.3 FORMULAS FOR CALCULATING HEAD LOSS IN PIPE

The Hazen-Williams equation is convenient for calculating head loss. For full flow, this equation, with a C factor of 150, predicts head loss with sufficient accuracy for nearly all water piping situations.

Fluids other than water require a more universal solution such as given by the Darcy-Weisbach equation. This section gives the information needed to solve these head loss problems for fluids such as crude oil and salt brine. Head loss for two-phase fluids such as sludges and slurries is not covered.

7.3.1 Hazen—Williams Equation (For Water Pipe, Full Flow)

An equation commonly used for calculating head loss in water piping is that published by Hazen and Williams. Solving for head loss, this equation becomes

$$H_L = 1046 \left[\frac{Q}{C\,\overline{ID}^{2.63}} \right]^{1.852}$$

Where HL = head loss (feet per 100 feet of pipe),

Q = discharge (gallons per minute), (U.S. gallon)

C = Hazen-Williams Factor (C = 150 for Bondstrand), and

$\overline{ID}$ = inside diameter of pipe (inches).

In International System (SI) units, this equation is

$$H_L = 1068 \left[\frac{Q}{C\ \overline{ID}^{2.63}} \right]^{1.852}$$

where H_L = head loss (meters per 100 meters of pipe),

Q = discharge (cubic meters per second),

C = Hazen—Williams factor (C = 150 for Bondstrand), and

$\overline{ID}$ = inside diameter of pipe (meters).

7.3.2 Darcy-Weisbach Equation (For All Fluids, Full Flow)

The solution of the Darcy-Weisbach equation is complicated by the fact that the Darcy friction factor, f, is itself a variable. Solutions for f may be obtained using handbooks, or by using a programmable calculator, for both laminar and turbulent flow conditions.

Figure 7-1 gives the head loss versus discharge for water flowing in Bondstrand pipe based on the Darcy-Weisbach equation

$$H_L = f \left[\frac{L}{\overline{ID}} \quad \frac{V^2}{2g} \right]$$

Where HL = frictional resistance (meters),

f = Darcy friction factor,

L = length of pipe run (meters),

$\overline{ID}$ = internal diameter of pipe (meters),

V = average velocity of fluid (meters per second), and

g = gravitational constant = 9.806 meters per second^2.

The frictional resistance is obtained in feet by the same equation if all units of length are changed to feet and the gravitational constant is changed to 32.2 feet per second^2. When using Figure 7-1, convert discharge in gal/mm to cu in/sec by multiplying by 0.0000631.

The variable Darcy friction factor can be determined for any fluid in the turbulent range of flows by use of the Moody equations.

$$f = 0.0055 \left[1 + \left[20{,}000 \frac{\epsilon}{\overline{ID}} + \frac{10^6}{R} \right]^{1/3} \right]$$

in which ϵ = pipe roughness (meters),

$$R = \frac{\mu\ \overline{ID}}{\mu} = \text{Reynold's Number,}$$

Where μ = kinematic viscosity of the fluid (square meters per second).

If the Reynold's Number falls below 2000, the flow can be assumed to be laminar. Then the Darcy friction factor becomes

$$f = \frac{64}{R}$$

Roughness Parameter — ϵ

The smoothness of the inside pipe surface over the life of Bondstrand pipe produces lower frictional head loss compared to most other piping materials. The lower head loss means lower pressures will be required to produce an equivalent discharge, thereby also conserving pumping energy.

Tests of Bondstrand pipe show that the roughness is 5.3 x 10^6 meters (1.7 x 10^6 feet). There is a high probability that this low level roughness will be sustained, and will not be increased due to corrosion and incrustation as often the case with steel piping, which may double in roughness under certain conditions.

Kinematic Viscosity of Fluid — μ

Increase in fluid viscosity leads to increased head loss. Table 7—I illustrates the effect of kinematic viscosity on head loss for several common fluids. Kinematic viscosity is defined as the absolute viscosity divided by the density. It varies with temperature. The kinematic viscosity for water at room temperature is 0.000001115 square meters per sec (0.000012 sq. ft per sec)

Figure 7-2 shows how head loss and flow are affected by kinematic viscosity. The transition between laminar flow and turbulent flow in 6-in. pipe is seen in the plot for a fluid having a kinematic viscosity of 0.001 square feet per second.

7.4 HEAD LOSS IN FITTINGS

Head loss for water flow in fittings 2 through 36 in. in diameter may be determined by the above methods after obtaining their equivalent pipe lengths using Figure 7-3. For example, find the equivalent pipe length (L_e) for water flowing through a 6-in. diameter elbow at a rate of 0.003 meters^3 per second. Beginning at the bottom of the chart given in Figure 7-3 at a flow of 0.003 meters^3 per second, proceed vertically to intersect the 6-in. diameter curve, and read L_e = 6 meters on the left ordinate. Multiply this value by the resistance coefficient, K, given for 90 degree elbows in Table 7-II to obtain equivalent pipe length,

$$L_e = 6 \times 0.5 = 3 \text{ meters.}$$

Head loss in the fitting is then determined as the head loss in this equivalent length of pipe. The resistance coefficients from Table 7-III may be used in similar fashion for reducers.

Although the Darcy friction factor, f, for water was used in the development of Figure 7-3, the equivalent pipe length obtained may then be used to estimate head loss for the actual fluid in the system.

With a known Darcy friction factor, the equivalent length of pipe for any size and type of fitting can be determined using the appropriate resistance coefficient, K, from Table 7-II and the equation

$$L_e = K\,\overline{ID}/f$$

provided L_e and $\overline{ID}$ are given in the same units.

Figure 7—1

Head Loss For Water as a Function of Flow Rate

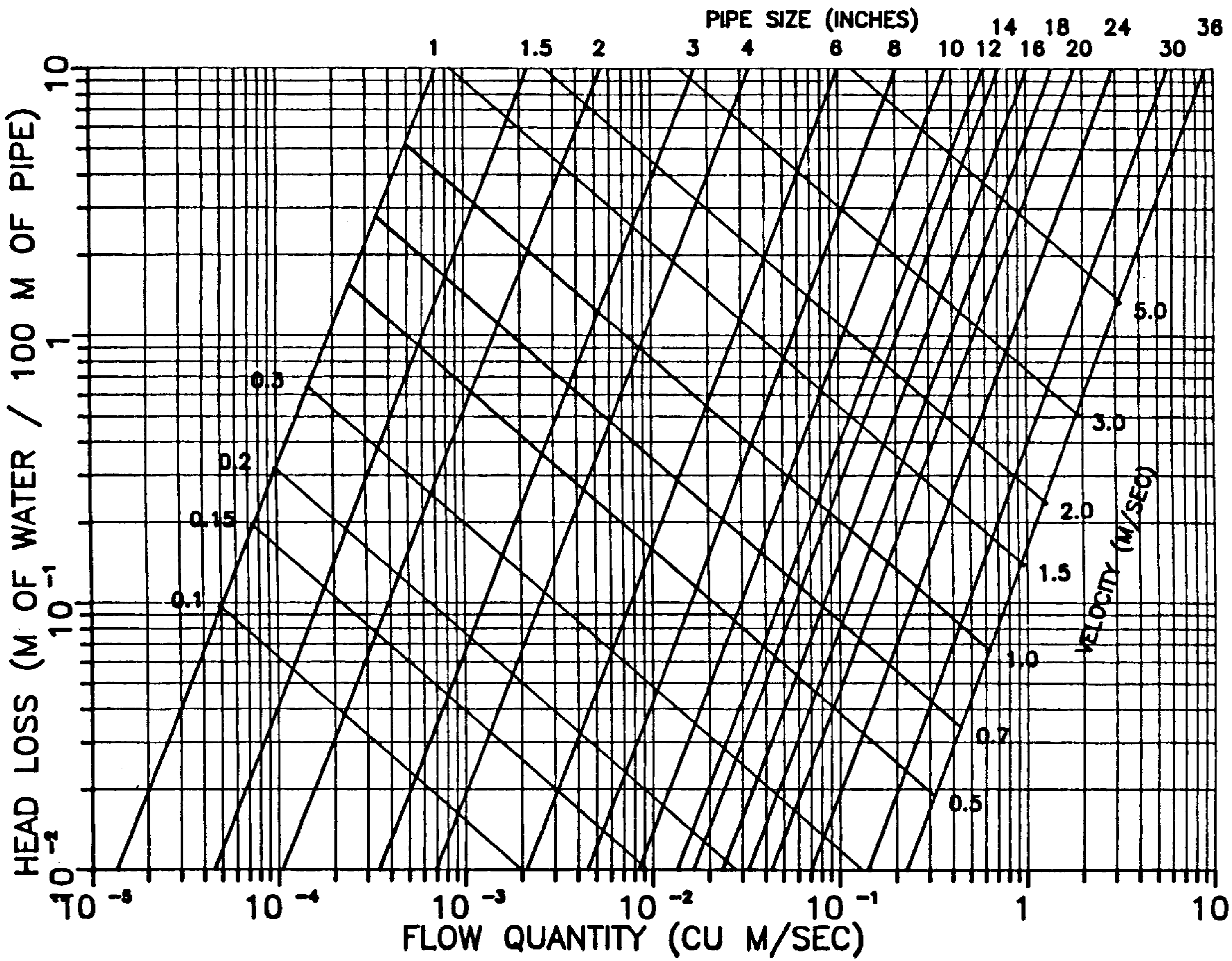


Figure 7—2

Effect of Kinematic Viscosity on Head Loss vs. Discharge for 6-inch Pipe Flowing Full

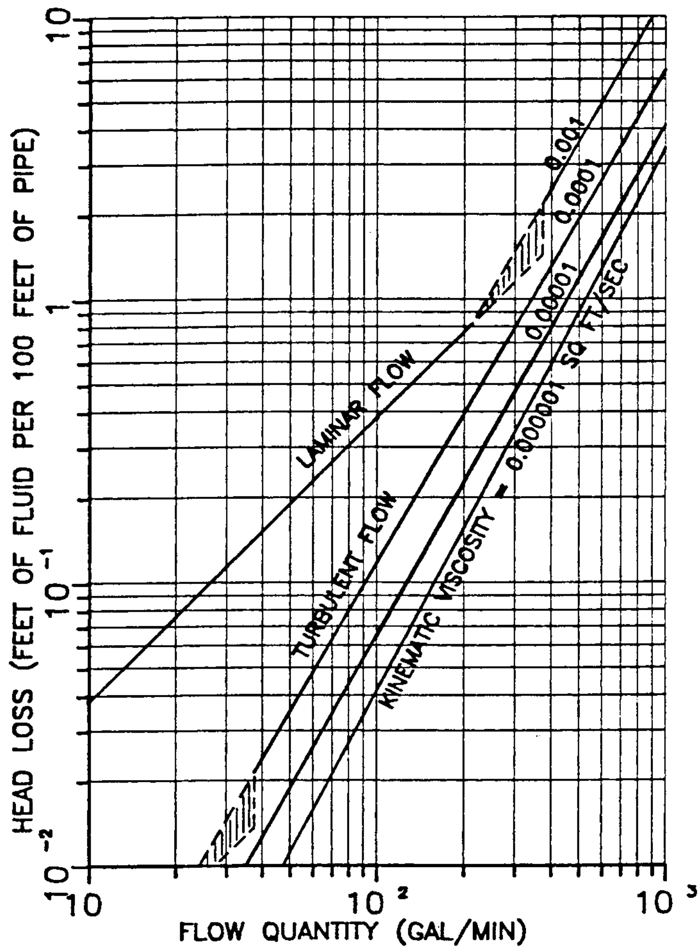


Table 7-I

Head Loss for Various Flowing at 500 GPM in a 6-Inch Bondstrand Marine Pipe

	Line Temperature F°	Specific Gravity	Kinematic Viscosity ft^2/sec	Head Loss/100 Ft of Pipe: Ft of Fluid	Head Loss/100 Ft of Pipe: Pounds/ sq. in.
Aviation Gasoline, 70° API	60	0.70	.0000072	1.17	0.36
Gasoline, 60° API	60	0.74	.0000086	1.21	0.39
JP-4 Fuel, 45° API	60	0.80	.000013	1.30	0.45
JP-7 Fuel, 45° API	60	0.80	.000028	1.51	0.52
Kerosene, 42° Baumé	60	0.81	.000027	1.49	0.53
Water	60	1.0	.000012	1.28	0.55
Water	140	0.98	.0000051	1.11	0.47
Brine, 20% NaCL	60	1.14	.000016	1.35	0.67
Hydrochloric Acid, 31.5%	86	1.15	.000016	1.35	0.67
Sulfuric Acid, 60%	60	1.50	.000050	1.70	1.10
Fuel Oil, No. 5, 30° Baumé	140	0.875	.00051	3.0	1.15
Crude Oil, 24.4° Baumé	140	0.91	.00012	2.07	0.81
Crude Oil, 15.2° Baumé	140	0.96	.00113	3.84	1.60

Figure 7-3
Equivalent Pipe Length of Fittings

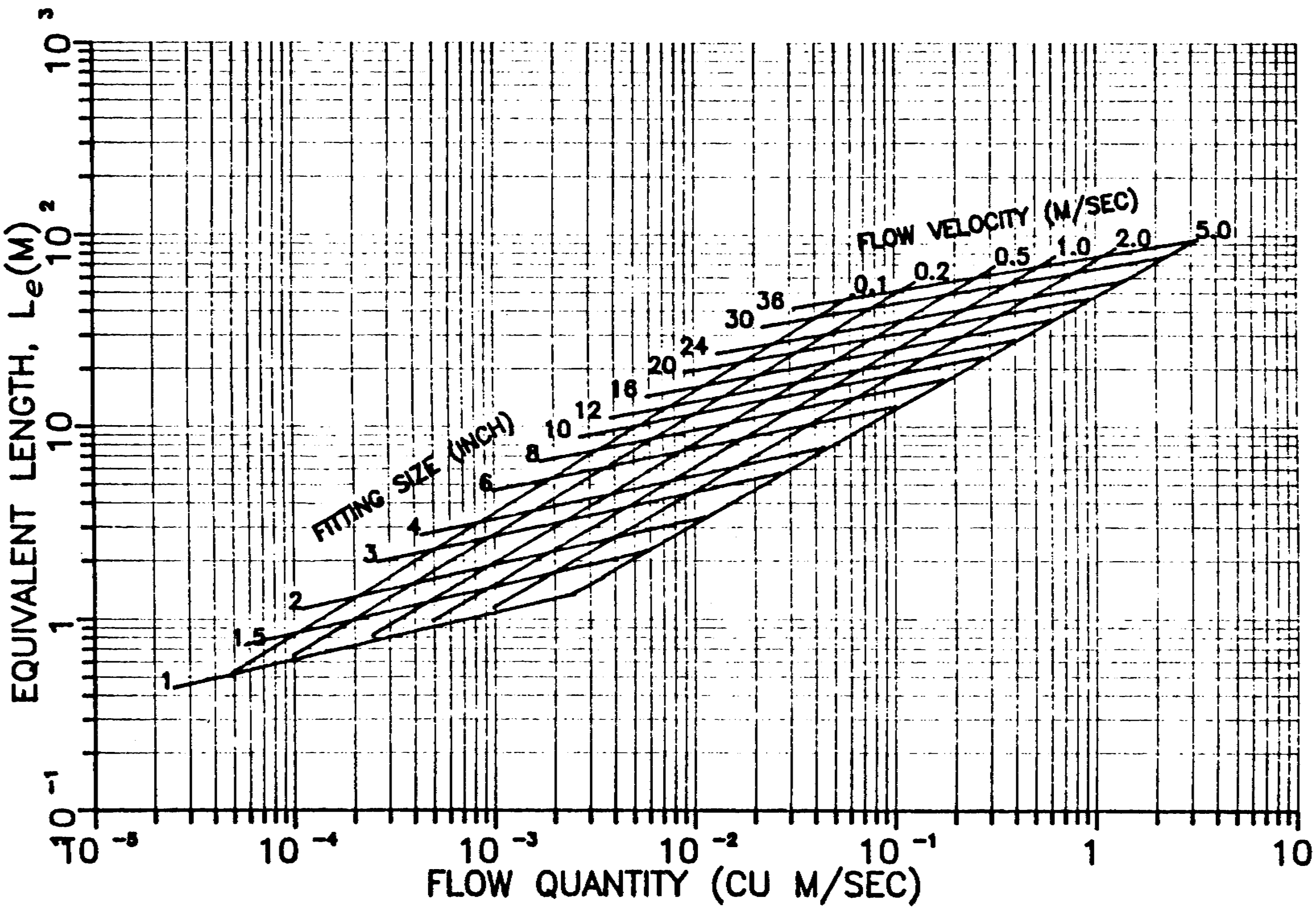


TABLE 7-II

Resistance Coefficients for Bondstrand Fittings and Metal Valves

Description	K
45º Elbow Standard	0.3
45º Elbow Single Miter	0.5
90º Elbow Standard	0.5
90º Elbow Single Miter	1.4
90º Elbow Double Miter	0.8
90º Elbow Triple Miter	0.6
180º Return Bend	1.3
Tees >T	0.4
>T	1.4
>T	1.7
Gate Valve Open	0.17
3/4 Open	0.9
1/2 Open	4.5
1/4 Open	24.0
Diaphragm Valve Open	2.3
3/4 Open	2.6
1/2 Open	4.3
1/4 Open	21.0
Globe Valve Bevelseal, Open	6.0
1/2 Open	9.5
Check Valve Swing	2.0
Disk	10.0
Ball	70.0

Note: Coefficients are for fittings with no net change in velocity.

TABLE 7-III
Resistance Coefficients for Bondstrand Reducers, Tapered Body

SIZE	K	SIZE	K
$1^{1/2}$ X 1	0.5	12 X 8	0.8
2 X 1	2.8	12 X 10	0.1
2 X $1^{1/2}$	0.3	14 X 10	0.12
3 X $1^{1/2}$	3.7	14 X 12	0.01
3 X 2	0.7	16 X 12	0.08
4 X 2	2.9	16 X 14	0.03
4 X 3	0.1	18 X 14	0.16
6 X 3	3.1	18 X 16	0.02
6 X 4	0.7	20 X 16	0.13
8 X 4	3.3	20 X 18	0.02
8 X 6	0.1	24 X 18	0.17
10 X 6	1.5	24 X 20	0.07
10 X 8	0.2	30 X 24	0.22

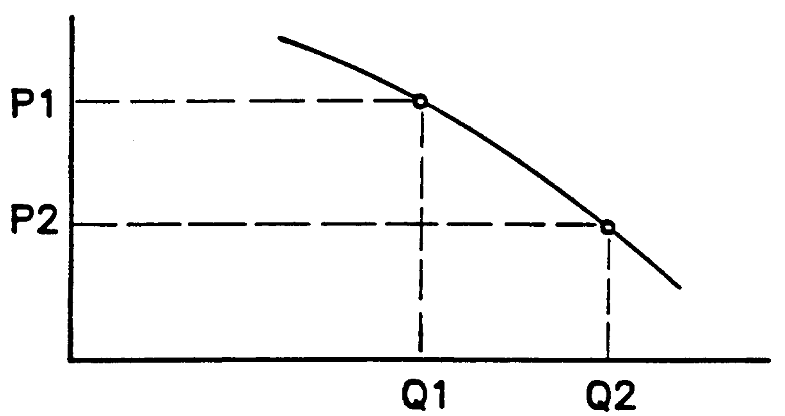
7.5 CARGO DISCHARGE TIME AND ENERGY SAVINGS

The advantage of low friction loss in Fiberglass smooth bore pipe has been explained in EB-19, “HEAD LOSS IN BONDSTRAND VERSUS STEEL.” This section will focus on another aspect of this topic, namely energy savings in cargo tank discharge, and how loading and unloading time can be reduced by using Bondstrand piping products.

7.5.1 Pump Flow Rate

Consider a typical pump operating at a certain pressure P_1 to overcome friction loss in the piping system as shown in Figure 7-4. At this pressure the pump will discharge a certain flow rate Q_1. This same pump will discharge a higher flow rate Q_2 if somehow the friction loss in the pipeline can be reduced, bringing the pump’s operating head down to a lower level, P_2. The increase in volume flow rate, as a result of the reduction in operating pressure, depends largely on the pump performance characteristics which vary from pump to pump. This flow variation with pressure can be found in the pump manufacturer’s literature, thus it is omitted from further discussion here.

Fig. 7-4



Pumping Pressure vs. Discharge

7.5.2 Full—Pipe Flow Of Water In Low—Friction Fiberglass Pipe

Let's now focus our discussion only to the pipeline and examine how low friction pipe can improve the volume flow rate of the system.

For example consider two pipelines - Schedule 40 steel and Bondstrand Series 2000M pipe - both designed to transport water 100 meters. We will compare the volume flow rate. The friction head loss in the pipelines can be calculated by the Hazen-Williams formula as stated before. In metric units:

$$H_L = 1068 \left[\frac{Q}{C\ \overline{ID}^{2.63}} \right]^{1.852}$$

Where H_L = head loss (meters per 100 meters of pipe)

Q = discharge (cubic meters per second),

C = Hazen-Williams Factor (C = 150 for Bondstrand), and

$\overline{ID}$ = inside diameter of pipe (meters).

With the same energy consumption rate to overcome the friction loss in the pipeline, the rate of discharge will be different due to the differences in friction coefficient in the pipe. In other words, using the same head loss for both pipe, we obtain:

$$H_L = 1068 \left[\frac{Q_{steel}}{C_{steel}\ \overline{ID}_{steel}^{2.63}} \right]^{1.852} = 1068 \left[\frac{Q_{BS}}{C_{BS}\ \overline{ID}_{BS}^{2.63}} \right]^{1.852}$$

Rearrange the above expression to show the flow rate in Bondstrand pipe in terms of flow rate in steel pipe:

$$Q_{BS} = Q_{steel} \left[\frac{C_{BS}}{C_{steel}} \right] \left[\frac{\overline{ID}_{BS}}{\overline{ID}_{steel}} \right]^{2.63}$$

Examining the above formula, we can conclude that for the same head loss, Fiberglass pipe will deliver more volume flow rate that that of the same nominal diameter steel pipe since the product of $\frac{C_{BS}}{C_{steel}}$ and $\frac{\overline{ID}_{BS}}{\overline{ID}_{steel}}$ is always greater than 1.0.

Table 7-IV lists the calculated value of the flow ratio Q_{BS} / Q_{steel} where C_{BS} = 150 and C_{steel} = 120 or 70. A "C" value of 120 represents a very slightly corroded steel pipe. A "C" value of 70 represents a severely corroded steel pipe.

Table 7-IV
Flow in Bondstrand and Steel Pipe for Same Head Loss

NPS (in)	NPS (mm)	Bondstrand Pipe ID (inches)	Steel Pipe ID (inches)	C=120 QBS/QSteel	C=70 QBS/QSteel
2	50	2.095	2.067	1.30	2.22
3	80	3.225	3.068	1.43	2.45
4	100	4.140	4.026	1.35	2.31
6	150	6.265	6.065	1.36	2.33
8	200	8.225	7.981	1.35	2.31
10	250	10.350	10.020	1.36	2.33
12	300	12.350	12.000	1.35	2.31
14	350	13.290	13.25	1.26	2.16
16	400	15.190	15.25	1.24	2.13
18	450	17.080	17.25	1.22	2.09
20	500	18.980	19.25	1.20	2.06
24	600	22.780	23.25	1.18	2.02

7.5.3 Flow Of Fluids Other Than Water

In Marine applications, however, most cargo tankers carry fluids other than water. In such cases, calculations of head loss are slightly more complicated because direct comparison of volume flow rates between the two pipes is not possible. Comparison of volume flow rate can only be done in steps as illustrated below:

Step 1:

The head loss of one pipeline, usually the steel line, is chosen as a standard for comparison. This is determined using the Darcy-Weisbach method as discussed before.

$$H_L = f \frac{L}{\overline{ID}} \frac{V^2}{2g}$$

Where HL = frictional resistance (meters),

f = Darcy friction factor,

L = length of pipe run (meters),

$\overline{ID}$ = internal diameter of pipe (meters),

V = average velocity of fluid (meters per second),

g = gravitational constant = 9.806 meters per second^2.

The variable Darcy friction factor can be determined for any fluid in the turbulent range by use of the Moody equation,

$$f = 0.0055 \left[1 + \left[20{,}000 \frac{\epsilon}{\overline{ID}} + \frac{10^6}{R} \right]^{1/3} \right]$$

in which ϵ = pipe roughness (meters), and

$$R = \frac{V\,\overline{ID}}{\mu} = \text{Reynold's Number,}$$

where μ = kinematic viscosity of the fluid (square meters per second).

Step 2:

From the head loss calculated in Step 1 above, the flow velocity (the only unknown quantity in the equation for Bondstrand system) can be found by trial and error. A programmable calculator will speed this calculation considerably. Subsequently, the volume flow rate can be easily determined.

For example, 1000 cubic meters of 1400F, 24.4 degree Baum~ crude oil with kinematic viscosity of 0.00001115 square meters per second is to be unloaded through a 1000-meter long standard Schedule 40, 8-in. diameter steel pipeline at a rate of 500 cubic meters per hour. How much time can be saved unloading the same amount of crude through Bondstrand Series 2000M, 8-in. pipeline?

	Steel Pipe	Bondstrand Pipe
Data Given	Schedule 40	Series 2000M
Inside Diameter (in)	0.2027	0.2089
Roughness (in)	0.0000457	0.0000053
Flow Velocity (m/sec)	4.30	To Be Found
Reynold's Number	78200	To Be Found

Step 1:

The total head loss is calculated for the steel pipeline.

$$H_L = .0055 \left[1 + \left(20000 \frac{0.0000457}{0.2027} + \frac{1000000}{78200} \right)^{1/3} \right] \frac{1000\ (4.30)^2}{.2027\ (2)\ 9.806}$$

H_L = 94 meters

Step 2:

With 94 meters of friction head loss, the flow velocity for Bondstrand piping system can be found from the equation.

$$94 = .0055 \left[1 + \left(20000 \frac{0.0000053}{0.2089} + \frac{1000000}{V} + \frac{0.0000115}{0.2089} \right)^{1/3} \right] \frac{1000\ V^2}{.2089\ (2)\ 9.806}$$

By trial and error V = 4.55 meters per second, and R = 85,250.

As illustrated in the above example, for the given conditions, Bondstrand Series 2000M 8-in. pipe will deliver 560 cubic meters per hour, emptying the tank in less than 1.8 hours, a 10% saving in both unloading time and energy.

It is important to note here that the roughness value of new steel was used. The difference in volume flow rate would have even been higher had the roughness value of old steel pipe been used in the calculation.

7.5.4 Energy Savings Using Bondstrand Fiberglass vs. Steel Piping

Users of piping products have long known that Fiberglass piping has far lower friction factors than carbon steel piping. It is equally important to recognize the energy cost savings which accrue over the life of the installed system as a result of the lower friction factors.

The largest savings is found simply in lower pumping costs, where the power consumption can often be cut in half. For example, let us assume a 6-in. line is to deliver 500 gallons per minute of water on a year-round basis and determine energy cost per 100 feet. At this flow the average velocity is about 5 feet per second. Over a 10-year service life, a Bondstrand line can be expected to maintain a Hazen-Williams "C" factor of 150, whereas for carbon steel the average "C" factor can be estimated to be about 110. In English units:

$$H_L = 1046 \left[\frac{Q}{C\ \overline{ID}^{2.63}} \right]^{1.852}$$

Where H_L = head loss (ft. per 100 ft. of pipe), Q = discharge (gpm),

$\overline{ID}$ = internal diameter of pipe (inches), and

C = Hazen-Williams frictional factor depending on smoothness of pipe bore.

For a 100 foot run in the example described above, this formula yields 1.28 feet for Bondstrand and 2.65 feet for schedule 40 carbon steel pipe. To overcome this head loss, the horsepower demand may be calculated as

For Bondstrand:

$$\frac{500 \text{ gpm x } 8.34 \text{ lb of water/gal x } 1.28 \text{ ft}}{33{,}000 \text{ ft-lb/mm/hp}} = .162 \text{ hp}$$

For Steel:

$$\frac{500 \text{ gpm x } 8.34 \text{ lb of water/gal x } 2.65 \text{ ft}}{33{,}000 \text{ ft-lb/mm/hp}} = .335 \text{ hp}$$

Then, the energy required for full-time operation for a one month period is:

For Bondstrand:

$$\frac{.162 \times 24\ \text{hr/day} \times 30\ \text{day/month}}{.80\ \text{efficiency}} = 146\ \text{hp-hr/month}$$

For Steel:

$$\frac{.335 \times 24\ \text{hr/day} \times 30\ \text{day/month}}{.80\ \text{efficiency}} = 301\ \text{hp-hr/month}$$

It is impossible to make a generalization on the cost of electricity on board ship which is dependent on the efficiency of the ship's plant; however, if we assume that the ship is connected to shore power, we could expect to pay approximately 10 cents per kilowatt-hour or 7.5 cents per horsepower-hour. This cost is significantly lower than ship-based generation. The cost per month is then

For Bondstrand:

146 hp-hr/month x U.S. $.075/hp-hr = U.S. $10.95/month/100 ft. of pipe

For Steel:

301 hp-hr/month x U.S. $.075/hp-hr = U.S. $22.58/month/100 ft. of pipe

Difference = U.S. $11.63

For a ship using 500 feet of Bondstrand fiberglass pipe the annual savings could be:

U.S.S11.63/month/100 ft. x 12 months x 500 ft. = U.S. $69,780 (Annual Savings)

The annual savings shown above for one ship during one year of operation can increase substantially if the owner implements the usage of fiberglass for all the vessels in his fleet.

If you add up this savings over a ten-year period for every hp-hr for every 100 feet the saving is very significant and Bondstrand pipe can be used for the life of the vessel while steel pipe probably must be replaced several times.

In addition to time and energy saving, there are also savings due to purchase and maintenance of significantly smaller pumps in terms of horsepower rating.

References

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APPENDIX A

USING METALLIC PIPE COUPLINGS TO JOIN BONDSTRAND

Over the years, metallic pipe couplings have proven to be reliable and economical in certain Bondstrand piping systems. However, when joining Bondstrand, the recommended procedure is somewhat different than when joining rigid pipe materials such as steel and ductile iron. This bulletin describes the joining of Bondstrand pipe using Viking Johnson Couplings* along with a brief review of the couplings' design, construction and operating features. Because of the similarity of design, the same recommendations generally apply also to the use of Rockwell** or Dresser*** couplings.

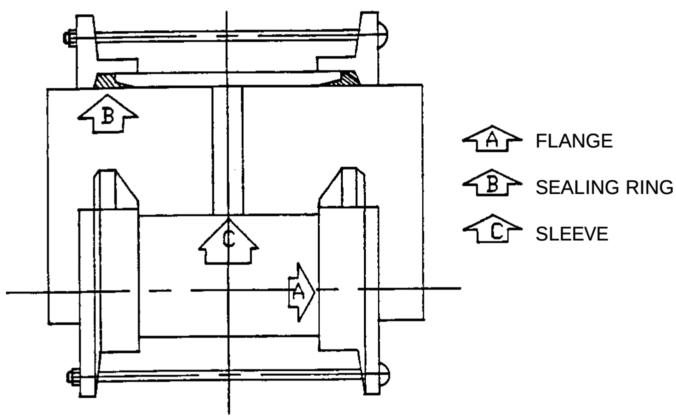
DESCRIPTION

Viking Johnson mechanical couplings are manufactured in many different sizes and configurations to meet many pipe joining requirements. Ease in close quarter installation and disassembly allow them to be used in many areas where other pipe jointing methods would be impractical. The elastomeric seals in the couplings help absorb movements such as length changes due to temperature or the flexing of a ship, and help dampen vibrations such as are produced by a pump.

The Viking Johnson Coupling consists of a cylindrical center sleeve, two end flanges, two elastomeric sealing rings and a set of 'D' neck cup-head bolts. (See Figure1)

Tightening the bolts pulls the end flanges together, compressing the sealing rings between the pipe wall and center sleeves, producing a flexible, reliable seal.

Fig. 1



a. Sealing Ring Materials

The grade 'T' ring is made from Nitrile and is, according to Viking Johnson literature the ring most commonly used. It is recommended for use on lines carrying gases, air, fresh and salt water, petroleum products, alkalies, sugar solutions and some refrigerants, and for temperatures from —20º to +100ºC (-4ºF to +212ºF). Other grades such as EPDM — 'E' Polychloroprene — 'V', Polyacrylic — 'A', Fluoroelastomer — '0', and Silicone, — 'L', are also available.

* Viking Johnson is a trade name of the Viking Johnson International division of the Victaulic Co. Plc — England
** Rockwell is a trade name of the Municipal and Utility Division of Rockwell International Corp.
*** Dresser is a registered trademark of Dresser manufacturing Division of Dresser Industries Inc.

DESCRIPTION (cont.)

b. Pressure Plating

Maximum pressure ratings of the Viking Johnson Couplings are determined on the basis of Barlow's formula using a working stress equal to two—thirds the minimum yield of the center sleeve material. All pressure ratings exceed the minimum requirements for 10 bar (150 psi) piping systems.

c. Chemical Resistance

Viking Johnson Couplings can serve in most chemical environments. This is accomplished by changing the type of sealing rings and using different types of protective coatings on the coupling.

d. Electrical Grounding

On special order, Viking Johnson provides a stud welded connection for grounding the center sleeve to the end flanges. Wires from the end flanges are bolted onto the stud on the center sleeve, and the connection is bolted down. Connecting the wiring on the center sleeve may be carried out prior to the assembly on the Bondstrand pipe ends.

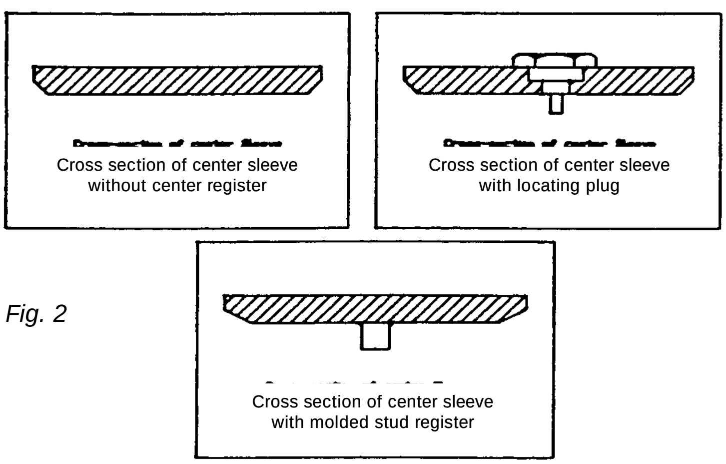
e. Locating Plug

Where there is any possibility of coupling movement along the pipe, due to repeated expansion and contraction or under vibration conditions, it is preferable to use a locating plug which centralizes the coupling over the pipe ends. If the coupling is to be slipped back along the pipe at a later date, the plug can be removed and subsequently refitted. Locating plugs are mandatory with most approval authorities when couplings are used on board ships. (See Figure 2).

JOINT FUNCTION

The sealing ring used in the Viking Johnson coupling is not intended to slide. The coupling will accommodate up to 9.5mm (3/8 in.) longitudinal pipe movement per joint as the rings deform (roll slightly) in response to such movement.

Important: Where pipe movement out of the coupling might occur, proper anchorage of the pipe must be provided.



Cross section of center sleeve without center register

Cross section of center sleeve with locating plug

Cross section of center sleeve with molded stud register

Fig. 2

Individual couplings must be protected against movements greater than 9.5mm (3/8 in.). Anchorage must be provided to prevent excessive accumulation of movement, particularly at all points which produce thrust, including valves, bends, branches and reducers.

LENGTH CHANGES IN BONDSTRAND

Bondstrand pipe lengths change due to both temperature and pressure. Estimate these changes by referring to Chapter 2 “Design for Expansion and Contraction” contained in this manual.

ASSEMBLY PROCEDURE

Joining of Bondstrand pipe using Viking Johnson Couplings is similar to joining of steel pipe, but there are important differences. You may need suitable coatings for the cut and sanded surfaces. (See step d. below). Also, you will need the following tools:

1. Torque wrench reading in increments of 5 foot—pounds or metric equivalent.
2. Hacksaw, saber saw or abrasive wheel.
3. Duster brush or clean rags.
4. Bondstrand pipe shaver or belt sander.

Although Bondstrand pipe can be supplied with prepared ends, you may need to cut pipe to length on site. If so, you will need one or more of the following:

1. For 100mm, 4-in. and smaller pipe, emery cloth strips to “shoeshine” pipe ends.
2. For 150mm to 300mm (6 to 12 in.) pipe - Bondstrand MBO Pipe Shaver (NOV FGS CC #34342) plus arbor sizes as required. Arbors used are same as for M74 shaver.
3. For 350 to 600mm (14 to 24 in.) pipe — Bondstrand M81 Pipe shaver (NOV FGS CC #34354).
4. For 350 to 900mm (14 to 36 in.) pipe - Bondstrand M81 Pipe shaver (NOV FGS CC #34355).

Caution: Be aware that the standard assembly instructions for these couplings are intended for rigid metallic pipe materials and MAY DAMAGE THE BONDSTRAND PIPE. Instead, follow this step- by-step procedure:

a. Cutting Pipe to Length

When necessary to cut a pipe to length, measure the desired length and scribe the pipe using a pipefitter’s wrap-around. Place the pipe in a vise, using 6mm (1/4 inch) thick rubber pad to protect pipe from damage. Cut pipe with hacksaw, saber saw or abrasive wheel. Pipe should be square within 3mm (1/8 in.). Use a disc grinder or file to correct squareness as required.

b. Sand Cut Ends of Pipe

End surfaces of the plain end pipe should be either hand sanded using a 40—50 grit aluminum oxide sanding surface or, if many ends are to be prepared, use a 6mm (1/4 inch) drill motor, 1700-2000 RPM, and flapper type sander available from NOV FGS. Be sure to remove all sharp edges by sanding the inside and outside edges of the pipe end. Do not touch the sanded surface with bare hands or other articles that would leave an oily film.

c. Prepare Gasket Sealing Surfaces

Machining the surface of Bondstrand pipe is not required for a tight seal between the gasket and pipe wall. However, the winding techniques used in the manufacture of Bondstrand fiberglass pipe sometimes produce a somewhat oversized outside diameter. This increase in diameter sometimes may not permit the Viking Johnson Coupling to slide over the pipe ends when installing plain-end pipe section.

d. Coat the Cut and Sanded Surfaces

Ends must be clean and dry. Select and apply a coating to the sanded end surfaces of the pipe and allow to dry thoroughly. A coating such as Amercoat 90, manufactured by NOV FGS Protective Coating Division, is suitable for water and other mildly corrosive services.

Note: On special order, NOV FGS can supply full-length Bondstrand pipe for couplings with ends prepared in accordance with steps b, c, and d.

e. Lubricate the Joining Surfaces

Clean and lubricate the sealing rings and the outside surface of the pipe with the coupling manufacturer's recommended lubricant. The ring lubricant makes it easier to slip the rings onto the pipe, and enables rings to seat properly when tightening bolts.

f. Mount and Assemble the Coupling

Slide the end flanges onto the pipe, followed by the lubricated sealing rings. Align the pipes, being careful not to bump or damage the pipe ends, and assemble the couplings over the center of the joint. The assembly of the coupling to Bondstrand fiberglass pipe should take place with the pipe supported in its final installation position.

g. Tighten the Bolts

Torque each bolt to 7 N-m (5 ft-lbs) in a diametrically opposite sequence. At 7 N-m (5 ft-lbs) torque, check to make sure that both end flanges are compressed evenly on the sealing rings. If the end flanges are not even, loosen the nuts and re-check alignment of pipe. Also check to make sure that the end flanges are not binding on the pipe wall or the center sleeve and that there is clearance between the pipe ends.

Caution: Excess torque can damage pipe. Instructions that accompany Viking Johnson Couplings show general assembly instructions and specify 70-90 foot-pounds (100-125 N-m) torque. This torque has been shown to damage Bondstrand pipe.

h. Check Bolt Torque

After each bolt has been tightened to the required torque, re-check the torque on all bolts in the same sequence. Bolts previously tightened may have relaxed as subsequent bolts were tightened.

TESTING

Be sure all pipe, fittings and appurtenances are properly and securely anchored before testing. Remember, the couplings themselves will not resist longitudinal load. Replace all air in the piping system with water and test to 1-1/2 times the operating pressure for four hours, or as required by the project specifications.

TROUBLE SHOOTING

If proper procedures have been followed, no difficulty should be experienced. If troublesome problems occur, try the following suggestions:

1. Loosen all bolts and nuts.
2. Check for alignment of assembly. Rebuild to correct alignment if out of alignment.
3. Check the alignment of assembly. Replace damaged rings.
4. Measure the diameter of the pipe at the ring location. This measurement should be within the limits shown on Table 1.

Table 1
Permissible Outside Diameter Limits at Pipe Ends for Metallic Pipe Couplings

Nominal Pipe Diameter		Nomiani Outside Diameter		Permissible Outside Diameters at Pipe Ends*			
				Min.		Max.	
mm	in.	mm	in.	mm	in.	mm	in.
50	2	60.3	2.375	59.5	2.344	61.1	2.406
80	3	88.9	3.50	88.1	3.469	89.7	3.531
100	4	114.3	4.50	113.5	4.469	115.1	4.531
150	6	168.3	6.625	167.5	6.594	169.8	6.687
200	8	219.1	8.625	218.1	8.586	220.6	8.687
250	10	273.1	10.75	272.1	10.711	274.6	10.812
300	12	323.9	12.75	322.9	12.711	325.5	12.813
350	14	356	14	354.0	13.938	357.2	14.062
400	16	406	16	404.8	15.938	408.0	16.062
450	18	457	18	455.6	17.938	458.8	18.062
500	20	508	20	506.4	19.938	509.6	20.062
550	22	559	22	557.2	21.938	560.4	22.062
600	24	610	24	608.0	23.938	611.2	24.062
650	26	660	26	658.8	25.938	662.0	26.062
700	28	711	28	709.6	27.938	712.8	28.062
750	30	762	30	760.4	29.938	763.6	30.062
800	32	813	32	811.2	31.938	814.4	32.062
850	34	864	34	862.0	33.938	865.2	34.062
900	36	914	36	912.8	35.938	916.0	36.062

Note: Tolerances apply only for a length of 6 inches back from pipe ends

STRAUB-FLEX COUPLINGS*

Straub-Flex couplings may be used as mechanical joints for Bondstrand pipe much like Dresser-type couplings. Tests of the Straub design show that the seal is effected without grinding or sanding of the pipe's outer surface. The coupling is suitable for fire, salt water and crude oil lines and various other services normally provided by Series 1600, 2000. 2000M, 6000 and 7000 piping, either suspended or buried. It may also be used with Series 4000 and 5000 piping in certain slurry applications.

The coupling design, shown in Figure 1, incorporates a stainless steel outer casing split longitudinally at one point on the circumference. The casing encloses a rubber gasket with a patented lip seal, which is pressed in place by a relatively low radial pressure. The coupling is installed on plain-end pipe using a torque wrench with a hex bit to tighten two socket-head cap screws. These features permit installation on Bondstrand pipe using the same bolt torques as recommended for steel pipe.

Straub-Flex couplings are not designed to withstand longitudinal forces. They allow 3/8-in. (10mm) longitudinal pipe movement per joint without slippage of the gasket lip on the pipe surface. Individual joints should be protected against movements greater than 3/8-in. (10mm) to prevent gasket wear. Anchorages must be provided to prevent excessive accumulation of movement, particularly at thrust points such as valves, turns, branches or reducers.

The rubber gasket both dampens vibration and allows flexing of joints such as in piping on a ship. With proper support the coupling also allows up to 2 degrees of angular movement. This added flexibility, along with the coupling's added weight, must be considered in the analysis of deflections and spans in suspended systems.

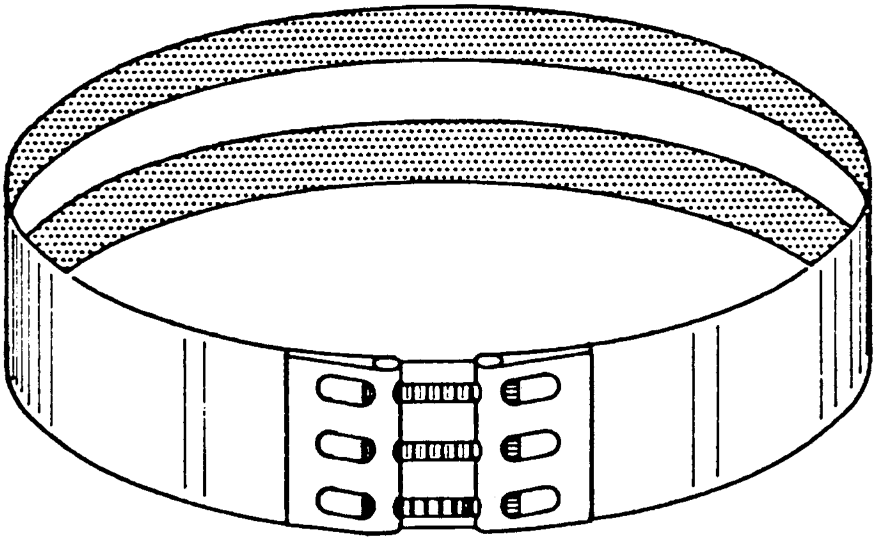


Fig. 3 Straub-Flex Coupling

* Straub. Flex is a trade name of Straub Kupplungen, AG, Wangs, Switzerland and Thornhill, Ontario, Canada.

MATERIALS

Casing

Straub-Flex Type LS couplings have type 304 stainless steel casings and galvanized steel lock bolts. Type LS Special couplings are made of the same materials but have thicker casings. Types 316 and 316L stainless steel casings and stainless steel lock bolts are available on special order.

Gaskets. Two synthetic rubber gaskets are available:

a. EPDM (ethylene propylene diene rubber)—a high quality synthetic rubber with excellent resistance to fresh or salt water, clean air, and sewage, and resistant to most moderately corrosive liquids in a pH range from 2 to 11. This rubber is not recommended for use with petroleum products.

b. Buna-N (nitrile rubber)—-a synthetic rubber for use with oil, gasoline, natural gas and most petroleum products.

PRESSURE RATING

All types of Straub-Flex couplings shown in Table 1 are rated for at least 150 psi pressure. Contact the manufacturer for possible lower ratings if stainless steel bolts are specified. Ratings include an allowance for test pressures up to 50 percent higher than rated pressure according to the manufacturer. Higher pressure ratings are available in all sizes.
The pressure ratings are for continuous service at 180ºF (82ºC) with the EPDM gasket, and for continuous services at 160ºF (71ºC) with the Buna-N gasket.

OPTIONAL PROTECTION SLEEVE**

Heat-shrinkable thermoplastic sleeves may be used to provide a moisture and soil barrier around the couplings after joint assembly. An adhesive inside the sleeve seals it against the pipe on the outside to encapsulate the coupling.

ELECTRICAL GROUNDING

A Straub-Flex coupling may act as a joint insulator. If electrical continuity is required across the pipe joint for Bondstrand Series 7000 pipe, a separate electrical bonding strip should be placed across the outside of the Straub-Flex casing, and connected to the pipe on both sides of the coupling.

LENGTH CHANGES IN BONDSTRAND

Bondstrand pipe changes length due to changes in temperature and pressure. Estimate these changes by referring to Chapter 2 “Design for Expansion and Contraction” contained in this manual.

** Heat-shrinkable sleeves are produced by the Pipe Production Division of Raychem Corp., Redwood City, CA., by Chemplast, Inc., Wayne, NJ, and outside the U.S. by Canusa Coating Systems, Ltd., Rexdale, Ontario, Canada.

ASSEMBLY PROCEDURE

Using Straub-Flex couplings, joining Bondstrand is similar to joining steel pipe, except for sealing cut pipe ends. Depending on chemical exposure, you may need a suitable coating to cover exposed glass fibers on the cut ends. It is usually not necessary to sand or shave the outer surface of Bondstrand pipe as the Straub couplings make a tight seal on the as-wound surface. Exceptions are given in step “c” of this procedure.

You may use the standard joining instructions for Straub-Flex couplings as used with steel pipe. You will need the following tools:

1. Torque wrench reading in increments of 5 ft-lbs (7 N-m.)

2. Hacksaw, saber saw or abrasive wheel.

3. Duster brush or clean rags.

Steps “b” and “d” given below are recommended for piping in which the cut pipe ends must be protected against chemical attack or abrasion. In slurry applications, the user should be aware that the joint cavity may fill with sediment, restricting flexibility.

a. Cut Pipe to Length

When cutting is necessary, measure the desired length and scribe the pipe using a pipefitter’s wraparound. Place the pipe in a vise, using 1/4-inch (6mm) thick rubber pad to protect pipe from damage. Cut pipe with hacksaw, saber saw or abrasive wheel. Pipe end cut should be square within 1/8-inch (3mm). Use a disc grinder or file to correct squareness as required.

b. Sand Cut Ends of Pipe

End surfaces of cut pipe should be sanded either by hand using a 40-50 grit aluminum oxide sanding surface or using a 1/4-in. (6mm) drill motor 1700-2000 RPM with a flapper-type sander available from NOV FGS. Be sure to remove all sharp edges by sanding the inside and outside edges of the pipe end. Do not touch the sanded surface with bare hands or articles that leave an oily film.

c. Prepare Gasket Sealing Surfaces

Machining the gasket sealing surfaces at the ends of Bondstrand pipe is not generally required for a tight seal between the gasket and pipe wall. However, two-inch (50mm) pipe will require shaving of the ends, since its average outside diameter of 2.42 in. (61.5mm) is larger than can be fitted by the two-inch Straub-Flex coupling (Article No. 005761).

The coupling manufacturer recommends that the difference in outside diameters of mating pipe ends be no greater than 0.12 in. (3mm), to avoid distortion of the coupling and damage to the cap screws while joining. Using a diameter tape, measure the outside diameters of pipe ends to ensure that this difference is not exceeded. If the difference is larger than permissible, milling or shaving of the larger end is necessary. Because Bondstrand Series 2000M and Series 7000 pipe in sizes 10 and 12 in. (250 and 300mm) have outside diameters larger than steel pipe, their ends must be shaved to mate to standard outside diameters of steel pipe and fittings.

d. Coat the Cut Ends and Gasket Sealing Surfaces (Lined Pipe Only)

Surfaces must be sanded, clean and dry for coating. Select and apply a coating to the cut ends and shaved gasket sealing surfaces of the pipe and allow to dry thoroughly. A coating covers
exposed glass fibers and is suitable for water and other mildly corrosive services. Bondstrand PSX™-34 adhesive may also be suitable.

Note: On special order, NOV FGS can supply full-length Bondstrand pipe for Straub couplings with ends prepared in accordance with steps b, c and d.

e. Fit the Coupling

With the pipe ends ready for joining, chalk a mark on each end at a distance equal to half the coupling width. Joining of the pipe should be done with the pipe supported in its final installation position.

Couplings are supplied loosely assembled. Slide the coupling onto the end of one pipe up to the chalk's mark. Align the second pipe end and slide it into the coupling, using care not to bump or damage the pipe ends. Center the coupling over the two pipe ends, leaving a small clearance between the pipe ends.

Note: Do not soap the inside surfaces of the gaskets or the outside surface of the pipe.

f. Tighten the Bolts

Using a torque wrench with a hex bit, alternately torque each of the two socket-head cap screws to the recommended torques. Ensure that there is clearance between pipe ends.

TESTING

Because Straub-Flex couplings do not resist longitudinal load, make sure all pipe, fittings and appurtenances are properly and securely anchored before testing. Replace all air in the system with water, and test to 1-1/2 times the operating pressure for four hours or as required by the project specifications.

TROUBLE SHOOTING

If proper procedures have been followed, no difficulty should be experienced. If a joint leaks, try the following:

1. Disassemble the leaky coupling and an adjacent coupling and remove a pipe section for examinaton of the rubber gasket and the pipe ends.

2. If the gasket is damaged, replace with another coupling.

3. If the pipe end is not within the diameter limits shown in Table 2, or has abnormally rough surface or grooves, sand the pipe end surfaces and reinstall the pipe.

Table 2
Application Data for Straub-Flex Couplings

Nominal Diameter (in.)	Nominal Diameter (mm)	Straub Type	Article No. (1)	Pressure Rating (3) psi.	Pressure Rating (3) Bar	Coupling Width (in.)	Coupling Width (mm)	Pipe Outside Diameter Minimum (in.)	Pipe Outside Diameter Minimum (mm)	Pipe Outside Diameter Maximum (in.)	Pipe Outside Diameter Maximum (mm)
2	50	Flex-1 L-LS	005761	225	16	3.0	76	2.25	57	2.40	61
3	80	Flex-1 L-LS	008792	225	16	3.7	94	3.43	87	3.62	92
4	100	Flex-1 L-LS	112117	225	16	3.7	94	4.41	112	4.60	117
6	150	Flex-1 L-LS	166171	225	16	4.3	108	6.54	116	6.73	171
8	200	Flex-2 L-LS (2)	214224	213	15	5.5	139	8.43	214	8.81	224
10	250	Flex-2 L-LS (2)	270280	178	12.5	5.5	139	10.63	270	11.02	280
12	300	Flex-2 L-X	323333	>225	>16	5.8	148	12.72	323	13.11	333
14	350	Flex-2 L-X	353363	>225	>16	5.8	148	13.90	353	14.30	363
16	400	Flex-2 L-X	404414	>225	>16	5.8	148	15.90	404	16.30	414
18	450	Flex-2 L-X	455465	>225	>16	5.8	148	17.90	455	18.30	465
20	500	Flex-2 L-X	505516	>225	>16	5.8	148	19.90	505	20.30	516
22	550	Flex-2 L-X	556566	>225	>16	5.8	148	21.90	556	22.30	566
24	600	Flex-2 L-X	608617	>225	>16	5.8	148	23.90	608	24.30	617
26	650	Flex-2 L-X	658668	>225	>16	5.8	148	25.90	658	26.30	668
28	700	Flex-2 L-X	709719	>225	>16	5.8	148	27.90	709	28.30	719
30	750	Flex-2 L-X	759770	>225	>16	5.8	148	29.90	759	30.30	770
32	800	Flex-2 L-X	810820	>225	>16	5.8	148	31.90	810	32.30	820
34	850	Flex-2 L-X	862871	>225	>16	5.8	148	33.90	862	34.30	871
36	900	Flex-2 L-X	912922	>225	>16	5.8	148	35.90	912	36.30	922

1. Article number gives OD range of coupling in millimetres.
2. 8 and 10 in. (200-250 mm) sizes must be ordered with special casing thickness because the standard coupling only provides (15 bar) and (12 bar) maximum pressure. Casing does provide > 225 psi (10 bar) minimum pressure rating.
3. Couplings with higher pressure ratings are available on special order.

APPENDIX B

GROUNDING OF SERIES 7000M PIPING

Electrical charges generated within flowing fluids with low conductivity such as liquid hydrocarbon fuels can cause hazardous static charges to build up on the surfaces of the pipe. To overcome this problem and still offer the advantages inherent in RTB piping, NOV FGS has developed special piping systems-Bondstrand Series 7000 and 7000M. These piping systems provide electrical continuity throughout by incorporating conductive elements into the structural wall of the pipe, flanges and the interior surface of the fittings, and through the use of a specially formulated adhesive which provides the conductivity required at the bonded joints.

Proper installation and grounding is important for the safe operation of Series 7000 and 7000M pipe when carrying these charge-generating fluids. This bulletin explains how these products are to be installed, grounded and checked to verify their electrical continuity.

ASSEMBLY OF PIPE

All Series 7000 and 7000M piping are assembled using electrically conductive Bondstrand PSX™-60 adhesive. This special two-component epoxy adhesive is supplied in kit form. Detailed application instructions are contained in "Bondstrand Assembly Instructions, PSX™-60 Epoxy Adhesive," FP827.

ADHESIVE MOUNTING OF GROUNDING SADDLE

Grounding saddles provide a positive method of electrically grounding the piping system. On the pipe, determine where the grounding saddle will be located. Using a flapper sander, sand until the surface gloss is removed from at least a 3-in. width around the pipe circumference as needed to fit the saddle on the area selected. This exposes the conductive elements in the pipe wall and produces a clean, fresh surface suitable for bonding the grounding saddle to the pipe surface.

Before bonding on saddle, place probes from a standard ohmmeter at least two in. apart on conductive elements exposed by sanding pipe surface. If measured resistance exceeds 10^6 ohms, more sanding is required.

If measured resistance is below 10^6 ohms, bond the grounding saddle onto the clean, dry surface within two hours using PSX™-60 Epoxy Adhesive. After continuity checks recommended herein, grounding cable must be attached to ship structure.

METALLIC FITTINGS

All metallic fittings must be individually grounded. Tees, elbows, etc. should be welded or otherwise connected directly to the ship or other grounding structure. Metallic mechanical joints such as Dresser or Straub must be grounded. If mechanical joints are used, at least one grounding saddle will be required for each length of pipe.

ELECTRICAL CONTINUITY CHECK

Prefabricated Spools.

This may be done in one of three ways:

a. Non-Flanged Prefabricated Spools.

After shop fabrications but before onboard installation and grounding, spools should be checked for electrical continuity. Sand lightly around the pipe surface at each end of the spool where the steel hose clamps will attach. Mount the two steel hose clamps over the prepared surface and measure the resistance between them as shown on Figure 1.

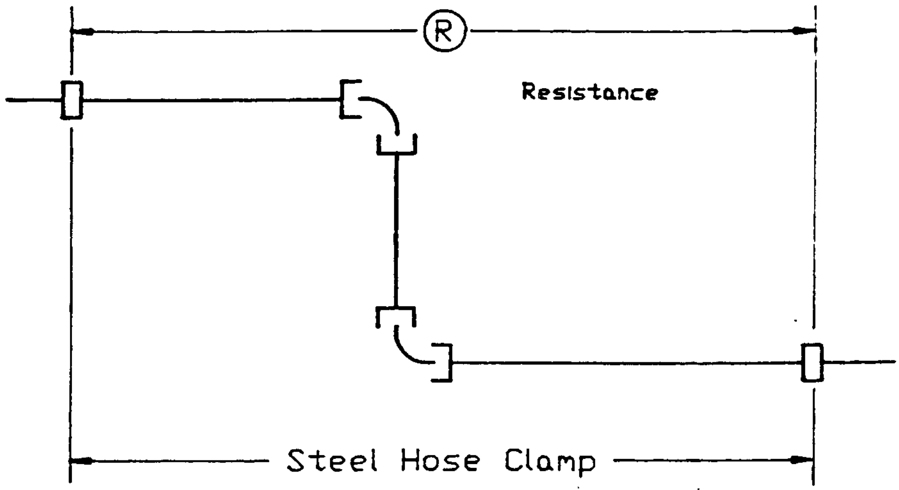


Fig. 1 Electrical Continuity Check Diagram for Non-flanged Prefabricate Spools

b. Flanged Prefabricated Spools.

Flange assemblies should be checked by placing a bolt with washer and nut through each of the flanges and tightening, then measuring the resistance between the flanges at each end of the assembly as shown on Figure 2.

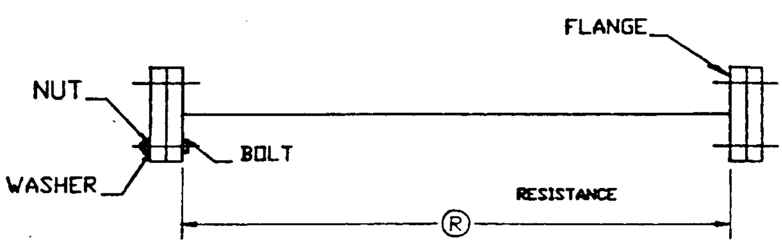


Fig. 2 Electrical Continuity Check Diagram for Flanged Prefabricate Spools

C. Flanged One End Only Spools.

This assembly should be checked by following the procedure established in b. above for the flanged end and the procedure established in a. above for the plain end as shown in Figure 3.

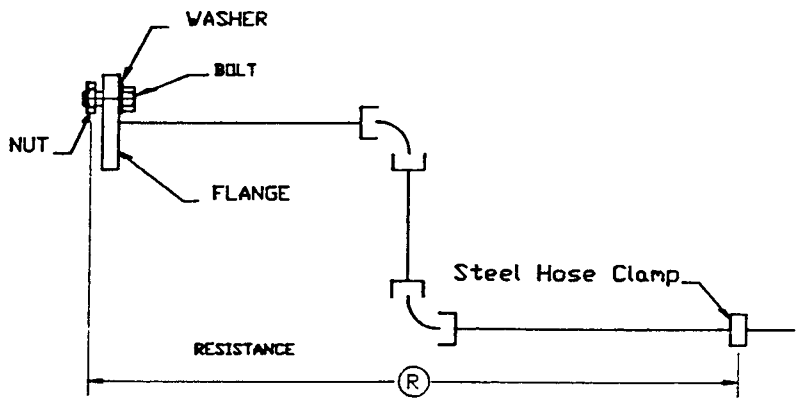


Fig. 3 Electrical Continuity Check Diagram for Flanged One End Only

Apply sufficient voltage between the hose clamps to measure the electrical resistance in the spool using a standard generator- type insulation tester* capable of applying up to 1,500 volts dc. The measured resistance should not exceed 10^6 ohms.

Onboard Check During New Construction.

Piping should be checked electrically as installation proceeds onboard ship. After mounting a grounding saddle (A) as shown on Figure 4, the length of piping from the grounding saddle to the end of the pipe run should be electrically insulated by placing a layer of nonconducting rubber (B) temporarily between the remaining unattached supports and the free end of the pipe.

Attach a steel hose clamp over the pipe surface at the free end and use the tester to measure the resistance between the hose clamp and the ship structure. Current must flow back through the pipe, fittings and joints to the nearest grounding support clamp to complete the circuit as shown in Figure 1. As before, the measured resistance must not exceed 10^6 ohms between any two grounding supports.

After the electrical continuity of the piping has been verified, the non-conducting rubber pads at the grounding supports should be removed. Proceed to bond the pipe into the remaining grounding saddle.

* NOV FGS recommends the use of a Megger Mark IV Insulation Tester, Cat. No. 211805, James G. Biddle Co., or equal.

Onboard Check During Drydock for Maintenance and Repair

Fiberglass piping systems using Series 7000 and 7000M pipe and fittings should be checked during each drydock inspection while the tanks are “gas freed” to ensure that the systems are still properly grounded. This can be done using either of the following procedures:

a. Electrically Isolated Piping

The straps attached to the grounding saddle utilized to ground the piping system must be disconnected and the pipe electrically isolated from the structure of the ship shown on Figure 4. Tightly fasten two steel hose clamps at opposite ends of the pipe spool being tested and measure the resistance between them using a standard generator—type insulation tester capable of applying 1,500 volts dc. The resistance should not exceed 10^6 ohms. Now attach one of the grounding cables to the structure of the ship and in like fashion check the resistance between the pipe and the structure of the ship.

Important: To ensure that each grounding saddle is functioning properly, no more than one grounding strap at a time should be connected to the ship’s structure during the test.

b. Grounded Piping

If it is impossible to electrically isolate the system, each section of pipe must be checked separately. This may be done by placing a steel hose clamp on each section of pipe (defined as a length between bonded joints) and measuring the resistance between it and the nearest grounding location as described above.

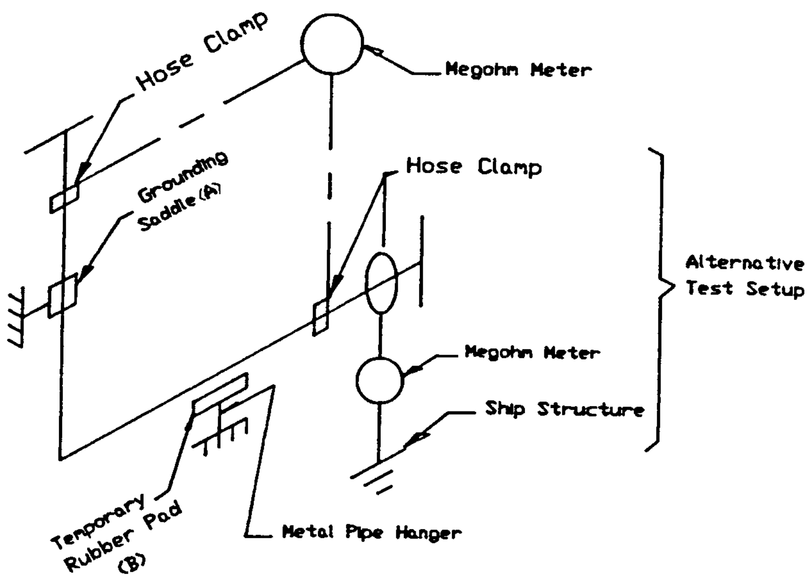


Fig. 4 Test Setup For Electrical Continuity Check of Piping During New Construction and Drydock Periods

APPENDIX C

SIZING OF SHIPBOARD PIPING

Shipyards and design agencies have used various methods to evaluate and select velocities for each application. These methods have yielded acceptable sizes, pressure drops and efficiency losses and have allowed adaptation of the nearest standard pipe size in the preliminary design stages.

The method discussed herein uses the inside diameter factor to calculate maximum velocities and flow in gallons per minute for Nominal Pipe Size (NPS) 1 to 36 with Iron Pipe Size (IPS) and Metric Cast Iron (MCI) internal diameters.

For Bondstrand fiberglass piping systems a maximum allowable velocity of 15 ft./sec. has been established. This is to prevent erosion which might occur at higher fluid velocities. Table 1 shows inside diameter factors

$$\left[\overline{ID}\right]^{1/2} \; ; \; \left[\overline{ID}\right]^{1/3} ; \text{ and } \left[\overline{ID}\right]^{2}$$

For NPS 1 to 36 IPS and MCI internal diameter configurations. Table 2 shows fourteen inside diameter functions for different shipboard piping systems.

Applying the IDF (inside diameter function) for a given piping system, maximum velocity value for different pipe sizes can be obtained as follows:

Example A:

Calculate the maximum velocity and maximum flow rate for a 6-in. IPS fiberglass pipe to be used in a feed discharge system.

IDF for feed discharge = $220\ \overline{ID}^{1/2}$ = (From Table 2)

I.D. Factor for 6 in. (IPS) = $\overline{ID}^{1/2}$ = 2.50 (From Table 1)

$$V(fpm) = 220 \times 2.50 = 550 \text{ fpm.}$$

$$V(fps) = \frac{550}{60}$$

$$= 9.17 \text{ fps (Max. allowable velocity)}$$

9.17 fps < 15 fps (Ok to use fiberglass)

To establish maximum flow rate:

$$Q(gpm) = \frac{\overline{ID}2 \times Vfpm}{24.51}$$

$$Q(gpm) = \frac{39.19 \times 550}{24.51}$$

$$Q(gpm) = 879.42\ (gpm)$$

Where:

Q(gpm) = Maximum (Gallons per minute) Flow Rate.

V(fpm) = Maximum Allowable Velocity (Feet per Minute)

$\overline{ID}^2$ = Pipe inside diameter (in^2) (See Table 1)

24.51 = Constant

Table 1

NPS	Bondstrand Inside Diameter IPS(in)	MCI(in)	$\overline{ID}^{1/2}$ IPS	MCI	$\overline{ID}^{2}$ IPS	MCI	$\overline{ID}^{1/3}$ IPS	MCI
1								
1¼								
2	2.09		1.45		4.37		1.28	
3	3.22		1.79		10.37		1.48	
4	4.14		2.04		17.14		1.61	
6	6.26		2.50		39.19		1.84	
8	8.22		2.87		67.57		2.02	
10	10.35		3.22		107.12		2.18	
12	12.35		3.51		152.52		2.31	
14	13.29	14.12	3.65	3.76	176.62	199.37	2.37	2.42
16	15.19	16.03	3.90	4.00	230.74	256.96	2.47	2.52
18	17.08	17.94	4.13	4.24	291.73	321.84	2.58	2.62
20	18.98	19.83	4.36	4.45	360.24	393.23	2.66	2.71
22	20.88	21.78	4.57	4.67	435.97	474.37	2.76	2.79
24	22.78	23.73	4.77	4.87	518.93	563.11	2.83	2.87
26	24.68	25.63	4.97	5.06	609.10	656.90	2.91	2.95
28	26.57	27.57	5.16	5.25	705.97	760.11	2.99	3.02
30	28.47	29.52	5.34	5.43	810.54	871.43	3.06	3.09
32	30.37	31.46	5.51	5.61	922.34	989.73	3.12	3.16
34	32.27	33.37	5.68	15.78	1041.35	1,113.56	3.18	3.22
36	34.17	35.31	5.85	15.94	1167.59	1,246.80	3.24	3.28

Example B:

Check for maximum velocity and maximum flow rate for a sea water discharge for 10-in. IPS.

IDF for water discharge = 300 $\overline{ID}^{1/2}$ = (From Table 2)

I.D. Factor for 10—inch (I.P.S.) = $\overline{ID}^{1/2}$ = 3.22 (From Table 1)

$$V(fpm) = 300 \times 3.22 = 966 \text{ fpm}$$

$$V(fps) = \frac{966}{60}$$

$$= 16.1 \text{ fps (Maximum allowable velocity)}$$

16.1 fps > 15 fps. (not recommended to use with fiberglass)

To establish maximum flow rate:

$$Q(gpm) = \frac{\overline{ID}^2 \times Vfpm}{24.51}$$

$$Q(gpm) = \frac{107.12 \ \times \ 96824.51}{24.51}$$

$$Q(gpm) = 4{,}221.87 \text{ gpm (Maximum Flow Rate)}$$

Where:

Q(gpm) = Maximum (Gallons per minute) Flow Rate.

V(fpm) = Maximum Allowable Velocity (Feet per Minute)

$\overline{ID}^2$ = Pipe inside diameter (in.^{2})

24.51 = Constant

Based on the required system flow rate, the correct pipe size can be determined by trial and error.

Table 2

MARINE PIPING SYSTEM		INSIDE DIAMETER FUNCTION (IDF)
Feed suction	Centrifugal pump	$125\ \overline{ID}^{1/2*}$ = Max. allowable velocity (fpm)
	Positive displacement pump	$175\ \overline{ID}^{1/2*}$ = "
Feed discharge		$220\ \overline{ID}^{1/2*}$ = "
Condenser suction		$55\ \overline{ID}^{1/3*}$ = "
Condenser discharge		$180\ \overline{ID}^{1/2*}$ = "
Fuel oil transfer suction		$70\ \overline{ID}^{1/2*}$ = "
Fuel oil transfer discharge		$110\ \overline{ID}^{1/2*}$ = "
Condensate from heating system		$30\ \overline{ID}^{1/2*}$ = "
Water suction		$190\ \overline{ID}^{1/2*}$ = "
Water discharge		$300\ \overline{ID}^{1/2*}$ = "
Lube oil suction		$9.5\ \overline{ID}^{2*}$ = "
Lube oil discharge		$25\ \overline{ID}^{2*}$ = "
Fuel oil discharge		$65\ \overline{ID}^{2*}$ = "
Hydraulic oil (power steering)		$850\ \overline{ID}^{1/2*}$ = "

* See Table 1 for inside diameter coresponding to the NPS selection.

Note: For bilge suction use V=400 fpm (feet per minute) for all NPS selections

APPENDIX D

Miscellaneous data

D.1 Adhesive Requirements (PSX^{tm}-34 ; PSX^{tm}-60)

The number of joints that can be made using 3 oz., 5 oz., or 8 oz. Kits of PSX^{tm}-34 and/or PSX^{tm}-60 are shown on the Table below.

Nominal Pipe Size	KIT SIZE		
	3 oz.	5 oz.	8 oz.
1	10	—	—
1.5	6	10	—
2	4	7	10
3	3	5	8
4	2	3	6
5	1	2	5
6	1	1	3
8	.50	1	2
10	.50	1	2
12	.50	1	1
14	—	.50	1
16	—	.50	1

Note:

a. Joint sizes 18 thru 36 require minimum of 2 persons to make up a joint.

b. Minimum required curing time with heating blanket is 45 minutes for all size joints.

D2. Rated Pressures, Volumes and Weights of Pipe

Nominal Diameter		Rated Internal Operating Pressure(1) at 150°F (66°C)		Rated External Pressure(2) at 150°F (66°C		Fluid Volume		Shipping Weight	
in	mm	psi	bar	psi	bar	gal/ft	m^3/m	lb/ft	kg/m
2	50	550	38	610	42.	0.18	0.0022	0.8	1.2
3	80	450	31	170	12.	0.42	0.0052	1.2	1.9
4	100	450	31	190	13.	0.70	0.0087	2.0	3.0
6	150	300	21	63	4.3	1.60	0.0199	3.1	4.6
8	200	300	21	63	4.3	2.76	0.0342	5.2	7.8
10	250	300	21	63	4.3	4.37	0.0542	8.1	12.
12	300	300	21	63	4.3	6.23	0.0772	11.	17.
Iron Pipe Size (IPS)									
14	350	300	21	63	4.3	7.21	0.0894	13.	20.
16	400	300	21	63	4.3	9.42	0.117	17.	25.
18	450	300	21	63	4.3	11.9	0.148	21.	32.
20	500	300	21	63	4.3	14.7	0.182	26.	39.
24	600	300	21	63	4.3	21.2	0.263	37.	56.
28	700	300	21	63	4.3	28.8	0.358	51.	75.
30	750	300	21	63	4.3	33.1	0.410	58.	86.
36	900	300	21	63	4.3	47.7	0.591	83.	123.
Metric Cast Iron Size									
14	350	300	21	63	4.3	8.14	0.101	15.	22.
16	400	300	21	63	4.3	10.5	0.130	19.	28.
18	450	300	21	63	4.3	13.1	0.163	24.	35.
20	500	300	21	63	4.3	16.1	0.200	29.	43.
24	600	300	21	63	4.3	23.0	0.285	41.	60.
28	700	300	21	63	4.3	31.0	0.385	54.	81.
30	750	300	21	63	4.3	35.6	0.441	62.	93.
36	900	300	21	63	4.3	50.9	0.631	88.	132.

Note:

1) System internal operating pressures may be limited by mechanical joints, fittings or anchoring requirements to values below the rating of the pipe itself.

2) Pipe design resists collapse due to combined internal suction head and external fluid pressure. For example, a 63-psi (4.3-bar) external pressure rating allows for 120 ft (37 m) of water plus a 75% (suction head) with a safety factor of 2 to minimum ultimate collapse pressure

APPENDIX E

PIPING SUPPORT FOR NON-RESTRAINED MECHANICAL JOINTS

This bulletin offers suggestions for supporting and anchoring Bondstrand piping systems joined with bolted coupling mechanical joints which do not offer axial restraint. These bolted couplings are the standard designs offered by Dresser, Viking- Johnson, Rockwell, Straub, R.H. Baker and others which seal by means of an elastomeric gasket or gland seal against the outside diameter of the pipe.

The flexibility allowed by bolted couplings must be accounted for in calculating allowable span lengths. Also, provisions for anchoring against hydrostatic thrusts must be incorporated into the design.

Span Recommendations

Recommended maximum spans for Bondstrand pipe joined with bolted couplings can be determined by use of the following equation:

$$L = 0.207 \left[\frac{E_I}{w} \right]^{1/4}$$

Where L = support spacing (ft),

E_I = beam stiffness psi (lb-in2), see Tables 4—3 and 4-4

w = Total uniformly distributed load (Ib/linear in.),

In metric units:

$$L = 0.0995 \left[\frac{E_I}{w} \right]^{1/4}$$

Where L = support spacing (in),

E_I = beam stiffness psi (kg-cm^2), see Tables 4—3 and 4-4

w = Total uniformly distributed load (kg/mm).

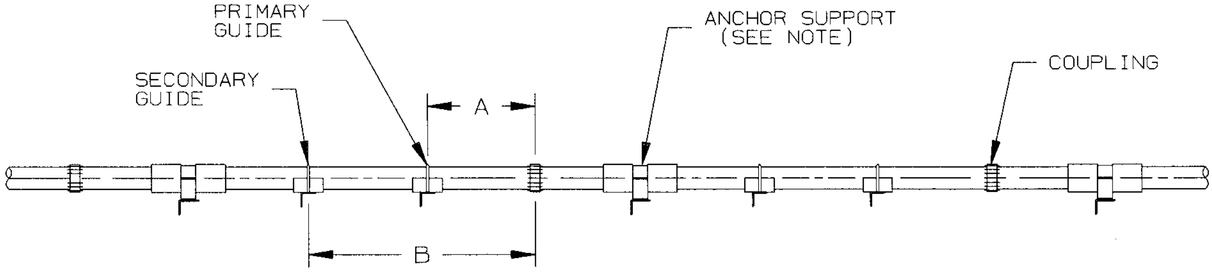
These spans are intended for normal horizontal piping support arrangements as shown in Figure 1; i.e., those which have no fittings, valves, or vertical runs incorporated within the span.

Anchoring Recommendations

Bolted couplings, not designed to withstand longitudinal forces, allow 3/8-in. (10mm) longitudinal pipe movement per joint without slippage of the gasket lip on the pipe surface. Individual joints should be protected against movements greater than 3/8-in. (10mm) to prevent gasket wear as well as preventing, in severe cases, the pipe from moving out of the coupling. Anchors must be provided at thrust points such as valves, turns, branches, or reducers, as well as at locations where excessive movement may occur (see Figure 1).

Figure 2 shows how mechanically coupled pipe should be supported and anchored at fittings. Supports must be designed to carry the weight of the pipe and its contents. Anchors are located at the terminal points of the piping system or where there is a change in direction and should be designed to withstand thrusts due to internal line pressure.

Fig. 1 Support Arrangements

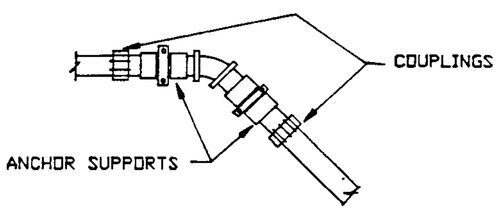


A = 4 x DIAMETER
B = 14 x DIAMETER

REMAINING SUPPORTS AND GUIDES SHOULD REMAIN WITHIN THE MAXIMUM ALLOWED FOR VARIOUS PIPE SIZE AND SERIES.

EACH PIPE LENGTH SHOULD BE ANCHORED AT LEAST ONCE TO KEEP PIPE ENDS FROM MOVING OUT OF COUPLINGS OR JAMMING TOGETHER AND ABRADING.

Note: Each Pipe length (L) should be anchored at least once to keep pipe ends from moving out of couplings or jamming together and abrading.



a. ANGULAR DIRECTION CHANGE

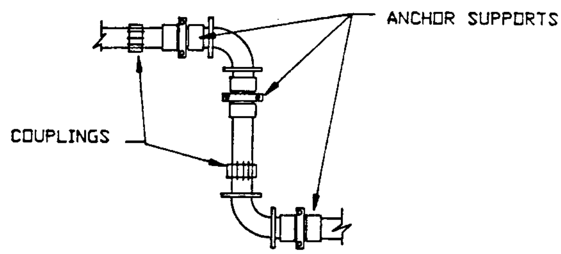
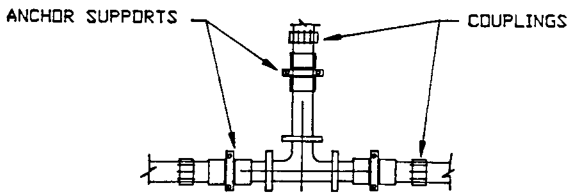


Fig. 2 Support and Anchors at Fitting

b. OFFSET.



c. BRANCH.

Note: Anchors may be affixed to pipe using saddles as shear conntectors or bolted to flanges

Conversions

1 psi = 6895 Pa = 0.07031 kg/cm^2
1 bar = 10^5 Pa = 14.5 psi = 1.02 kg/cm^2
1 MPa = 10^6 Pa = 145 psi = 10.2 kg/cm^2
1 GPa = 10^9 Pa = 145,000 psi = 10,200 kg/cm^2
1 in = 25.4 mm
1 ft = 0.3048 m
1 lb·in = 0.113 N·m
1 in^4 = 4.162 x 10^{-7}m^4
1 ft/sec = 0.304 m/sec
1 gpm = 6.31 x 10^{-7} m^3/sec
°C = $^5/_9$ (°F - 32)

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