

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION



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FOREWORD

Since 1958, when the Expansion Joint Manufacturer's Association (EJMA®) first published these Standards, continuing technological improvements in the application and design of Expansion Joints have been reported through the cooperative efforts of its association members by expanding the scope and content of this publication. Founded three years earlier in 1955, the Expansion Joint Manufacturer's Association began with a group of companies experienced in the application, design, and fabrication of Expansion Joints. The first EJMA® Standard edition was, of necessity, somewhat brief and covered only applications involving axial movement. But as research and extensive testing results were catalogued, more detailed design data has been included in the EJMA® Standard. The EJMA® Standards are intended for application to metallic bellows expansion joints having only the convolution shapes shown in the Standards and having convolution welds only in the meridional direction with the exception of the bellows attachment welds.

The EJMA® Technical Committee is dedicated to continuously improving the utility and technical content of the Standards. Suggestions and comments from industry users are welcomed and should be forwarded to the Secretary of this Association in writing.

It is important to note that the EJMA® Standard is a trade association document containing recommendations for application of expansion joint products and in-depth technical information for use in designing expansion joint products. It is not a manufacturing standard or a quality assurance document. The type of non-destructive examination and the extent of quality assurance testing to be applied to given product should be addressed by other documents such as the ASME B31.3 Piping Code, the ASME Pressure Vessel Code, or another user-provided specification. The Standard does not limit or dictate the manufacturing process to be used for construction of expansion joints, nor does it establish specific engineering requirements deemed necessary for the safe application, design, and manufacture of Expansion Joints. If there is a strong preference for a certain type of manufacturing process, the user should provide this information. Industry users are cautioned that these Standards should not be considered as a design handbook, and must not replace sound engineering judgment, education, and experience.

As of this writing, the EJMA® Standard thoroughly covers the design of expansion joint bellows elements. However, the Standard does not cover the design of hardware associated with restraint of pressure thrust. Pressure thrust restraint hardware is as important as the bellows element in the design and fabrication of an expansion joint assembly. Users are strongly advised to obtain documented design results for bellows elements and pressure thrust restraint hardware for any critical application.

NO WARRANTY EXPRESSED OR IMPLIED

The engineering Standards herein are recommended by the Expansion Joint Manufacturers Association, Inc. to assist users, engineers, architects and others who specify, design and install Expansion Joints in piping systems to obtain the most efficient service from Expansion Joint installations. These Standards are based upon sound engineering principles, research and field experience in the manufacture, design, installation and use of Expansion Joints. These Standards may be subject to revision as further investigation or experience may show is necessary or desirable. Utilization of these Standards remains entirely optional. Nothing herein shall constitute a warranty of any kind, expressed or implied. Accordingly, all warranties of whatever nature, expressed or implied, are herewith specifically disclaimed and disavowed.

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SECTION 1 – SCOPE, DEFINITIONS, AND NOMENCLATURE

1.1 SCOPE

The EJMA® Standards are only intended for application to metallic bellows expansion joints.

1.2 DEFINITION OF TERMS

The Expansion Joint Manufacturers Association, Inc. has adopted the following definitions of Expansion Joint components and related equipment.

ANGULAR ROTATION

The displacement of the longitudinal axis of the Expansion Joint from its initial straight line position into a circular arc. Angular rotation is occasionally referred to as "rotational movement." This is not torsional rotation which is described further in this section.

AXIAL COMPRESSION

The dimensional shortening of an Expansion Joint along its longitudinal axis. Axial compression has been referred to as axial movement, traverse or compression.

AXIAL EXTENSION

The dimensional lengthening of an Expansion Joint along its longitudinal axis. Axial extension has been referred to as axial movement, traverse, elongation or extension.

BELLOWS

The flexible element of an Expansion Joint consisting of one or more convolutions and the end tangents with $L_b / D_b \leq 3$, $nt \leq 0.375$ in. (9.53 mm), r_{ic} and $r_{ir} \geq 3t$, $|r_{ic}-r_{ir}| \leq 0.2r_m$, with no more than five plies.

CONTROL RODS

Devices, usually in the form of rods or bars, attached to the Expansion Joint assembly whose primary function is to distribute the movement between the two bellows of a universal Expansion Joint. Control rods are not designed to restrain bellows pressure thrust.

CONVOLUTION

The smallest flexible unit of a bellows. The total movement capacity of a bellows is proportional to the number of convolutions.

COVER

A device used to provide limited protection of the exterior surface of the bellows of an expansion joint from foreign objects or mechanical damage. A cover is sometimes referred to as a shroud.

DIRECTIONAL ANCHOR

A directional or sliding anchor is one which is designed to absorb loading in one direction while permitting motion in another. It may be either a main or intermediate anchor, depending upon the application involved. When designed for the purpose, a directional anchor may also function as a pipe alignment guide. In the design of a directional anchor, an effort should be made to minimize the friction between its moving or sliding parts, since this will reduce the loading on the piping and equipment and insure proper functioning of the anchor.

DOUBLE EXPANSION JOINT

A double Expansion Joint consists of two bellows joined by a common connector which is anchored to some rigid part of the installation by means of an anchor base. The anchor base may be attached to the common connector either at installation or at time of manufacture. Each bellows acts as a single Expansion Joint and absorbs the movement of the pipe section in which it is installed independently of the other bellows. Double Expansion Joints should not be confused with universal Expansion Joints.

EQUALIZING AND REINFORCING RINGS

Devices used on some expansion joints fitting in the roots of the convolutions. The primary purpose of these devices is to reinforce the bellows against internal pressure. Equalizing rings are made of cast iron, steel, stainless steel or other suitable alloys and are approximately "T" shaped in cross section. Reinforcing or root rings are fabricated from tubing or solid round bars of carbon steel, stainless steel or other suitable alloys.

EXPANSION JOINTS

Any device containing one or more bellows used to absorb dimensional changes, such as those caused by thermal expansion or contraction of a pipeline, duct or vessel.

FLANGED ENDS

The ends of an expansion joint equipped with flanges for the purpose of bolting the expansion joint to the mating flanges of adjacent equipment or piping (See Section 9.3).

GIMBAL EXPANSION JOINT

A gimbal Expansion Joint is designed to permit angular rotation in any plane by the use of two pairs of hinges affixed to a common floating gimbal ring. The gimbal ring, hinges and pins must be designed to restrain the thrust of the Expansion Joint due to pressure and extraneous forces, where applicable.

HINGED EXPANSION JOINT

A hinged Expansion Joint contains one bellows and is designed to permit angular rotation in one plane only by the use of a pair of pins through hinge plates attached to the Expansion Joint ends. The hinges and hinge pins must be designed to restrain the thrust of the Expansion Joint due to pressure and extraneous forces, where applicable. Hinged Expansion Joints should be used in sets of two or three to function properly.

IN-LINE PRESSURE BALANCED EXPANSION JOINT

An in-line pressure balanced Expansion Joint is designed to absorb axial movement and/or lateral deflection while restraining the pressure thrust by means of tie devices interconnecting the line bellows with outboard compensating bellows also subjected to line pressure. Each bellows set is designed to absorb the axial movement and usually the line bellows will absorb the lateral deflection. This type of Expansion Joint is used in a straight run of piping.

INTERMEDIATE ANCHOR

An intermediate anchor is one which must withstand the bellows thrust due to flow, spring forces, and all other piping loads, but not the thrust due to pressure.

An intermediate anchor base for connection to the anchor structure can be furnished as an integral part of a single or double Expansion Joint, if desired. The Expansion Joint manufacturer must be advised of the magnitude and direction of all forces and moments which will be imposed upon the anchor base, so that it can be adequately designed to suit the specific application.

INTERNAL SLEEVE

A device which minimizes contact between the inner surface of the bellows of an expansion joint and the fluid flowing through it (See Section 4.9 for application). These devices have also been referred to as liners or baffles.

INTERNALLY GUIDED EXPANSION JOINT

An internally-guided Expansion Joint is designed to provide axial guiding within the Expansion Joint by incorporating a heavy internal guide sleeve, with or without the use of bearing rings. The use of such Expansion Joints will assure installation without initial lateral or angular misalignment and can be installed in pipelines where reverse flow will be encountered. The use of an internally-guided Expansion Joint does not eliminate the necessity of using adequate external pipe guides in accordance with the instructions given in Section 2.10. Its use will not prevent bellows instability.

LATERAL DEFLECTION

The relative displacement of the two ends of an Expansion Joint perpendicular to its longitudinal axis. This has been referred to as lateral offset, lateral movement, parallel misalignment, direct shear, or transverse movement.

LIMIT RODS

Devices, usually in the form of rods or bars, attached to the expansion joint assembly whose primary function is to restrict the bellows movement range (axial, lateral and angular) during normal operation. In the event of a main anchor failure, they are designed to prevent bellows over-extension or over-compression while restraining the full pressure loading and dynamic forces generated by the anchor failure.

MAIN ANCHOR

A main anchor is one which must withstand the full bellows thrust due to pressure, flow, spring forces, and all other piping loads.

A main anchor base for connection to the anchor structure can be furnished as an integral part of a single or double Expansion Joint, if desired. The Expansion Joint manufacturer must be advised of the magnitude and direction of all forces and moments which will be imposed upon the anchor base, so that it can be adequately designed to suit the specific application.

MOTION INDICATORS

Devices attached to an Expansion Joint for the purpose of indicating the movement of the Expansion Joint. These devices are useful in determining if the piping system is behaving as planned and if the actual movements being imposed upon the bellows are within the limits of the original design criteria. An example of motion indicators used on hinge or gimbal hardware is an indicator attached to the hinge pin with an angular scale attached to the hinge arm. This allows one to quickly determine the extent of angular offset.

Another common example of motion indicators is found on slotted hinge assemblies. With the hinge pin used as an indicator, permanent marks are scribed upon the hinge hardware to record the original cold position. The relative distance between the pin and the cold position mark can then be used to determine the movements imposed upon the bellows.

PANTOGRAPH LINKAGES

A scissors-like device. A special form of control rod attached to the expansion joint assembly whose primary function is to positively distribute the movement equally between the two bellows of the universal joint throughout its full range of movement. Pantograph linkages, like control rods, are not designed to restrain pressure thrust.

PIPE ALIGNMENT GUIDE

A pipe alignment guide is a form of framework fastened to some rigid part of the installation which permits the pipe line to move freely only along the axis of the pipe. For further information, see the definition of planar pipe guide below.

PIPE SECTION

A pipe section is that portion of a pipeline between two anchors. All dimensional changes in a pipe section must be absorbed between these two anchors.

PLANAR PIPE GUIDE

A planar pipe guide permits transverse movement and/or bending of the pipeline in one plane. It is commonly used in applications involving lateral deflection or angular rotation resulting from "L" or "Z" shaped piping configurations.

PRESSURE BALANCED EXPANSION JOINT

A pressure balanced Expansion Joint is designed to absorb axial movement and/or lateral deflection while restraining the pressure thrust by means of tie devices inter-connecting the flow bellows with an opposed bellows also subjected to line pressure.

PURGE CONNECTIONS

Purge connections, where required, are usually installed at the sealed end of each internal sleeve of an expansion joint for the purpose of injecting a liquid or gas between the bellows and the internal sleeve to keep the area clear of erosive and corrosive media and/or solids that could pack the convolutions. Purging may be continuous, intermittent or just on start-up or shut down, as required. These are sometimes called aeration connections.

RATED MOVEMENT

The maximum amount of movement (axial extension, axial compression, lateral deflection, angular rotation, or any combination thereof) which an Expansion Joint is capable of absorbing. This rating may be different for each size, type and make of Expansion Joint and is established by the manufacturer.

SHIPPING DEVICES

Rigid support devices installed on an expansion joint to maintain the overall length of the assembly for shipment. These devices may also be used to precompress, pre-extend or laterally offset the bellows. See Section 8.2. They should not be used to resist pressure thrust during testing.

SINGLE EXPANSION JOINT

The simplest form of Expansion Joint, of single bellows construction, for the purpose of absorbing any combination of the three basic movements of the pipe section in which it is installed.

SLOTTED HINGES

Devices installed as diametrically opposed pairs on an Expansion Joint permitting axial and one plane angular movement. Slotted hinges can be designed to perform as control devices, distributing movements between two bellows of a universal Expansion Joint but do not restrain pressure thrust. They may also be designed as limiting devices that restrict the bellows movement range and restrain the full pressure loading and dynamic forces generated by an anchor failure. These devices can be used to transmit extraneous loads and forces such as system dead weight, wind loads, and seismic loads that are transverse to the Expansion Joint axis.

STABILIZER

A device, internally or externally attached to the Expansion Joint assembly, whose primary function is to increase the stability of a universal Expansion Joint assembly.

SWING EXPANSION JOINT

A swing Expansion Joint is one containing two bellows joined by a common connector designed to absorb lateral deflection and/or angular rotation in one plane. Pressure thrust and extraneous forces are restrained by the use of a pair of swing bars, each of which is pinned to the Expansion Joint ends.

TANGENT REINFORCEMENT

A reinforcing member located around the circumference of the bellows tangent for the purpose of reducing excessive pressure stresses which could lead to circumferential yielding.

TANGENTS

The straight un-convoluted portions at the end of the bellows.

TIE RODS

Devices, usually in the form of rods or bars, attached to the expansion joint assembly whose primary function is to continuously restrain the full bellows pressure thrust during normal operation while permitting only lateral deflection. Angular rotation can be accommodated only if two tie rods are used and located 90° opposed to the direction of rotation.

TORSIONAL ROTATION

The twisting of one end of the Expansion Joint with respect to the other end about its longitudinal axis. This twisting generally produces extremely high shear stresses in the bellows. For this reason it is extremely important that special hardware be used to limit the amount of torsional shear stress in the bellows. The equations in Section 4.13.4 may be used as a guide in calculating this stress.

UNIVERSAL EXPANSION JOINT

A universal Expansion Joint is one containing two bellows joined by a common connector for the purpose of absorbing any combination of the three basic movements: axial movement, lateral deflection and angular rotation. Universal Expansion Joints are usually furnished with control rods to distribute the movement between the two bellows of the Expansion Joint and stabilize the common connector. This definition does not imply that only a universal Expansion Joint can absorb combined movement.

WELD ENDS

The ends of an expansion joint equipped with pipe suitably beveled for welding to adjacent equipment or piping.

1.3 NOMENCLATURE

A_c = Cross sectional metal area of one bellows convolution, in.² (mm²)

$$= \left[2\pi(r_m) + 2\sqrt{\left[\frac{q}{2} - 2(r_m)\right]^2 + [w - 2(r_m)]^2} \right] t_p n \quad \text{for round bellows}$$

$$= \left[2\pi(r_m) + 2\sqrt{\left[\frac{q}{2} - 2(r_m)\right]^2 + [w - 2(r_m)]^2} \right] t \quad \text{for rectangular bellows}$$

A_e = Bellows effective area, corresponding to the mean diameter of the convolutions of the Expansion Joint, in.² (mm²)

$$= \pi \frac{(D_m)^2}{4}$$

A_f = Cross sectional metal area of one reinforcement fastener, in.² (mm²)

A_p = Internal area of pipe, in.² (mm²)

A_r = Cross sectional metal area of one bellows reinforcing member, in.² (mm²)

A_{tc} = Cross sectional metal area of one tangent collar, in.² (mm²)

A_{tp} = Cross sectional metal area of the pipe based on length L_p , in.² (mm²)

A_{tr} = Cross sectional metal area of the reinforcing ring based on length L_r , in.² (mm²)

B_1 = Factor used in specific design calculations to relate toroidal bellows convolution segment behavior to a simple strip beam

B_2 = Factor used in specific design calculations to relate toroidal bellows convolution segment behavior to a simple strip beam

B_3 = Factor used in specific design calculations to relate toroidal bellows convolution segment behavior to a simple strip beam

C_a = 2.0 when tangent is fully supported against the pressure

= 1.5 when tangent is not fully supported against the pressure

C_c = Factor used to account for curvature of tangent collar

$$= -0.2431 + 0.0168n_g + 0.3024n_g^2$$

C_d = Factor used in specific design calculations to relate U-shaped bellows convolution segment behavior to a simple strip beam

C_f = Factor used in specific design calculations to relate U-shaped bellows convolution segment behavior to a simple strip beam

C_m = Material strength factor at temperatures below the creep range

= 1.5 for bellows in the annealed condition (without cold work)

= 1.5 Y_{sm} (1.5 min., 3.0 max.) for bellows in the as-formed condition (with cold work)

C_p = Factor used in specific design calculations to relate U-shaped bellows convolution segment behavior to a simple strip beam

C_r = Convolution height factor

$$= 0.36 \ln \left(\frac{w}{e} \right)$$

C_{sf} = Stress concentration factor derived from manufacturer's fatigue test results. It is a function of corner configuration and weld joint efficiency.

C_{sp} = Stress concentration factor derived from manufacturer's fatigue test results. It is a function of the effect of applied pressure.

- C_w = Longitudinal weld joint efficiency factor from applicable code. Subscripts b , c , f , p , and r denote the bellows, reinforcement collar, fastener, pipe, and reinforcing ring material, respectively.
- C_θ = Column instability pressure reduction factor based on imposed angular rotation
 = Lesser of R_θ or 1.0 for single bellows
 = 1.0 for universal bellows
- D_b = Inside diameter of cylindrical tangent and bellows convolutions, in. (mm)
- D_c = Mean diameter of bellows tangent reinforcing collar, in. (mm)
 = $D_b + 2nt + t_c$
- D_i = Pipe inside diameter, in. (mm)
- D_m = Mean diameter of bellows convolutions, in. (mm)
 = $D_b + w + nt$ for “U” profile
- D_n = Tie rod nut or welded tie rod ring outside diameter, in. (mm)
- D_p = Mean diameter of pipe or hinge pin outside diameter, in. (mm)
- D_r = Mean diameter of reinforcing ring, in. (mm)
- E = Modulus of Elasticity at design temperature, unless otherwise specified, for material, psi (MPa). Subscripts b , c , f , p , s , and r denote the bellows, reinforcement collar, fastener, pipe sleeve, and reinforcing ring material, respectively.
- F = Axial force required to move a single convolution axially the amount of e_x , lbf (N)
- F_a = Axial force at the end of the convoluted length of an Expansion Joint resulting from axial deflection x , lbf (N)
- F_g = Axial force per tangent collar gusset, lbf (N)
 = $\frac{1}{n_g} \left[0.25\pi (D_m^2 - D_b^2) P + e_c f_w \right]$ below the creep range
 = $\frac{0.25\pi P}{n_g} (D_m^2 - D_b^2)$ in the creep range
- F_l = Lateral force from all the tie rods, lbf (N)
- F_t = Total axial force on all the restraint hardware including pressure thrust and all external loads, lbf (N)
- G = Modulus of Rigidity at design temperature for material, psi (MPa)
- H = Resultant total internal pressure force acting on the bellows and reinforcement for one convolution, lbf (N)
 = $PD_m q$
- H_t = Hold time at temperature between cycles, hours

I = Moment of inertia of rectangular bellows convoluted cross-section, in⁴ (mm⁴)

$$= N \left[\frac{t(2w-q)^3}{48} + 0.4qt(w-0.2q)^2 \right] \text{ for "U" profile}$$

$$= N \left[\frac{t(w-2r_m)^2 \sqrt{4(w-2r_m)^2 + (q-4r_m)^2}}{12} + 1.6r_mt(w-0.7r_m)^2 \right] \text{ for "V" profile}$$

I_p = Moment of inertia of pipe cross section, in.⁴ (mm⁴)

K_2 = Inplane instability factor

$$= \frac{S_2}{P}$$

K_4 = Inplane instability factor

$$= \frac{C_p}{2n} \left[\frac{w}{t_p} \right]^2$$

K_f = Forming method factor

= 1 for expanding mandrel or roll forming

= 0.6 for hydraulic, elastomeric, or pneumatic tube forming

K_r = Circumferential stress factor

= The greater of the following but not less than 1.0

$$\frac{2(q+e_x) + K_\theta e_\theta + e_y}{2q} \text{ where } e_x \text{ and } e_y \text{ are based on axial extension concurrent with pressure P}$$

$$\frac{2(q-e_x) + K_\theta e_\theta + e_y}{2q} \text{ where } e_x \text{ and } e_y \text{ are based on axial compression concurrent with pressure P}$$

K_s = Shape factor for cross section (see Table III)

$$= \frac{Nw}{2I} \left[\frac{t(w-2r_m) \sqrt{4(w-2r_m)^2 + (q-4r_m)^2}}{4} + 3.1416r_mt(w-0.7268r_m) \right] \text{ for rectangular bellows}$$

K_{sr} = Overall bellows spring rate, lbf/in. (N/mm)

$$= \frac{f_i}{N}$$

K_u = Factor establishing relationship between equivalent axial displacement per convolution due to lateral deflection and the ratio $L_u/(2L_b)$

K_{um} = Factor for determining the moment reaction for a universal expansion joint with angular rotation about one end

$$= \frac{0.0123 + 1.958R_u^{-1.1528}}{2.9359 + R_u^{-1.1528}}$$

$K_{u\theta}$ = Factor for determining the moment and equivalent axial movement for a universal expansion joint with angular rotation about one end

$$= \frac{-0.6042 + 2R_u^{1.1598}}{0.3914 + R_u^{1.1598}}$$

K_{uv} = Factor for determining the lateral force for a universal expansion joint with angular rotation about one end

$$= 0.7713(1.2876)^{\frac{1}{R_u}} R_u^{-1.0641}$$

K_{θ} = Angular rotation internal pressure effect factor

$$= \frac{e_{\theta} + e_{yp}}{e_{\theta}} \text{ for single bellows}$$

$$= 1.0 \text{ for universal bellows}$$

$K_{\theta l}$ = Lateral Deflection Pitch Change Factor

$$= 1 + 0.24y \left(\frac{L_b}{D_m} \right)^{1.33} \text{ for single bellows, in.}$$

$$= 1 + 0.0094y \left(\frac{L_b}{D_m} \right)^{1.33} \text{ for single bellows, mm}$$

$$= 1.0 \text{ for universal bellows}$$

L_b = Bellows convoluted length, in. (mm)

$$= Nq$$

L_c = Bellows tangent collar length, in. (mm)

L_d = Maximum length from the attachment weld to the center of the first convolution for externally attached bellows, in. (mm)

L_f = Effective length of one reinforcing ring fastener, in. (mm)

L_g = Maximum distance across the inside opening of a toroidal convolution considering all movements, in. (mm)

L_l = Mean length of long side of rectangular bellows, in. (mm)
= long inside length + convolution height

L_{ml} = Effective length of long side, in. (mm)

$$= \frac{L_l}{3} \left(\frac{3L_s + L_l}{L_s + L_l} \right)$$

L_{ms} = Effective length of short side, in. (mm)

$$= \frac{L_s}{3} \left(\frac{3L_l + L_s}{L_s + L_l} \right)$$

L_p = Effective pipe length, in. (mm)

$$= \frac{1}{3} \sqrt{D_p t_{pe}}$$

L_{pm} = Minimum required pipe length having thickness t_{pe} , in. (mm)

$$= 1.5 \sqrt{D_p t_{pe}}$$

L_r = Effective reinforcing ring length, in. (mm)

$$= \frac{1}{3} \sqrt{D_r t_r}$$

L_{rt} = Overall length of the reinforcing ring, in. (mm)

L_s = Mean length of short side of rectangular bellows, in. (mm)
= short inside length + convolution height

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L_t = Bellows tangent length, in. (mm)

L_{tm} = Maximum length of bellows tangent that extends past the collar, in. (mm)

$$= 1.5 \sqrt{\frac{nt^2 S_{ab}}{P}}$$

L_{tr} = Length of tie rods between attachment locations, in. (mm)

L_u = Distance between outermost ends of the convolutions in a universal Expansion Joint, in. (mm)

$$L^* = L_u - L_b$$

M_{Li} = Moment at the ends of the convoluted length of an Expansion Joint resulting from lateral deflection, y_l , parallel to the long side, lbf-in (N•mm)

M_{Ls} = Moment at the ends of the convoluted length of an Expansion Joint resulting from lateral deflection, y_s , parallel to the short side, lbf-in (N•mm)

M_l = Moment at the ends of the convoluted length of an Expansion Joint resulting from lateral deflection, y , lbf-in (N•mm)

M_p = Maximum resisting frictional moment from both hinge pins (in-lbs)

M_θ = Moment at the ends of the convoluted length of an Expansion Joint resulting from angular rotation, θ , lbf-in (N•mm)

$M_{\theta l}$ = Moment at the ends of the convoluted length of an Expansion Joint resulting from angular rotation, θ_l , of the long side, lbf-in (N•mm)

$M_{\theta s}$ = Moment at the ends of the convoluted length of an Expansion Joint resulting from angular rotation, θ_s , of the short side, lbf-in (N•mm)

N = Number of convolutions in one bellows

N_c = Fatigue life, number of cycles to failure

P = Pressure, psig (MPa)

P_d = Design pressure based on the most severe conditions, whether operational or test, psig (MPa)

P_{sc} = Limiting internal design pressure based on column instability, psig (MPa)

P_{si} = Limiting design pressure based on inplane instability and local plasticity, psig (MPa)

P_t = Test pressure, psig (MPa).

R = Ratio of the internal pressure force resisted by the bellows to the internal pressure force resisted by the reinforcement. Use R_1 or R_2 as designated in the equations.

$$= R_1 \text{ for integral reinforcing members} = \frac{A_c E_b}{A_r E_r}$$

$$= R_2 \text{ for reinforcing members joined by fasteners} = \frac{A_c E_b}{D_m} \left(\frac{L_f}{A_f E_f} + \frac{D_m}{A_r E_r} \right)$$

R_s = Spherical radius of spherical washers, in. (mm)

$$R_u = \frac{L_u}{2L_b}$$

R_θ = Limiting internal design pressure ratio for single bellows

$$= \frac{1.18N^2(q \pm e_x)^2}{\pi^2 D_m K_{\theta l} \sin(\theta/2)(L_b \pm x)} \quad \text{with imposed angular rotation}$$

where $+e_x$ and $+x$ are axial extension; $-e_x$ and $-x$ are axial compression

$$= 1.0 \text{ with no imposed angular rotation}$$

S_a = Allowable material stress at design temperature, unless otherwise specified, from the applicable code, psi. (MPa). Subscripts b , c , f , p , and r denote bellows, reinforcement collar, fastener, pipe, and reinforcing member material.

S_c = Allowable stress of pipe/vessel material at test temperature, psi. (MPa)

S_h = Allowable stress of pipe/vessel material at design temperature, psi. (MPa)

S_y = Yield strength at design temperature, unless otherwise determined, of the actual bellows material after completion of bellows forming and any applicable heat treatment, psi. (MPa)

$$= \frac{0.67 C_m S_{ym} S_{yh}}{S_{yc}}$$

S_{yc} = Yield strength at room temperature of the bellows material in the annealed condition from the applicable code or standard reference, psi. (MPa)

S_{yh} = Yield strength at design temperature of the bellows material in the annealed condition from the applicable code or standard reference, psi. (MPa)

S_{ym} = Yield strength at room temperature of the actual bellows material in the annealed condition from the certified test report, psi. (MPa)

$T_{inst.}$ = Installation temperature, °F (°C)

$T_{max.}$ = Maximum design temperature, °F (°C)

$T_{min.}$ = Minimum design temperature, °F (°C)

V_{Ll} = Lateral force at the ends of the convoluted length of the Expansion Joint resulting from lateral deflection, y_l , in a direction parallel to the long side, lbf (N)

V_{Ls} = Lateral force at the ends of the convoluted length of the Expansion Joint resulting from lateral deflection, y_s , in a direction parallel to the short side, lbf (N)

V_l = Lateral force at the ends of the convoluted length of the Expansion Joint resulting from lateral deflection, y , lbf (N)

W = Elevated temperature weld joint strength reduction factor from applicable design code. Subscripts b , c , and r denote bellows, reinforcement collar, and reinforcing ring material, respectively.

W_{cs} = Total dead weight of the center spool including pipe, refractory, insulation, attachments, and media, lbf (N)

X, Y, Z = Lengths in coordinate directions

Y_{sm} = Yield strength multiplier

$$= 1 + 9.94 \cdot 10^{-2}(K_f \varepsilon_f) - 7.59 \cdot 10^{-4}(K_f \varepsilon_f)^2 - 2.4 \cdot 10^{-6}(K_f \varepsilon_f)^3 + 2.21 \cdot 10^{-8}(K_f \varepsilon_f)^4 \text{ for austenitic stainless steel}$$

$$= 1 + 6.8 \cdot 10^{-2}(K_f \varepsilon_f) - 9.11 \cdot 10^{-4}(K_f \varepsilon_f)^2 + 9.73 \cdot 10^{-6}(K_f \varepsilon_f)^3 - 6.43 \cdot 10^{-8}(K_f \varepsilon_f)^4 \text{ for nickel alloys}$$

$$= 1 \text{ for other materials. Higher values may be used if supported by test data.}$$

Z_c = Section modulus of tangent collar about the neutral axis in the lateral direction, in.³ (mm³)

- e = Total equivalent axial movement per convolution, in. (mm)
 e_c = Equivalent axial compression per convolution, in. (mm)
 e_e = Equivalent axial extension per convolution, in. (mm)
 e_x = Axial movement per convolution resulting from imposed axial movement, x . This movement may be measured as compression or extension, in. (mm).
 e_y = Axial movement per convolution resulting from imposed lateral deflection, y , in. (mm)
 e_{yl} = Axial movement per convolution for a rectangular bellows resulting from imposed lateral deflection, y , in a direction parallel with the long side, in. (mm)
 e_{yp} = Axial movement per convolution resulting from internal pressure on a single bellows with imposed angular rotation, in. (mm)
 e_{ys} = Axial movement per convolution for a rectangular bellows resulting from imposed lateral deflection, y , in a direction parallel with the short side, in. (mm)
 e_θ = Axial movement per convolution resulting from imposed angular rotation, θ , in. (mm)
 $e_{\theta l}$ = Axial movement per convolution for a rectangular bellows resulting from imposed angular rotation, θ , in a direction parallel with the long side, in. (mm)
 $e_{\theta s}$ = Axial movement per convolution for a rectangular bellows resulting from imposed angular rotation, θ , in a direction parallel with the short side, in. (mm)
 f_c = Factor for modification of the lower bound of the fatigue curve
 = 1 when providing EJMA calculations unless otherwise indicated by specification
 f_i = Bellows theoretical initial axial elastic spring rate per convolution, lbf/in. (N/mm) of movement per convolution. Subscripts u , r , and t denote unreinforced, reinforced, and toroidal bellows respectively.
 f_w = Bellows working spring rate of movement per convolution, lbf/in. (N/mm)
 = f_i for $S_t \leq 1.5S_y$
 = $0.67f_i$ for $S_t > 1.5S_y$
 g = Acceleration due to gravity, 32.2 ft/sec² (9.8 m/sec²)
 k = A factor which considers the stiffening effect of the attachment weld and the end convolution on the pressure capacity of the bellows tangent
 = $\frac{L_t}{1.5\sqrt{D_b t}}$ If $k \geq 1$, use $k = 1$
 m = Mass, lbm (kg)
 n = Number of bellows material plies of thickness, t
 n_g = Number of equally spaced gussets per tangent collar
 q = Convolution pitch, the distance between corresponding points of any two adjacent convolutions in a bellows, in. (mm)
 r = Mean radius of toroidal bellows convolution, in. (mm)
 r_{ic} = The crest convolution inside radius, in. (mm)
 r_{ir} = The root convolution inside radius, in. (mm)
 r_m = Mean radius of bellows convolution, in. (mm)
 = $\frac{r_{ic} + r_{ir} + nt}{2}$
 t = Bellows nominal material thickness of one ply, in. (mm)
 t_c = Bellows tangent reinforcing collar material thickness, in. (mm)

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$$t_e = 0.778t \left(\frac{Nq + 2L_t}{Nw} \right)^{\frac{1}{3}} \text{ in. (mm)}$$

Note: if tangent is fully supported against the pressure, set $L_t = 0$

t_p = Bellows material thickness for one ply, corrected for thinning during forming, in. (mm)

$$= t \sqrt{\frac{D_b}{D_m}} \text{ for bellows formed from tubes with inside diameter equal to } D_b$$

= t for rectangular expansion joint rails

t_{pe} = Pipe thickness, in. (mm)

t_r = Reinforcing ring thickness, in. (mm)

v = Velocity of media flow, ft/sec (m/sec)

w = Convolution height (see Figures 4.13, 4.14, and 5.9), in. (mm)

x = Applied axial movement in compression or extension, in. (mm)

y = Applied lateral deflection, in. (mm)

y_{bml} = Bellows beam mode deflection due to pressure at the center of long span and mid-point of bellows live length, in. (mm)

y_{bms} = Bellows beam mode deflection due to pressure at the center of short span and mid-point of bellows live length, in. (mm)

y_l = Applied lateral deflection in a direction parallel with the long side, in. (mm)

y_s = Applied lateral deflection in a direction parallel with the short side, in. (mm)

α = Inplane instability stress interaction factor

$$= 1 + 2\delta^2 + (1 - 2\delta^2 + 4\delta^4)^{0.5}$$

δ = Inplane instability stress ratio

$$= \frac{K_4}{3K_2}$$

ε_f = Bellows forming strain (%)

$$= 100 \sqrt{\left[\ln \left(1 + \frac{2w}{D_b} \right) \right]^2 + \left[\ln \left(1 + \frac{nt_p}{2r_m} \right) \right]^2} \quad \text{For bellows formed from tubes with an inside diameter of } D_b$$

μ_p = Coefficient of static friction for the hinge pin connections

μ_s = Coefficient of static friction for the spherical washers

θ = Applied angular rotation per individual bellows, rad

θ_c = Angle of rotation for an unrestrained center spool, rad

θ_l = Applied angular rotation per individual bellows in a plane parallel with the long side, rad

θ_s = Applied angular rotation per individual bellows in a plane parallel with the short side, rad

θ_u = Angle of the universal expansion joint centerline with respect to horizontal, rad

ν = Poisson's Ratio

ρ = Density of fluid, lbm/in.³ (kg/m³)

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SECTION 2 - SELECTION AND APPLICATIONS**2.1 SELECTION OF EXPANSION JOINTS**

The first step in the selection of Expansion Joints is to choose tentative locations for the pipe anchors. Any piping system, regardless of its complexity, can be divided into a number of individual expanding pipe sections having relatively simple configurations (ie: straight runs, "L" shaped bends, "Z" shaped bends and other means), by means of anchors. The number of pipe anchors selected, as well as their locations, will depend upon the piping configuration, the amount of expansion which can be accommodated by a single Expansion Joint, the availability of structural members suitable for use as anchors, the location of various pipe fittings, the location of connected equipment, the location of branch connections and other considerations.

The major pieces of connected equipment such as turbines, pumps, compressors, heat exchangers, reactors, and similar devices can be considered as anchors in most applications. It is usually necessary to supplement these equipment anchor points by locating additional anchors at valves, at changes in the direction of the pipe, at blind ends of pipe and at major branch connections. It is generally advisable to start out with the assumption that the use of single and double Expansion Joints in straight axial movement will provide the simplest and most economical layout, unless there are obvious advantages to be gained from another approach.

After the anchor points have been tentatively located, the resulting pipe configurations should be reviewed to determine whether they conform to the standard pipe sections shown in Sections 2.2 and 2.10. At this point, consideration should be given to the relative merits of systems utilizing single and double Expansion Joints for axial movement only, as opposed to those utilizing universal, pressure balanced, hinged and gimbal Expansion Joints. A final decision on anchor locations and the types of Expansion Joints to be used can only be made after a comparison of various alternative solutions. Cost, the ability to comply with cyclic life and force requirements, space restrictions, and similar items should be considered.

The next step is to calculate the actual change in length of each leg of each individual pipe section due to temperature changes. The minimum and installation temperatures are assumed to be 70° F unless otherwise specified. An allowance, added by the system designer, should then be included in the actual calculated movements to account for the following possibilities:

- (a) The minimum and/or installation temperatures used in the design calculations may have been based on the erroneous assumption that the metal temperature of the pipe is the same as the ambient temperature.
- (b) During erection of the piping, it may be necessary to relocate some of the anchor points because of construction problems encountered at the job site.
- (c) During operation the system may be subject to a different temperature range than the designer anticipated, particularly during start-up.

Refer to Appendix J Example 11 for a sample calculation.

2.2 SELECTION FOR AXIAL MOVEMENT ONLY

(For an explanation of the symbols used in the diagrams, refer to Appendix B.)

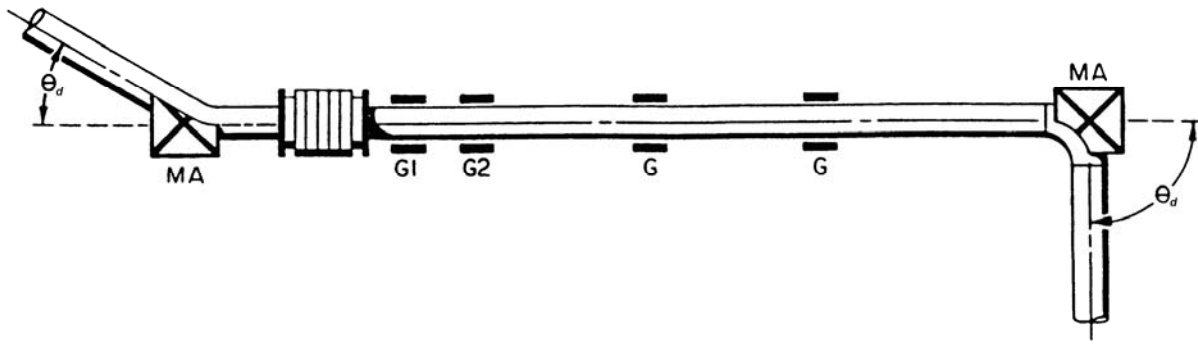


FIGURE 2.1

Figure 2.1 typifies good practice in the use of a single Expansion Joint to absorb axial pipe line expansion. Note the use of one Expansion Joint between two main anchors (MA), the nearness of the Expansion Joint to an anchor, the closeness of the first alignment guide (G1), the spacing between the first alignment guide and the second alignment guide (G2), and the spacing of intermediate guides (G) along the balance of the line. See Figures 2.30 and 2.31, and/or equation (2-7).

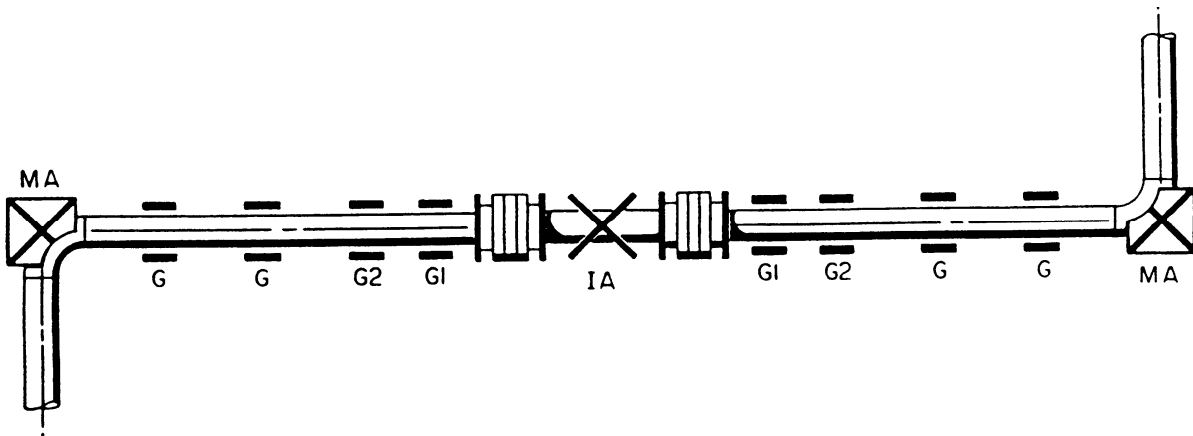


FIGURE 2.2

Figure 2.2 typifies good practice in the use of a double Expansion Joint to absorb axial pipe line expansion. Note the addition of the intermediate anchor (IA) which, in conjunction with the two main anchors, divides the pipe line into individual expanding sections, so that there is only one Expansion Joint between any two anchors. Note also the closeness of the first alignment guide (G1) to each Expansion Joint, the spacing between the first alignment guide and the second alignment guide (G2) and the spacing of intermediate guides (G) along the balance of each pipe section. See Figures 2.30 and 2.31 and/or equation (2-7).

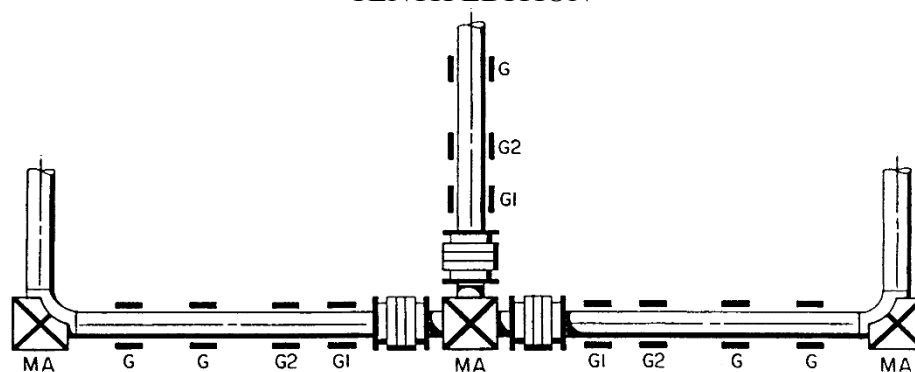


FIGURE 2.3

Figure 2.3 typifies good practice in the use of Expansion Joints to absorb axial pipe line expansion in a pipe line with a branch connection. The anchor at the junction, which in this case is a tee, is a main anchor (MA) designed to absorb the thrust from the Expansion Joint in the branch line. Note the nearness of each Expansion Joint to an anchor, the closeness of each first alignment guide (G1), the spacing between the first alignment guide and the second alignment guide (G2) and the spacing of intermediate guides (G) along the balance of each pipe section. See Figures 2.30 and 2.31 and/or equation (2-7).

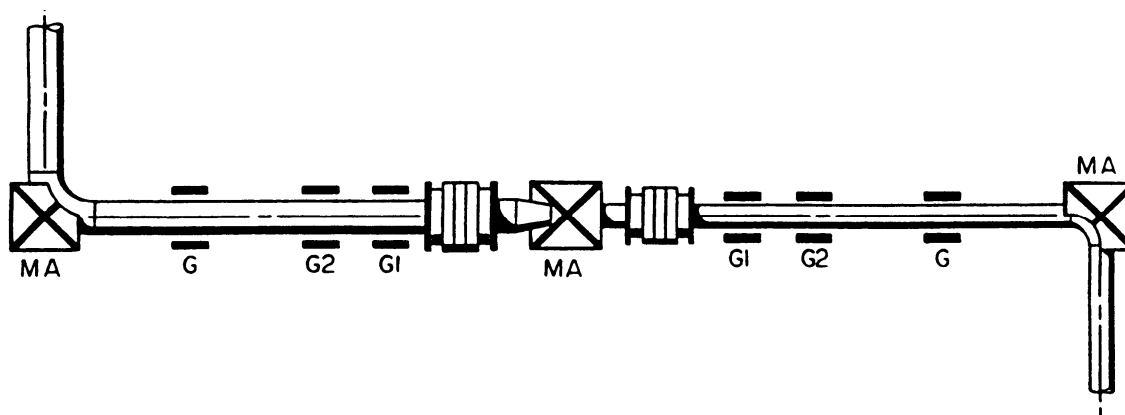


FIGURE 2.4

Figure 2.4 typifies good practice in the use of Expansion Joints to absorb axial pipe line expansion in a pipe line containing a reducer. The anchor at the reducer is a main anchor (MA) designed to absorb the difference in the thrusts of the Expansion Joints on each side of the reducer. Note the nearness of each Expansion Joint to an anchor, the closeness of each first alignment guide (G1), the spacing between the first alignment guide and the second alignment guide (G2) and the spacing of intermediate guides (G) along the balance of each pipe section. See Figures 2.30 and 2.31 and/or equation (2-7).

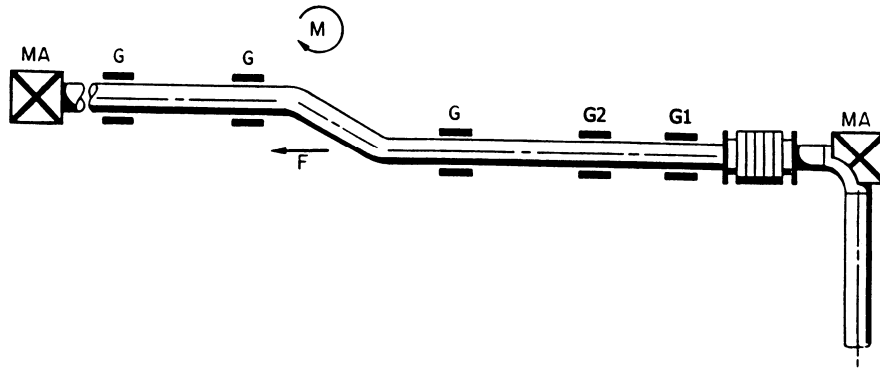


FIGURE 2.5

Figure 2.5 shows the application of a single Expansion Joint to a pipe line containing an offset. It should be noted that applications of this type are not usually recommended and will perform satisfactorily only within certain limits. As in Figure 2.1, the line is provided with main anchors at each end to absorb the pressure, movement loading, and guide friction. Where the line contains an offset, this load must first be transmitted through the offset leg, resulting in a moment on the piping. Where the line size is small, the offset appreciable, or where the pressure and movement forces are relatively high, this configuration may result in over-stressing, or distortion of the piping and guides.

Note the nearness of the Expansion Joint to an anchor (MA), the closeness of the first alignment guide (G1), the spacing between the first alignment guide and the second alignment guide (G2) and the spacing of intermediate guides (G) along the balance of the line. Guides should be installed near both ends of the offset leg to minimize the effects of the bending moment on the system. For spacing of other guides, see guide chart Figure 2.31, and/or equation (2-7).

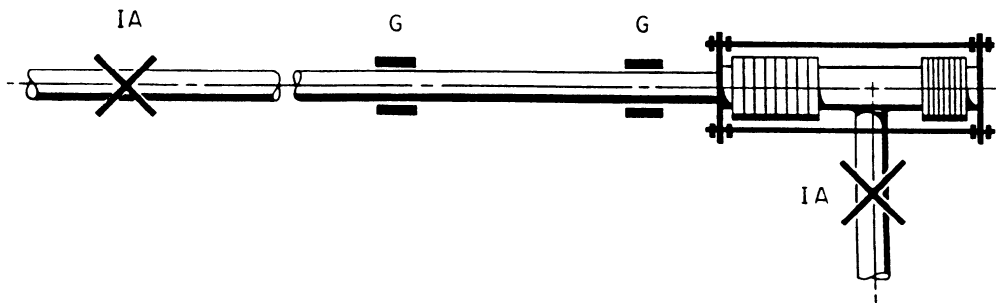


FIGURE 2.6

Figure 2.6 typifies good practice in the use of a pressure balanced Expansion Joint to absorb axial pipe line expansion. Note that the Expansion Joint is located at a change in direction of the piping and that the elbow and the end of the pipe line are secured by intermediate anchors. Since the pressure thrust is absorbed by the Expansion Joint itself, and only the forces required to deflect the Expansion Joint are imposed on the piping, a minimum of guiding is required. Frequently, directional guiding adjacent to the Expansion Joint, as shown, may suffice. In long, small-diameter pipe lines, additional guiding may be necessary.

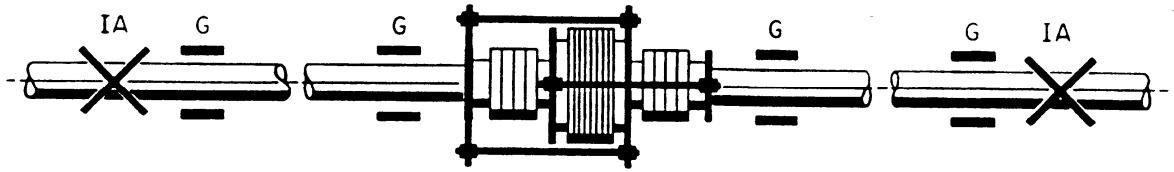


FIGURE 2.7

Figure 2.7 shows the use of an in-line pressure balanced Expansion Joints to absorb axial pipe line movements in a long, straight piping run. By utilizing this arrangement the two anchors shown are relieved of pressure loading and are designed as intermediate anchors. Since the piping is relieved of compressive pressure loading, a minimum of guiding is required, primarily to direct the thermal expansion of the piping into the Expansion Joints in an axial direction.

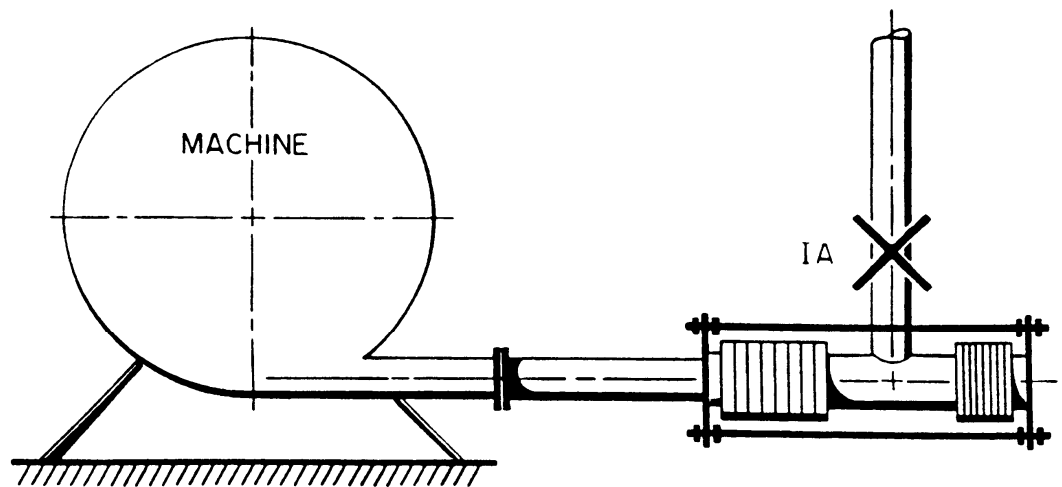


FIGURE 2.8

Figure 2.8 typifies good practice in the use of a pressure balanced Expansion Joint to absorb the thermal expansion of equipment such as turbines, pumps, compressors, etc. The primary function of the Expansion Joint is to minimize loading upon the equipment casing. Note that only an intermediate anchor is required at the change of piping direction and that, if the Expansion Joint is located immediately adjacent to the machine, no guiding is required. Care should be taken to provide sufficient flexibility in both the flow bellows and the balancing bellows, so that the forces required to compress the Expansion Joint do not exceed loading limits for the equipment as established by the equipment manufacturer. See Section 2.6 for further information.

2.3 SELECTION FOR LATERAL DEFLECTION, ANGULAR ROTATION AND COMBINED MOVEMENTS

The selection and proper application of Expansion Joints for lateral deflection, angular rotation and combined movements, involves the evaluation of a number of variables. These can include the piping configuration, the operating conditions, desired cyclic life, load limitations upon piping and equipment, and available supporting structure. In some cases, two or more types of Expansion Joints may be suitable for a particular application. The selection then becomes purely an economic one. More frequently one or the other of the available designs possesses unique characteristics which make it particularly suitable for a given application.

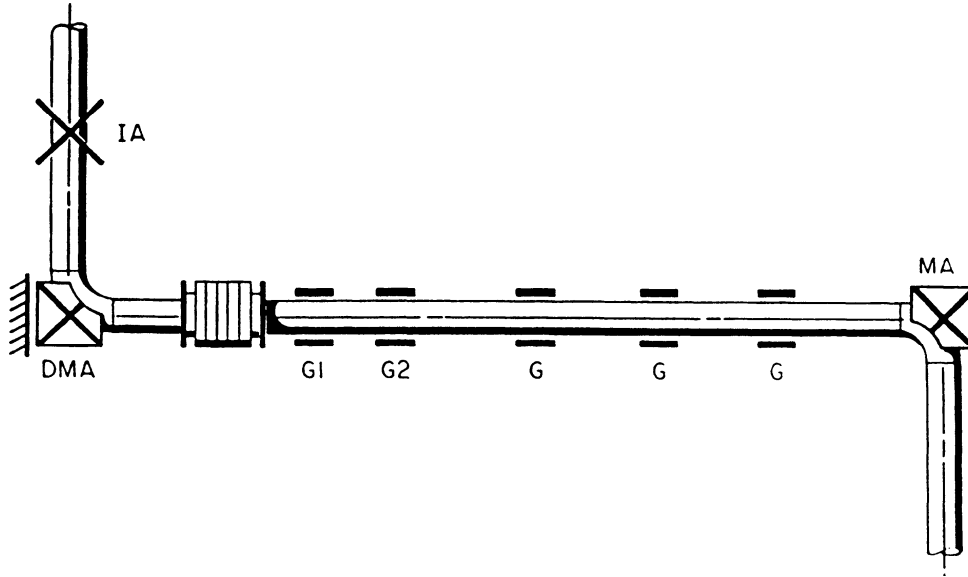
2.4 APPLICATIONS USING SINGLE EXPANSION JOINTS

FIGURE 2.9

The single Expansion Joint is usually considered first for any application because it offers the lowest Expansion Joint cost. Figure 2.9 shows a typical application of a single Expansion Joint absorbing combined axial movement and lateral deflection. The system closely resembles the arrangements shown for axial movement only in the preceding section.

The Expansion Joint is located at one end of the long piping leg with main anchors at each end and guides properly spaced for both movement control and protection of the piping against buckling. The anchor at the left end of the line is a directional main anchor (DMA) which, while absorbing the main anchor loading in the direction of the Expansion Joint axis, permits the thermal expansion of the short piping leg to act upon the Expansion Joint as lateral deflection. Because the main anchor loading exists only in the piping segment containing the expansion joint, the anchor at the end of the shorter piping leg is an intermediate anchor.

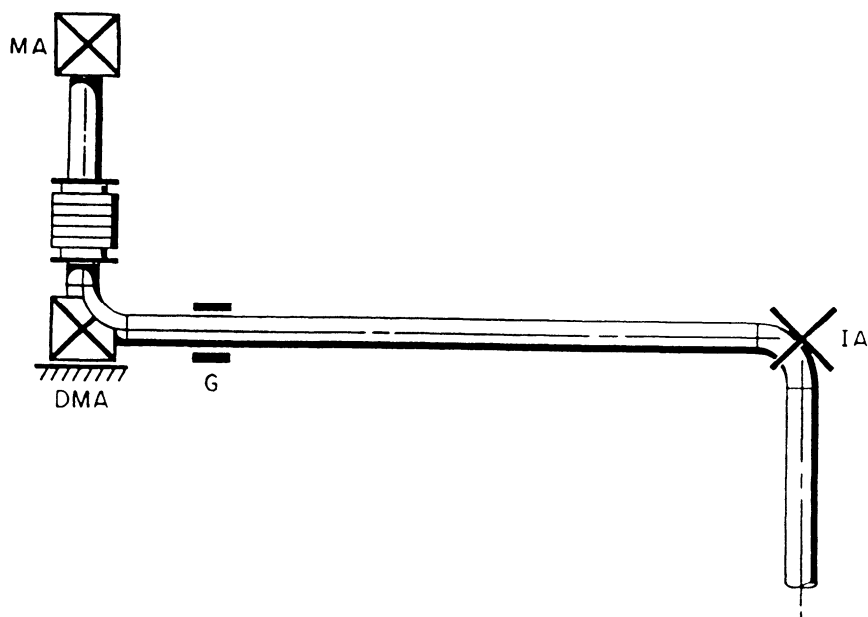


FIGURE 2.10

Figure 2.10 shows an alternate arrangement in which the Expansion Joint is installed in the short piping leg and the principal expansion is absorbed as lateral deflection. The longer piping leg is free of compressive pressure loading and requires only an intermediate anchor and directional guiding. The functions of the directional main anchor and the pipe guide may be combined in a single device.

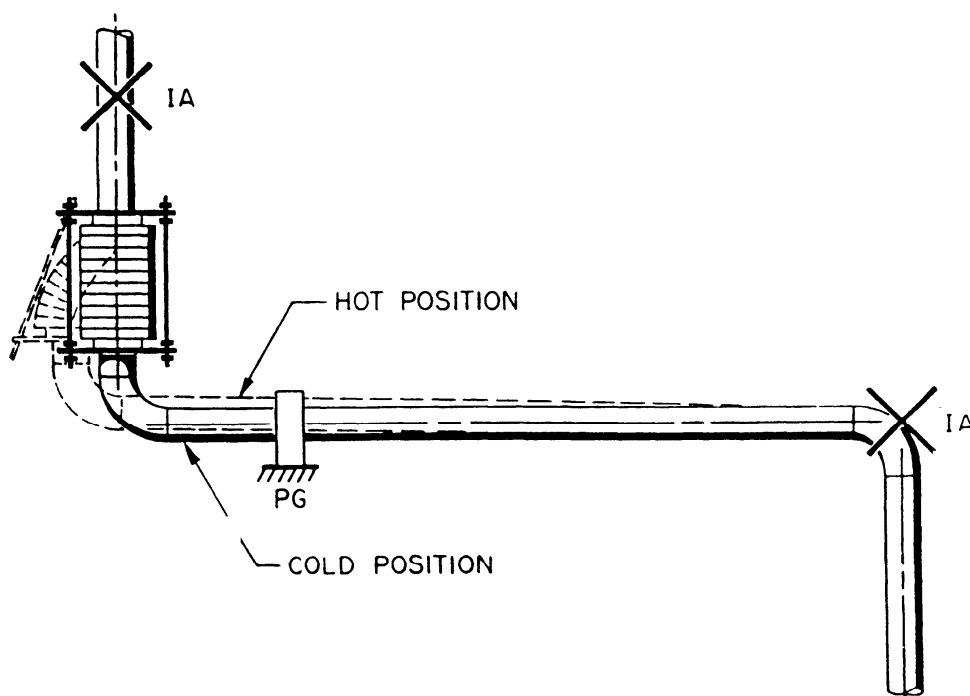


FIGURE 2.11

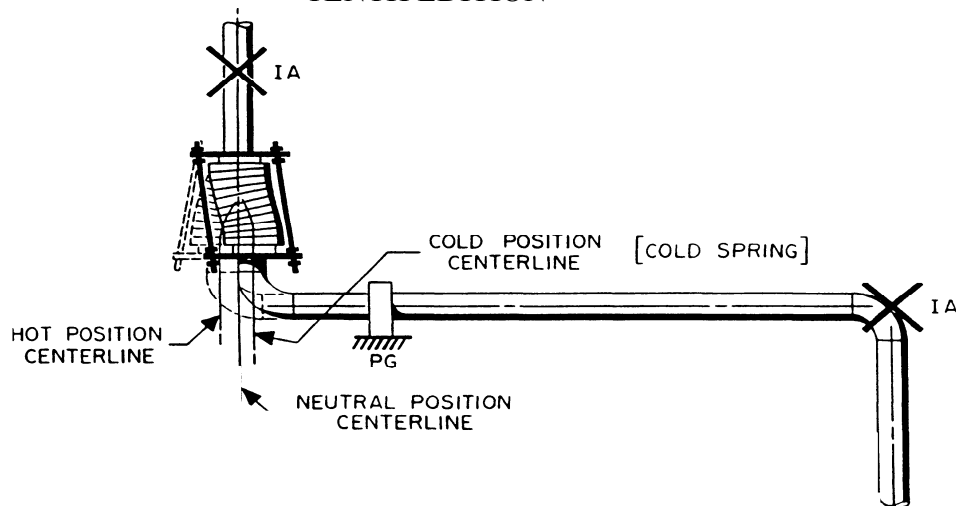


FIGURE 2.12

Figures 2.11 and 2.12 represent modifications of Figure 2.10 in which the main anchors at either end of the Expansion Joint are replaced by tie rods. Where the piping configuration permits, the use of tie rods adjusted to prevent axial movement frequently simplifies and reduces the cost of the installation. Because of these tie rods, the Expansion Joint is not capable of absorbing any axial movement other than its own thermal expansion. The thermal expansion of the piping in the shorter leg is, as a result, imposed as deflection on the longer piping leg. Where the longer piping leg is not sufficiently flexible and where the dimension of the shorter leg is suitable, tie rods may be installed spanning the entire short leg so that no deflection is imposed on the longer run from this source.

Where appreciable amounts of lateral deflection are imposed upon the Expansion Joint, some shortening of the Expansion Joint results from the displacement of the tie rods as shown in Figure 2.11. Care should be taken to insure that sufficient piping flexibility exists to absorb this deflection and that adequate clearances are provided in the guide to permit deflection of the piping. The amount of this deflection can be minimized by cold springing the Expansion Joint in the lateral direction as shown in Figure 2.12.

The principal restriction upon the use of single Expansion Joints for lateral deflection or combined axial movement and lateral deflection is the limited amount of lateral deflection which such an Expansion Joint can absorb. The allowable lateral deflection is directly proportional to the ratio of convoluted length to diameter which, in turn, is restricted by considerations of stability and manufacturing limitations. While eminently suitable for applications such as Figure 2.9 where the principal movement is axial, the relatively small available lateral movement severely limits the type of application illustrated in Figures 2.10, 2.11 and 2.12. Where operating pressures and temperatures are high, or where availability of suitable structures precludes the use of main anchors and multiple guides, the application shown in Figure 2.9 may not be feasible and another type of Expansion Joint may result in far more economical installation.

2.5 APPLICATIONS USING UNIVERSAL EXPANSION JOINTS

The universal Expansion Joint is particularly well adapted to the absorption of lateral deflection. In addition, this design may be used to absorb axial movement, angular rotation or any combination of the three. A common application of the universal Expansion Joint is its use as a tied Expansion Joint in a 90 degree piping offset with the tie rods adjusted to prevent external axial movement. Two such applications are shown in Figures 2.13 and 2.14.

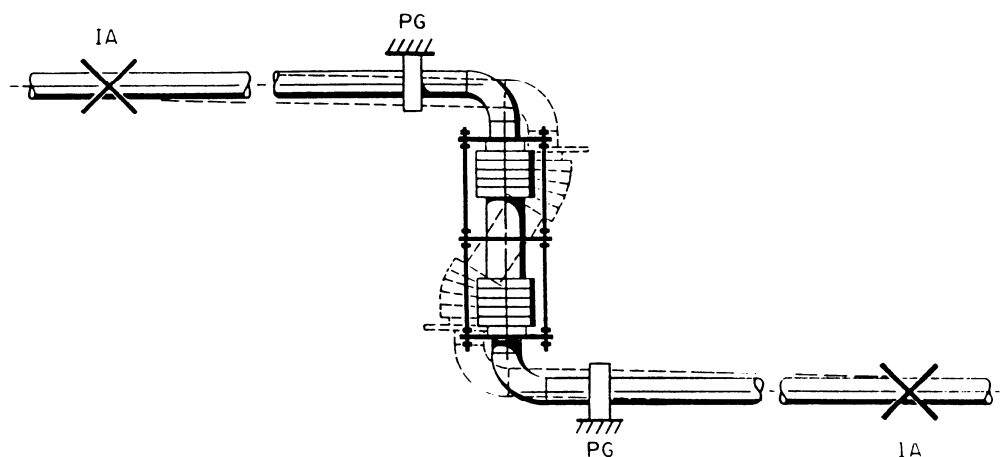


FIGURE 2.13

Figure 2.13 shows a tied universal Expansion Joint used to absorb lateral deflection in a single plane "Z" bend. Where dimensionally feasible, the Expansion Joint should be designed to fill the entire offset leg so that its expansion is absorbed within the tie rods as axial movement. The tie rod should be extended to the elbow center line when practical. The thermal movement of the horizontal lines is absorbed as lateral deflection by the Expansion Joint.

Both anchors are intermediate anchors since the pressure loading is absorbed by the tie rods. Only directional guiding is required since the compressive load on the pipe consists only of the force necessary to deflect the Expansion Joint. Any thermal expansion of the offset leg external to the tie rods, such as that of the elbows at either end, must be absorbed by bending of the horizontal pipe legs. Provision should be made in the design of the guides to allow for both this deflection and the reduced length of the Expansion joint in its deflected position. In addition, particularly in the case of long universal Expansion Joints under high pressure, additional allowance may be necessary to compensate for stretching of the tie rods under load. The Expansion Joint manufacturer should be consulted for recommended minimum guide clearances.

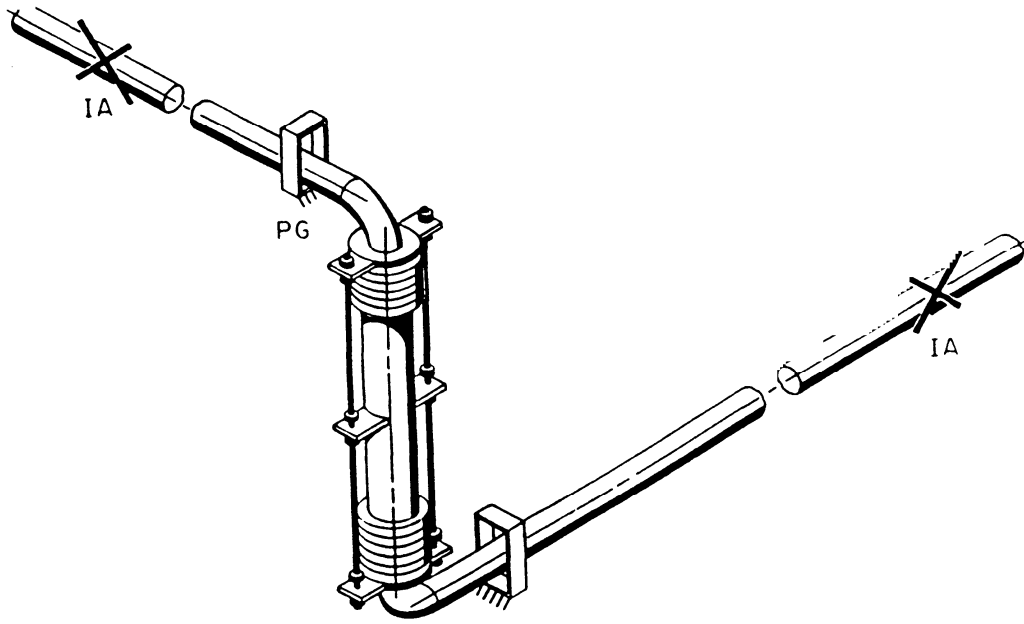


FIGURE 2.14

Figure 2.14 shows a typical application of a tied universal Expansion Joint in a three-plane "Z" bend. Since the universal Expansion Joint can absorb lateral deflection in any direction, the two horizontal piping legs may lie at any angle in the horizontal plane.

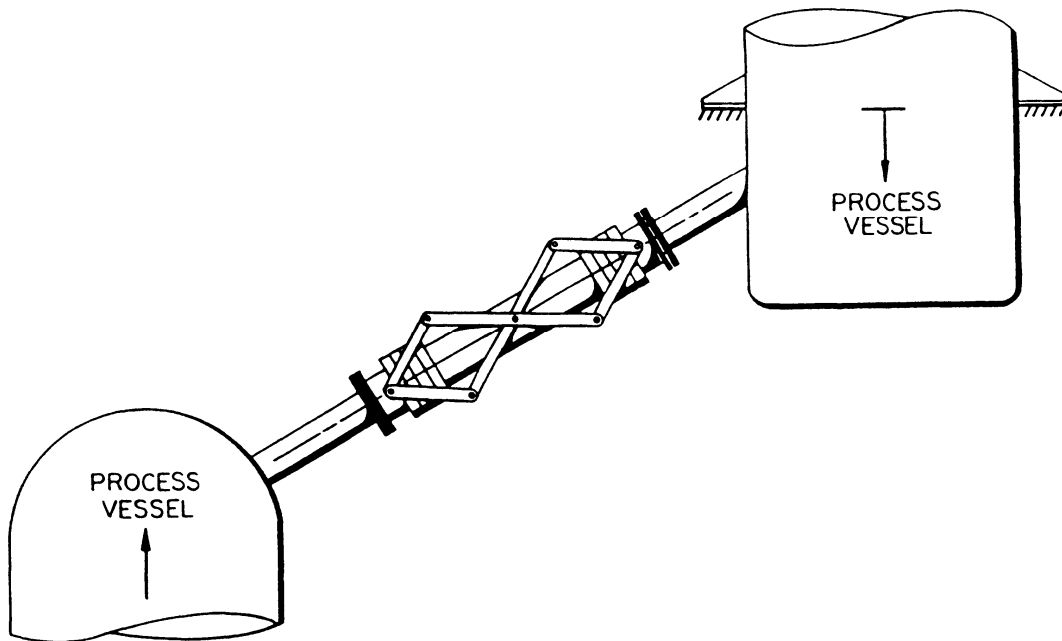


FIGURE 2.15

In cases where a universal Expansion Joint must absorb axial movement other than its own thermal growth, it cannot function as a tied Expansion Joint and must be used in combination with main anchors to absorb pressure loading.

One such case is shown in Figure 2.15. The relative expansion between the two vessels results in both axial movement and lateral deflection on the Expansion Joint. Both vessels must be designed to absorb main anchor loading. Control rods or pantographic linkages may be used to distribute the movement between the bellows and control their movements.

Numerous variations are possible in the design of universal Expansion Joints. Rods, pantographic linkages, slotted hinges or external structural members may be used in a horizontal installation, for example, where it is desirable to support the center pipe section of the Expansion Joint independently of the bellows. In a single plane system, the rods may be replaced by two bars with pinned connections at either end of the Expansion Joint. This construction is so commonly used that it has been given the standard nomenclature of "Swing Expansion Joint". In some cases two sets of short control rods, each set spanning one of the two bellows in the universal Expansion Joint are used instead of the overall rods shown in most of the illustrations. This arrangement is frequently used where the Expansion Joint must absorb axial movement and where the control rods are used for control and stability and not for absorption of pressure loading. This can result when the universal Expansion Joint is very long in relation to its diameter, or a large number of convolutions are used at each bellows of the Expansion Joint, or where the Expansion Joint is subject to external forces.

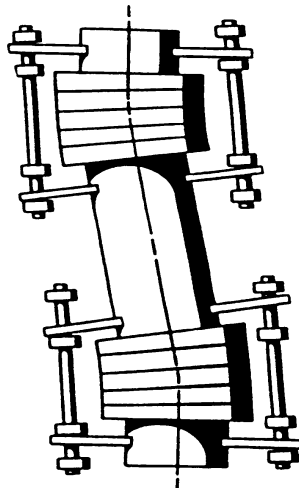


FIGURE 2.16A

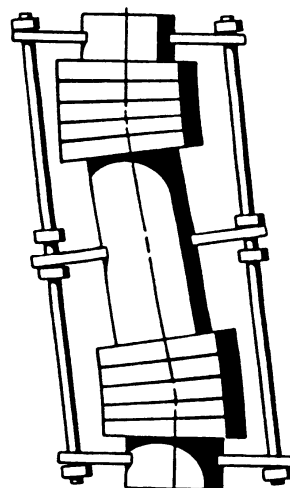


FIGURE 2.16B

It may be desirable to incorporate control devices in the Expansion Joint to prevent excessive displacement of the bellows and the relatively free pipe section between them. Figures 2.16A and 2.16B show two forms of controls which may be used for this purpose. In Figure 2.16A, short rods are used spanning each of the bellows in the Expansion Joint. Stops are provided on the rods so that, once the Expansion Joint has reached its rated lateral deflection, the stops will be engaged by members rigidly fastened to the pipe portions of the Expansion Joint.

Figure 2.16B shows a similar device adapted to an Expansion Joint with overall rods. The rod stops are engaged by a plate or lug attached to the center pipe portion and movement of this part beyond its design deflection is restrained. In order to obtain maximum control from these devices, the stops are usually oriented to lie in the plane of resultant movement of the Expansion Joint, affording maximum leverage as well as greater sensitivity to small movement. Devices of

this nature are usually furnished by the manufacturer dependent upon the design characteristics of the Expansion Joint.

Despite the versatility of the universal Expansion Joint, its use is sometimes precluded by the configuration of the piping, the operating conditions or even by manufacturing and transportation limitations. It may be undesirable or impossible to fabricate, ship to the job site and install a universal Expansion Joint which would span the full length of the offset where, for example, the length of the offset leg in a "Z" bend is extremely long. When the Expansion Joint is very long in relation to its diameter, the flexibility of overall rods may reduce the effectiveness of the control so that the center pipe section becomes unstable. Other types of Expansion Joints may offer a more desirable solution when such limits are encountered.

2.6 APPLICATIONS USING PRESSURE BALANCED EXPANSION JOINTS

The pressure balanced Expansion Joint is used most frequently in applications similar to those shown for the single Expansion Joint, but where pressure loading upon piping or equipment is considered excessive or objectionable. The major advantage of the pressure balanced design is its ability to absorb externally imposed axial movement without imposing pressure loading on the system. The force resulting from the bellows spring rate is not eliminated. In fact, it is usually increased over that of a single Expansion Joint, since both the flow bellows and the balancing bellows must be compressed or elongated and the combined axial force acts upon the piping or equipment. Since the forces to move the bellows are generally of a low order of magnitude, these are usually not objectionable, except in cases involving extremely light equipment with close clearance moving parts which might be affected by small forces.

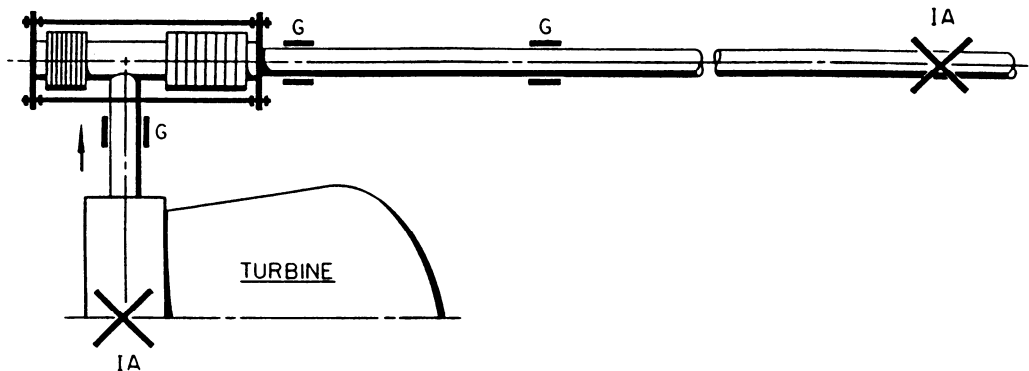


FIGURE 2.17

Figure 2.17 shows a typical application of a pressure balanced Expansion Joint for combined axial movement and lateral deflection. Both the anchor at the end of the piping run and that on the turbine are intermediate anchors and only directional guiding is required. By proper design, the guide directly above the turbine can be made to absorb the axial movement forces of the Expansion Joint without imposing these on the turbine. The only force imposed on the turbine is that which is required to deflect the Expansion Joint laterally.

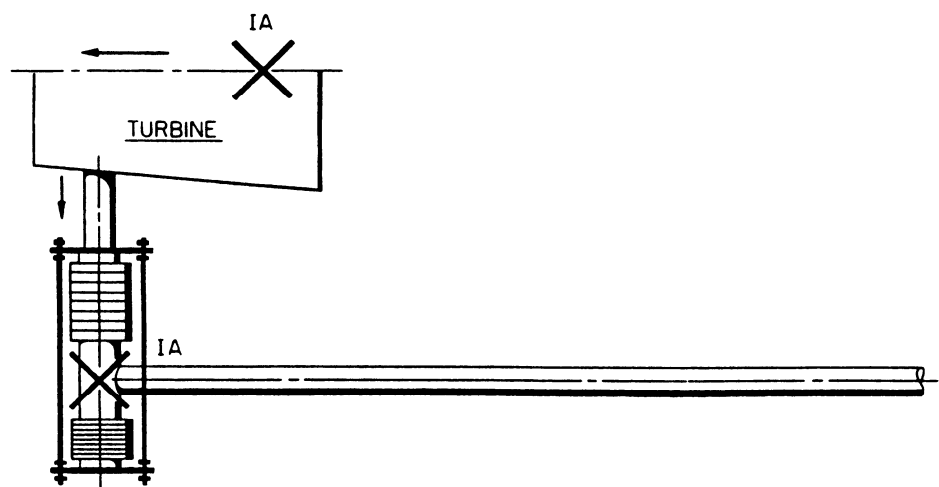


FIGURE 2.18

Figure 2.18 shows another turbine application but, in this case, the anchor point of the turbine is located some distance from the Expansion Joint and the expansion of the turbine between its anchor and the Expansion Joint is absorbed as lateral deflection. An intermediate anchor is used at the center fitting of the Expansion Joint. Since the Expansion Joint is located close to the turbine, guiding between the turbine and Expansion Joint is not required.

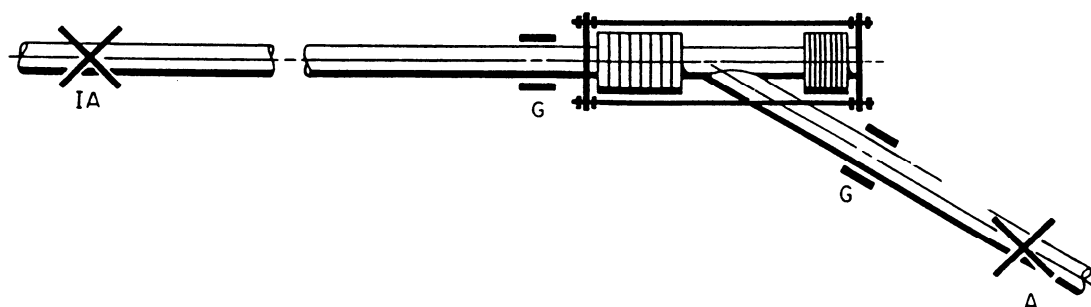


FIGURE 2.19

Figure 2.19 shows that a pressure balanced Expansion Joint can be used at changes in direction other than 90 degrees. In this case, the growth of the longer piping run is absorbed as axial movement on the Expansion Joint, while the thermal expansion of the offset piping run introduces both axial and lateral components or deflection on the Expansion Joint. Only intermediate anchors are required at the ends of the lines and directional guiding is used. The guide on the offset run may be used to absorb the axial movement forces of the Expansion Joint, if the piping is not sufficiently stiff to transmit this directly to the intermediate anchor.

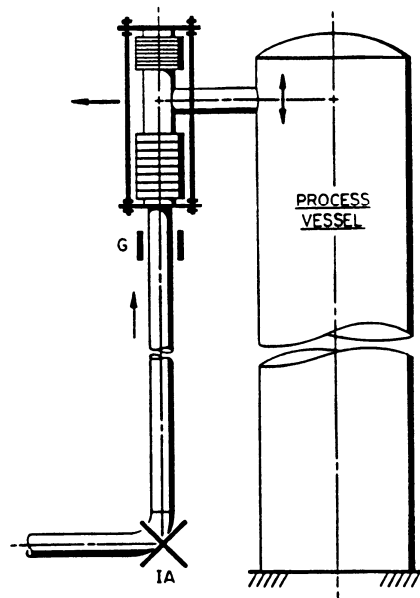


FIGURE 2.20

Figure 2.20 shows a common application for which a pressure balanced Expansion Joint is well suited. Under various process conditions, the vessel and the vertical pipe may expand at different rates. By installing a pressure balanced Expansion Joint as shown, the differential vertical movement is absorbed as axial movement on the Expansion Joint and the thermal expansion from the center line of the process vessel to the piping is absorbed as lateral deflection. The piping may then be secured by an intermediate anchor at the bottom and furnished with a directional guide adjacent to the Expansion Joint.

In many cases, no external structure is available at the upper elevation of the process vessel and the guide must be connected to the vessel itself. Using this arrangement may result in some bending load upon the piping, especially where the vessel is tall and is subject to wind loading deflection or similar effects. Where the guide is attached to a rigid external structure, the Expansion Joint must be designed to absorb wind loading deflection, and other similar loading, as lateral deflection.

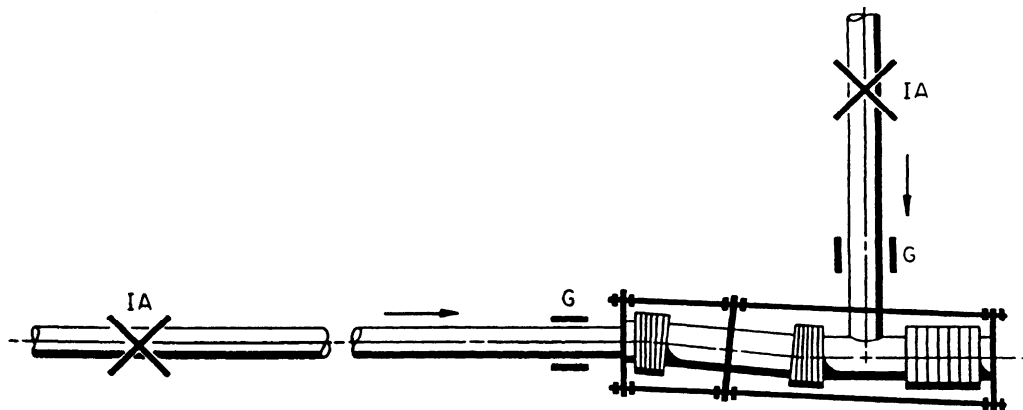


FIGURE 2.21

Where large amounts of lateral deflection are involved, a pressure balanced universal Expansion Joint must be used. In this design, two bellows are used in the flow end of the Expansion Joint and a single bellows in the balancing end. Normally, as shown in Figure 2.21, the balancing

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bellows will be subjected only to axial movement if the tie rods are properly designed to rotate or pivot at their attachment points.

In order for a pressure balanced Expansion Joint to function properly, the pressure thrust restrained by the tie rods must exceed the axial movement forces of the Expansion Joint. In a large diameter, low pressure application, it may be impossible to utilize the pressure balanced Expansion Joint to eliminate the pressure loading or, at best, the effect may be uncertain. In such cases, some other Expansion Joint design must be considered. Pressure balanced Expansion Joints are not recommended for use in services where the pressure equalizing connection between the flow bellows and the balancing bellows may become plugged or blocked by the flowing medium or by contaminants. Where flow considerations permit, this problem may be overcome by the use of a tee as a center fitting of the Expansion Joint, rather than an elbow. In some cases, the pressure for the balancing end of the Expansion Joint has been introduced from a separate pressure source, but this is considered somewhat hazardous. A control failure or even a slow control response might result in partial or full pressure loading being imposed upon the piping or equipment, thus overcoming the initial reason for using the pressure balanced Expansion Joint.

The pressure balanced Expansion Joint is used to relieve loads on equipment such as pumps, compressors and turbines. In many cases, the cost of the pressure balanced Expansion Joint will be negligible when compared to the cost of additional equipment, piping and building space which would be necessary for safe functioning of the equipment without the Expansion Joint.

2.7 APPLICATIONS USING HINGED EXPANSION JOINTS

Hinged Expansion Joints are usually used in sets of two or three, to absorb lateral deflection in one or more directions in a single plane piping system. Each individual Expansion Joint in such a system is restricted to pure angular rotation by its hinges. Each pair of hinged Expansion Joints, separated by a segment of piping, will act in unison to absorb lateral deflection in much the same manner as a swing or universal Expansion Joint in a single plane application. For a given angular rotation of the individual Expansion Joints, the amount of lateral deflection which a pair of hinged Expansion Joints can absorb is directly proportional to the distance between their hinge pins. In order to utilize the Expansion Joints most efficiently, this distance should be made as large as possible.

Expansion Joint hinges are normally designed to absorb the full pressure thrust of the Expansion Joint and may be designed to support the weight of piping and equipment, wind loads or similar externally applied forces. Where such external forces are anticipated, their direction and magnitude must be indicated to the Expansion Joint manufacturer so that the hinges can be adequately designed to withstand these forces.

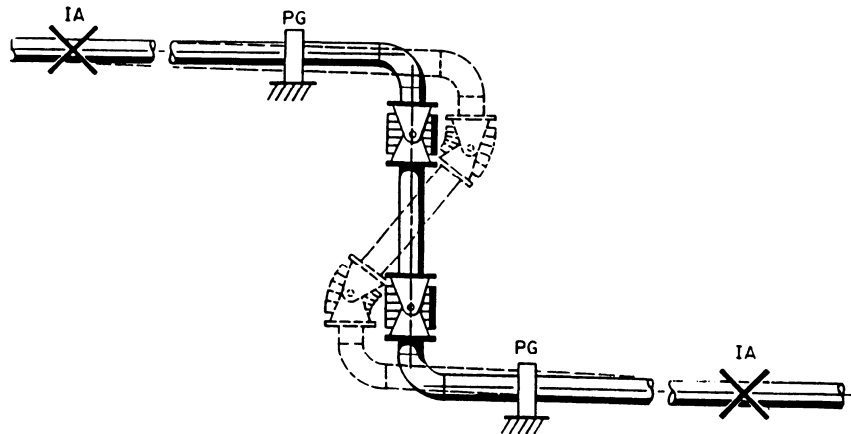


FIGURE 2.22

Figure 2.22 illustrates the use of a two-hinge system to absorb the major thermal expansions in a single-plane "Z" bend. Since the pressure thrust is absorbed by the hinges on the Expansion Joints, only intermediate anchors are required at each end of the piping system. The thermal expansion of the offset section containing the Expansion Joints must be absorbed by bending of the piping legs perpendicular to that segment, since the Expansion Joints are restricted to pure angular rotation by their hinges and cannot extend or compress. The amount of bending deflection imposed on each of the two long piping legs may be controlled by proper design of guides and supports. Where one long leg is sufficiently flexible to absorb the full thermal growth of the offset leg, the other long leg may be controlled to permit longitudinal movement only. The planar guides shown at the ends of the long piping runs near the elbows are intended to maintain the plane of the piping system only and must allow for the bending deflections of the long piping legs. In calculating guide clearances, consideration shall be given to the fact that the thermal expansion of the offset piping leg containing the Expansion Joints will be partially offset by the reduction in length resulting from the displacement of the center pipe section. The latter effect may be neglected only where the distance between hinge pins is very large and the lateral displacement small. This effect can be minimized by cold springing the Expansion Joints 50% of the full rated deflection.

Because of the ability of the hinges to transmit loads, support of a hinged piping system can frequently be simplified. Assuming that Figure 2.22 is an elevation view and that the upper piping leg is sufficiently flexible to absorb the total expansion of the vertical leg, it would be possible to use sliding supports on the lower horizontal run to support its weight and restrict it to longitudinal movement only. By utilizing the rigidity of the hinges, a substantial portion of the weight of the upper horizontal leg may also be carried on these lower supports. It should be noted that the sliding supports nearest the vertical leg must be designed to resist the force required to deflect the piping. Spring supports must be used throughout the length of the upper horizontal leg where bending occurs. Beyond that point, sliding supports may be used.

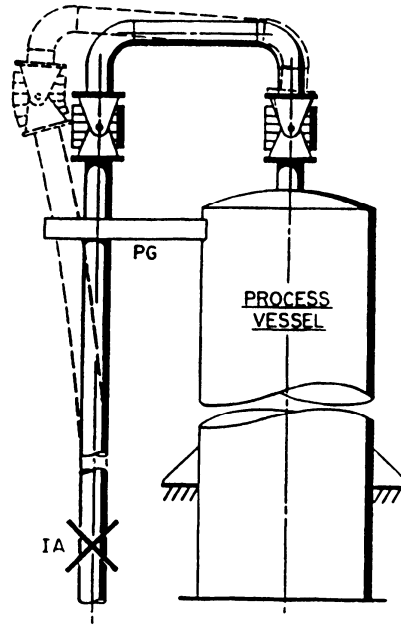


FIGURE 2.23

In locating hinged Expansion Joints for more efficient use, it should be noted that the hinges need not be co-linear in order to function properly. Figure 2.23 illustrates a two-hinge Expansion Joint system similar to the pressure balanced Expansion Joint application of Figure 2.20. In this case, the Expansion Joints will absorb only the differential vertical growth between the vessel and pipe riser. Any horizontal movement due to piping expansion, vibration and wind loads will be absorbed by bending of the vertical pipe leg. A planar guide may be installed near the top of the vessel to protect the hinged Expansion Joints from wind loads at right angles to the plane of the piping.

The anchor shown at the bottom of the riser is an intermediate anchor only, since the pressure load is absorbed by the Expansion Joint hinges. This anchor must be capable of withstanding the forces created by bending of the riser. Depending upon the dimensions and weight of the piping system, complete support may be obtained from the process vessel and from the intermediate anchor. If additional supports are required, spring type supports should be used. The vertical piping may be cold sprung to reduce bending stresses, utilizing the hinges to withstand the cold spring force.

Where the piping in a single plane system is not sufficiently flexible to absorb the bending deflections involved in a two hinge system, or where the loads resulting from such bending exceed the allowable limits for connected equipment, a system of three hinged Expansion Joints may be used. Figure 2.24 illustrates a system of three hinged Expansion Joints in a single plane "Z" bend. The thermal expansion of the offset piping section is absorbed by the action of Expansion Joints B and C. It is therefore evident that Expansion Joint B must be capable of absorbing the total of the rotations of Expansion Joints A and C.

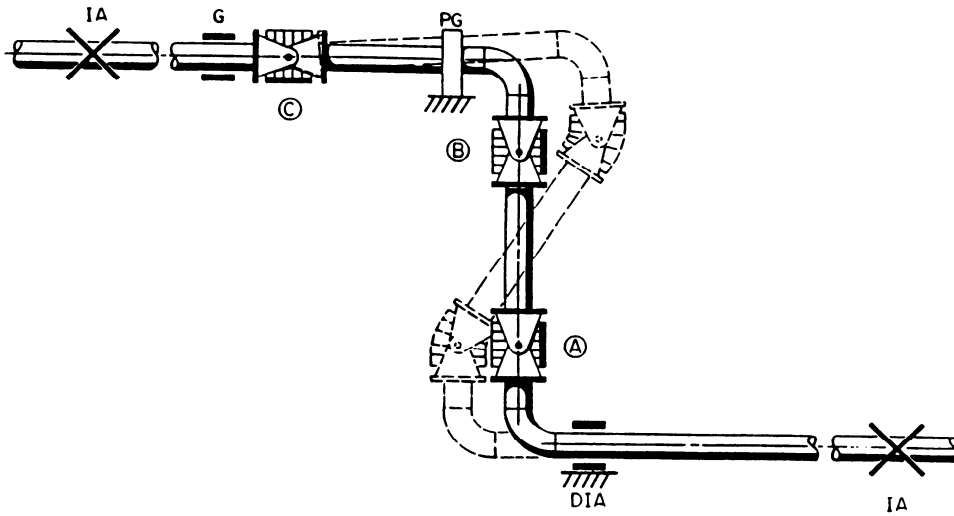


FIGURE 2.24

As in the previous cases, the anchors at the ends of the piping system are intermediate anchors only. In this case, all deflection is absorbed by the Expansion Joints and negligible pipe bending loads will be imposed upon these anchors. Where the distance between the anchor at the left and the first hinged Expansion Joint C is large, a pipe guide should be installed adjacent to the Expansion Joint, as shown in Figure 2.24. This pipe guide will minimize bending of the pipe section between Expansion Joint C and the left hand anchor which might otherwise result from the moment required to rotate the Expansion Joint. One or more additional guides may be used to maintain the plane of the piping system and relieve the hinges of bending forces which may be created by external loads. Support of the piping system may be accomplished in various ways, utilizing available supporting structures with greatest efficiency. It is essential that spring supports be used to permit free movement of the piping between the Expansion Joints.

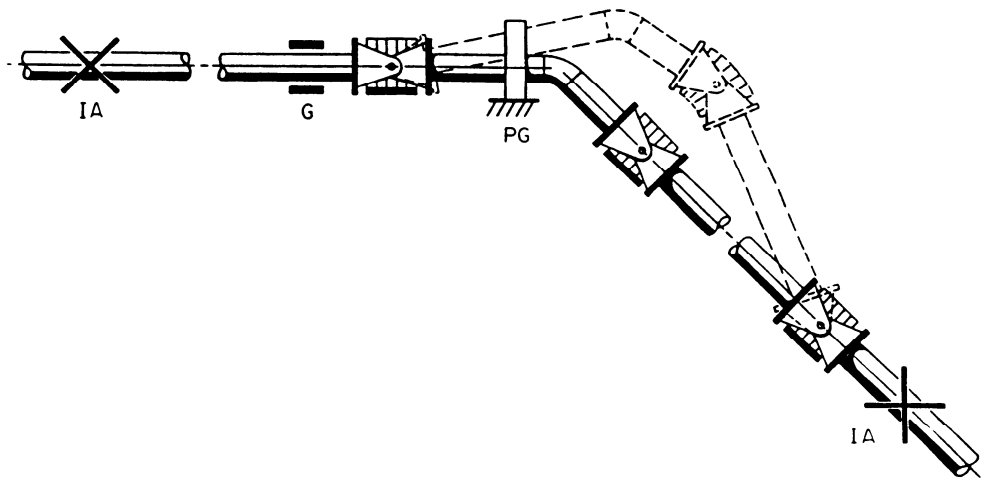


FIGURE 2.25

Figure 2.25 illustrates the principle that systems of hinged Expansion Joints may be used in other than 90° bends. Only intermediate anchors and planar guides are required.

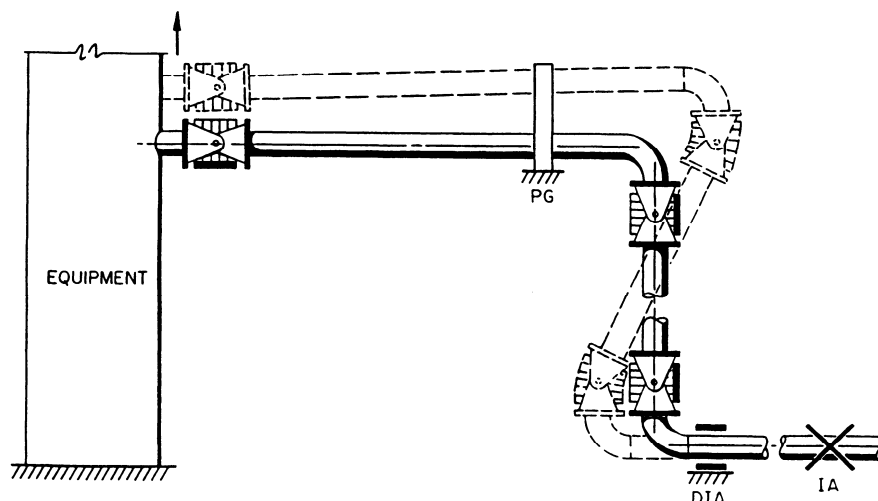


FIGURE 2.26

A hinged Expansion Joint system may be used effectively in applications involving movement other than the pure thermal growth of piping. Figure 2.26 illustrates an application combining the thermal expansion of a piping system with the single plane movements of a piece of connected equipment. So long as all movements are restricted to a single plane, the behavior of the Expansion Joint system is quite similar to that of the system shown in Figure 2.24. An intermediate anchor is required at one end of the piping, while the equipment serves as an intermediate anchor at the opposite end. The displacements of the equipment are added to those of the piping to evaluate the movements of the Expansion Joints. Planar guide clearances in the plane of the piping must be adequate to allow for the equipment movement as well as the piping rotations.

Some advantages of hinged Expansion Joints are compact size and structural rigidity. By the use of these individual units, it is frequently possible to compensate for the thermal expansion of irregular and complex piping configurations which might preclude the use of other types of Expansion Joints. Because of the ability of the hinge structure to transmit loads, piping systems containing hinged Expansion Joints impose minimum forces on the pipe anchors. Such systems may be supported at virtually any point which does not interfere with the free movement of the system.

2.8. CALCULATION OF ANGULAR ROTATION IN A 3 HINGE PIPING SYSTEM

The application of a 3 hinge piping system is described in Section 2.7. This section provides a standard method to calculate the angular movements in each of the three hinge joints. Refer to Appendix J for a sample problem to illustrate the calculations.

Diagram the piping system as shown below. All lengths of pipe outside of points A and C should be added algebraically to compute the thermal growth from $A - A^1$ and $C - C^1$.

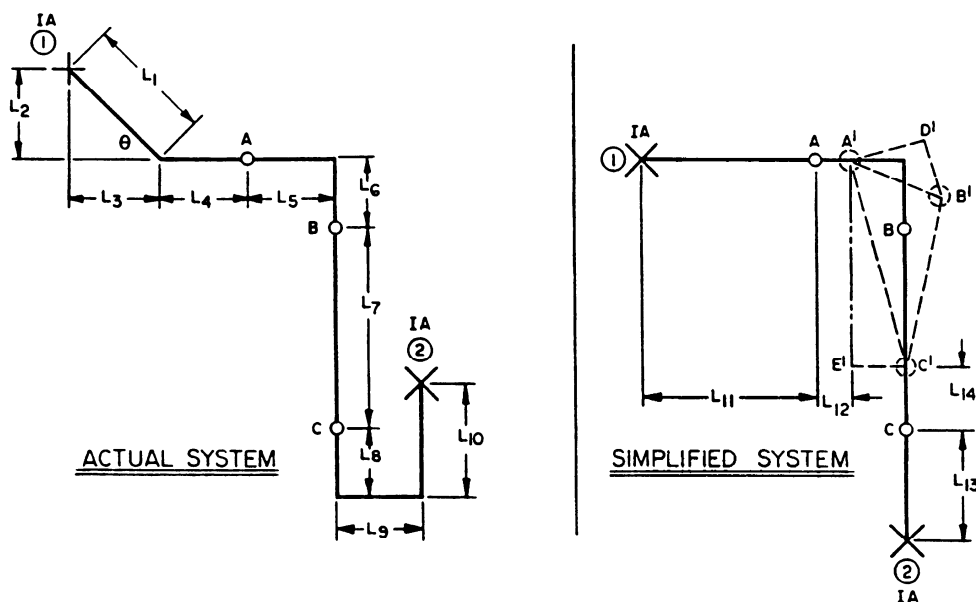


FIGURE 2.27

GIVEN

$$L_1 = \underline{\hspace{2cm}} \text{ in. (mm)} \quad L_6 = \underline{\hspace{2cm}} \text{ in. (mm)} \quad \delta = \underline{\hspace{2cm}} \text{ in. (mm)/in. (mm)}$$

$$\theta = \underline{\hspace{2cm}} \text{ deg.} \quad L_7 = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$L_8 = \underline{\hspace{2cm}} \text{ in. (mm)} = \frac{\text{Tabulated values from Table I}}{1200}$$

$$L_4 = \underline{\hspace{2cm}} \text{ in. (mm)} \quad L_9 = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$L_5 = \underline{\hspace{2cm}} \text{ in. (mm)} \quad L_{10} = \underline{\hspace{2cm}} \text{ in. (mm)}$$

MOVEMENT CALCULATIONS

$$L_2 = (L_1)(\sin \theta) = \underline{\hspace{2cm}} \text{ in. (mm)} \quad L_3 = (L_1)(\cos \theta) = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$L_{13} = (L_2) + (L_8) - (L_{10}) = \underline{\hspace{2cm}} \text{ in. (mm)} \quad L_{11} = (L_3) + (L_4) + (L_9) = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$L_{14} = (L_{13})(\delta) = \underline{\hspace{2cm}} \text{ in. (mm)} \quad L_{12} = (L_{11})(\delta) = \underline{\hspace{2cm}} \text{ in. (mm)}$$

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$$A^1E^1 = L_6 + L_7 - L_{14} = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$E^1C^1 = L_5 - L_{12} = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$A^1C^1 = \left[(A^1E^1)^2 + (E^1C^1)^2 \right]^{1/2} = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$A^1 = TAN^{-1} \left[\frac{(E^1 C^1)}{(A^1 E^1)} \right] =$$

$$C^1 = 90^\circ - A^1 =$$

$$A^1 D^1 = (L_5)(1 + \delta) = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$D^1B^1 = (L_6)(1 + \delta) = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$A^1B^1 = \left[(A^1D^1)^2 + (D^1B^1)^2 \right]^{1/2} = \underline{\hspace{2cm}} \text{ in. (mm)}$$

$$A^1 = TAN^{-1} \left[\frac{(D^1 B^1)}{(A^1 D^1)} \right] =$$

$$B^1 = 90^\circ - A^1 =$$

$$B^1C^1 = (L_7)(1 + \delta) \quad = \quad \text{in. (mm)}$$

$$B^1 = \cos^{-1} \left[\frac{(A^1 B^1)^2 + (B^1 C^1)^2 - (A^1 C^1)^2}{(2)(A^1 B^1)(B^1 C^1)} \right] =$$

$$C^1 = \cos^{-1} \left[\frac{(A^1 C^1)^2 + (B^1 C^1)^2 - (A^1 B^1)^2}{(2)(A^1 C^1)(B^1 C^1)} \right] =$$

$$A^1 = 180^\circ - B^1 - C^1 =$$

$$\Sigma =$$

[illegible]

CALCULATED ANGULAR MOVEMENTS

$$\Delta A = \sum A^1 - 90^\circ = \text{deg.}$$

$$\Delta B = 180^\circ - \sum B^1 = \quad \text{deg.}$$

$$\Delta C = \sum C^1 - 90^\circ = \quad \text{deg.}$$

$$\Delta B = \Delta A + \Delta C \text{ (Check)} = \text{deg.}$$

Refer to Appendix J Example 7 for a sample calculation.

2.9 APPLICATIONS USING GIMBAL EXPANSION JOINTS

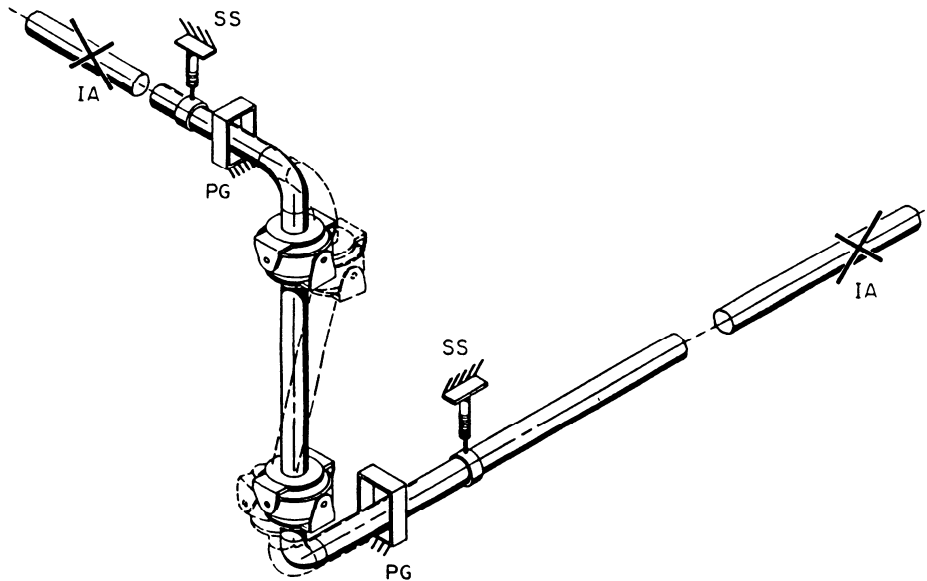


FIGURE 2.28

Just as hinged Expansion Joints may offer great advantages in single plane applications, gimbal Expansion Joints are designed to offer similar advantages in multi-plane systems. The ability of the gimbal Expansion Joint to absorb angular rotation in any plane is most frequently applied by utilizing two such units to absorb lateral deflection. An application of this type is shown in Figure 2.28. Since the pressure loading is absorbed by the gimbal structure, intermediate anchors only are required. Planar guides are provided to restrict the movement of each piping leg. As in the case of hinged Expansion Joints, the location of pipe supports is simplified by the load carrying ability of the gimbal structure. Since, in a two gimbal system, the growth of the vertical pipe leg will be absorbed by bending of the long legs, spring supports (SS) may be required on either or both of these. Guides must be designed to allow for the thermal expansion of the leg containing the Expansion Joints and for the shortening of this leg due to deflection.

Where it is impossible or undesirable for the piping to absorb the growth of the offset leg, a system consisting of two gimbal and one hinged Expansion Joint may be used as shown in Figure 2.29. The gimbal Expansion Joints function in unison to absorb the combined movements of the upper and lower legs, while the hinged Expansion Joint and the upper gimbal Expansion Joint act in combination to absorb deflection of the offset leg. Since the expansion of the offset leg takes place in one plane only, the use of the simpler hinged Expansion Joint is justified.

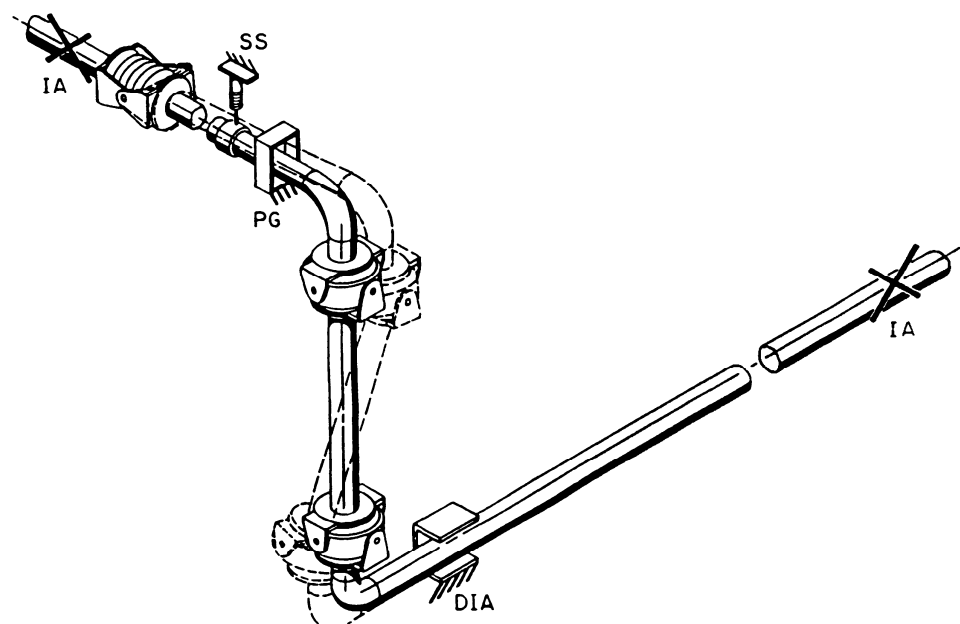


FIGURE 2.29

The advantages of using gimbal Expansion Joint systems are similar to those previously mentioned for systems containing hinged Expansion Joints. Greater flexibility of usage is possible since gimbal Expansion Joints are not restricted to single plane systems.

2.10 ANCHOR, GUIDE, AND SUPPORT REQUIREMENTS

2.10.1 PIPE ANCHORS

It is the purpose of any pipe anchor to divide a pipe line into individual expanding sections. Since thermal growth cannot be restrained, it then becomes the function of pipe anchors to limit and control the amount of movement which Expansion Joints, located between these anchors, must absorb.

Major pieces of connected equipment such as turbines, pumps, compressors, heat exchangers, and reactors may function as anchors. The design of such equipment must anticipate this loading. Additional pipe anchors are usually located at valves, at changes in the direction of the pipe, at blind ends of pipe, and at major branch connections. Expansion Joints must be provided in each of the individual pipe sections to provide adequate flexibility. See Section 2.2 and 2.3 for typical Expansion Joint applications.

DO NOT INSTALL MORE THAN ONE "SINGLE" EXPANSION JOINT BETWEEN THE TWO ADJACENT ANCHORS IN ANY STRAIGHT PIPE SECTION. Where expansion loops are used in the same line with an Expansion Joint, the section of pipe containing the loop must be isolated from the section containing the Expansion Joint by means of anchors.

Pipe anchors, their attachment, and the structures to which they are attached must be designed to withstand the forces acting upon them. Methods are given in the following paragraphs for determining the major forces to which anchors are subjected, and it is on the basis of these major forces that anchors are classified as intermediate or main anchors. The system designer must realize that additional indeterminate forces can be imposed on both intermediate and main anchors. All components of the anchor should be designed to a conservative stress level.

Determination of the magnitude of the major forces acting on anchors as set forth in the following paragraphs are forces acting axially. Consideration should also be given to possible lateral forces in arriving at a suitable anchor design.

The difference in cost of an adequately and inadequately designed anchor is nominal at installation. Anchor failure can cause damage which is far more costly than that of the more conservative design.

2.10.1.1 INTERMEDIATE PIPE ANCHORS

An intermediate pipe anchor must be designed to withstand the forces and moments imposed upon it by each of the pipe sections to which it is attached. These consist of the forces and/or moments required to deflect the Expansion Joint or Joints the full rated movement and the frictional forces due to pipe alignment guides, directional anchors and supports. Note that an intermediate anchor is not intended to withstand the pressure thrust. This force is absorbed by other anchors, by devices on the Expansion Joints such as tie rods, swing bars, hinges, gimbals, and other hardware, or, as in the case of a double Expansion Joint, is balanced by an equal pressure force acting in the opposite direction.

In certain applications, it may be necessary to consider the weight of the pipe, fittings, insulation and flowing medium, as well as various other forces and moments such as those resulting from wind loading or bending of one or more pipe sections. The net loading on the anchor can be calculated by a summation of the moments about the anchor point and by the vector addition of all forces acting upon it.

2.10.1.1.1 CALCULATION OF INTERMEDIATE ANCHOR LOADS FOR APPLICATIONS INVOLVING STRAIGHT PIPE SECTIONS ONLY (See Figure 2.2)

The force F_m required to extend or compress the Expansion Joint its full rated movement is a function of each manufacturer's design and is affected by the material, shape, depth and thickness of the bellows convolutions. This value should be obtained from the manufacturer of the Expansion Joint under consideration.

The frictional force F_g due to pipe alignment guides is a function of the design and number of alignment guides used in each pipe section and should be obtained from the manufacturer of the pipe alignment guides under consideration.

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Assuming that the weight of the pipe line and its contents is carried by supports, the total force acting on the intermediate anchor is then given by the formula:

$$F_{ia} = F_{m1} + F_{g1} + F_{m2} + F_{g2} \quad (2-1)$$

Where:

F_{m1} = The force required to extend or compress the Expansion Joint located immediately to the right of the intermediate anchor in Figure 2.2, lbf (N)

F_{g1} = The frictional force in the pipe alignment guides installed on the pipe section to the right of the intermediate anchor in Figure 2.2, lbf (N)

F_{m2} = The force required to extend or compress the Expansion Joint located immediately to the left of the intermediate anchor in Figure 2.2, lbf (N)

F_{g2} = The frictional force in the pipe alignment guides installed on the pipe section to the left of the intermediate anchor in Figure 2.2, lbf (N)

If the pipe is the same diameter on both sides of the intermediate anchor, and if the guides on both pipe sections are similar in number and design, F_{m2} and F_{g2} will be equal to F_{m1} and F_{g1} respectively, but opposite in sign. Thus, F_{ia} will be equal to zero. However, it is possible that the pipe line may heat up gradually from one end, thereby causing one of the pipe sections to expand before the other. It is, therefore, considered good practice to design the intermediate anchor to resist the forces exerted by one of the two pipe sections

$$(F_{ia} = F_{m1} + F_{g1}).$$

2.10.1.1.2 CALCULATION OF INTERMEDIATE ANCHOR LOADS FOR APPLICATIONS INVOLVING LATERAL DEFLECTION AND ANGULAR ROTATION

When lateral deflection and angular rotation are present, the loads imposed on an intermediate anchor will vary with each individual application, since they are dependent upon the piping configuration, the number and type of supports, the lengths of the various pipe legs, the types of Expansion Joints used, the weight of the pipe, fittings, insulation and flowing medium, and the magnitude of extraneous forces imposed by wind loading, bending of the piping, etc. Because of the large number of variables involved, it is not practical to establish formulas for calculating the loading of these anchors. Section 4.6 describes the methods used in calculating the forces and moments required to move Expansion Joints in lateral deflection or angular rotation and Section 2.2.1 gives a general description of the forces and moments which will be imposed on the anchors for certain typical Expansion Joint applications. Using this information as a guide, one can calculate the various forces and moments acting on any piping system, regardless of its complexity. The net load on the anchor can then be calculated by a summation of the moments about the anchor point and by the vector addition of all forces acting upon it.

2.10.1.2 MAIN PIPE ANCHORS

A main anchor is one which is installed at any of the following locations in a piping system containing one or more unrestrained Expansion Joints: (a) at a change in direction of flow, (b) between two Expansion Joints of different size installed in the same straight run, (c) at the entrance of a side branch containing an unrestrained Expansion Joint into the main line, (d) where a shut-off or pressure reducing valve is installed in a pipe run between two Expansion Joints, and (e) at a blind end of pipe.

A main pipe anchor must be designed to withstand the forces and moments imposed upon it by each of the pipe sections to which it is attached. In the case of a pipe section containing one or more Expansion Joints, these will consist of the full line thrust due to pressure and flow, the forces and/or moments required to deflect the Expansion Joint or Joints the full rated movement, and the frictional forces due to pipe alignment guides, directional anchors and supports. In certain applications, it may be necessary to consider the weight of the pipe, fittings, insulation and flowing medium, as well as various other forces and moments resulting from wind loading, bending of one or more pipe sections, etc. The net loading on the anchor can be calculated by a summation of the moments about the anchor point and by the vector addition of all forces acting upon it.

2.10.1.2.1 CALCULATION OF MAIN ANCHOR LOADS FOR APPLICATIONS INVOLVING STRAIGHT PIPE SECTIONS CONTAINING A BRANCH LINE (See center anchor in Figure 2.3)

$$F_s = A_e P_d \quad (2-2)$$

where:

F_s = The static thrust due to pressure in the Expansion Joint, lbf (N)

The forces F_m and F_{pg} may be calculated as outlined before for an intermediate anchor. Then, assuming that the weight of the pipe line and its contents is carried by supports; the total force imposed on the main anchor F_{ma} by any one pipe section will be:

$$F_{ma} = F_s + F_m + F_{pg} = A_e P_d + F_m + F_{pg} \quad (2-3)$$

To determine the net load on the anchor, it is necessary to add vectorially the forces imposed upon it by each of the three pipe sections to which it is attached.

2.10.1.2.2 CALCULATION OF MAIN ANCHOR LOADS FOR APPLICATIONS INVOLVING STRAIGHT PIPE SECTIONS CONTAINING EXPANSION JOINTS OF DIFFERENT DIAMETERS (See center anchor in Figure 2.4)

$$F_{s1} = (A_{e1} - A_{e2}) P_d \quad (2-4)$$

where:

A_{e1} = Bellows effective area of large pipe section, in.² (mm²)

A_{e2} = Bellows effective area of small pipe section, in.² (mm²)

Here again, we must consider the difference in the forces required to extend or compress the Expansion Joints and the difference in the frictional forces due to pipe alignment guides and supports. The total force on the main anchor will be:

$$\begin{aligned} F_{ma} &= F_{s1} + F_{m1} + F_{g1} - F_{m2} - F_{g2} \\ &= (A_{e1} - A_{e2})(P_d) + (F_{m1} - F_{m2}) + F_{g1} - F_{g2} \end{aligned} \quad (2-5)$$

where:

F_{m1} = The force required to extend or compress the Expansion Joint in the large pipe section, lbf (N)

F_{m2} = The force required to extend or compress the Expansion Joint in the small pipe section, lbf (N)

F_{g1} = The frictional force in the pipe alignment guides on the large pipe section, lbf (N)

F_{g2} = The frictional force in the pipe alignment guides on the small pipe section, lbf (N)

2.10.1.2.3 CALCULATION OF MAIN ANCHOR LOADS FOR APPLICATIONS INVOLVING ANCHORS AT PIPE BENDS AND ELBOWS (See Figure 2.1)

In the case of an anchor located at a pipe bend or elbow, it is necessary to consider the forces imposed by the pipe sections on both sides of the anchor. Assuming that each section contains an Expansion Joint, the line thrust due to pressure ($F_s = A_e P_d$) and the forces, F_m and F_g , explained previously, become biaxial components and must be added vectorially. In addition, the effect of the centrifugal thrust at the elbow, F_{ct} , due to flow must be considered.

$$F_{ct} = \frac{24 A_p \rho v^2}{g} \sin \frac{\theta_d}{2}, \text{ lbf} \quad (2-6)$$

$$F_{ct} = \frac{1.96 \times 10^{-5} A_p \rho v^2}{g} \sin \frac{\theta_d}{2}, \text{ N} \quad (2-6M)$$

θ_d = Angle of pipe bend, deg

ρ = Density of fluid, lbm./in.³ (kg/m³)

For US units, the constant (24) includes a units conversion factor of 12.

For Metric units, the constant (1.96×10^{-5}) includes a units conversion factor of 9.8×10^{-6} .

2.10.2 PIPE GUIDES AND GUIDING

Correct alignment of the adjoining pipe is of vital importance in the proper functioning of an Expansion Joint. Although Expansion Joints are designed and built for long and satisfactory life, maximum service will be obtained only when the pipe line has the recommended number of guides and is anchored and supported in accordance with good engineering practice. Proper supporting of the pipe line is required not only to support the live and dead loads imposed on the line but also to provide support for the Expansion Joint at each of its attachments. Pipe guides are necessary to insure proper application of

movement to the Expansion Joint and to prevent buckling of the line. Buckling may be caused by a combination of two conditions: (1) the flexibility of the Expansion Joint, and (2) the internal pressure loading on the pipe which causes it to act like a column loaded by the pressure thrust of the Expansion Joint. A typical application for pipe guiding is shown in Figure 2.1.

Application of planar pipe guides is shown in Figures 2.11 through 2.14. These guides allow the piping to deflect in order to compensate for the change in length of the Expansion Joint in its deflected position, while directing the thermal growth into the Expansion joint. These guides do not restrain the Expansion Joint ends against rotation in any plane. This restraint is a criterion for stability of most single and universal tied joints when subject to internal pressure. In general, if the torsional and/or bending flexibility of the attached piping is such that the pipe end attached to the Expansion Joint will bend or rotate more than 1.5 degrees when subjected to a force equal to 10% of the full pressure end load of the bellows applied perpendicular to the pipe centerline in any direction, consideration should be given to the use of further guiding to restrain bending and/or torsional rotation in the pipe.

Proper design of both pipe alignment guides (G) and planar pipe guides (PG) should contain sufficient clearance between the fixed and moving parts of the alignment guide to insure proper guiding without introducing excessive frictional forces. The first two alignment guides immediately adjacent to each side of the Expansion Joint should be circumferential to the pipe. Most commercially available alignment guides are acceptable, though some designs require installation procedures that, unless followed with extreme care, destroy the intended guiding features of the unit. Alignment guides made from roller supports may be used when a minimum of three (3) rollers equally spaced around the circumference of the pipe are provided; four (4) rollers at 90° intervals are preferable. Planar pipe guides must be designed with additional clearance in one direction to permit the intended lateral deflection and/or bending of the pipe to take place.

A U-bolt, pipe hanger, or single-roller support, which only supports the weight of the line, must not be considered as a substitute for either a proper pipe alignment guide or a planar guide. Materials from which pipe alignment guides and planar pipe guides are made must provide strength and rigidity under design operating conditions and be sufficiently resistant to corrosion and wear to prevent eventual malfunction of the guide. Test data has shown that the first and second pipe alignment guides nearest the Expansion Joint can be subjected to lateral forces averaging 7%, and as high as 15%, of the total force exerted on the main anchor F_{ma} . The lateral force was developed with consideration for the unknowns associated with actual field installation but primarily reflects an accentuation of the original allowable pipe bow between supports. Although field bolting of pipe alignment and planar pipe guides to the rigid parts of the installation is preferable, field welding is acceptable provided inaccuracies or excessive weld shrinkage do not destroy the effectiveness of these guides. Since properly spaced alignment guides will be considerably distant from the Expansion Joint, it is impractical to require that the guides all be fixed to the same rigid structure, but design of the total system must assure that no relative shifting of alignment guides and Expansion Joint will occur from ground settlement or other environmental conditions. Suitable pipe alignment and planar pipe guides may be obtained from reliable manufacturers of this type of equipment. It should be noted that the effectiveness of pipe alignment and planar pipe guides can be destroyed by improper installation. Consequently, care must be taken to insure proper alignment of the guide itself. In applications involving axial movement

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only, the use of a single pipe alignment guide should be avoided since it may act as a fulcrum imposing lateral deflection or angular rotation on an Expansion Joint. However, in certain applications involving lateral deflection or angular rotation, a single pipe guide may be adequate. For further information see Section 2.3.

In locating the pipe alignment guides for applications involving axial movement only, it is generally recommended that the Expansion Joint be located close to an anchor and that the first pipe guide be located a maximum distance of four pipe diameters from the end of the bellows. This arrangement will provide proper movement guiding as well as proper support for each end of the Expansion Joint. The distance between the first pipe guide and the second must be a maximum of fourteen (14) pipe diameters. The recommended maximum spacing of intermediate pipe guides along the balance of a standard weight carbon steel pipe line is determined from Figure 2.31. For any known pressure and pipe size, the recommended maximum guide spacing can be determined by using the following procedure: First, locate the specified pressure P_d at the bottom of the chart and follow this pressure line vertically upwards to its intersection with the diagonal line representing the specified pipe size. Next, move horizontally to the guide spacing column on the side of the chart and select the recommended maximum spacing. As an example, the recommended maximum spacing of intermediate pipe alignment guides along the balance of a 6 inch (152 mm) pipe line containing an Expansion Joint under a pressure of 122 psig. (0.84MPa) is 43 feet (13.1 m). The first guide would be located a maximum distance of 2 feet (0.61 m) from the Expansion Joint and the second guide would be located a maximum distance of 7 feet (2.1 m) from the first guide. See Section 2.3 for recommendations regarding guiding of pipe lines subjected to lateral deflection and angular rotation.

Maximum intermediate guide spacing for any pipe material or thickness shall be calculated using the following formula:

$$L_g = 0.131 \sqrt{\frac{E_p I_p}{P_d A_e \pm f_i e_x}} \quad \text{ft} \quad (2-7)$$

$$L_g = 0.00157 \sqrt{\frac{E_p I_p}{P_d A_e \pm f_i e_x}} \quad \text{m} \quad (2-7M)$$

Note: When bellows is compressed in operation, use $(+)|f_i e_x|$; when extended, use $(-)|f_i e_x|$.

Guide spacing for standard wall carbon steel pipe may also be calculated in lieu of using Figure 2.31. Caution: This figure is based on average spring rate and bellows effective area with bellows inside diameters that equal pipe outside diameters. The formula is based on one half the critical length of a pinned-pinned Euler column.

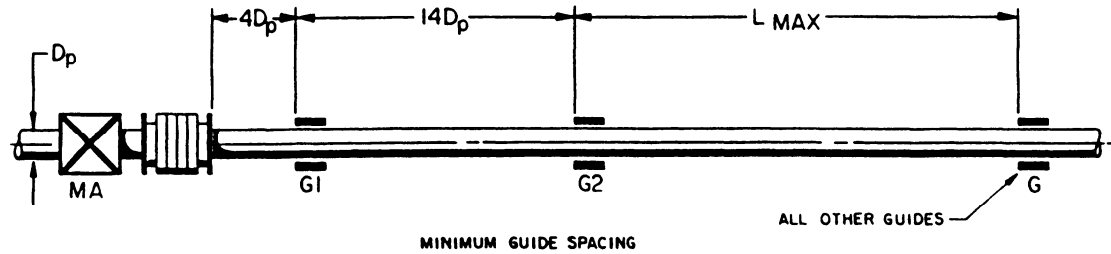


FIGURE 2.30

Note: The recommendations given for pipe anchors and guides represent the minimum requirements for controlling pipelines which contain expansion joints and are intended to protect the expansion joint and pipe system from abuse and failure. However, additional pipe supports are often required between the pipe guides in accordance with accepted piping practices.

2.10.3 PIPE SUPPORTS

A pipe support is any device which permits free movement of the piping and carries the total weight of in line equipment such as valves, meters, Expansion Joints, and the weight of the contained fluid. Pipe supports cannot be substituted for pipe alignment guides or planar pipe guides. Pipe rings, U-bolts, roller supports, and spring hangers are some examples of conventional pipe supports. These devices cannot control the direction of pipe line movement as does a pipe alignment guide or a planar pipe guide. The recommendations given previously for pipe anchors and guides represent the minimum requirements for controlling pipe lines containing Expansion Joints and are intended to protect the Expansion Joints and piping from abuse and possible damage. Additional pipe supports are usually required between guides in accordance with standard piping practice.

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Recommended maximum spacing of intermediate pipe guides for applications involving axial movement only of bellows Expansion Joints. Values based on standard weight carbon steel pipe, see equation (2-7) for other types of pipe. Applicable for bellows inside diameter less or equal to pipe outside diameter. The first pipe guide must be located within a distance of four pipe diameters from the end of the bellows and the second guide must be located within a distance of fourteen pipe diameters from the first guide.

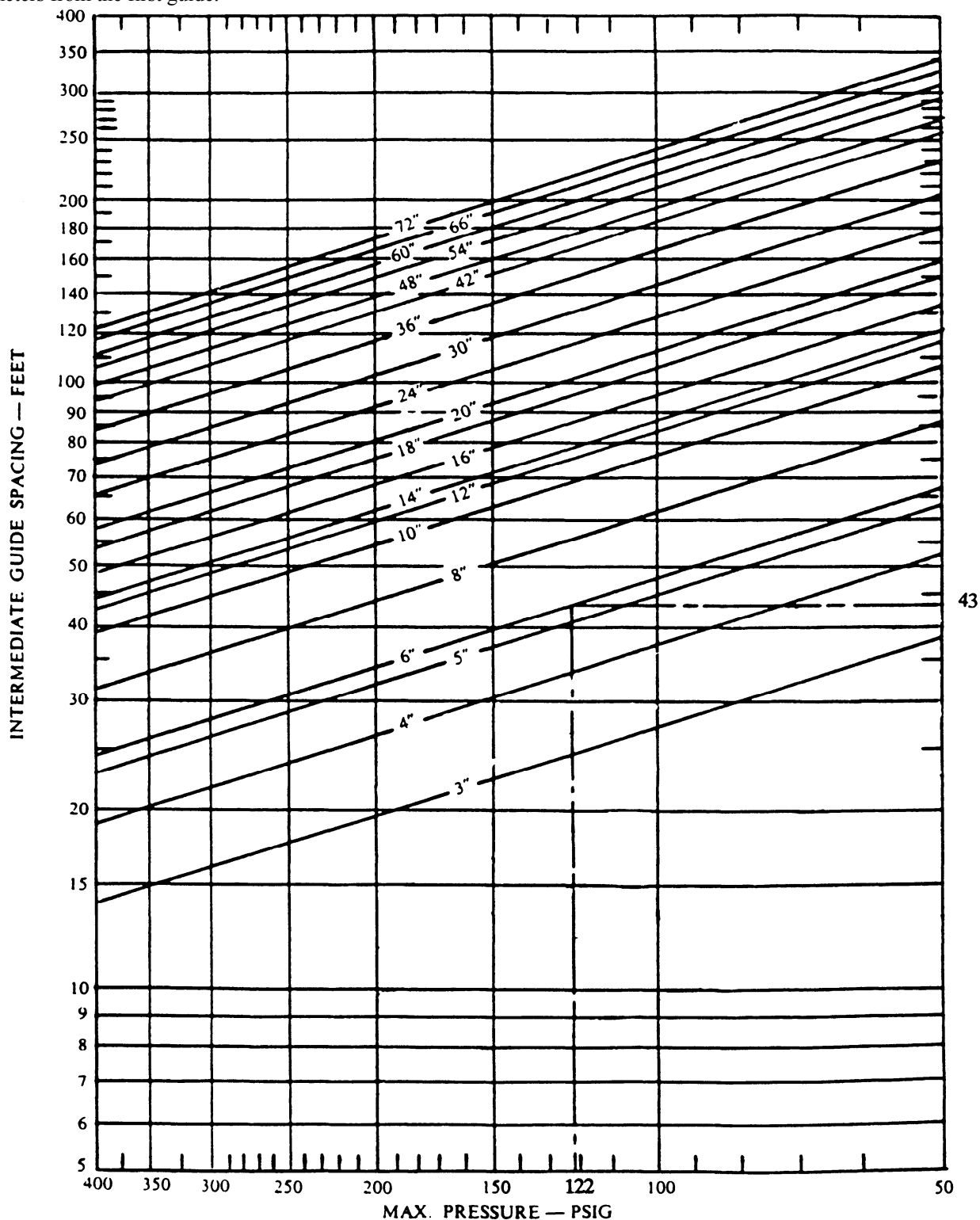


FIGURE 2.31

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Recommended maximum spacing of intermediate pipe guides for applications involving axial movement only of bellows Expansion Joints. Values based on standard weight carbon steel pipe, see equation (2-7) for other types of pipe. Applicable for bellows inside diameter less or equal to pipe outside diameter. The first pipe guide must be located within a distance of four pipe diameters from the end of the bellows and the second guide must be located within a distance of fourteen pipe diameters from the first guide.

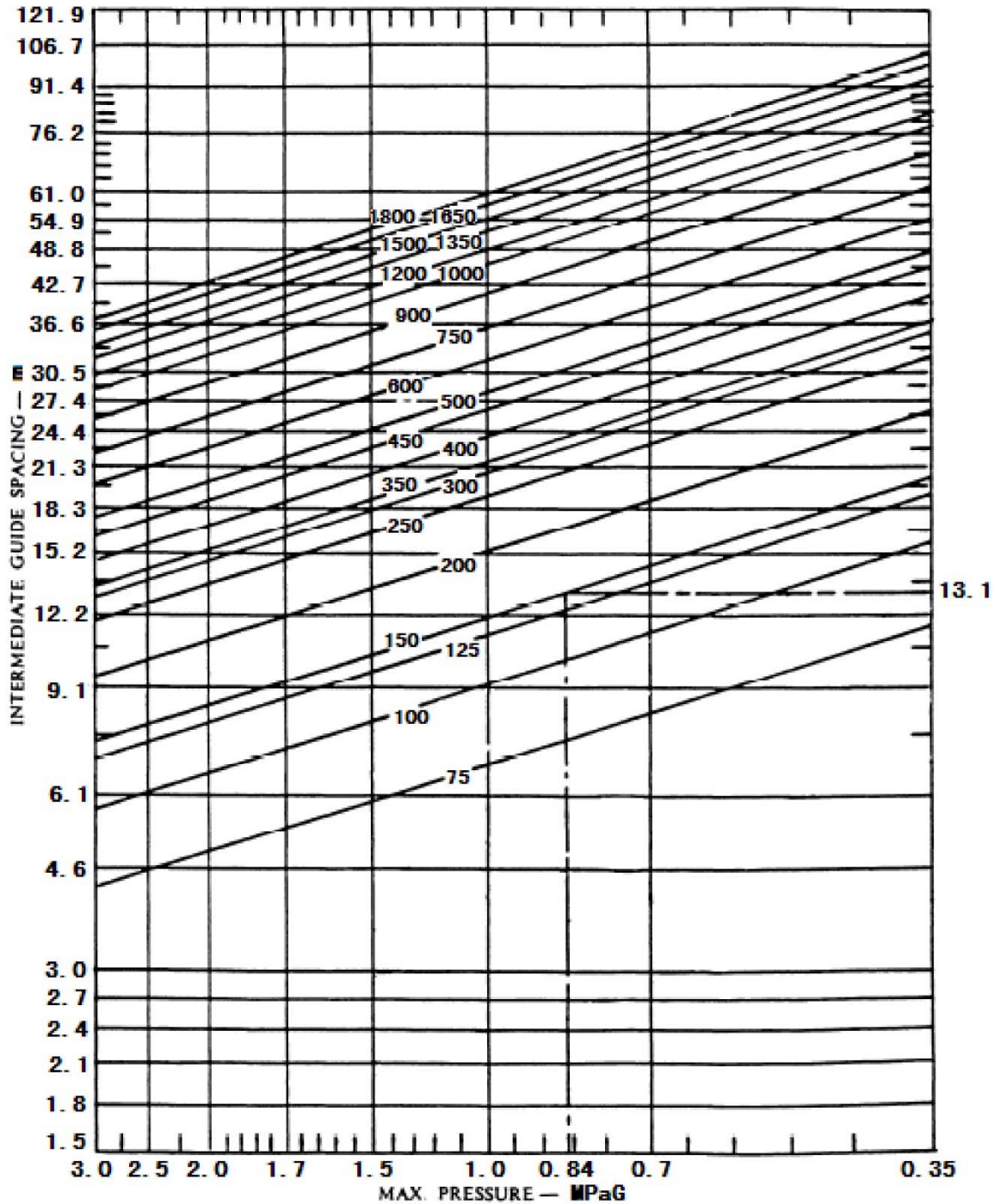


FIGURE 2.31M

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**SECTION 3 – SAFETY RECOMMENDATIONS FOR PIPING SYSTEMS CONTAINING
BELLOWS EXPANSION JOINTS**

Bellows Expansion Joints are employed in piping systems to absorb differential thermal expansion while containing the system pressure. They are being successfully utilized in refineries, chemical plants, fossil and nuclear power systems, heating and cooling systems, and cryogenic plants. Typical service conditions have pressures ranging from full vacuum to 1000 psig (6.9 MPa) and temperatures from -420 to 1800 °F (-251 to 982 °C). Such Expansion Joints fall into the category of a highly engineered product. The system operating characteristics, the Expansion Joint design and manufacturing quality, and the installation, test and operating procedures must all be considered for all Expansion Joint installations.

Unlike most commonly used piping components, a bellows is constructed of relatively thin gage material in order to provide the flexibility needed to absorb mechanical and thermal movements expected in service. This requires design, manufacturing quality, handling, installation and inspection procedures which recognize the unique nature of the product.

In general, the most reliable and safe bellows Expansion Joint installations have always involved a high degree of understanding between the user and manufacturer. With this basic concept in mind, this section was prepared in order to better inform the user of those factors which many years of experience have shown to be essential for the successful installation and performance of piping systems containing bellows Expansion Joints. Additional detailed information can be found in other sections of these Standards.

3.1 DESIGN SPECIFICATION

A. A design specification shall be prepared for each Expansion Joint application.

B. In preparing the Expansion Joint design specification it is imperative that the system designer completely review the piping system layout, flowing medium, pressure, temperature, and movements. The standard Expansion Joint Specification Sheets published in Appendix A can be used as a guide. Particular attention shall be given to the following items:

a. The piping system shall be reviewed to determine the location and type of Expansion Joint most suitable for the application. The EJMA Standards provide numerous examples to assist the user in this effort. The availability of supporting structures for anchoring and guiding of the line, and the direction and magnitude of thermal movements to be absorbed will have a definite bearing on the type and location of the Expansion Joint. **TORSIONAL ROTATION OF THE BELLOWS SHOULD BE AVOIDED.** Where torsional rotation cannot be avoided, refer to Section 4.13.4.

b. The bellows material shall be specified and must be compatible with the flowing medium, the external environment and the operating temperature. Particular consideration shall be given to possible corrosion including stress corrosion. The 300 series stainless steels may be subject to chloride ion stress corrosion. High nickel alloys are subject to caustic induced stress corrosion. The presence of sulfur may also be detrimental to such nickel alloys.

The material chosen shall also be compatible with any water treatment or pipeline cleaning chemicals. In some cases, leachates from insulating materials can be a source of corrosion.

c. Internal sleeves shall be specified in all applications involving flow velocities which could induce resonant vibration in the bellows or cause erosion of the convolutions resulting in substantially reduced bellows life. See Section 4.9.

d. The system design pressure and test pressure shall be specified realistically without adding arbitrary safety factors. Excess bellows material thickness required for overstated

pressures may produce an adverse effect on the bellows fatigue life. In the case of extreme high temperature operating conditions, it may not be practical to test the Expansion Joint to a pressure of more than 1.5 times the design pressure, see Section 7.2.1. This is due to the various materials employed in the Expansion Joint, temperature gradient utilized in design, pressure stability criteria, anchor strength and other considerations. The manufacturer must be consulted.

e. The maximum, minimum and installation temperatures shall be accurately stated. Where the ambient temperature can vary significantly during pipe line construction, pre-positioning of the Expansion Joint at installation may be required. See Appendix J.

f. The Expansion Joint manufacturer shall be advised if the Expansion Joint will be insulated and the manner by which the Expansion Joint will be insulated in order to properly design the component parts.

g. The movements to be absorbed by the Expansion Joint shall include not only piping elongation or contraction, but also movement of attached vessels, anchors, and the possibility of misalignment during installation. Unless included in the design requirements, misalignment of the Expansion Joint must be avoided. Where movements are cyclic, the number of cycles expected shall be specified. As in the case of pressure, the movement specified must be realistic. An excessive safety factor can result in an Expansion Joint which is highly flexible and could have reduced stability under pressure.

h. If the flowing medium can pack or solidify, provisions shall be made to prevent entrapment or solidification of the material in the convolutions which could result in damage to the Expansion Joint or pipeline.

i. Internal sleeves are usually installed in the direction of flow. If the stagnant flow medium trapped behind the sleeve is undesirable, drain holes in the sleeve or purge connections shall be specified. Where back flow will be encountered, an extra heavy sleeve shall be specified to prevent buckling of the sleeve and possible damage to the bellows.

j. The predicted amplitude and frequency of external mechanical vibrations to be imposed on the bellows, such as caused by reciprocating or pulsating machinery, shall be specified. The Expansion Joint must be designed to avoid the resonant vibration of the bellows to preclude the possibility of sudden fatigue failure. Field modifications to the Expansion Joint or other system components may be necessary.

C. The piping system drawings shall specify the location of all anchors, guides, supports and fixed points. Considerable information to assist the system designer in this regard is provided in these Standards. See Section 2.10. Both the anchors and guides must be suitable for the highest pressures to be applied to the system. (NOTE: IN MOST CASES THE TEST PRESSURE WILL BE SIGNIFICANTLY HIGHER THAN THE SYSTEM OPERATING PRESSURE.)

D. The system designer shall specify those special features which best accomplish personnel protection in his particular system. Piping systems containing high pressure and/or hazardous materials which are located in close proximity to personnel shall be provided with additional safety features which will protect such personnel in the event of a failure in the system. Expansion Joints can be furnished with special features including, but not limited to, the following:

a. Extra heavy covers which would serve to impede the effect of a jet flow produced by a failure; however, such covers will not prevent the escaping medium from expanding and filling the surroundings in which it is located.

- b.** Limit rods designed for dynamic loading can be employed to restrain the longitudinal pressure thrust in the event of an anchor failure. Such rods would normally remain completely passive until the anchor restraint is removed.
 - c.** A two ply or two concentric bellows design may be employed with each ply or bellows designed to contain the full line pressure. The annular space between the plies or concentric bellows can be monitored continuously for leakage by means of suitable instrumentation. A change in pressure in the annulus could be used to detect bellows leakage. (See Section 9.1).
- E.** The system designer shall provide for the accessibility of components such as anchors and Expansion Joints in the piping system for periodic inspection after initial start up.

3.2 EXPANSION JOINT DESIGN

The Expansion Joint design shall conform to the requirements of these Standards, the ASME/ANSI Piping Codes and the ASME Boiler and Pressure Vessel Codes as applicable. The design of structural attachments shall be in accordance with accepted methods, based on elastic theory. Circular bellows design shall be based on the equations contained in Section 4.13 with substantiating test data as stated in Section 4.12. Rectangular bellows design can be evaluated based on the equations contained in Section 5.

3.3 EXPANSION JOINT MANUFACTURING QUALITY

The Expansion Joint manufacturer shall comply with the requirements of Section 6. Each manufacturer shall be required to furnish, on request, a copy of his Quality Assurance Manual.

3.4 INSTALLATION

- A.** The necessary steps for installing all Expansion Joints shall be preplanned. The installers shall be made aware of these steps as well as the special instructions furnished by the manufacturer. Section 8.3, as well as the individual instructions tags furnished by the manufacturer with the Expansion Joint, provides information necessary to the proper handling and installation of Expansion Joints.
- B.** The most critical phases of the Expansion Joint installation are as follows:
 - a.** Care shall be exercised to prevent any damage to the thin bellows section, such as dents, scores, arc strikes and weld splatter.
 - b.** No movement of the Expansion Joint (compression, extension, lateral offset, rotation) due to piping misalignment, for example, shall be imposed which has not been anticipated and designed into the movement capability of the Expansion Joint. If such movements are imposed, this can result in system malfunction, damage to the bellows or other components in the system. Specifically, cyclic life can be substantially reduced, forces imposed on adjacent equipment may exceed their design limits, internal sleeve clearances may be adversely affected, and the pressure capacity and stability of the bellows may be reduced.
 - c.** Any field pre-positioning shall be performed in accordance with specific instructions which include both the direction and magnitude of movement.
 - d.** Anchors, guides and pipe supports shall be installed in strict accordance with the piping system drawings. Any field variances from planned installation may affect proper functioning of the Expansion Joint and must be brought to the attention of competent design authority for resolution.
 - e.** The Expansion Joint, if provided with internal sleeves, shall be installed with the proper orientation with respect to flow direction.

f. Once the pipeline anchors or other fixed points are in place, the piping is properly supported and guided and the Expansion Joint installed, the shipping devices should be removed in order to allow the Expansion Joint to compensate for changes in ambient temperature during the remainder of the construction phase.

3.5 POST INSTALLATION INSPECTION PRIOR TO SYSTEM PRESSURE TEST

A. A careful inspection of the entire piping system shall be made with particular emphasis on the following:

- a.** Are anchors, guides and supports installed in accordance with the system drawings?
- b.** Is the proper Expansion Joint in the proper location?
- c.** Are the Expansion Joint flow direction and pre-positioning correct?
- d.** Have all of the Expansion Joint shipping devices been removed?
- e.** If the system has been designed for a gas, and is to be tested with water, has provision been made for proper support of the additional dead weight load on the piping and Expansion Joint? Some water may remain in the bellows convolutions after the test. If this is detrimental to the bellows or system operation, means shall be provided to remove such water.
- f.** Are all guides, pipe supports and the Expansion Joints free to permit pipe movement?
- g.** Has Expansion Joint been damaged during handling and installation?
- h.** Is Expansion Joint misaligned? This can be determined by measuring the joint overall length, inspection of the convolution geometry, and checking clearances at critical points on the Expansion Joint and at other points in the system.
- i.** Are the bellows and other movable portions of the Expansion Joint free of foreign material?

3.6 INSPECTION DURING AND IMMEDIATELY AFTER SYSTEM PRESSURE TESTS

WARNING: Extreme care must be exercised while inspecting any pressurized system or component.

A. A visual inspection of the system shall include checking for the following:

- a.** Evidence of leakage or loss of pressure.
- b.** Distortion or yielding of anchors, Expansion Joint hardware, the bellows and other piping components.
- c.** Any unanticipated movement of the piping due to pressure.
- d.** Evidence of instability (squirm) in the bellows.
- e.** The guides, Expansion Joints and other movable parts of the system shall be inspected for evidence of binding.
- f.** Any evidence of abnormality or damage shall be reviewed and evaluated by competent design authority.

3.7 PERIODIC IN-SERVICE INSPECTION

WARNING: Extreme care must be exercised while inspecting any pressurized system or component.

A. SCOPE

This section will serve as a guide for periodic inspection and review of metallic bellows expansion joints that are in service. Criteria are presented for evaluation of their suitability for continued safe operation. Frequency of service will be at owner's discretion.

B. IDENTIFICATION AND RECORDING

An expansion joint record system should be established to identify and characterize the design and operating conditions for each bellows expansion joint. The record should include tag number, process service, design and operating pressure, temperature, flow direction, fluid velocity, materials of construction, and engineering design data such as pipe size, number of convolutions, bellows wall thickness and number of plies, presence of an internal sleeve, lateral, axial, and angular movements, and design basis cycle life. The record should allow documentation of all design reviews and inspections.

C. DESIGN REVIEW

- a.** Review all bellows expansion joints periodically to confirm that current service conditions are compatible with bellows design capabilities. Changes in pressure, temperature, fluid composition, frequency of thermal and pressure cycling, and possible exposure to vibration from external means or pulsating pressure should also be reviewed and compared to the original design basis of the expansion joint. Considerations should also be given to upset or short time conditions not originally anticipated.
- b.** If current process conditions and movements imposed upon the expansion joint are within the bellows expansion joints' ratings, and there has been no corrosion, damage, or permanent distortion of the bellows, no further analysis is needed. If, however pressure, temperature, or movement exceeds the expansion joints' ratings, or if there has been excessive deformation of the convolutions, an expansion joint manufacturer should be contacted and an engineering analysis should be made to determine whether the expansion joint should be replaced.
- c.** After an initial design review of existing installations, a periodic review program should be established. The frequency of the review will depend service and environmental conditions, the potential for process changes, and the critical or hazardous nature of the service.

D. INSPECTION AND EVALUATION

- a.** All bellows expansion joints in service should be inspected periodically for mechanical damage; distortion caused by overpressure, overextension or overcompression, cracking, cracking of the bellows attachment weld, corrosion, restriction of movement from foreign material, and any others signs that might indicate premature failure.
- b.** Two Ply Testable Bellows Expansion Joints - Any bellows expansion joint designed as a two ply testable (redundant ply) and equipped with warning equipment such as a pressure gauges or pop-up detection devices should be inspected on more frequent intervals, since these expansion joints are normally specified and intended for more critical service.

c. Consider replacement of the bellows expansion joint based upon inspection results, the expected number of cycles in the plant, the design cyclic basis of the bellows, and the hazardous nature of the service.

d. The frequency of the inspection depends on the nature of the service and conditions as discussed in 4.9.1-g, and the potential for mechanical damage, vibration, and corrosion. Bellows handling extremely hazardous or lethal material should be inspected regularly.

e. The following can be used as a guide in making the inspection:

Bellows thickness. Inspect for pitting or thinning.

Bellows deformation. Inspect the bellows while in service for squirm or excessive movement.

Bellows surface. Inspect the bellows surface for the following conditions:

Wrinkles. Wrinkles are an indication that torsion has been applied to the bellows either in operation or installation. If wrinkles are present, the bellows should be replaced and the new bellows expansion joint should have measures such as hinges or round gimbals installed to prevent torsion from being induced.

Dents. Dents can reduce the life of a bellows significantly, depending upon the radius of curvature of the dented surface. Sharp dents with small radius of curvature in the convolution are more harmful than dents with a large radius of curvature.

Weld splatter. Weld splatter can be detrimental to bellows performance. If weld splatter is present, contact manufacturer for recommendations and consider possible replacement.

Foreign material. Foreign material, such as scraps of metal, wood, nuts and bolts, etc., could possibly interfere with the normal movement of the bellows. Remove any such foreign material.

Scratches. Scratches on the surface of the bellows can act as stress risers and can reduce the bellows cycle life. A scratch that runs circumferentially is more harmful than a scratch that runs longitudinally or radially.

f. Attachments to the expansion joint

In severe service applications, attachment of lugs and rings is also a potential source of cracks and these areas should be checked by dye penetrant, magnetic particle or ultrasonic NDE for any incipient cracking. Any insulation should be replaced in accordance with manufacturer's drawings.

Inspect tie rods, hinges, lugs, and rings for any distortion. Tie rods, hinges, and gimbals should be insulated in accordance with manufacturer's drawings.

E. SYSTEM OPERATION

A record shall be maintained of any changes in system operating conditions (such as pressure, temperature, thermal cycling, water treatment) and piping modifications. Any such change shall be reviewed by competent design authority to determine its effect on the performance of the anchors, guides and Expansion Joints.

F. TYPICAL CAUSES OF EXPANSION JOINT FAILURE

Bellows expansion joints will give many years of satisfactory service when they are properly designed and manufactured for specified piping system conditions. Failures can occur for many reasons, but experience has shown that certain causes of failure fall into fairly distinct categories. The following are some typical causes:

a. Shipping and handling damage. Examples:

Denting or gouging of bellows from being struck by hard objects (tools, chain falls, forklifts, adjacent structures, etc.)

Improper stacking for shipping or storage.

Insufficient protection from weather or other adverse environmental conditions.

b. Improper installation and insufficient protection during and after installation.

Examples:

Joints with internal liners installed in reverse direction with respect to flow.

Installing a joint in a location other than as prescribed by the installation drawings.

Premature removal of shipping devices.

Springing of bellows to make up for piping misalignment.

Insufficient protection from mechanical damage due to work in the surrounding area.

Insufficient protection of bellows during nearby welding operations.

Failure to remove shipping devices before system operation.

c. Improper anchoring, guiding, supporting of the piping system.

d. Anchor failure in service.

e. Bellows corrosion. Examples:

Improper selection of bellows material for the flowing medium and/or adverse external environment. Specifically, chlorides leaching from insulation, have been frequently the cause of stainless steel bellows corrosion. Stress corrosion cracking (consult material manufacturer for proper selection).

f. System over-pressurization (in-service or hydrotest).

g. Bellows vibration (mechanical or flow-induced) resulting in fatigue failure.

h. Excessive bellows movement (axial, lateral, and angular movement greater than design values).

i. Bellows erosion. Example:

Bellows without internal liner installed in a system having a very high velocity and/or erosive flowing medium.

j. Packing of particulate matter in bellows convolutions which inhibits proper movement of the bellows.

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SECTION 4 – CIRCULAR EXPANSION JOINT DESIGN

4.1 MOVEMENT EQUATIONS

Expansion Joints may be subjected to axial movement, angular rotation, lateral deflection, or any combination of these. Figure 4.2 shows a single bellows Expansion Joint subjected to axial movement only. Note that the total applied movement is absorbed by a uniform displacement of all the convolutions. This also applies to dual bellows assemblies such as universal, swing and universal pressure balanced Expansion Joints.

$$e_x = \frac{x}{N} \quad (\text{For a single bellows Expansion Joint}) \quad (4-1)$$

$$e_x = \frac{x}{2N} \quad (\text{For a dual bellows Expansion Joint}) \quad (4-2)$$

In equation (4-2) above, the value of x should include the thermal expansion of the center pipe nipple connecting the two bellows. This may be a significant factor in applications involving long center pipe nipples, or a large differential between the minimum and maximum design temperatures. When the center pipe nipple is anchored, as it is in a double Expansion Joint (see Section 1.2), each end of the assembly should be treated as a single Expansion Joint. In such a case, equation (4-1) will apply and the value of x should include the thermal expansion of that portion of the center pipe nipple which is located between the anchor base and the bellows in question.

Figure 4.3 illustrates that an Expansion Joint bellows absorbs pure angular rotation by extending uniformly on one side and compressing uniformly on the other. The movement of any convolution may be expressed as:

$$e_\theta = \frac{\theta D_m}{2N} \quad (\text{For a single bellows Expansion Joint}) \quad (4-3)$$

$$e_\theta = \frac{\theta D_m}{4N} \quad (\text{For a dual bellows Expansion Joint}) \quad (4-4)$$

As illustrated in figures 4.4 and 4.5, lateral deflection of an Expansion Joint is, in reality, a special case of angular rotation. The two bellows in a universal type Expansion Joint, or each end of the bellows of a single type Expansion Joint, rotate in opposite directions to produce the total lateral deflection y . Unlike the case of pure angular rotation, lateral deflection results in unequal movement distribution over the bellows, the amount of displacement increasing with the distance from the center of the Expansion Joint. This applies to both single and universal type Expansion Joints. Since we are concerned only with the maximum displacement per convolution which may be imposed upon any convolution in the Expansion Joint, the following equations are arranged to arrive at the maximum displacement figure. The displacement per convolution resulting from applied lateral deflection y is as follows (see Figure 4.1):

$$e_y = \frac{3D_m y}{N(L_b \pm x)} \quad \text{for a single bellows Expansion Joints,} \quad (4-5)$$

$$e_y = \frac{3D_m}{2N L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* \pm x/2)} y \quad \text{for universal Expansion Joints,} \quad (4-6)$$

The positive sign is valid for axial extension and the negative one for axial compression.

The growth of the center pipe nipple may be significant in certain Expansion Joint applications, consequently, the value of x given in equation (4-6) should be adjusted to include the axial component of this growth. In most applications, the center pipe nipple will rotate through a very small angle, so the lateral component can usually be neglected.

The angle of rotation for an unrestrained center spool is given by:

$$\theta_c = \frac{3(L^*/L_b)}{1 + 3(L^*/L_b)^2} \cdot \frac{y}{L_b} \quad (4-7)$$

A one convolution single bellows is highly resistant to shear loading and should not be used to absorb imposed lateral deflection.

For single bellows with initial angular rotation, the maximum movement per convolution due to internal pressure is

$$e_{yp} = \frac{\pi D_m K_{\theta l} P \sin(\theta/2)(L_b \pm x)}{4f_i} \quad \begin{array}{l} \text{where } +x \text{ is axial extension} \\ \text{and } -x \text{ is axial compression} \end{array} \quad (4-8)$$

4.2 COMBINING MOVEMENTS

The effects of combined movement may be calculated as follows:

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_{\theta} + |e_x| \\ e_{\theta} K_{\theta} + |e_x| \end{array} \right\} \quad (4-9)$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_{\theta} - |e_x| \\ e_{\theta} K_{\theta} - |e_x| \end{array} \right\} \quad (4-10)$$

where x is axial compression and y and θ occur in the same plane. Where x is extension, reverse the signs for e_x in the above equations. When y and θ do not occur in the same plane, they must be added vectorially and combined with e_x to find the maximum values of e_c and e_e .

All bellows are rated by the manufacturer in terms of maximum allowable axial displacement per convolution, e_c and e_e . These values are established by the physical limitation of bellows movement capability. The design of every Expansion Joint must be such that the total displacement per convolution from all sources does not exceed the rated values:

$$e_c(\text{calculated}) \leq e_c(\text{rated}) \leq e_c(\text{max}) \quad (4-11)$$

$$e_e(\text{calculated}) \leq e_e(\text{rated}) \leq e_e(\text{max}) \quad (4-12)$$

When bellows with equalizing rings are used, an additional calculation must be made to assure that there will be no interference between adjacent rings when the bellows is in the deflected position. The combined movement, e_c , is to be calculated using the outside diameter of the equalizing rings in the formulas for e_y and e_θ in place of D_m . The e_c (calculated) must not exceed the space between adjacent equalizing rings in the cold position.

The following limits should be observed to prevent excessive movements which could permanently damage the bellows:

$$e_c(\text{max}) = q - 2r_m - nt \quad \text{or distance between adjacent equalizing rings} \\ \text{whichever is less}$$

$$e_e(\text{max}) = 6r_m - q$$

Figures 4.3, 4.4, and 4.5 illustrate that as an Expansion Joint is rotated or deflected laterally, the configuration of the bellows changes appreciably.

It should be noted that one side of the bellows attains a larger projected area than on the opposite side. When pressure is applied, unbalanced forces are set up which tend to distort the Expansion Joint further. To control the effect of this factor, a limit is established by the manufacturer upon the amount of angular rotation and/or lateral deflection which may be imposed upon the Expansion Joint.

4.3 MOVEMENT RANGE

The equivalent axial movement range per convolution, (e), results from the movement of an Expansion Joint from its initial position in the piping system to the operating position under consideration. When an Expansion Joint is installed without lateral or angular cold spring, e is the greater of e_c or e_e as calculated from the initial to the operating position under consideration. When cold springing is involved the e_c and e_e due to the cold spring must be added algebraically to the e_c and e_e due to movement from the neutral to the operating position in order to obtain the maximum movement range, e . The value of e for each condition is used in the calculation of bellows deflection stress range in order to evaluate bellows fatigue life. See Section 4.12.1.5.

Refer to Appendix J Example 9 for a sample calculation.

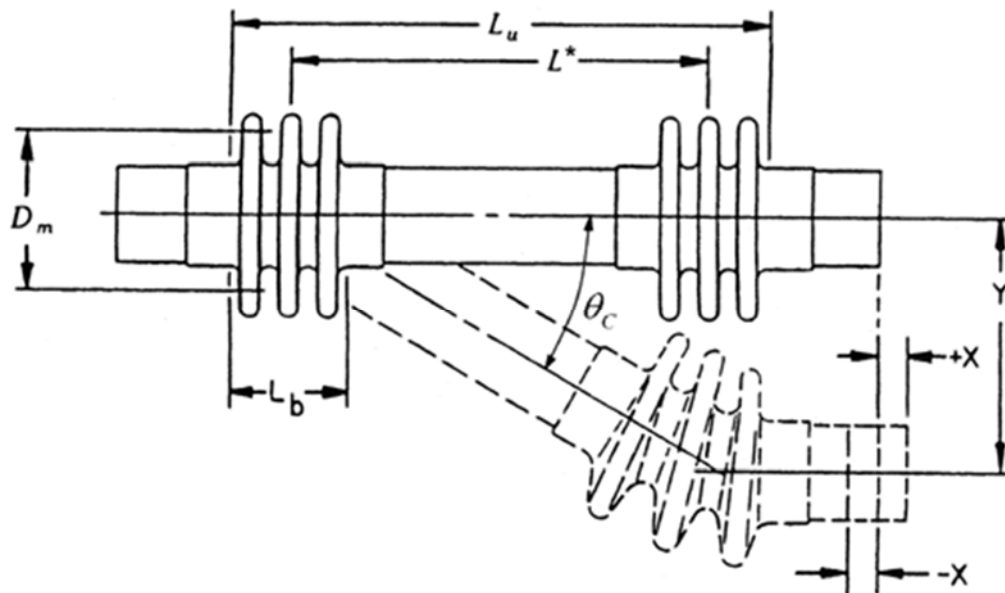


FIGURE 4.1

4.4 UNIVERSAL CIRCULAR EXPANSION JOINT MOVEMENTS

The unrestrained non-cyclic movements of a universal expansion joint centerspool due to dead weight should be considered in the design. The movements applied to **each** bellows may be calculated as follows:

$$x = \frac{W_{cs} \sin \theta_u N}{2f_i} \quad (\text{for axial movement}) \quad (4-13)$$

$$y = \frac{W_{cs} \cos \theta_u N (L_b \pm x)^2}{3f_i D_m^2} \quad (\text{for lateral movement}) \quad (4-14)$$

The above movements should be combined with the other design movements to confirm that the total movements per convolution e_c and e_e do not exceed the maximum value e_c (max) and e_e (max). In addition, the calculated total stress range (St) based on the above movements at the design pressure must be less than $1.5 C_m S_{ab}$. The dead weight of the centerspool may be supported by devices such as pantograph linkages and slotted hinges.

4.5 COLD SPRING OF CIRCULAR EXPANSION JOINTS

The term "Cold Springing," as defined by the Piping Designer, entails pre-straining of the elements of a piping system at the time of installation, so that the thermal stresses in the piping in the operating position are appreciably reduced. As applied to Expansion Joints, the purpose of cold springing may be considerably different, although the mechanism is basically the same.

"Cold Springing" is defined as the lateral or angular offset of the ends of an Expansion Joint when installed and should not be confused with the terms "pre-compressing," "pre-extending" or "presetting." These latter terms apply to the adjustment of an Expansion Joint in an axial direction to allow for specified amounts of axial compression or axial extension within the limits e_c and e_e established by the manufacturer. In some cases it may not be practical to cold spring an Expansion Joint at the factory. The reasons for "Cold Springing" an Expansion Joint are described below.

4.5.1 FORCE REDUCTION

In a wide range of present day applications, the force required to deflect an Expansion Joint is of significant importance. Where the Expansion Joint is used to relieve loading on sensitive equipment, or anchor structures are limited to extremely small loads, cold springing the Expansion Joint at installation will effect a reduction in the maximum deflection force value of as much as 50%. In other cases, 100% cold spring may be used to provide minimum lateral deflection forces at the operating position.

4.5.2 STABILITY

Figures 4.3, 4.4, and 4.5 illustrate the positions assumed by bellows subjected to angular rotation and/or lateral deflection. In all cases, the movement is achieved by rotation of the convolutions, so that one side is extended and the other compressed. It has been noted previously that a bellows displaced in this manner, when subjected to internal pressure, is

acted upon by an unbalanced pressure force or couple which, if sufficiently large, could result in distortion of the bellows.

Because the magnitude of the unbalanced pressure force or couple is proportional to the internal pressure and the displacement of the convolutions, a reduction in either of these values will improve the stability of the Expansion Joint. By cold springing the Expansion Joint 50% at installation, the maximum displacement per convolution is reduced by half and, consequently, the Expansion Joint becomes far more stable than would be the case if it were deflected fully in one direction. For this reason, where Expansion Joints are subject to large amounts of lateral deflection, or where operating pressures are relatively high, the Expansion Joint manufacturer may require that the Expansion Joint be installed in a cold sprung condition.

4.5.3 COMPONENT CLEARANCES

Where an Expansion Joint is furnished with internal sleeves, external covers, or tie devices spanning the bellows, these components must be designed with adequate clearances to accommodate the lateral deflection or angular rotation of the Expansion Joint. The amount of clearance required is directly proportional to the displacement and, if the Expansion Joint is cold sprung 50%, these clearances can be reduced to a minimum. By cold springing, internal sleeves of maximum diameter can be furnished, the overall diameter of an Expansion Joint incorporating external covers or tie devices minimized, and the design of external structures simplified.

4.6 FORCES AND MOMENTS (See Appendix H)

4.6.1 FORCE AND MOMENT CALCULATION

In order to evaluate the loads upon piping, supports, or equipment, it is necessary to determine the forces and moments required to move an Expansion Joint. For this reason, the catalogs of most Expansion Joint manufacturers contain force data for the standard designs offered. This data is expressed as the force required to move a convolution to the rated axial movement established by the manufacturer. For convenience, it is desirable to divide this force by the rated movement to obtain a bellows resistance factor or working spring rate, f_w . (Refer to Section 4.12.1.7 for further discussion of f_w). Having determined this factor, the moments and forces required to move an Expansion Joint may be calculated as follows:

$$F_a = f_w e_x \quad (4-15)$$

$$M_l = \frac{f_w D_m e_y}{4} \quad (\text{for lateral movement}) \quad (4-16)$$

$$M_\theta = \frac{f_w D_m e_\theta}{4} \quad (\text{for angular rotation}) \quad (4-17)$$

$$V_l = \frac{f_w D_m e_y}{2(L_b \pm x)} \quad (\text{for lateral movement of a single bellows}) \quad (4-18)$$

$$V_l = \frac{f_w D_m e_y}{2(L_u \pm x)} \quad (\text{for lateral movement of a universal bellows}) \quad (4-19)$$

Figures 4.2 through 4.5 show the forces and moments applied on the expansion joints to produce static equilibrium for the various types of movement.

The preceding relationships are applicable to all Expansion Joints. It should be noted, that every equation is dependent upon data which must be supplied by the Expansion Joint manufacturer. For standard designs, all necessary data is available in the catalogs of the individual manufacturer, or can be obtained on request. **IN NO CASE SHOULD DATA OF ONE MANUFACTURER BE APPLIED TO THE PRODUCT OF ANOTHER SINCE, DUE TO FUNDAMENTAL DESIGN DIFFERENCES, THESE FACTORS MAY VARY.**

NOTE: "x", "y", and "θ" are all to be expressed from the initial installed position of the Expansion Joint to the position under consideration. When cold spring is involved or when there are several sets of thermal conditions to be considered (system at operating temperature and system shutdown in a sub-zero ambient, for example), the x , y , and θ deflections should be determined for each condition and separate e_x , e_y , e , e_c , e_e , and e calculations made for each condition as described in Section 5.4.

4.6.2 RESTRAINT HARDWARE FORCE AND MOMENT CALCULATIONS

4.6.2.1 TIE RODS

With no lateral offset, tie rods are normally parallel to the longitudinal axis passing through the ends of the expansion joint. With lateral offset, the tie rods angulate with respect to that axis. When pressure thrust is applied, tie rod angulation generates a lateral force that is opposite to the direction of the lateral offset. The resulting lateral force for the expansion joint with tie rods without spherical washers is given by the following equation

$$F_l = \frac{F_t (y + D_n)}{L_{tr}} \quad (4-20)$$

When spherical washers are used, the maximum lateral force results from both the pressure thrust and washer friction as given by the following equations:

$$F_l = F_t \left[\frac{y}{L_{tr}} + \frac{2R_s \mu_s}{L_{tr} + 2R_s} \right] \quad \text{when the outer washer is convex} \quad (4-21)$$

$$F_l = F_t \left[\frac{y}{L_{tr}} + \frac{2R_s \mu_s}{L_{tr} - 2R_s} \right] \quad \text{when the outer washer is concave} \quad (4-22)$$

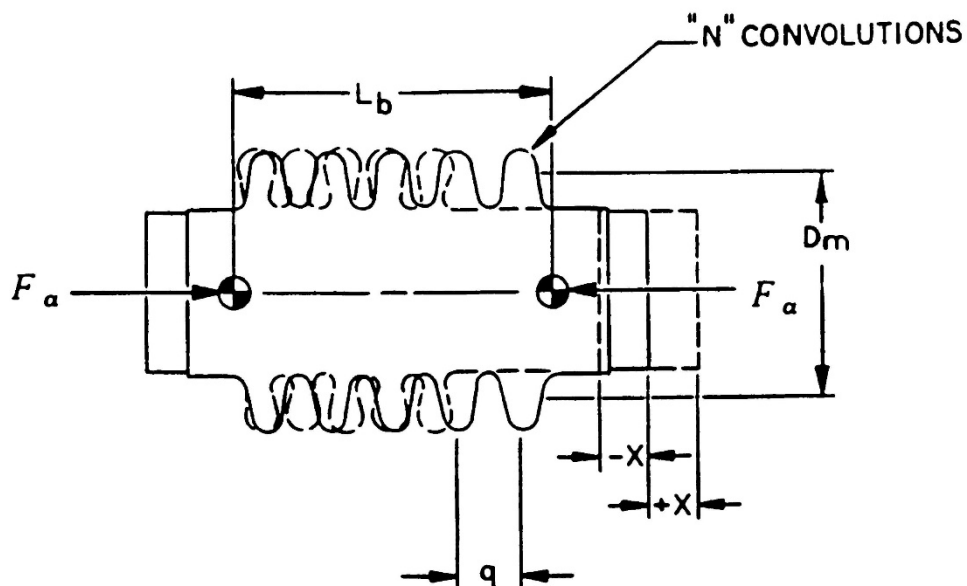
The lateral force due to friction occurs only when the expansion joint is moving or just starting to move.

4.6.2.2 HINGES AND GIMBALS

The rotation of hinge and gimbal hardware is resisted by pin friction. The maximum resisting frictional moment for the expansion joint is given by

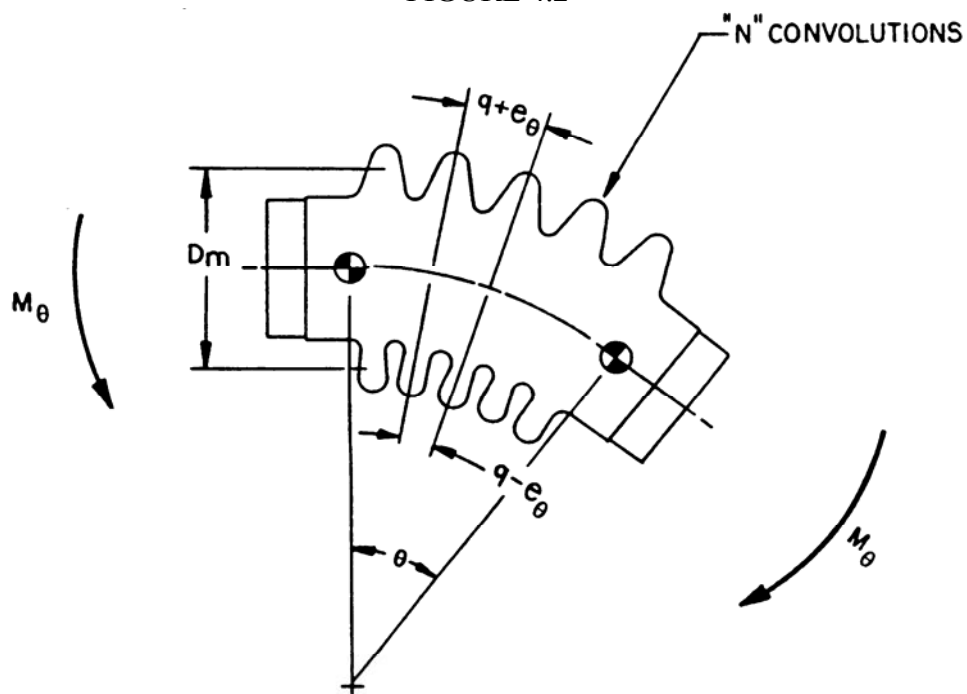
$$M_p = \frac{\mu_p F_t D_p}{2} \quad (4-23)$$

The frictional moment occurs only when the expansion joint is moving or just starting to move. Any contact between adjacent side plate surfaces will increase the resisting moment.



AXIAL MOVEMENT (SINGLE EXPANSION JOINT)

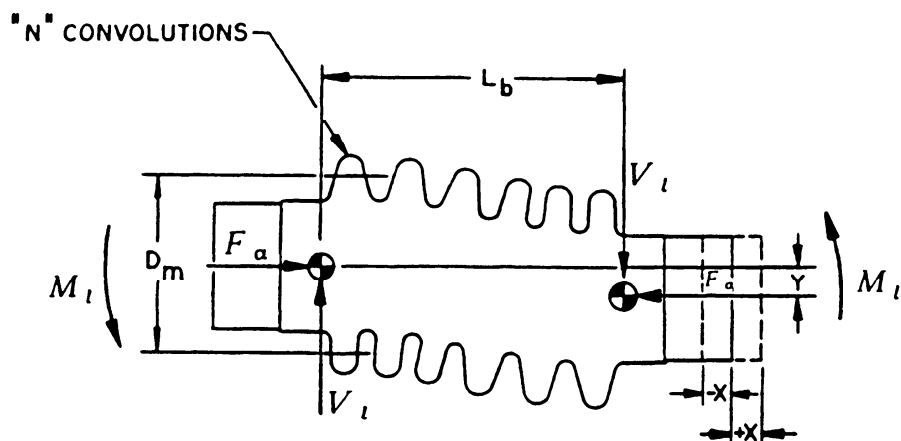
FIGURE 4.2



ANGULAR ROTATION (SINGLE EXPANSION JOINT)

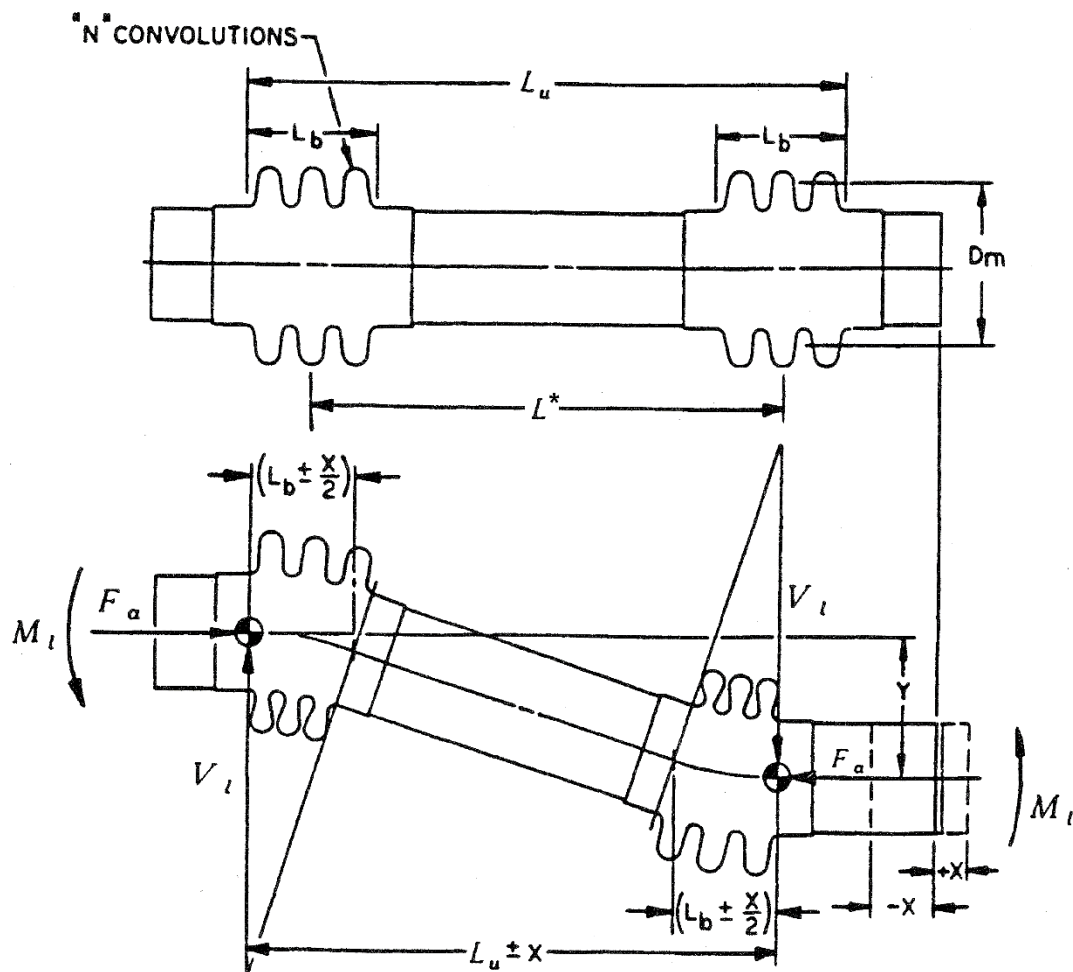
FIGURE 4.3

$\odot$ POINT OF APPLICATION OF EXTERNAL FORCES AND MOMENTS



LATERAL DEFLECTION (SINGLE EXPANSION JOINT)

FIGURE 4.4



LATERAL DEFLECTION (UNIVERSAL EXPANSION JOINT)

FIGURE 4.5

⊙ POINT OF APPLICATION OF EXTERNAL FORCES AND MOMENTS

4.7 MAXIMUM AXIAL COMPRESSION BASED ON INSTABILITY

A long bellows or a series of unguided interconnected bellows may sometimes buckle when compressed. Buckling occurs when the lateral stiffness is insufficient to resist the lateral forces generated by axial compression of the bellows.

The max. axial compression movement per convolution based on instability is:

$$e_x = \frac{1.25 D_m^2}{N^2 q}$$

N = Total number of convolutions in all unguided interconnected bellows.

4.8 EXPANSION JOINT FLANGE LOADING CONSIDERATIONS

Typically a flange connection is required to withstand the axial thrust that is produced during operation of a piping system as shown in Figure 4.6. The axial force that results from the pressure being applied against the elbow is restrained by the flange bolts, thus creating a force and moment on the flange at the flange connection attempting to unseat the gasket.

However, when an unrestrained expansion joint is employed as shown in Figure 4.7, the flange loading conditions change dramatically. In order to keep the expansion joint from freely extending, a main anchor is normally utilized to restrain the elbow. This main anchor will also carry the pressure thrust that results from the pressurization of the system and release this load from the flange bolts. In this instance, the loading on the flange due to pressure is a compressive load that is equal to the $(F_s - F_p)$ as shown in Figure 4.7. This compressive load is in addition to that normally applied due to gasket seating.

Refer to Appendix J Examples 1-6 and 8 for sample calculations.

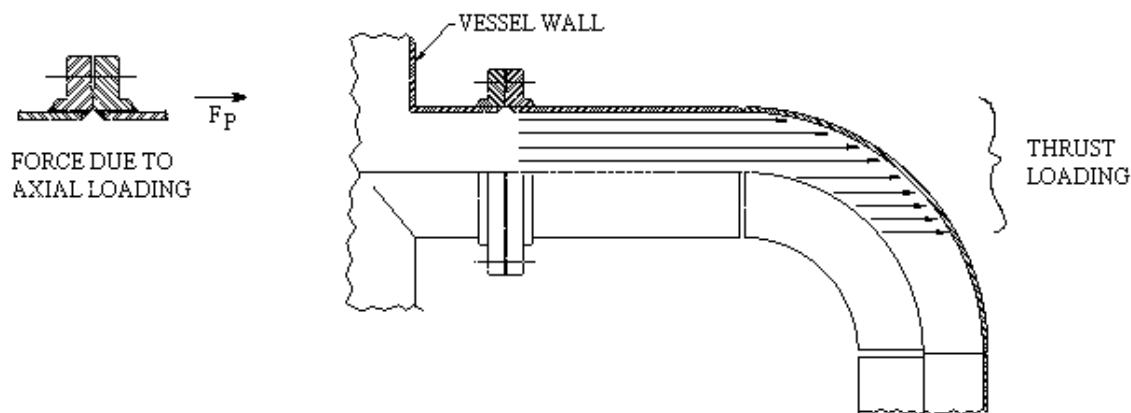


FIGURE 4.6

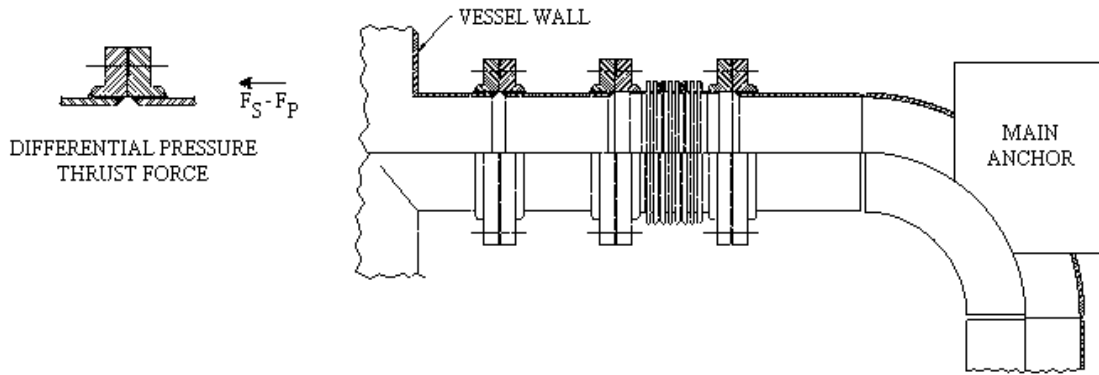


FIGURE 4.7

4.9 VIBRATION

Metal bellows can be used in applications where the vibration is of high frequency and low amplitude. They are not suitable for vibrations where the frequency is low and the amplitude high, such as those resulting from reciprocating machines. Vibrations which are the result of pressure pulses can not be removed by the installation of an Expansion Joint, since the pressure pulses are transmitted beyond the Expansion Joint through the flow media. In this case, a pulsation dampener is required.

The piping system designer should insure that vibration loads in his piping system will not be detrimental to the function of the bellows. In reducing or eliminating vibration effects the designer may wish to consider the use of external dampening devices or system mass adjustments.

Where flow velocities are high, turbulent flow generated within the bellows section or turbulence originating upstream of the bellows may induce vibration. To minimize this phenomenon, an internal sleeve must be used. Refer to Section 4.10 for specific recommendations.

Theoretical natural frequencies of single bellows and dual bellows assemblies for axial and lateral vibration may be calculated using the following equations.

4.9.1 SINGLE BELLOWS

When vibration is present and the frequency is known, the bellows shall be designed so that its natural frequency (f_n) and higher modes do not coincide with the system frequency. To avoid a resonant response in the bellows, the bellows natural frequency shall be less than 2/3 of the system frequency or greater than 2 times the system frequency.

Axial Vibration: (Accordion Mode)

$$f_n = C_n \sqrt{\frac{K_{sr}}{m}} \text{ (hertz)} \quad (4-24)$$

m = Mass of the bellows including reinforcement, lbm (kg). For liquid media, include the mass of liquid contained only between the convolutions.

C_n = A constant used in the calculation of single bellows axial and lateral vibration frequencies. Use C_1 for natural or fundamental frequency, C_2 for first harmonic, etc.

$$n = 1, 2, 3, 4, 5 \dots$$

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Values of " C_n " for first 5 modes (US units)

Number of Convolutions	$\underline{C_1}$	$\underline{C_2}$	$\underline{C_3}$	$\underline{C_4}$	$\underline{C_5}$
1	8.84	-	-	-	-
2	9.57	17.7	23.1	-	-
3	9.71	18.8	26.5	32.5	36.2
4	9.76	19.1	27.8	35.4	41.6
5	9.78	19.3	28.4	36.8	44.2
6	9.80	19.4	28.7	37.5	45.7
7	9.80	19.5	28.9	38.0	46.6
8	9.81	19.5	29.0	38.3	47.2
9	9.81	19.5	29.1	38.5	47.6
10	9.81	19.6	29.2	38.7	47.9
11 & above	9.82	19.6	29.2	38.8	48.1

Values of " C_n " for first 5 modes (Metric units)

Number of Convolutions	$\underline{C_1}$	$\underline{C_2}$	$\underline{C_3}$	$\underline{C_4}$	$\underline{C_5}$
1	14.23	-	-	-	-
2	15.41	28.50	37.19	-	-
3	15.63	30.27	42.66	52.32	58.28
4	15.71	30.75	44.76	56.99	66.97
5	15.75	31.07	45.72	59.24	71.16
6	15.78	31.23	46.20	60.37	73.57
7	15.78	31.39	46.53	61.18	75.02
8	15.79	31.39	46.69	61.66	75.99
9	15.79	31.39	46.85	61.98	76.63
10	15.79	31.55	47.01	62.30	77.12
11 & above	15.81	31.55	47.01	62.46	77.44

Lateral Vibration: (Beam Mode)

$$f_n = \frac{C_n D_m}{L_b} \sqrt{\frac{K_{sr}}{m}} \text{ (hertz)} \quad (4-25)$$

m = Mass of the bellows including reinforcement, lbm. (kg). For liquid media, include the mass of a column of fluid of diameter D_m and length L_b .

Values of " C_n " for first 5 modes (US units)

$\underline{C_1}$	$\underline{C_2}$	$\underline{C_3}$	$\underline{C_4}$	$\underline{C_5}$
24.8	68.2	133	221	330

Values of " C_n " for first 5 modes (Metric units)

$\underline{C_1}$	$\underline{C_2}$	$\underline{C_3}$	$\underline{C_4}$	$\underline{C_5}$
39.93	109.80	214.12	355.79	531.27

4.9.2 DUAL BELLOWS (Universal Expansion Joint)

Resonant vibration of dual bellows assemblies may be very severe, particularly at lower frequencies, resulting in large displacements of the center spool pipe. In tied assemblies, unanticipated lateral and axial movements of the center pipe may be controlled with hardware to limit vibratory motion. To avoid a resonant response in the bellows, the bellows natural frequency shall be less than 2/3 of the system frequency or greater than 2 times the system frequency. Alternatively, test methods may be used to determine expansion joint frequency response. Higher modes or harmonics do not occur in a spring mass system such as the dual bellows. The individual bellows in a dual assembly should also be checked for vibration response as a single bellows. Natural frequencies for axial, lateral, or "rocking" vibration may be calculated using the following equations:

Axial Vibration:

$$f_n = 4.43 \sqrt{\frac{K_{sr}}{m}} \text{ hertz} \quad (4-26)$$

$$f_n = 7.13 \sqrt{\frac{K_{sr}}{m}} \text{ hertz} \quad (4-26M)$$

m = Mass of the spool pipe + one bellows including reinforcement + any attachments to the spool pipe including liners, covers, trunnions, lugs, nozzles, refractory, and insulation, lbm (kg). For liquid media, include the mass of liquid contained only between the convolutions of one bellows.

Lateral Vibration: Ends of spool pipe in phase.

$$f_n = \frac{5.42D_m}{L_b} \sqrt{\frac{K_{sr}}{m}} \text{ hertz} \quad (4-27)$$

$$f_n = \frac{8.73D_m}{L_b} \sqrt{\frac{K_{sr}}{m}} \text{ hertz} \quad (4-27M)$$

m = Mass of the spool pipe + one bellows including reinforcement + any attachments to the spool pipe including liners, covers, trunnions, lugs, nozzles, refractory, and insulation, lbm (kg). For liquid media, include the mass of a column of fluid of diameter D_m and length $(L_u - L_b)$.

Rocking Vibration: Lateral vibration with ends of spool pipe out of phase; one end up and one end down.

$$f_n = \frac{9.38D_m}{L_b} \sqrt{\frac{K_{sr}}{m}} \text{ hertz} \quad (4-28)$$

$$f_n = \frac{15.1D_m}{L_b} \sqrt{\frac{K_{sr}}{m}} \text{ hertz} \quad (4-28M)$$

m = Mass of the spool pipe + one bellows including reinforcement + any attachments to the spool pipe including liners, covers, trunnions, lugs, nozzles, refractory, and insulation, lbm. (kg). For liquid media, include the mass of a column of fluid of diameter D_m and length $(L_u - L_b)$.

The rocking vibration natural frequency equation is based on having the center of gravity located at the center of the spool pipe. Attachments to the spool pipe may shift the center of gravity off center and reduce the natural frequency below the calculated value.

NOTE: A properly designed, close tolerance, pantographic linkage could be used to suppress possible vibration tendencies in a universal expansion joint assembly.

4.9.3 INTERNAL SLEEVES - CIRCULAR EXPANSION JOINTS

The natural frequency of an internal sleeve at the design temperature with one end rigidly attached may be calculated using the following equation:

$$f_n = \frac{10.886}{L_{is}} \sqrt{\frac{t_{is} E_{is}}{D_{is}}} \text{ hertz} \quad (4-29)$$

$$f_n = \frac{3329.93}{L_{is}} \sqrt{\frac{t_{is} E_{is}}{D_{is}}} \text{ hertz} \quad (4-29M)$$

L_{is} = Length of internal sleeve, in. (mm)

t_{is} = Thickness of internal sleeve, in. (mm)

E_{is} = Modulus of elasticity of internal sleeve at design temperature, psi (MPa)

D_{is} = Mean diameter of the internal sleeve, in. (mm)

4.10 INTERNAL SLEEVES - CIRCULAR EXPANSION JOINTS

4.10.1 CRITERIA FOR DETERMINING THE NEED FOR INTERNAL SLEEVES

Internal sleeves shall be specified for all Expansion Joints in the following cases:

- a. When flow velocities are high and could induce resonant vibration of the bellows (see Section 4.10.2).
- b. When it is necessary to hold friction losses to a minimum and smooth flow is desired.
- c. When there is a possibility of erosion, as in lines carrying catalyst or other abrasive media, heavy gauge sleeves must be used. At no time should the relatively thin bellows be directly exposed to erosion.
- d. When there is reverse flow, heavy gauge sleeves may be required, or the use of telescopic sleeves may be appropriate.
- e. For high temperature applications to decrease the temperature of the bellows and enable the bellows metal to retain its higher physical properties. The annular area between the bellows and liner may be packed with ceramic fiber insulation, or a gas purge may be installed to further reduce the bellows effective temperature.
- f. Internal sleeves should not be used where high viscosity fluids such as tars are being transmitted, since these fluids may cause "packing up," "coking" and "caking" which, may cause premature Expansion Joint failure. Where the fluid is such that purging will effectively prevent the "packing up," internal sleeves may be used in conjunction with purge connections.

4.10.2 LIMITS FOR FLOW VELOCITIES

This paragraph gives limits for flow velocities which can be tolerated by the bellows without using an internal sleeve based on the effect of flow induced vibration.

- a. Values for allowable velocities V_{alw} were found either by practical experience (see Table 4.10-1) or from empirical data; see equation (4-30). Flow energy and bellows ply interaction are considered.

TABLE 4.10-1 ALLOWABLE FLOW VELOCITY

Fluid	Liquids					Gases				
Number of plies n	1	2	3	4	5	1	2	3	4	5
Nominal diameter in.	Allowable flow velocity V_{alw} in ft/sec ¹									
2	4	6	7	8	9	8	11	14	16	18
4	7	10	12	14	16	16	23	28	32	36
≥ 6	10	14	17	20	22	24	34	42	48	54
¹ Velocity values to be interpolated for intermediate nominal diameters										

TABLE 4.10-1M ALLOWABLE FLOW VELOCITY

Fluid	Liquids					Gases				
Number of plies <i>n</i>	1	2	3	4	5	1	2	3	4	5
Nominal diameter mm	Allowable flow velocity v_{alw} in m/sec ¹									
50	1.22	1.83	2.13	2.44	2.74	2.44	3.35	4.27	4.88	5.49
100	2.13	3.05	3.66	4.27	4.88	4.88	7.01	8.53	9.75	10.97
≥ 150	3.05	4.27	5.18	6.10	6.71	7.32	10.36	12.80	14.63	16.46
¹ Velocity values to be interpolated for intermediate nominal diameters										

Specific applications shall be evaluated by the following equation which includes a safety factor of 1.33:

$$v_{alw} = 1.35 \ q \ K_i \sqrt{\frac{K_{sr}}{m_{eff}}} n \ , \text{ ft/sec} \quad (4-30)$$

$$v_{alw} = 0.026 \ q \ K_i \sqrt{\frac{K_{sr}}{m_{eff}}} n \ , \text{ m/sec} \quad (4-30M)$$

where

K_i is the influence factor depending on the flow media

1 for liquids

2 for gases.

m_{eff} is the effective mass of the bellows including reinforcement, and the mass of liquid contained between the convolutions, lbm (kg).

Where an internal sleeve is not provided the allowable flow velocities shall not be greater than:

25 ft/sec (7.6 m/sec) for liquids

65 ft/sec (19.8 m/sec) for gases.

- b.** The flow velocity through the bellows or internal sleeve must consider the characteristics of the upstream flow, especially the conditions generated from elbows, tees, valves, and cyclonic devices. The local maximum flow velocity through the bellows or internal sleeve is:

$$v_{max} = K_t \ v_{ave} \ , \text{ where} \quad (4-31)$$

K_t is the flow acceleration factor given in Table 4.10-2

v_{ave} is the calculated average flow velocity inside of the bellows or the internal sleeve, ft/sec (m/sec)

TABLE 4.10-2 FLOW ACCELERATION FACTOR

K_t	Length of upstream straight pipe ¹	Upstream components
1.0	$\geq 10 D_i$	Any
1.5	$< 10 D_i$	1 or 2 elbows
2.0	$< 10 D_i$	3 or more elbows
2.5	$< 10 D_i$	1 valve, tee, or cyclonic device
4.0	$< 10 D_i$	2 or more valves, tees, or cyclonic devices
¹ Between components and bellows		

All causes for increased flow velocity at the bellows shall be considered in the calculations.

To avoid a resonant condition in a bellows without an internal sleeve, the maximum flow velocity v_{max} must not be higher than the allowable flow velocity v_{alw} .

4.10.3 DESIGN RECOMMENDATIONS FOR INTERNAL SLEEVES

- a. To minimize the possibility of flow induced vibration of the inner sleeve a minimum sleeve thickness shall be designed. Sleeve length, flow velocity, and media temperature can increase the minimum internal sleeve thickness. Thickness increase factors shall be calculated in accordance with the following equations
- b. The following increased minimum internal sleeve thickness t_s for the application shall be utilized:

$$t_s = C_l C_v C_t t_{s, \min}, \text{ where} \quad (4-32)$$

the empirically based minimum sleeve thickness $t_{s, \min}$ is given in Table 4.10-3

the length factor is defined as:

$$C_l = \begin{cases} 1 & \text{if } L_{sl} \leq 18 \text{ in} \\ \sqrt{L_{sl}/18} & \text{if } L_{sl} > 18 \text{ in} \end{cases}, \text{ with } L_{sl} \text{ the sleeve length}$$

$$C_l = \begin{cases} 1 & \text{if } L_{sl} \leq 450 \text{ mm} \\ \sqrt{L_{sl}/450} & \text{if } L_{sl} > 450 \text{ mm} \end{cases}, \text{ with } L_{sl} \text{ the sleeve length}$$

the velocity factor is defined as:

$$C_v = \begin{cases} 1 & \text{if } v_{max} \leq 100 \text{ ft/sec} \\ \sqrt{v_{max}/100} & \text{if } v_{max} > 100 \text{ ft/sec} \end{cases}, \text{ with } v_{max} \text{ the max. flow velocity}$$

$$C_v = \begin{cases} 1 & \text{if } v_{max} \leq 30 \text{ m/sec} \\ \sqrt{v_{max}/30} & \text{if } v_{max} > 30 \text{ m/sec} \end{cases}, \text{ with } v_{max} \text{ the max. flow velocity}$$

according to Section 4.10.2.

the temperature factor is given by:

$$C_t = \begin{cases} 1 & \text{if } T_{max} \leq 300^\circ \text{ F} \\ E_{sc}/E_{sh} & \text{if } T_{max} > 300^\circ \text{ F} \end{cases}, \text{ with } T_{max} \text{ the maximum media temperature,}$$

$$C_t = \begin{cases} 1 & \text{if } T_{max} \leq 150^\circ \text{ C} \\ E_{sc}/E_{sh} & \text{if } T_{max} > 150^\circ \text{ C} \end{cases}, \text{ with } T_{max} \text{ the maximum media temperature,}$$

E_{sc} is modulus of elasticity at 300°F (150°C), psi (MPa)

E_{sh} is modulus of elasticity of sleeve at design temperature, psi (MPa)

- c. Where lateral deflection or rotation is present, the internal sleeve must be sufficiently smaller in diameter to provide clearance between the outside diameter of the sleeve and the inside diameter of the bellows or pipe. If the reduction of inside diameter is unacceptable, an oversize bellows or alternate expansion joint design must be used. Cold spring can sometimes be used to provide the necessary clearance (See Section 4.5.3).
- d. Drain holes should be provided for vertical installations where liquid could become trapped inside the sleeve.
- e. Internal sleeves designed only to minimize the possibility of flow induced vibration shall not be considered as substitutes for internal guide sleeves described in Section 1.2.
- f. The internal sleeve material should normally be the same as the bellows material. Other materials may be used provided they are suitable for the application.

TABLE 4.10-3 MINIMUM INTERNAL SLEEVE THICKNESS

Nominal Expansion Joint Diameter Inches	Minimum Sleeve Thickness t_s , min Inches
2-3	0.024
4-10	0.036
12-24	0.048
26-48	0.060
50-72	0.075
> 72	0.090

TABLE 4.10-3M MINIMUM INTERNAL SLEEVE THICKNESS

Nominal Expansion Joint Diameter mm	Minimum Sleeve Thickness t_s , min mm
50-80	0.61
100-250	0.91
300-600	1.22
650-1200	1.52
1250-1800	1.91
> 1800	2.29

4.11 EXTERNAL COVERS – CIRCULAR EXPANSION JOINTS

External covers shall be specified for all expansion joints based on the following criteria.

4.11.1 FLOW INDUCED VIBRATION

When the vortex shedding frequency from bellows due to external flow is close to the natural frequency of vibration of the bellows, this can cause damage due to resonant interaction. The lowest axial and lateral flow velocities that may induce resonant vibration in the bellow may be calculated using the following equations:

a) Lowest freestream axial velocity over bellows

$$V_{axial} = 4.6w\sqrt{\frac{K_{sr}}{m}} \text{ ft/sec} \quad (4-33)$$

$$V_{axial} = 0.089w\sqrt{\frac{K_{sr}}{m}} \text{ m/sec} \quad (4-33M)$$

where

m = Mass of bellows including media fluid, lbm (kg)

b) Lowest freestream lateral velocity over bellows

$$V_{lateral} = 2.0 \left(\frac{D_m^2}{L_b} \right) \sqrt{\frac{K_{sr}}{m}} \text{ ft/sec} \quad (4-34)$$

$$V_{lateral} = 0.039 \left(\frac{D_m^2}{L_b} \right) \sqrt{\frac{K_{sr}}{m}} \text{ m/sec} \quad (4-34M)$$

where

m = Mass of bellows including media fluid, lbm (kg)

A cover must be used when the actual freestream velocity over the bellows exceeds 75% of the corresponding values from either of the above equations.

4.11.2 DRAG FORCE

The non-cyclic movement of the single bellows due to the drag force produced by lateral flow over the outside of the bellows should be considered in the design. This movement should be combined with the other design movements to confirm that the total movements per convolution e_c and e_e do not exceed the maximum value e_c (max) and e_e (max). In addition, the calculated total stress range (St) based on the lateral movement with the design pressure must be less than $1.5 C_m S_{ab}$. The lateral movement may be calculated as follows:

$$y = \frac{\rho V^2 N (L_b \pm x)^3}{46368 f_i D_m} \text{ in}$$

$$y = \frac{\rho V^2 N (L_b \pm x)^3}{10^7 f_i D_m} \text{ mm}$$

where

ρ = Density of the fluid flowing over the bellows, lbm/ft³ (kg/m³)

V = Flow velocity of the fluid, ft/sec (m/sec)

A cover must be used when the design criteria cannot be met.

4.12 BELLOWS DESIGN

The design of a bellows is complex in that it involves an evaluation of pressure capacity, stress due to deflection, fatigue life, spring forces and instability (squirm). Instability is unique in that users do not generally recognize that internal pressure can cause a bellows to buckle in a manner similar to a column subjected to compressive loading. The determination of an acceptable design is further complicated by the numerous variables involved such as diameter, material thickness, pitch, height, number of plies, method of reinforcement, manufacturing technique, material type, and heat treatment. In many cases, the design for a particular application will involve a compromise of conflicting requirements. For example: high pressure necessitates a bellows constructed of thick material while low forces require a thinner material.

Several noteworthy theoretical stress analyses of bellows have been developed, each of which has inherent limitations. The analyses are normally based on assumptions which approximately predict the true behavior of a bellows. The assumptions usually consist of an idealized bellows configuration, a uniform thickness, a homogeneous and isotropic material, and elastic behavior. These assumptions are not precisely correct for most applications. A bellows usually operates in the plastic stress region and cold work, due to forming, alters the mechanical properties of the material. A few investigators have employed computerized analysis techniques to more accurately consider the effect of thickness and shape variations as well as plasticity. This procedure is obviously more complex than a simple elastic analysis and yet does not fully solve the design problem in the absence of experimental verification.

The major stresses in a bellows result from the effects of pressure and deflection. Normally the deflection stresses are higher than the pressure stresses, are generally above the yield point of the bellows material, and are meridional (longitudinal) in direction.

Pressure produces circumferential (hoop) membrane stress in the bellows tangent and convolutions. Both meridional membrane and bending stresses are also produced in the convolutions by pressure.

A toroidal cross section is superior for high pressure capacity, but is limited to small deflections. Conversely, a U-shaped cross section permits greater deflection but has a lower pressure capacity for the same material thickness. One method of providing a combination of high internal pressure capacity and large deflection is the use of external reinforcement of the U-shaped bellows. The external reinforcement offers circumferential restraint and supports the root radius against collapse from internal pressure loading. The pressure capacity of a bellows can also be increased by the use of multi-ply construction or by increasing the thickness of the bellows; however, the latter can significantly reduce the bellows fatigue life.

Fatigue life of a bellows is influenced by the combined stress range induced by pressure and deflection. The fatigue life of the bellows for a given configuration and material thickness will be a function of the imposed pressure and deflection.

The spring forces exerted by a deflected bellows may be critical. A deep convolution with a thin wall will deflect with less force than a shallow convolution with a thick wall.

A bellows design should always be based on the actual bellows metal temperature expected during operation. This temperature may be less than the media temperature.

It should be understood that bellows geometry as well as forming methods vary widely throughout the industry with no one configuration and forming method necessarily superior for all design conditions.

4.12.1 PARAMETERS AND CRITERIA AFFECTING BELLOWS DESIGN

This Standard contains a series of equations intended to provide the users and designers of Expansion Joints with a meaningful method for evaluating the various parameters affecting bellows design.

The equations in Section 4.13 can be used to design a bellows for specific pressure and cyclic movement conditions only if they have been correlated with actual bellows test data in accordance with Section 4.12.1.8. Modifying factors may be used by a manufacturer provided that they are in accordance with this test data. When the available test data is not sufficient to verify the equations for a specific application, an individual bellows design may be considered acceptable for specific pressure temperature and movement conditions when a history of successful operation of a similar bellows size and configuration for identical or more severe service can be demonstrated to the satisfaction of the purchaser. Determination of the suitability of the design may include the use of the equations in Section 4.13 on a parametric basis.

4.12.1.1 UNREINFORCED BELLOWS

The equations for unreinforced bellows are based on those shown in Atomics International Report NAA-SR-4527 "Analysis of Stresses in Bellows, Part 1, Design Criteria and Test Results," with modifications and additions by the Association to reflect the experience of the members. These equations are based on elastic shell theory and consider the parameters involved for bellows of the "U" shaped configuration.

The equations shown in Section 4.13.1 are taken from the Atomics International report with modifications such that the calculated stresses in equations (4-35), (4-36), (4-37), (4-38) and (4-39) can be directly compared to the bellows material allowable stress at design temperature published in the ASME Piping Codes and the ASME Boiler and Pressure Vessel Codes. The system designer **MUST** identify the specific design code to the Expansion Joint manufacturer. Contact the Expansion Joint manufacturer for designs governed by other codes. An unreinforced bellows is shown in Figure 4.13.

4.12.1.2 REINFORCED BELLOWS

The equations for reinforced bellows are based on those shown in Atomics International Report NAA-SR-4527 "Analysis of Stresses in Bellows, Part 1, Design Criteria and Test Results," with modifications and additions by the Association to reflect the experience of the members. These equations are based on elastic shell theory and consider the parameters involved for bellows of the "U" shaped configuration.

The equations shown in section 4.13.2 are based on the Atomics International Report for unreinforced bellows. Equations (4-52), (4-53), (4-54), (4-55) and (4-58) reflect the increased strength and stiffness of the convolution due to the reinforcing member. Equations (4-46), (4-47), (4-49), (4-50), (4-51), (4-52) and (4-53) as modified may be used to calculate stresses which can be directly compared to the bellows material allowable stress at design temperature published in the ASME Piping Codes and the ASME Boiler and Pressure Vessel Codes. The system designer **MUST** identify the specific design code to the Expansion

Joint manufacturer. Contact the Expansion Joint manufacturer for designs governed by other codes. A reinforced bellows is shown in Figure 4.14. An externally reinforced bellows with external pressure shall be treated as an unreinforced bellows.

Pressure on the convolution sidewall will apply an axial load on the end collar tending to push the collar away from the convolutions. This force will be equal to the internal pressure times the area difference between the bellows mean diameter and the bellows inside diameter. External restraints must be provided to resist this load and hold the collar in position.

a. TOROIDAL BELLOWS

The equations for toroidal bellows shown in Section 4.13.3 are taken from *Design and Analysis of Piping, Pressure Vessels, and Components*, ASME PVP Vol. 120, 1987, Pgs. 99-106. A toroidal bellows is shown in Figure 4.15.

Pressure on the convolution sidewall will apply an axial load on the end collar tending to push the collar away from the convolutions. This force will be equal to the internal pressure times the area difference between the bellows mean diameter and the bellows inside diameter. External restraints must be provided to resist this load and hold the collar in position.

4.12.1.3 INTERNAL PRESSURE CAPACITY

Excessive hoop stress in the straight cylindrical end tangents of a bellows will cause circumferential yielding. This stress is calculated by a modification of the Barlow equation. For unreinforced bellows, a factor "k" is included which considers the stiffening effects of the attachment weld and the end convolution. When required, the straight tangent of unreinforced bellows can be reinforced by collars. The equations apportion the stress in the tangent and collar in relation to their respective cross sectional areas and material properties.

Excessive hoop stress in the convoluted section of the bellows can produce circumferential yielding and possible rupture. As in any cylindrical shell, this stress is inversely proportional to the cross sectional area. All equations apportion the stress between the bellows and any reinforcing members in relation to their respective cross sectional areas and material properties. Factors have been included to account for the effect of movement on the hoop stress.

Excessive meridional pressure stress in the convoluted section of a U-shaped bellows will produce bulging of the sidewall. Any gross change in the convolution shape will decrease the space between convolutions, and the ability of the bellows to absorb movement. Such changes in the shape may also affect the fatigue life. Excessive meridional pressure stress in a toroidal bellows will produce meridional yielding and possible rupture.

4.12.1.4 DEFLECTION STRESS

The stress in the convoluted section of the bellows due to deflection is given by equations (4-40), (4-41), (4-54), (4-55), (4-68) and (4-69). Typical calculated stress range values are 50,000 to 500,000 psi (350 to 3,500 MPa). These values are not true stresses, since they exceed the elastic limit of the material. They are meaningful when correlated with actual test results in evaluating fatigue life.

4.12.1.5 FATIGUE LIFE EXPECTANCY

The fatigue life expectancy of an Expansion Joint is affected by various factors such as: operating pressure, operating temperature, the material from which the bellows is made, the movement per convolution, the thickness of the bellows, the convolution pitch, and the depth and shape of the convolution. Any change in these factors will result in a change in the life of the Expansion Joint.

The fatigue life expectancy can be defined as the total number of complete cycles which can be expected from the Expansion Joint based on data tabulated from tests performed at room temperature under simulated operating conditions. A cycle is defined as one complete movement from the initial position in the piping system to the operating position and back to the initial position. Fatigue life is dependent upon the maximum stress range to which the bellows is subjected, the maximum stress amplitude being a far less significant factor.

Expansion Joints can be specially designed for very high cyclic life. When this is required, the Expansion Joint manufacturer must be advised of the estimated number of cycles required.

The equations given for fatigue life should only be used when the actual bellows metal temperature under operating conditions is below the creep range. Fatigue life calculations for actual bellows metal temperatures in the creep range must be substantiated by high temperature test data or history of successful operation of a similar bellows size and configuration for identical or more severe service.

a. **FATIGUE LIFE**

The fatigue life of a bellows is a function of the sum of the meridional pressure stress range and the total meridional deflection stress range. The deflection stress range must be based on the total equivalent axial movement range as discussed in Section 4.3.

The number of cycles to failure may be evaluated by equations (4-41) and (4-49). The constants are derived from graphs of the total stress range versus number of cycles to failure from actual fatigue tests of a series of bellows of similar materials at room temperature evaluated by a best fit continuous curve. These equations are meant to predict the average fatigue life for the bellows design and do not contain any curve modifications or factors of safety. Certain codes and standards will invoke the use of curve modifications and or safety factors that account for the normal effects of size, surface finish, and scatter of the data. Therefore, the design cycle life should realistically represent the estimated number of operating cycles. An overly conservative estimate of cycles can result in an increased number of convolutions and an Expansion Joint more prone to instability.

b. CUMULATIVE FATIGUE

An Expansion Joint may be required to withstand a number of stress cycles such as those produced by the start up and shut down of the system. While these cycles usually control the fatigue life of the bellows, there may be instances where other conditions assume importance in determining the overall or cumulative fatigue life of the bellows. As an example, one condition may generate 1,000 cycles of stress variation from zero to 160,000 psi (1,100 MPa) and another condition 10,000 cycles of stress variation from zero to 50,000 psi (350 MPa). The procedure described in the following paragraphs illustrates the method used in evaluating the fatigue damage in a bellows when it is subjected to a variety of conditions during its lifetime. This method is based on Miner's Hypothesis¹ which is generally accepted as sufficiently accurate for predicting the effect of cumulative fatigue. The method assumes that a stress versus fatigue life curve has been established for the type of bellows under consideration.

c. CONCURRENT CONDITIONS

Concurrent conditions are those that occur at exactly the same time and frequency producing a combined stress range. The stress ranges for each condition are superimposed to give cases of combined stress range for the applicable number of cycles. Truly concurrent conditions are rare in practice. If the conditions described in the above example are concurrent, the cumulative effect can be evaluated using the following steps:

Step 1: Superimpose the stress range of each condition for the applicable number of cycles to find each case as follows:

$$\begin{aligned}
 \text{Case 1:} \quad n_1 &= 1000 \text{ cycles} \\
 S_{t1} &= |160,000| + |50,000| \quad (|1100| + |350|) \\
 &= 210,000 \text{ psi (1450 MPa) stress range} \\
 \text{Case 2:} \quad n_2 &= 10,000 - 1000 = 9000 \text{ cycles} \\
 S_{t2} &= |0| + |50,000| \quad (|0| + |350|) \\
 &= 50,000 \text{ psi (350 MPa) stress range}
 \end{aligned}$$

Step 2: For each stress range S_{t1} , S_{t2} , ..., use the applicable fatigue curve to find the number of cycles to failure N_1 , N_2 , ... for each case.

Step 3: For each case, calculate the usage factor U_1 , U_2 , ... where $U_1 = n_1/N_1$, $U_2 = n_2/N_2$, etc.

Step 4: Calculate the cumulative usage factor where $U = U_1 + U_2 + \dots$

Step 5: The cumulative usage factor U shall not exceed 1.0.

¹ Miner, Milton A., "Cumulative Damage in Fatigue." Journal of Applied Mechanics, Sept., 1945.

d. INDEPENDENT CONDITIONS

Independent conditions are those that do not occur at exactly the same time or frequency. Independent conditions are not superimposed. Most conditions in practice are independent. If the conditions described in the above example are independent, the cumulative effect can be evaluated using the following steps:

Step 1: Summarize the stress range and cycles for each condition to find the cases as follows:

Case 1:	$n_1 = 1000$ cycles
	$S_{t1} = 160,000$ psi (1100 MPa) stress range
Case 2:	$n_2 = 10,000$ cycles
	$S_{t2} = 50,000$ psi (350 MPa) stress range

Step 2: For each stress range S_{t1} , S_{t2} , ..., use the applicable fatigue curve to find the number of cycles to failure N_1 , N_2 , ... for each case.

Step 3: For each case, calculate the usage factor U_1 , U_2 , ... where $U_1 = n_1/N_1$, $U_2 = n_2/N_2$, etc.

Step 4: Calculate the cumulative usage factor where $U = U_1 + U_2 + \dots$

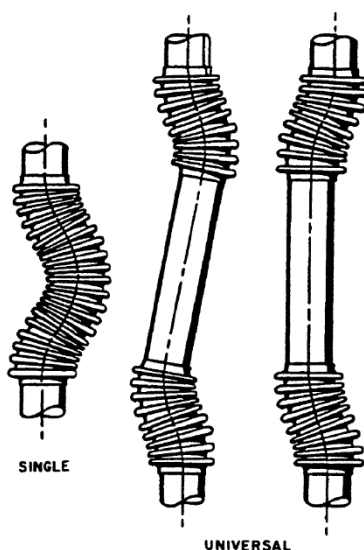
Step 5: The cumulative usage factor U shall not exceed 1.0.

e. CYCLE LIFE EXPECTANCY AT HIGH TEMPERATURES

When the actual bellows metal temperature is high, cycle life can be affected by factors other than just the deflection stress range. Metallurgical changes can make the material more sensitive to the microscopic flaws which lead to fatigue failures. When temperatures are high enough, creep strains and cyclic deflection strains can interact to further reduce the cycle life. The number of cycles to failure can be evaluated using the method given in Appendix G.

4.12.1.6 BELLOWS STABILITY

Excessive internal pressure may cause a multi-convolution bellows to become unstable and squirm. Squirm is detrimental to bellows performance in that it can greatly reduce both fatigue life and pressure capacity. The two most common forms are column squirm and in-plane squirm. Column squirm is defined as a gross lateral shift of the center section of the bellows. It results in curvature of the bellows centerline as shown in Figure 4.9.



COLUMN SQUIRM

FIGURE 4.9

This condition is most associated with bellows which have a relatively large length-to-diameter ratio and is analogous to the buckling of a column under compressive load.

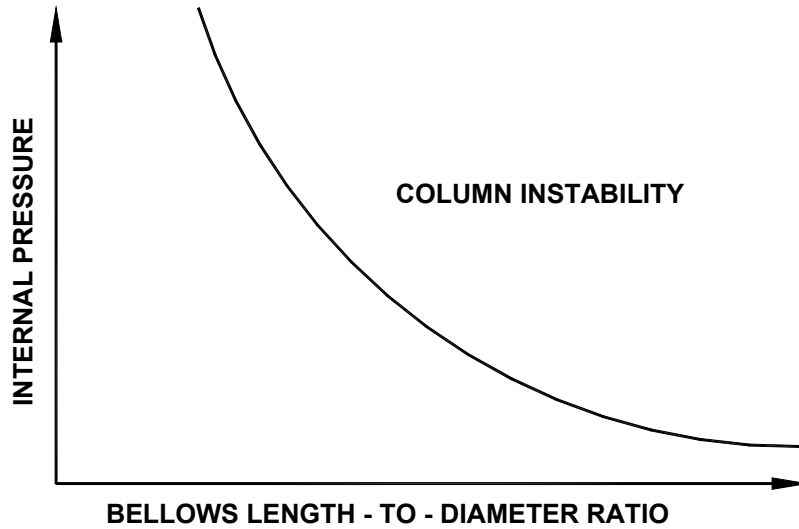


FIGURE 4.10

Figure 4.10 depicts the critical column squirm pressure for a series of bellows having the same diameter, thickness and convolution profile. See equation (4-43), (4-57), or (4-71) for a method of evaluating a bellows for column squirm. Factors have been included to account for the effect of movements on the column squirm pressure.

The equations assume that each end of the expansion joint is rigidly supported (fixed). For other end conditions, the limiting design pressure should be evaluated as follows:

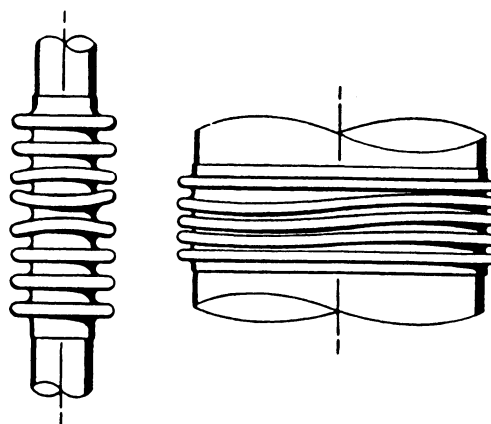
Fixed/Pinned - $.5P_{sc}$

Pinned/Pinned - $.25P_{sc}$

Fixed/Laterally Guided - $.25P_{sc}$

Fixed/Free - $.06P_{sc}$

It should be noted that external pressure does not produce column squirm. When a bellows is subjected to external pressure, its pressure capacity can be verified by the method discussed in Section 4.13.



IN-PLANE SQUIRM

FIGURE 4.11

In-plane squirm is defined as a shift or rotation of the plane of one or more convolutions such that the plane of these convolutions is no longer perpendicular to the axis of an unreinforced bellows. It is characterized by tilting or warping of one or more convolutions as shown in Figure 4.11. This condition is predominantly associated with high meridional bending stress and the formation of plastic hinges at the root and crest of the convolutions. It is most common in bellows which have a relatively small length-to-diameter ratio. See Equation (4-44) for a method of evaluating an unreinforced bellows for in-plane squirm.

To prevent bellows squirm under test conditions, the test pressure should be less than or equal to 1.5 times the limiting design pressure based on column or in-plane instability using room temperature material properties. In addition, the test fixture should duplicate the as-installed condition as closely as possible.

The equations given for squirm should only be used when the actual bellows metal temperature under operating conditions is below the creep range. Squirm calculations for actual bellows metal temperatures in the creep range must be substantiated by high temperature test data or history of successful operation of a similar bellows size and configuration for identical or more severe service.

These equations include factors such that the ratio between the limiting design pressure and the critical squirm pressure is approximately 2.25 for column squirm and 1.75 for inplane squirm.

When a universal type expansion joint is subjected to lateral offset, the internal pressure produces a force that tends to rotate the centerspool. This force is resisted by the stiffness of the bellows. If the force is sufficiently high, instability can occur. A method for evaluating this mode of instability is given in *Metallic Bellows and Expansion Joints* - 1989, ASME PVP Vol. 168, Pgs 41-43.

4.12.1.7 BELLOWS SPRING RATE

The force required to deflect a bellows axially is a function of the dimensions of the bellows and the material from which it is made. The curve of force vs. deflection for most bellows indicates motion extending into the plastic range as shown by the solid line in Figure 4.12. The first portion of the curve is a straight line as the bellows is deflected through its elastic range. As bellows deflection continues and extends into the plastic range, the force vs. deflection relationship becomes non-linear until the point of maximum deflection is reached. The restraining force at maximum deflection will decrease if the bellows is exposed to temperatures in the creep range.

When the restraining force is released, the curve again becomes linear until the applied force is zero at which point the residual deflection of the bellows still has a positive value. To return the bellows to its initial position, a restoring force must be applied in the opposite direction as shown by the curve below the abscissa.

Line A in Figure 4.12 represents the bellows theoretical initial elastic spring rate, f_i . This value can be determined analytically with reasonable accuracy from equations based on elastic theory. The bellows theoretical initial elastic spring rate, f_i is calculated in accordance with Equations (4-45), (4-58), and (4-72).

Lines B and C represent bellows resistance factors or working spring rates, f_w , for bellows with operating deflections in the plastic range.

The use of the initial elastic spring rate in place of the working spring rate for a bellows whose deflection extends into the plastic range predicts forces which can be considerably higher than actual. This is recognized to be a problem and various methods have been used to obtain more accurate results. Line B, drawn from the origin to the point of maximum force and deflection, is used as the bellows working spring rate, f_w but has the disadvantage of underestimating the actual force over the full range. Line C, drawn from the point of maximum force and deflection to the value of the restoring force required to return the bellows to zero deflection, becomes line C' when transferred to the origin. A working spring rate based on line C' can be used. This reduces the discrepancy between the indicated and true values although the difference can still be significant.

For the great majority of applications, the manufacturers published spring rates have proved satisfactory. However, when the critical nature of a particular application warrants more precise knowledge of the bellows working spring rate, the user should require the manufacturer to supply information as to the means by which his data was developed. In special cases, prototype testing to determine the precise load vs. deflection characteristics of a particular bellows design may be necessary.

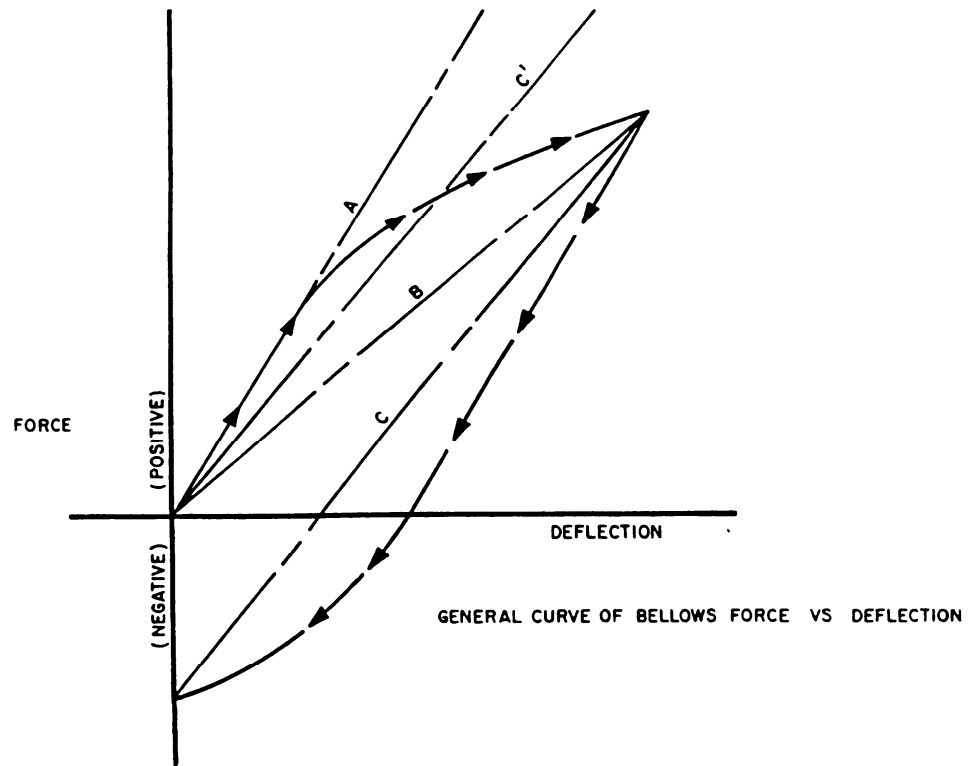


FIGURE 4.12

4.12.1.8 CORRELATION TESTING

The equations in Sections 4.13.1, 4.13.2, and 4.13.3 can be employed to design a bellows if they have been correlated with actual test results to demonstrate predictability of rupture pressure, meridional yielding, squirm and cycle life for a consistent series of bellows of the same basic design (unreinforced and reinforced bellows are considered as separate designs). A minimum of five meridional yield-rupture tests on bellows of varying sizes, with not less than three convolutions, are required to verify Equations (4-36), (4-37), (4-38), (4-39), (4-45), (4-52), (4-53), and (4-67). A minimum of ten squirm tests on bellows of varying diameters and number of convolutions are required to verify Equations (4-43), (4-57) and (4-71). A minimum of twenty-five fatigue tests on bellows of varying diameters, thicknesses, convolution profiles are required to construct a fatigue life versus combined stress plot. The effects of pressure shall be considered in the fatigue tests. The test bellows must be representative of typical bellows design and manufacturing processes.

4.12.1.9 BELLOWS HEAT TREATMENT

Heat treatment after forming can have a detrimental effect on bellows pressure capacity. It is not normally considered beneficial for fatigue life to either stress relieve or anneal after forming. The necessity for this form of heat treatment is the responsibility of the purchaser and shall be considered individually.

4.13 DESIGN EQUATIONS

4.13.1 DESIGN EQUATIONS FOR UNREINFORCED BELLOWS

Bellows Tangent Circumferential Membrane Stress Due to Pressure

$$S_1 = \frac{P(D_b + nt)^2 L_t E_b k}{2(nt E_b L_t (D_b + nt) + t_c k E_c L_c D_c)} \quad (4-35)$$

Collar Circumferential Membrane Stress Due to Pressure

$$S'_1 = \frac{PD_c^2 L_t E_c k}{2(nt E_b L_t (D_b + nt) + t_c k E_c L_c D_c)} \quad (4-36)$$

Bellows Circumferential Membrane Stress Due to Pressure

$$S_2 = \frac{PD_m K_r q}{2A_c} \quad (4-37)$$

Bellows Meridional Membrane Stress Due to Pressure

$$S_3 = \frac{Pw}{2nt_p} \quad (4-38)$$

Bellows Meridional Bending Stress Due to Pressure

$$S_4 = \frac{P}{2n} \left(\frac{w}{t_p} \right)^2 C_p \quad (4-39)$$

Note: The above stresses should be evaluated for pressure capacity as follows:

$$S_1 \text{ \& } S_2 \leq C_{wb} W_b S_{ab} \quad S'_1 \leq C_{wc} W_c S_{ac}$$

$$S_3 + S_4 \leq C_m S_{ab} \quad (\text{Below the Creep Range})$$

$$S_3 + (S_4/1.25) \leq S_{ab} \quad (\text{In the Creep Range})$$

Bellows Meridional Membrane Stress Due to Deflection

$$S_5 = \frac{E_b t_p^2 e}{2w^3 C_f} \quad (4-40)$$

Bellows Meridional Bending Stress Due to Deflection

$$S_6 = \frac{5E_b t_p e}{3w^2 C_d} \quad (4-41)$$

Note: Modulus of elasticity, E_b , in Equations (4-40) and (4-41) is at room temperature.

Fatigue Life

$$S_t = 0.7 (S_3 + S_4) + (S_5 + S_6) \quad (4-42)$$

See Figure 4.20 for fatigue equation and constants. .

Limiting Internal Design Pressure Based on Column Instability for Single Bellows (both ends rigidly supported). Refer to Section 4.12.1.6.

$$P_{sc} = \frac{0.34\pi C_\theta f_{iu}}{N^2 q} \quad (4-43)$$

For universal expansion joints, N = total number of convolutions in both bellows for calculation of P_{sc} .

Limiting Design Pressure Based on Inplane Instability and Local Plasticity at Temperatures Below the Creep Range

$$P_{si} = \frac{1.3 A_c S_y}{K_r D_m q \sqrt{\alpha}} \quad (4-44)$$

Bellows Theoretical Axial Elastic Spring Rate per Convolution

$$f_{iu} = 1.7 \frac{D_m E_b t_p^3 n}{w^3 C_f} \quad (4-45)$$

4.13.2 DESIGN EQUATIONS FOR REINFORCED BELLOWS

Bellows Tangent Circumferential Membrane Stress Due to Pressure

$$S_1 = \frac{P(D_b + nt)^2 L_d E_b}{2[(nt L_t + A_c/2) E_b (D_b + nt) + A_{tc} E_c D_c]} \quad (4-46)$$

Collar Circumferential Membrane Stress Due to Pressure

$$S'_1 = \frac{P(D_c)^2 L_d E_c}{2[(nt L_t + A_c/2) E_b (D_b + nt) + A_{tc} E_c D_c]} \quad (4-47)$$

Collar Circumferential Bending Stress Due to Pressure

$$S''_1 = \frac{F_g n_g D_c}{4\pi C_c Z_c} \quad (4-48)$$

Bellows Circumferential Membrane Stress Due to Pressure

$$S_2 = \frac{H}{2A_c} \left(\frac{R}{R+1} \right) K_r \quad (4-49)$$

 $R = R_1$ for integral reinforcing members $R = R_2$ for reinforcing members joined by fasteners

Note: In the case of reinforcing members which are made in sections and joined by fasteners in tension, this equation assumes that the structure used to retain the fastener does not bend so as to permit the reinforcing member to expand diametrically. In addition, the end reinforcing members must be restrained against the longitudinal annular pressure load of the bellows.

Reinforcing Member Circumferential Membrane Stress Due to Pressure

$$S'_2 = \frac{H}{2A_r} \left(\frac{1}{R_1 + 1} \right) K_r \quad (4-50)$$

Note: In the case of equalizing rings, this equation provides only the simple membrane stress and does not include the bending stress caused by the eccentric fastener location. These stresses can be determined by elastic analysis and/or actual tests.

Fastener Membrane Stress Due to Pressure

$$S''_2 = \frac{H}{2A_f} \left(\frac{1}{R_2 + 1} \right) K_r \quad (4-51)$$

Bellows Meridional Membrane Stress Due to Pressure

$$S_3 = \frac{.76P(w - r_m)}{2nt_p} \quad (4-52)$$

Bellows Meridional Bending Stress Due to Pressure

$$S_4 = \frac{.76P}{2n} \left(\frac{w - r_m}{t_p} \right)^2 C_p \quad (4-53)$$

Note: The above stresses should be evaluated for pressure capacity as follows:

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$$\begin{aligned} S_1 \& S_2 \leq C_{wb} W_b S_{ab} & S'_2 \leq C_{wr} W_r S_{ar} & S'_1 + S''_1 \leq K_s C_{wc} W_c S_{ac} \\ S'_1 \leq C_{wc} W_c S_{ac} & S''_2 \leq S_{af} \end{aligned}$$

$$\begin{aligned} S_3 + S_4 &\leq C_m S_{ab} \quad (\text{Below the Creep Range}) \\ S_3 + (S_4/1.25) &\leq S_{ab} \quad (\text{In the Creep Range}) \end{aligned}$$

Bellows Meridional Membrane Stress Due to Deflection

$$S_5 = \frac{E_b t_p^2 e}{2(w - r_m)^3 C_f} \quad (4-54)$$

Bellows Meridional Bending Stress Due to Deflection

$$S_6 = \frac{5E_b t_p e}{3(w - C_r r_m)^2 C_d} \quad (4-55)$$

Note: Modulus of elasticity, E_b in Equations (4-54) and (4-55) is at room temperature.

Fatigue Life

$$S_t = 0.9 (0.7 (S_3 + S_4) + (S_5 + S_6)) \quad (4-56)$$

See Figure 4.20 for fatigue equation and constants.

Limiting Internal Design Pressure Based on Column Instability for Single Bellows with Reinforcing Rings (both ends rigidly supported). Refer to Section 4.12.1.6. For bellows with equalizing rings see Figure 4.14 and consult the manufacturer.

$$P_{sc} = \frac{0.3\pi C_\theta f_{ir}}{N^2 q} \quad (4-57)$$

For universal expansion joints, N = total number of convolutions in both bellows for calculation of P_{sc}

Bellows Theoretical Axial Elastic Spring Rate per Convolution

$$f_{ir} = 1.7 \frac{D_m E_b t_p^3 n}{(w - C_r r_m)^3 C_f} \quad \begin{array}{l} \text{valid for column stability under} \\ \text{operating conditions} \end{array} \quad (4-58)$$

$$f_{ir} = 1.7 \frac{D_m E_b t_p^3 n}{(w - r_m)^3 C_f} \quad \begin{array}{l} \text{valid for force calculation and for} \\ \text{test conditions in neutral position} \end{array} \quad (4-59)$$

4.13.3 DESIGN EQUATIONS FOR TOROIDAL BELLOWS

Bellows Tangent Circumferential Membrane Stress Due to Pressure

$$S_1 = \frac{P(D_b)^2 L_d E_b}{2(D_c E_c A_{tc})} \quad (4-60)$$

Collar Circumferential Membrane Stress Due to Pressure for Externally Attached Bellows

$$S_1' = \frac{PD_c L_d}{2A_{tc}} \quad (4-61)$$

Collar Circumferential Bending Stress Due to Pressure for Externally Attached Bellows

$$S_1'' = \frac{F_g n_g D_c}{4\pi C_c Z_c} \quad (4-62)$$

Pipe Circumferential Membrane Due to Pressure for Internally Attached Bellows

$$S_1''' = \frac{PD_p(L_p + L_g / 2 + nt)}{2A_{tp}} \quad (4-63)$$

Bellows Circumferential Membrane Stress Due to Pressure

$$S_2 = \frac{Pr}{2nt_p} \quad (4-64)$$

Reinforcing Ring Circumferential Membrane Stress Due to Pressure

$$S_2' = \frac{PD_r(L_{rt} + L_g + 2nt)}{2A_r} \quad \text{if } L_{rt} \leq \frac{2}{3}\sqrt{D_r t_r} \quad (4-65)$$

$$S_2' = \frac{PD_r(L_r + L_g / 2 + nt)}{2A_{rr}} \quad \text{if } L_{rt} > \frac{2}{3}\sqrt{D_r t_r} \quad (4-66)$$

Bellows Meridional Membrane Stress Due to Pressure

$$S_3 = \frac{Pr}{nt_p} \left(\frac{D_m - r}{D_m - 2r} \right) \quad (4-67)$$

Note: The above stresses should be evaluated for pressure capacity as follows:

$$\begin{aligned} S_1 \ \& \ S_2 \leq C_{wb} W_b S_{ab} & S_3 \leq S_{ab} & S_1' + S_1'' \leq K_s C_{wc} W_c S_{ac} \\ S_1' \leq C_{wc} W_c S_{ac} & S_2' \leq C_{wr} W_r S_{ar} & S_1''' \leq C_{wp} W_p S_{ap} \end{aligned}$$

Bellows Meridional Membrane Stress Due to Deflection

$$S_5 = \frac{E_b t_p^2 e}{34.3r^3} B_1 \quad (4-68)$$

Bellows Meridional Bending Stress Due to Deflection

$$S_6 = \frac{E_b t_p e}{5.72 r^2} B_2 \quad (4-69)$$

Note: Modulus of elasticity E_b , in Equations (4-68) and (4-69) is at room temperature.

Fatigue Life

$$S_t = 3S_3 + S_5 + S_6 \quad (4-70)$$

See Figure 4.20 for fatigue equation and constants.

Limiting Internal Design Pressure Based on Column Instability for Single Bellows (both ends rigidly supported).

$$P_{sc} = \frac{0.3\pi C_\theta f_{it}}{N^2 r} \quad (4-71)$$

For universal expansion joints, N = total number of convolutions in both bellows for calculation of P_{sc}

Bellows Theoretical Axial Elastic Spring Rate per Convolution

$$f_{it} = \frac{D_m E_b t_p^3 n}{10.92 r^3} B_3 \quad (4-72)$$

4.13.4 BELLOWS TORSION - UNREINFORCED/REINFORCED BELLOWS

The following equations may be used as a guide in calculating the shear stress and deflection due to torsion about the centerline for one bellows.

$$S_s = \frac{2M_t}{nt\pi D_b^2} \quad \text{(Limited to } 0.25 S_{ab} \text{ or other value determined by testing)} \quad (4-73)$$

$$\theta_t = \frac{4M_t L_{dt} N}{\pi G n t D_b^3} \quad (4-74)$$

Note: Refer to Section 2.10.2 for design recommendation regarding bellows torsion.

S_s = Shear Stress, psi (MPa)

M_t = Torque lbf-in (N·mm)

L_{dt} = Developed length of one convolution, in. (mm)

$$= .571q + 2w$$

θ_t = Angle of twist, rad

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4.14 BENCHMARK CALCULATIONS

The benchmark calculations shown below are based on the design equations given in Sections 4.13.1, 4.13.2, and 4.13.3. They shall be used to verify the accuracy of computer programs used to perform bellows design calculations.

Input Variables	Bellows Type			
	1	2	3	4
	Single Unreinforced	Universal Unreinforced	Single Reinforced	Single Toroidal
D_b	24	24	24	24
$D_b + 2(w + nt)$	27	27	27	N/A
D_m	25.5	25.5	25.5	27.496
t	.03	.03	.03	.05
n	2	2	2	1
r_m	.25	.25	.25	N/A
q	1	1	1	3.4625
N	12	4	8	2
L_t	1	1.25	1.25	N/A
t_c	N/A	.25	.375	1
L_c	N/A	1	N/A	N/A
L_u	N/A	36	N/A	N/A
A_r	N/A	N/A	.15	2.282
A_{tc}	N/A	N/A	.43	1.27
Z_c	N/A	N/A	.0872	.271
n_g	N/A	N/A	8	12
f_w	8417	8417	11084	10528
r	N/A	N/A	N/A	1
D_r	N/A	N/A	N/A	25.1
L_d	N/A	N/A	1.75	2.106
L_g	N/A	N/A	N/A	.5
t_r	N/A	N/A	N/A	1
P	40	100	100	500
x (Comp)	1	1	1	.75
x (Ext)	.5	.5	.5	0
y	.06	2	0	0
Θ (Deg)	5	0	5	0
E_b (RT)	28.3e6	28.3e6	28.3e6	28.3e6
E_b (DT)	25.8e6	25.8e6	25.8e6	25.8e6
E_r (DT)	N/A	N/A	26.6e6	26.6e6
E_c (DT)	N/A	27.3e6	27.3e6	27.3e6
S_y	59726	59726	59726	59726

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Input Variables	Bellows Type			
	1	2	3	4
	Single Unreinforced	Universal Unreinforced	Single Reinforced	Single Toroidal
Results				
S_1	6301	4367	3296	8991
S_1'	N/A	4741	3615	10406
S_1''	N/A	N/A	12343	12646
S_2	2645	6744	3810	5352
S_2'	N/A	N/A	3929	9521
S_3	495	1237	777	11123
S_4	18009	45022	23367	N/A
S_5	689	1130	1713	2529
S_6	109067	178830	183747	87936
S_t	122709	212342	182125	123836
N_c	74218	4342	8920	70222
P_{sc}	93	210	183	3703
P_{si}	128	128	N/A	N/A
f_i	12562	12562	16544	15714
C_p	0.7357	0.7357	0.7357	N/A
C_f	1.4701	1.4701	1.4701	N/A
C_d	1.5326	1.5326	1.5326	N/A
B_1	N/A	N/A	N/A	3.7463
B_2	N/A	N/A	N/A	1.0146
B_3	N/A	N/A	N/A	2.3731
K_r	1.0417	1.0625	1.0625	N/A
e	0.2525	0.4140	0.3541	0.3750

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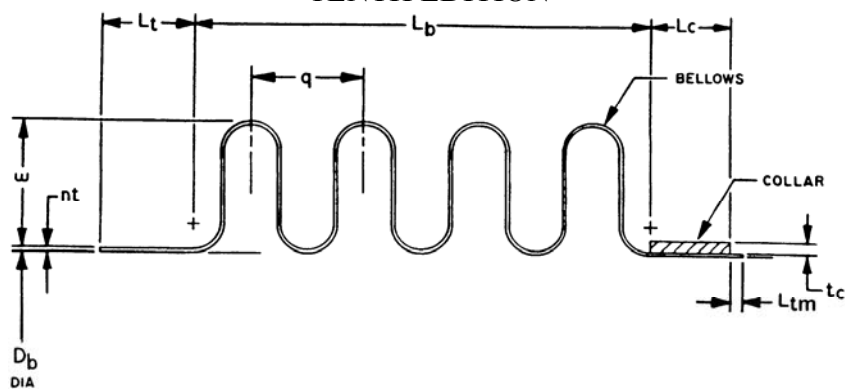
Input Variables	Bellows Type – Metric Values			
	1	2	3	4
	Single Unreinforced	Universal Unreinforced	Single Reinforced	Single Toroidal
D _b	609.6	609.6	609.6	609.6
D _b + 2(w + nt)	685.8	685.8	685.8	N/A
D _m	647.7	647.7	647.7	698.4
t	0.762	0.762	0.762	1.27
n	2	2	2	1
r _m	6.35	6.35	6.35	N/A
q	25.4	25.4	25.4	87.95
N	12	4	8	2
L _t	25.4	31.75	31.75	N/A
t _c	N/A	6.35	9.53	25.4
L _c	N/A	25.4	N/A	N/A
L _u	N/A	914.4	N/A	N/A
A _r	N/A	N/A	96.77	1472.26
A _{tc}	N/A	N/A	277.42	819.35
Z _c	N/A	N/A	1428.95	4440.89
n _g	N/A	N/A	8	12
f _w	1473.82	1473.82	2879.87	1843.35
r	N/A	N/A	N/A	25.4
D _r	N/A	N/A	N/A	637.54
L _d	N/A	N/A	44.45	53.49
L _g	N/A	N/A	N/A	12.7
t _r	N/A	N/A	N/A	25.4
P	0.28	0.69	0.69	3.45
x (Comp)	25.4	25.4	25.4	19.05
x (Ext)	12.7	12.7	12.7	0
y	1.52	50.8	0	0
Θ (Deg)	5	0	5	0
E _b (RT)	1.952e5	1.952e5	1.952e5	1.952e5
E _b (DT)	1.779e5	1.779e5	1.779e5	1.779e5
E _r (DT)	N/A	N/A	1.834e5	1.834e5
E _c (DT)	N/A	1.883e5	1.883e5	1.883e5
S _y	411.90	411.90	411.90	411.90

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Input Variables	Bellows Type – Metric Values			
	1	2	3	4
	Single Unreinforced	Universal Unreinforced	Single Reinforced	Single Toroidal
Results				
S_1	43.45	30.11	22.72	61.98
S_1'	N/A	32.69	24.93	71.76
S_1''	N/A	N/A	85.10	87.21
S_2	18.24	46.50	26.27	36.91
S_2'	N/A	N/A	27.09	65.66
S_3	3.41	8.53	5.36	76.71
S_4	124.17	310.43	161.12	N/A
S_5	4.75	7.80	11.82	17.45
S_6	752.02	1233.49	1267.39	606.54
S_t	846.08	1464.55	1395.74	854.11
N_c	74317	4341	5416	70188
P_{sc}	0.64	1.45	1.26	25.53
P_{si}	0.88	0.88	N/A	N/A
f_i	2201	2201	2897.54	2752
C_p	0.7357	0.7357	0.7357	N/A
C_f	1.4701	1.4701	1.4701	N/A
C_d	1.5326	1.5326	1.5326	N/A
B_1	N/A	N/A	N/A	3.7463
B_2	N/A	N/A	N/A	1.0146
B_3	N/A	N/A	N/A	2.3731
K_r	1.0417	1.0625	1.0625	N/A
e	6.41	10.52	8.99	9.53

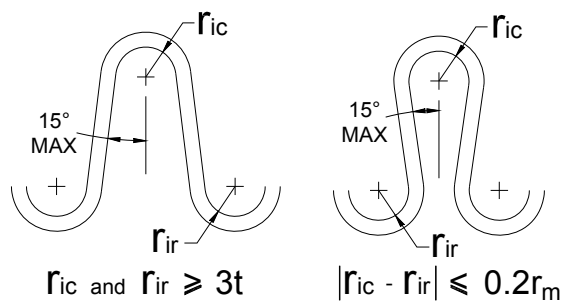
Notes:

1. The C_p , C_f , and C_d factors are taken from Appendix I. The interpolation method is in accordance with Section I-2.
2. The B_1 , B_2 , and B_3 factors are taken from Appendix I with linear interpolation.
3. The lateral and angular movements are assumed to be concurrent with axial compression only.
4. Fatigue life is based on the equations in figure 4.20, where material class is 1, and $f_c=1$.
5. Reinforcing members and collars are integral with no fasteners.
6. The notation 2e6 is equivalent to 2,000,000.
7. RT is room temperature and DT is design temperature. The design temperature is below the creep range.
8. One movement cycle consists of the following sequence:
 - a. Axial compression together with all lateral and angular movements.
 - b. Return to initial position.
 - c. Axial extension (if applicable)
 - d. Return to initial position.
9. The Single Toroidal bellows is externally attached.



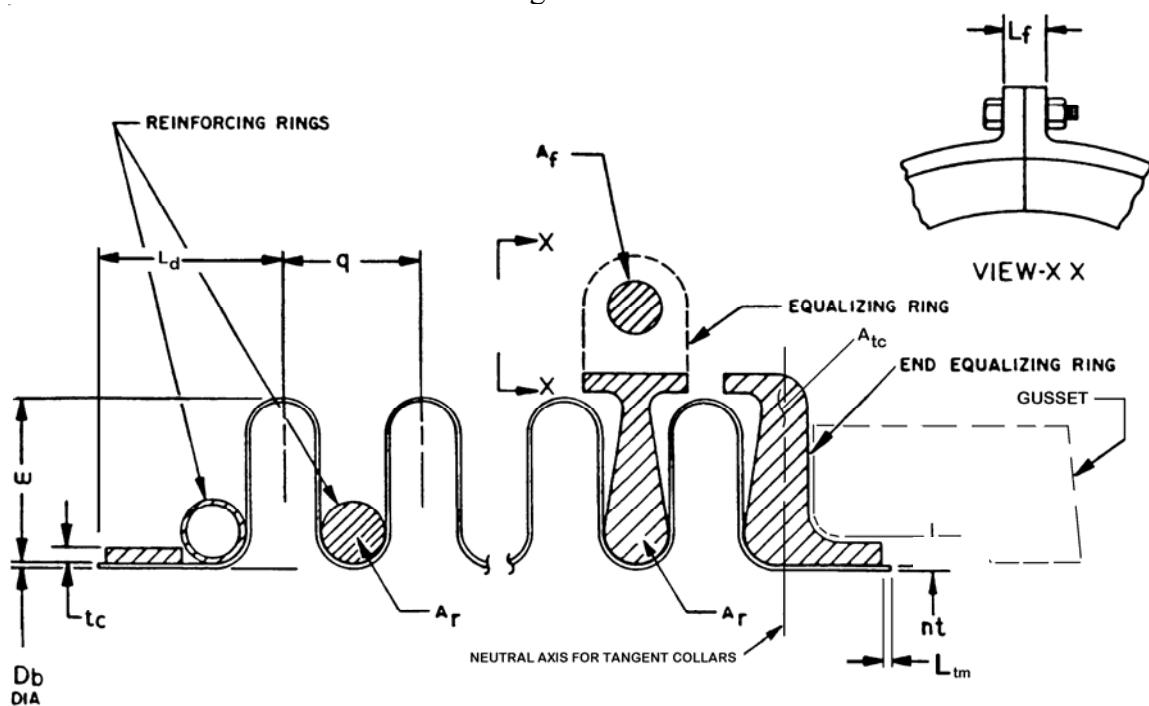
UNREINFORCED BELLOWS

Figure 4.13



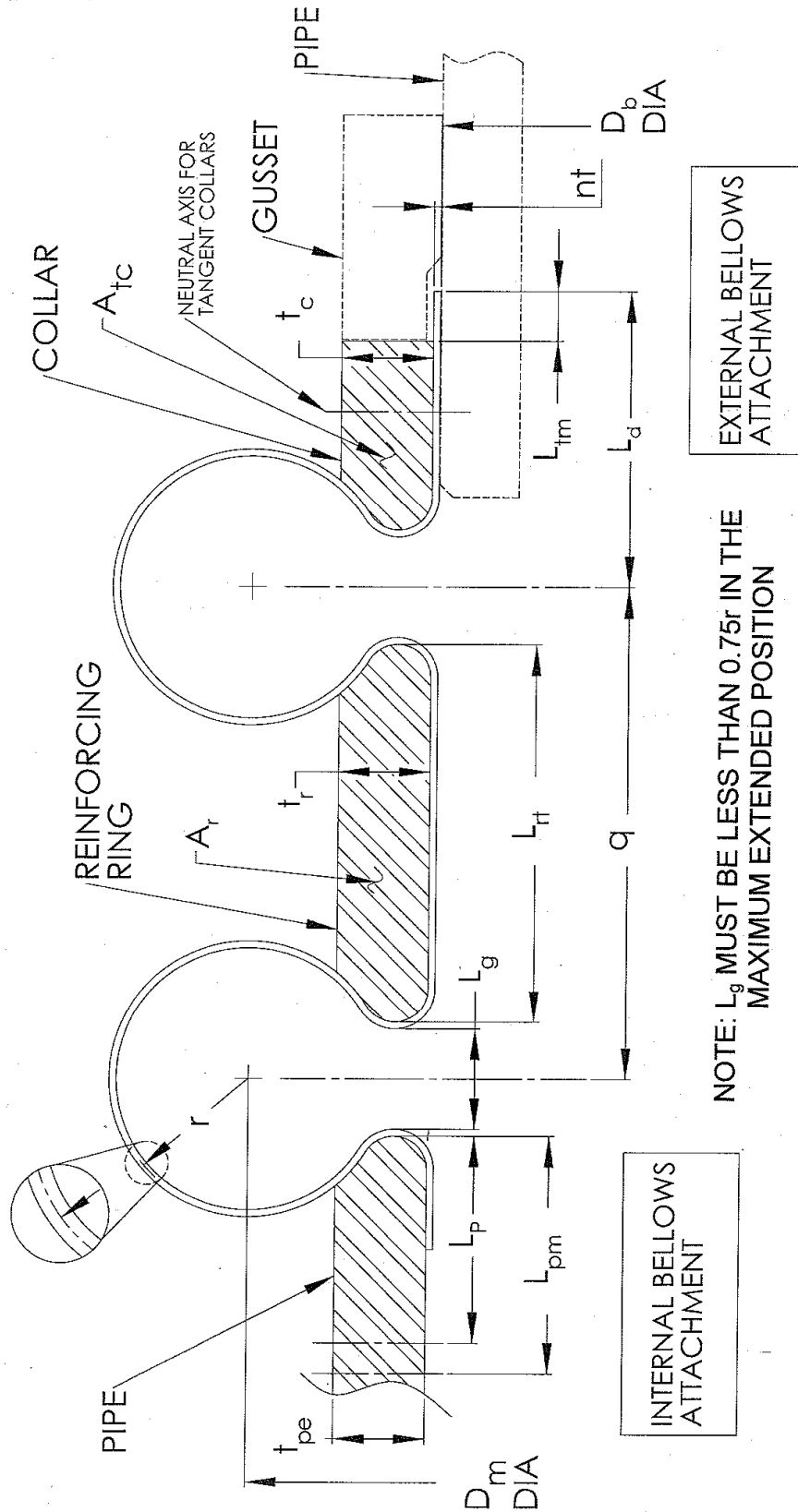
AS-FORMED CONVOLUTION PROFILE

Figure 4.13a



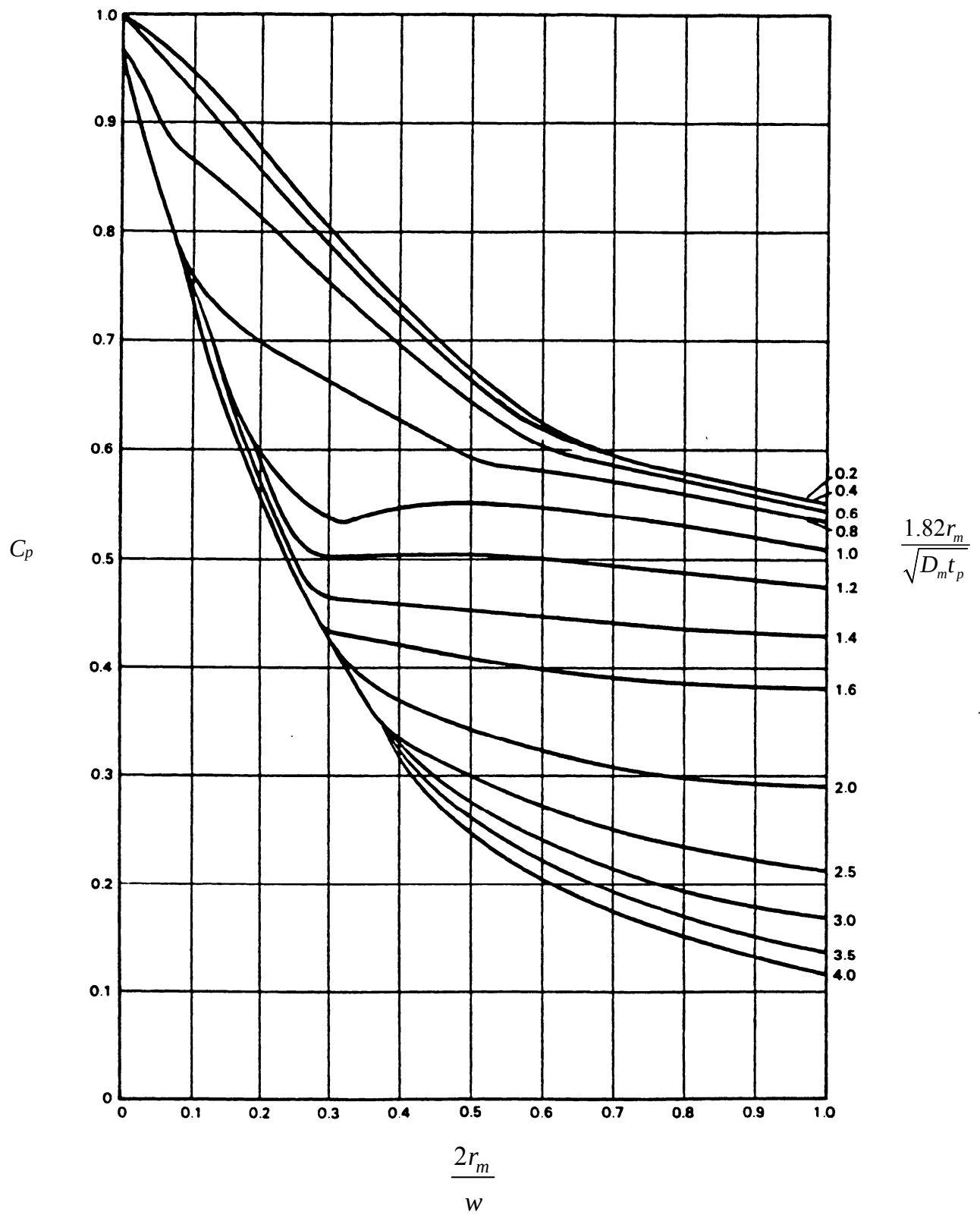
REINFORCED BELLOWS

Figure 4.14



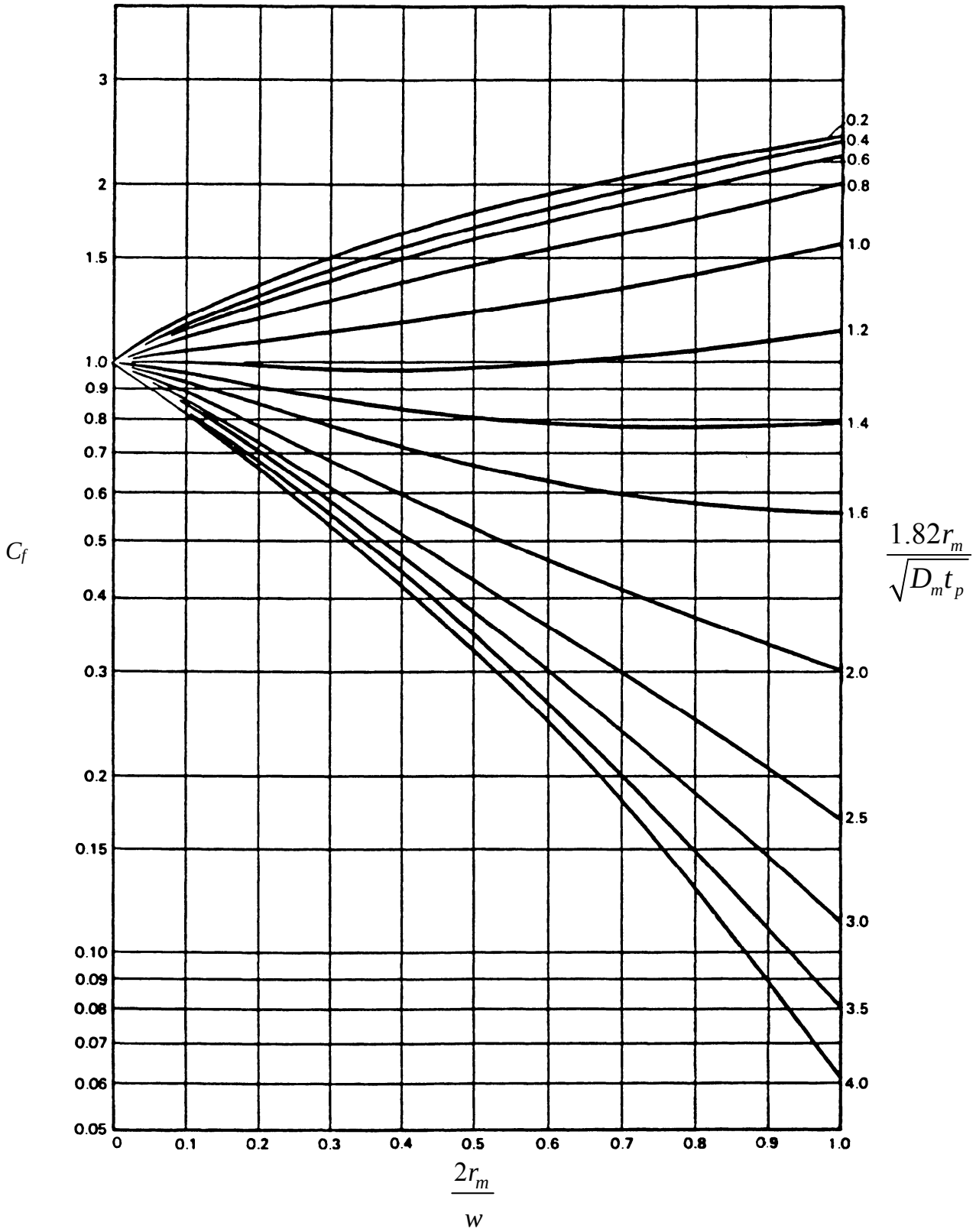
TOROIDAL BELLOWS
Figure 4.15

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C_p for Convolute Bellows
FIGURE 4.16

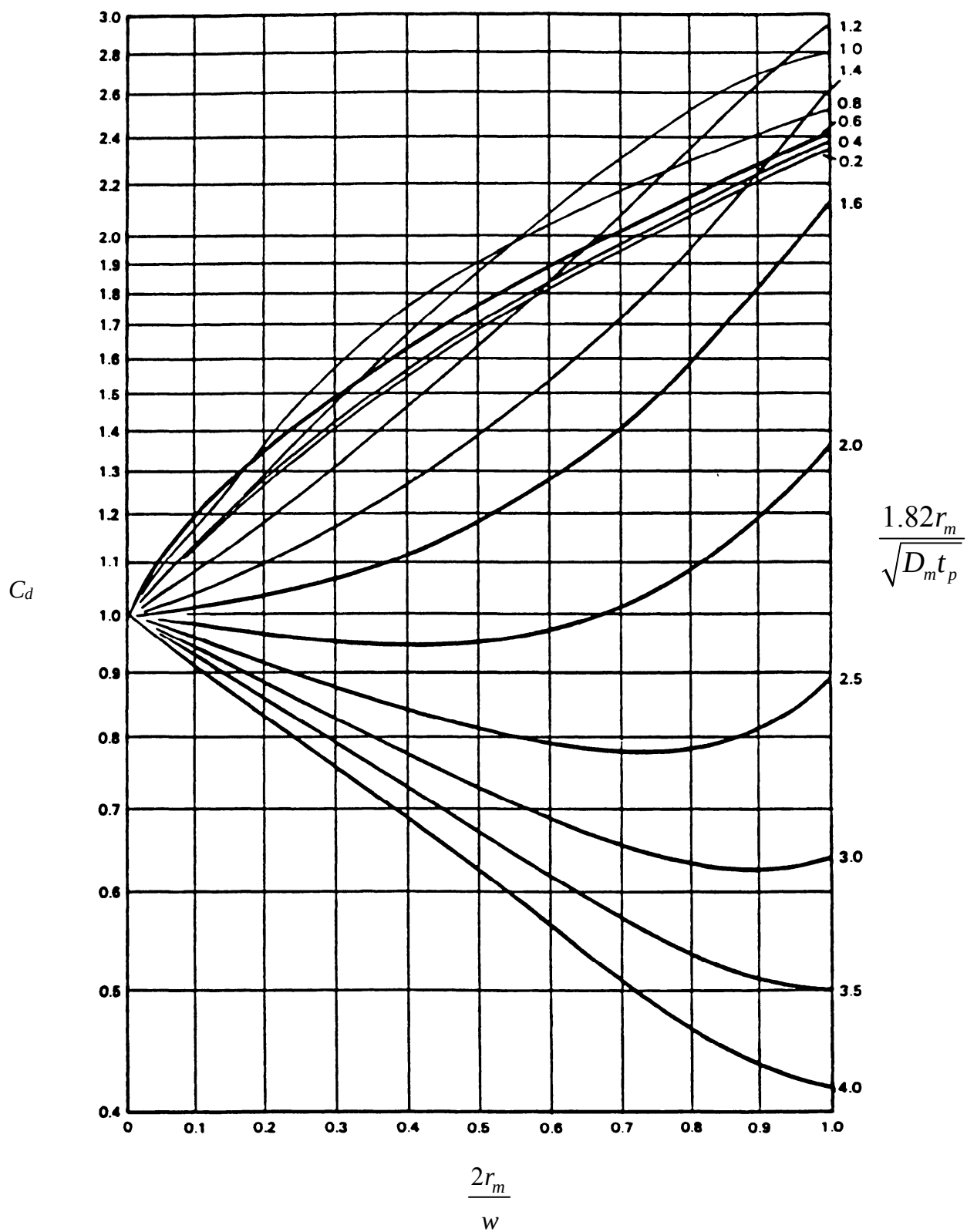
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C_f for Convolute Bellows

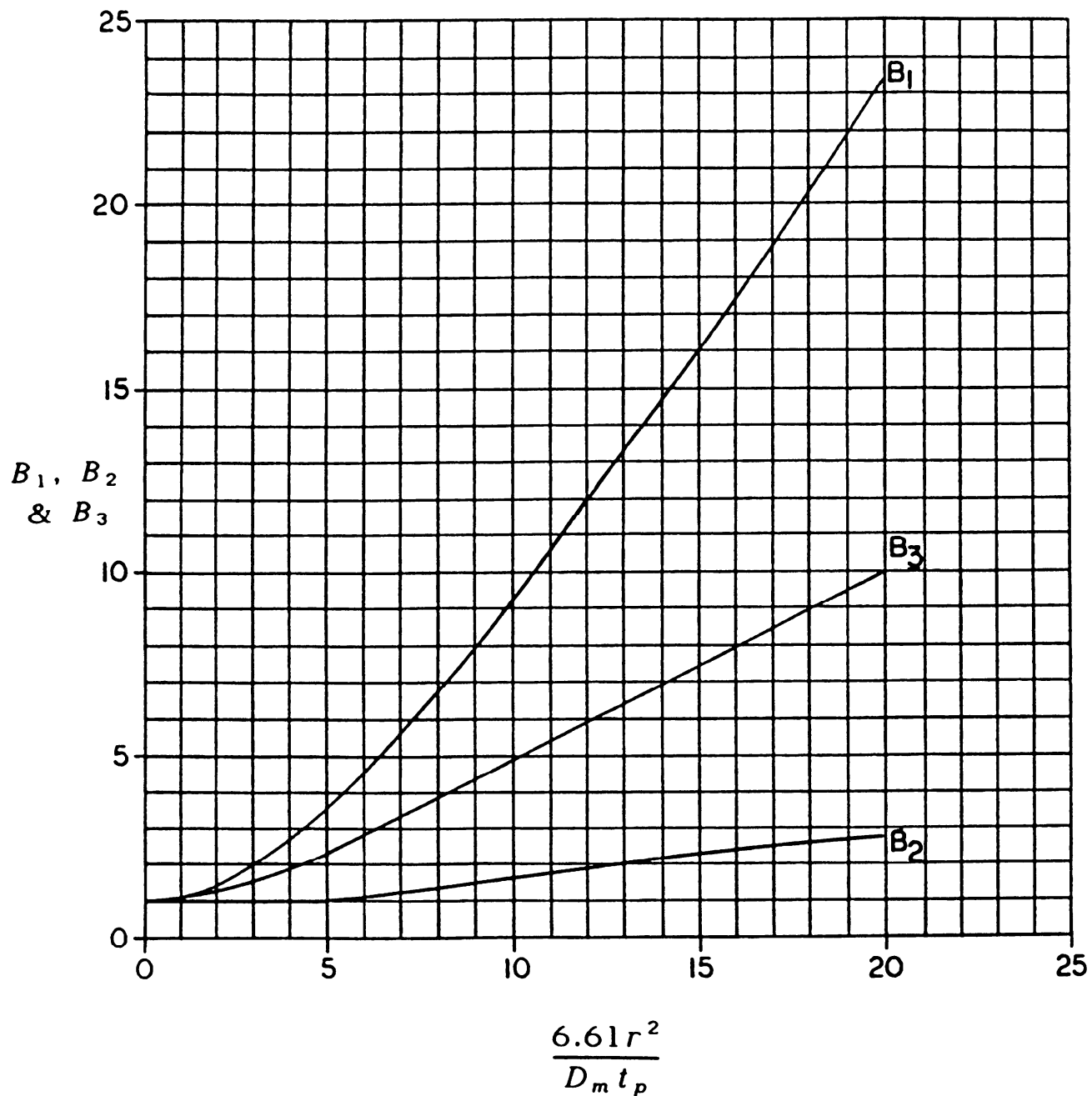
FIGURE 4.17

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
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C_d for Convolute Bellows

FIGURE 4.18



$B_1, B_2 \text{ \& } B_3$ FOR TOROIDAL BELLOWS

FIGURE 4.19

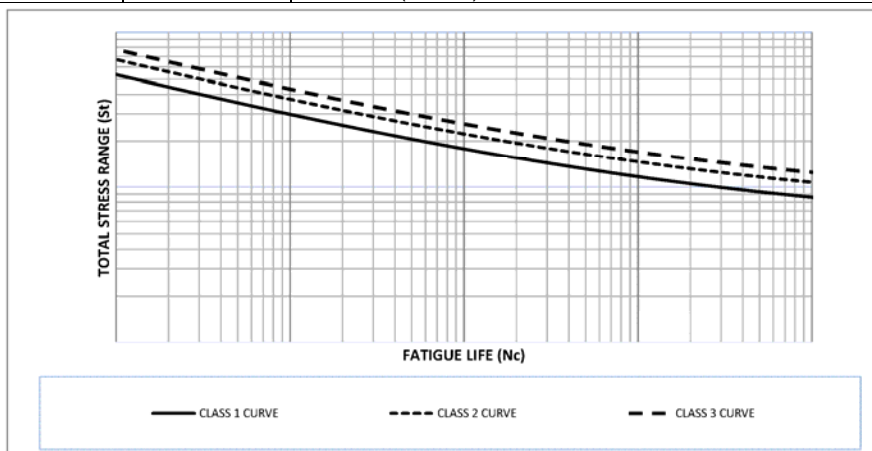
STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
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FIGURE 4.20

Unreinforced, Reinforced, and Toroid fatigue life calculations

$$N_c = \left(\frac{c}{\frac{S_t}{f_c} - b} \right)^{3.4} \quad \text{Where } S_t \text{ is PSI}$$

$$N_c = \left(\frac{c}{\frac{145S_t}{f_c} - b} \right)^{3.4} \quad \text{Where } S_t \text{ is Mpa}$$

Material Class	Manufacturing Constants		Material Grades, UNS (EN)
	c	b	
1	1.86E+06	54,000	• Austenitic stainless steels - S3xxx (1,43xx to 1,49xx) • Special nickel-chromium alloys - N08800 (1,4876) • High heat or corrosion resistant nickel alloys - N08810 (1,4958), N06600 (2,4816), N04400 (2,4360), N08811 (1,4949)
2	2.33E+06	67,500	• Corrosion resistant nickel-molybdenum-chromium alloys - N06455 (2,4610), N10276 (2,4819), N08825 (2,4858)
3	2.70E+06	78,300	• High strength nickel-chromium alloys - N06625 (2,4856)



- These equations are based on average curves but also allow for the inclusion of factor f_c , for modification to the lower bound of the curve and also to allow for the addition of a safety factors to ensure minimum fatigue life required for pressure and fluid containment applications such as ASME and PED.
- Unless otherwise indicated by specification, a traditional value, $f_c=1$, is used when providing EJMA calculations.
- All calculations for fatigue in accordance with this section shall indicate the value used for f_c .
- These curves are intended to predict fatigue life for as-formed or annealed bellows at temperatures below the bellows material creep range.
- They are considered valid in the range of $10E+02$ to $10E+06$ cycles, due to the limited data available for the very low and very high cyclic ranges.

Examples

S_t	f_c	N_c	Material Class	Reference
122,709 PSI (846 Mpa)	1	74,216	1	EJMA, Correction Factor 0%
	0.9	40,106		Correction Factor 10%
	0.5	2,278		Correction Factor 50%
	1	335,874	2	EJMA, Correction Factor 0%
	0.9	158,591		Correction Factor 10%
	0.5	6,284		Correction Factor 50%
	1	1,162,078	3	EJMA, Correction Factor 0%
	0.9	467,605		Correction Factor 10%
	0.5	12,834		Correction Factor 50%

4.15 EFFECT OF EXTERNAL PRESSURE

When an Expansion Joint is used in a system which is externally pressurized or operating under vacuum conditions, the design approach is similar to that for an internally pressurized system. There are several significant factors which must be understood and properly evaluated.

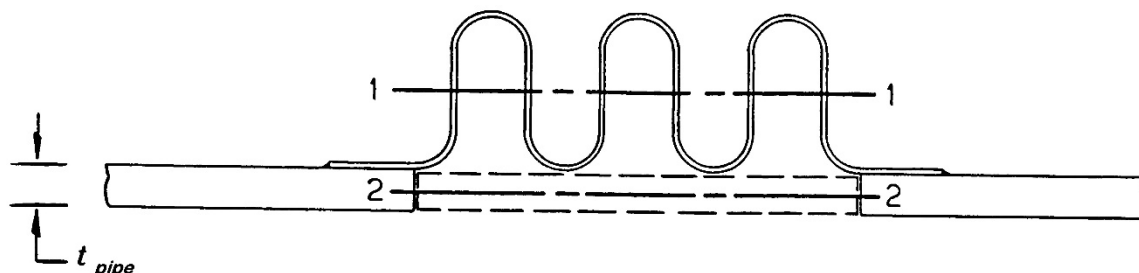


FIGURE 4.21

4.15.1 The cylindrical pressure-containing portions of the Expansion Joint (end connections, center connector between the pair of bellows in a universal Expansion Joint, for example), can be evaluated using the methods described in Section VIII, Division I of the ASME Boiler and Pressure Vessel Code. The moment of inertia of a single bellows element (I_{1-1} in Figure 4.21) is given by equation (4-75). The moment of inertia for the section of pipe that the bellow replaces (I_{2-2} in Figure 4.21) is given by equation (4-76). If I_{1-1} multiplied by the modulus of elasticity ratio is equal to or greater than I_{2-2} , as given by equation (4-78), the bellows is considered to be equivalent to the pipe for the elastic buckling analysis. The pipe including the bellows elements and the center connector for a universal expansion joint (if applicable) may be considered a continuous length and the stiffening requirements evaluated on that basis. If I_{1-1} times the modulus of elasticity ratio is less than I_{2-2} , the bellows is not considered to be equivalent to the pipe for elastic buckling analysis. The ends of the pipe on either side of the bellows and on both sides of the center connector for a universal expansion joint (if applicable) shall be evaluated as having free ends unless stiffening rings are provided adjacent to the bellows.

$$I_{1-1} = N \left[\frac{nt_p (2w - q)^3}{48} + 0.4qnt_p (w - 0.2q)^2 \right] \text{ in.}^4 \text{ (mm}^4\text{)} \quad (4-75)$$

$$I_{2-2} = \frac{L_b (t_{pipe})^3}{12(1 - \nu^2)} \text{ in.}^4 \text{ (mm}^4\text{)} \quad (4-76)$$

$$I_{1-1} \frac{E_b}{E_p} \geq I_{2-2} \quad (4-77)$$

where t_{pipe} = Nominal pipe wall thickness being used less manufacturing tolerances and corrosion allowance or minimum pipe wall thickness from external buckling analysis for the maximum length between stiffening elements that includes the bellows, in. (mm).

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The proper design of any externally pressurized system requires evaluation of the system as a whole. Designing such systems one segment at a time may lead to either misapplications or uneconomical designs.

4.15.2 The external pressure circumferential buckling capacity of the bellows convoluted length may be evaluated by methods such as that shown in the ASME Code, as a cylinder having a length equal to the bellows convoluted length, L_b and a thickness equal to that of an element having a moment of inertia I_{1-1} . If the bellows tangent length is unsupported internally it may be evaluated as a short cylinder supported by the connecting pipe on one end and the bellows convolutions on the other end.

4.15.3 Unreinforced and reinforced bellows with external pressure shall be designed for pressure capacity using equations (4-35), (4-37), (4-38), and (4-39). External bellows reinforcing members and external tangent collars are not included in the calculations for external pressure capacity. The design of toroidal bellows with external pressure is not covered by this standard.

4.15.4 Pressure thrust absorbing members such as tie rods, hinges, gimbals may be evaluated in the same manner as for an internally pressurized system. The effects of compression loading on long slender members must be taken into account.

4.15.5 For multi-ply unreinforced and reinforced bellows, the values of n and w used in the equations for determining external pressure capacity shall be based only on the plies that actively resist the external pressure. In the case of two ply designs, the following method may be used to determine the active plies and the external design pressure for the active plies:

If $P_m \leq \bar{P}$ then both plies are active and $P_e = P_o - P_i$ (zero if negative)

If $P_m > \bar{P}$ then only the inner ply is active and $P_e = P_m - P_i$

where $\bar{P}$ = Mean pressure, psia (MPaa)
 = $(P_o + P_i)/2$
 P_m = Pressure between the plies, psia (MPaa)
 P_o = Pressure outside the bellows, psia (MPaa)
 P_i = Pressure inside the bellows, psia (MPaa)
 P_e = External Design pressure, psi (MPa)

With respect to external buckling, the preceding method may be conservative and may be modified if substantiated by manufacturer's experience.

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SECTION 5 – RECTANGULAR EXPANSION JOINT DESIGN

The following sections describe the various movements, forces, and moments which occur at the interface of rectangular bellows and the associated ducting system. The movements are identical in all respects to those imposed on circular expansion joints and are defined in Section 1.2 of these standards. Further, the method of analysis of determining forces and moments resulting from these movements is also identical to circular bellows. Therefore, the same nomenclature can be used, with the exception that the terms L_l and L_s (mean length of long or short sides) is substituted for D_m (mean diameter).

The summary of equations which follows is the rectangular counterpart of the circular bellows. The explanation for the use of these equations is found in Section 4.6.1.

5.1 MOVEMENT EQUATIONS

Rectangular Expansion Joints may be subjected to axial movement, angular movement, lateral deflection or any combination of these.

- a.** Axial movement for single bellows Expansion Joint

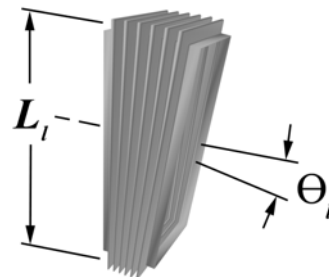
$$e_x = \frac{x}{N} \quad (5-1)$$

- b.** Axial movement for universal bellows Expansion Joints.

$$e_x = \frac{x}{2N} \quad (5-2)$$

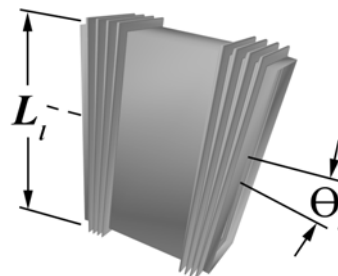
- c.** Equivalent axial movement per convolution for single or universal bellows with angular rotation.

$$e_{\theta l} = \frac{\theta_l L_l}{2N}$$



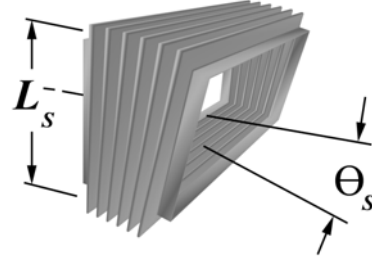
SINGLE BELLOWS
FIGURE 5.1

$$e_{\theta l} = \frac{\theta_l L_l}{4N}$$



UNIVERSAL BELLOWS
FIGURE 5.2

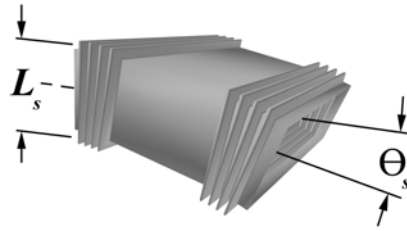
$$e_{\theta s} = \frac{\theta_s L_s}{2N}$$



(5-5)

SINGLE BELLOWS
FIGURE 5.3

$$e_{\theta s} = \frac{\theta_s L_s}{4N}$$

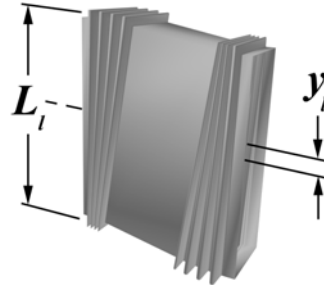


(5-6)

UNIVERSAL BELLOWS
FIGURE 5.4

- d. Equivalent axial movement per convolution for universal bellows with lateral movement.
For lateral movement in a direction parallel with the long side:

$$e_{yl} = \frac{3L_t}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* \pm x/2)} y_l$$

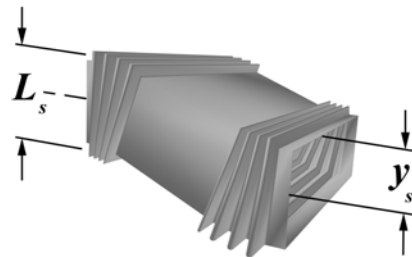


(5-7)

UNIVERSAL BELLOWS
FIGURE 5.5

For lateral movement in a direction parallel with the short side:

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* \pm x/2)} y_s$$



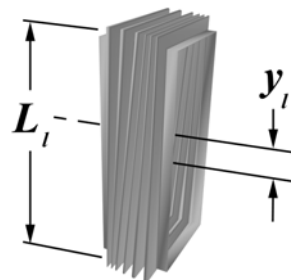
(5-8)

UNIVERSAL BELLOWS
FIGURE 5.6

- e. Equivalent axial movement per convolution for single bellows with lateral movement.

For lateral movement in a direction parallel to the long side:

$$e_{yl} = \frac{3L_l y_l}{N(L_b \pm x)}$$

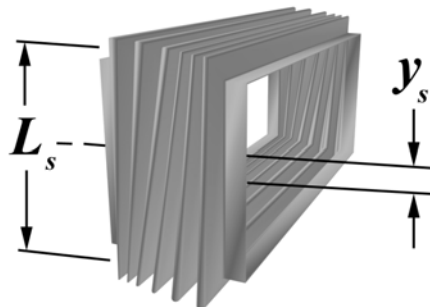


(5-9)

SINGLE BELLOWS
FIGURE 5.7

For lateral movement in a direction parallel to the short side:

$$e_{ys} = \frac{3L_s y_s}{N(L_b \pm x)}$$



(5-10)

SINGLE BELLOWS
FIGURE 5.8

5.2 COMBINING MOVEMENTS

Rectangular bellows differ from round bellows in the manner that total equivalent axial movement per convolution is determined. Where vector analysis is permissible for analyzing compound lateral and angular movement for round bellows, it is not permissible for rectangular bellows. The maximum total equivalent motion occurs at the corners of the bellows element and is equal to the algebraic sum of the equivalent axial motions for the lateral and angular movements that are parallel to either the long or the short sides. The equivalent axial movement per convolution for movement parallel to the long side is calculated separately from that for the short side. These separate movements are combined to determine the total equivalent axial compression or extension as follows:

$$e_c = e_{yl} + e_{ys} + e_{\theta l} + e_{\theta s} + |e_x| \quad (5-11)$$

$$e_e = e_{yl} + e_{ys} + e_{\theta l} + e_{\theta s} - |e_x| \quad (5-12)$$

where x is axial compression and the plane of deflection of y_l and θ_l is perpendicular to the plane of y_s and θ_s . Where x is extension, the signs of e_x should be reversed in the above equations. The design of every expansion joint must be such that the total displacement per convolution from all sources does not exceed the rated values, that is:

$$\begin{aligned} e_c(\text{calculated}) &\leq e_c(\text{rated}) \\ e_e(\text{calculated}) &\leq e_e(\text{rated}) \end{aligned}$$

Rated movements should be obtained from the expansion joint manufacturer.

5.3 MOVEMENT RANGE

The total equivalent axial movement range per convolution (e) is that which results from the movement of an Expansion Joint from its initial position in the ducting system to the operating position. When an Expansion Joint is installed without lateral or angular cold spring, e is the greater of e_c or e_e as calculated from the initial to the operating position under consideration. Care must be exercised when evaluating rectangular bellows. Maximum values for e_c or e_e must be found by calculating all movements in their respective planes. Vector addition can only be applied to round bellows. When cold springing is involved, the e_c or e_e due to the cold spring must be added algebraically to the e_c or e_e due to movement from the neutral to the operating position in order to obtain the maximum movement range, e .

Refer to Appendix J Example 10 for a sample calculation.

5.4 FORCE AND MOMENT CALCULATIONS

In order to evaluate the loads acting upon ducting, supports, or equipment, it is frequently necessary to determine the forces and moments required to move an Expansion Joint. For this reason, the catalogs of most Expansion Joint manufacturers contain force data for the standard designs offered. This data frequently is expressed as the force required to move a single convolution to the rated axial movement established by the manufacturer. For convenience, it is desirable to divide this force by the rated movement to obtain a convolution resistance factor or working spring rate f_w in pounds per inch (newtons per millimeter) of movement per convolution. Having determined this factor, the moments and forces required to move a rectangular Expansion Joint may be calculated using the equations as follows:

$$F_a = f_w e_x \quad (5-13)$$

$$M_{Ll} = \frac{f_w L_{ml} e_{yl}}{2} \quad \text{(For lateral deflection parallel to the long side)} \quad (5-14)$$

$$M_{Ls} = \frac{f_w L_{ms} e_{ys}}{2} \quad \text{(For lateral deflection parallel to the short side)} \quad (5-15)$$

$$M_{\theta l} = \frac{f_w L_{ml} e_{\theta l}}{2} \quad \text{(For angular rotation of the long side)} \quad (5-16)$$

$$M_{\theta s} = \frac{f_w L_{ms} e_{\theta s}}{2} \quad \text{(For angular rotation of the short side)} \quad (5-17)$$

$$V_{Ll} = \frac{f_w L_{ml} e_{yl}}{L_b} \quad \text{(For lateral deflection parallel to the long side of a single expansion joint)} \quad (5-18)$$

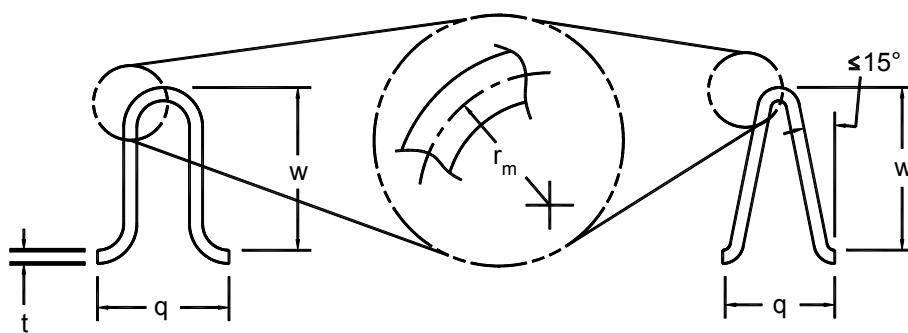
$$V_{Ls} = \frac{f_w L_{ms} e_{ys}}{L_b} \quad \text{(For lateral deflection parallel to the short side of a single expansion joint)} \quad (5-19)$$

$$V_{Ll} = \frac{f_w L_{ml} e_{yl}}{L_u} \quad \text{For lateral deflection parallel to the long side of a universal expansion joint} \quad (5-20)$$

$$V_{Ls} = \frac{f_w L_{ms} e_{ys}}{L_u} \quad \text{For lateral deflection parallel to the short side of a universal expansion joint} \quad (5-21)$$

The preceding relationships are applicable to all rectangular Expansion Joints. It should be noted, however, that every equation is dependent upon data which must be supplied by the Expansion Joint manufacturer. For standard designs, all necessary data is available in the catalogs of the individual manufacturers, or is obtainable from them. IN NO CASE SHOULD DATA OF ONE MANUFACTURER BE APPLIED TO THE PRODUCT OF ANOTHER SINCE, DUE TO FUNDAMENTAL DESIGN DIFFERENCES, THESE FACTORS MAY VARY.

5.5 DESIGN EQUATIONS



"U" SHAPE

"V" SHAPE

TYPICAL CONVOLUTION SHAPES

Other shapes must be individually analyzed.

FIGURE 5.9

The equations shown below pertain only to the single ply convolution shapes shown in Figure 5.9.

Bellows Longitudinal Membrane Stress Due To Pressure

$$S_{7l} = \frac{PL_s q}{2A_c} \quad (\text{long side}) \quad (5-22)$$

$$S_{7s} = \frac{PL_t q}{2A_c} \quad (\text{short side}) \quad (5-23)$$

Note: If $N=1$, set S_{7l} and $S_{7s} = 0$.

Bellows Longitudinal Bending Stress Due To Pressure

$$S_{8la} = \frac{PNqL_t^2 w}{24I} \quad (\text{long side}) \quad (5-24)$$

$$S_{8lb} = \frac{P(Nq + 2L_t)^2}{2t^2} \quad (5-25)$$

If $S_{8la} \leq 1.33K_s S_{ab}$, then $S_{8l} = S_{8la}$

If $S_{8la} > 1.33K_s S_{ab}$, then $S_{8l} = S_{8lb}$

$$S_{8sa} = \frac{PNqL_s^2 w}{24I} \quad (\text{short side}) \quad (5-26)$$

$$S_{8sb} = \frac{P(Nq + 2L_t)^2}{2t^2} \quad (5-27)$$

If $S_{8sa} \leq 1.33K_s S_{ab}$, then $S_{8s} = S_{8sa}$

If $S_{8sa} > 1.33K_s S_{ab}$, then $S_{8s} = S_{8sb}$

Notes:

If the tangent is fully supported against the pressure, set $L_t = 0$.

If $N=1$, set S_{8l} and $S_{8s} = 0$.

Bellows Meridional Bending Stress Due To Pressure (Sidewall)

$$S_9 = \frac{P}{2} \left(\frac{w}{t} \right)^2 \left(1.0 - \frac{1.3r_m}{w} \right) \quad (5-28)$$

Bellows Meridional Bending Stress Due To Pressure (Tangent)

$$S_{11} = \frac{0.938P(L_t)^2}{t^2} \quad (5-29)$$

Note: the above stresses should be evaluated for pressure capacity as follows:

$$S_{7l} \text{ and } S_{7s} \leq S_{ab}$$

Below the creep range

$$S_{7l} + S_{8l} \leq 1.33K_s S_{ab} \text{ when } S_{8la} \leq 1.33K_s S_{ab}$$

$$S_{7s} + S_{8s} \leq 1.33K_s S_{ab} \text{ when } S_{8sa} \leq 1.33K_s S_{ab}$$

$$S_{8l} \leq C_a S_{ab} \text{ when } S_{8la} > 1.33K_s S_{ab}$$

$$S_{8s} \leq C_a S_{ab} \text{ when } S_{8sa} > 1.33K_s S_{ab}$$

$$S_9 \leq 1.5S_{ab}$$

$$S_{11} \leq 1.5S_{ab}$$

In the creep range

$$S_{7l} + \frac{S_{8l}}{1.25} \text{ and } S_{7s} + \frac{S_{8s}}{1.25} \leq S_{ab}$$

$$S_9 \leq 1.25S_{ab}$$

$$S_{11} \leq 1.25S_{ab}$$

Bellows Meridional Bending Stress Due To Deflection

$$S_{10} = \frac{5E_b t e}{3w^2 (1.0 + 3r_m / w)} \quad (5-30)$$

Note: Modulus of elasticity, E_b , in equation (5-30) is at room temperature.

Bellows Deflection Due To Pressure (beam mode) At Center of Span and Midpoint of Bellows Length

$$y_{bmla} = \frac{PNqL_l^4}{384E_bI} \quad (5-31)$$

$$y_{bmlb} = \frac{P(Nq + 2L_t)^4}{32E_b t_e^3} \quad (5-32)$$

If $S_{8la} \leq 1.33K_s S_{ab}$, then $y_{bml} = y_{bmla}$

If $S_{8la} > 1.33K_s S_{ab}$, then $y_{bml} = y_{bmlb}$

$$y_{bmsa} = \frac{PNqL_s^4}{384E_bI} \quad (5-33)$$

$$y_{bmsb} = \frac{P(Nq + 2L_t)^4}{32E_b t_e^3} \quad (5-34)$$

If $S_{8sa} \leq 1.33K_s S_{ab}$, then $y_{bms} = y_{bmsa}$

If $S_{8sa} > 1.33K_s S_{ab}$, then $y_{bms} = y_{bmsb}$

Notes:

If the tangent is fully supported against the pressure, set $L_t = 0$.

If $N=1$, set y_{bml} and $y_{bms} = 0$.

Fatigue Life

$$N_b = \left(\frac{c}{St - b} \right)^a \quad (5-35)$$

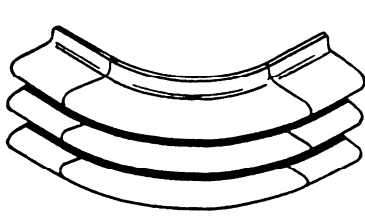
where a , b , and c are material and manufacturing constants. Fatigue data must be furnished by individual manufacturers. $S_t = C_{sp} S_9 + C_{sf} S_{10}$

Bellows Theoretical Axial Elastic Spring Rate

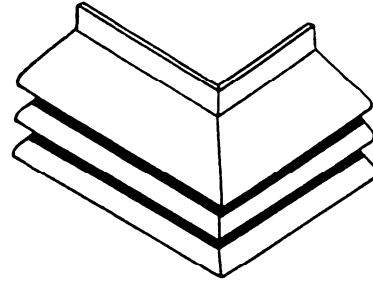
$$f_i = \frac{E_b t^3 (L_l + L_s)}{w^3 (1.0 + 3.4 r_m / w)} \quad (5-36)$$

The effect of corner configuration (see Figure 5.10) is not considered equation 5-36. Corner configuration will not significantly effect the spring rate performance when the length of the shortest side exceeds $10w$; $L_s / w > 10$. When this value is less than 10, consult the bellows manufacturer for information.

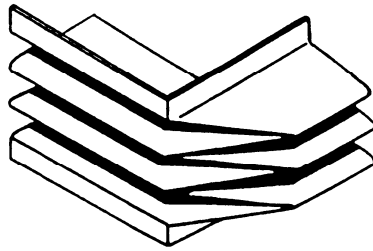
Longitudinal bending stress and mid-point deflection can be reduced by the addition of intermediate supports along the span.



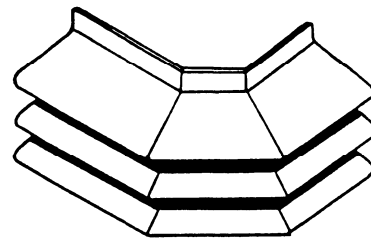
ROUNDED CORNER



SINGLE MITER CORNER



CAMERA CORNER



DOUBLE MITER CORNER

TYPICAL CORNER CONFIGURATIONS

FIGURE 5.10

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION

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SECTION 6 – QUALITY ASSURANCE AND BELLOWS FORMING METHODS

This section describes the minimum quality control program requirements for a manufacturer of metallic bellows type expansion joints inclusive of the product design and compliance to customer specifications. These requirements pertain to the inspections and tests necessary to substantiate product conformance to drawings, specifications and contract requirements. The program shall assure systematic and adequate quality control throughout all areas of contract performance; for example, product development, material selection, fabrication, processing, assembly, inspection, testing, delivery preparation and shipment, storage and maintenance, for which comprehensive written procedures shall be used and maintained and made available for customer review if requested. These quality control program requirements shall apply when a customer specification identifies an expansion joint to be "designed and manufactured to The Standards of the Expansion Joint Manufacturers Association". These section requirements shall be in addition to and shall not conflict with any other contractual agreements.

6.1 GENERAL

An effective and economical quality control program shall be developed, considering the manufacturer's facilities and products. The necessary scope and detail of the program shall depend upon the complexity of the work being performed and on the size and capabilities of the manufacturer. All supplies and services under the contract, whether manufactured or performed within the manufacturer's plant or at any other source, shall be controlled at all points necessary to assure conformance to the contractual requirements. The program shall provide for the prevention and prompt detection of non-conformities and for timely and positive corrective action. The following is a guide to the features which shall be included in the written description of the manufacturer's quality control program and shall be pertinent to both shop and field work.

6.2 AUTHORITY AND RESPONSIBILITY

Effective management for quality shall be clearly prescribed by the manufacturer. Personnel in charge of the design, manufacturing, testing, and quality functions shall have sufficient and well defined responsibilities, the authority, and organizational freedom to identify and evaluate quality problems and to initiate, recommend, or provide solutions. Management shall regularly review the status and adequacy of the quality control program. The quality program shall be certified and monitored by an internationally recognized standards authority.

6.3 QUALITY ASSURANCE ORGANIZATION

An organization chart showing the relationship between management, engineering, purchasing, manufacturing, inspection, and quality control is required to reflect the actual organization. The purpose of this chart is to identify and associate the various organizational groups within the particular function for which they are responsible.

6.4 DRAWINGS, DESIGN CALCULATIONS, AND SPECIFICATION CONTROL

The quality control program shall establish comprehensive written procedures which will assure that the latest applicable drawings, design calculations, specifications, and manufacturing processes required by the contract, as well as authorized changes, are in use for manufacture, examination, inspection, and testing. The manufacturer shall assure that requirements for the effectivity point of changes are met, and that obsolete drawings and change requirements are recalled and replaced from all points of issue and use. The manufacturer shall maintain a record of all customer approved drawings, specifications, and all drawing revisions pertinent to the contract provisions.

6.5 MATERIALS AND MATERIALS CONTROL

A. Supplier's materials and products shall be subject to receiving inspection to the extent necessary to assure that the material is properly identified and has documentation including required certificates of compliance or material test reports showing conformance to the manufacturer's contractual specifications. The quality control program shall assure that raw materials to be used in fabrication or processing of products conform to the applicable physical, chemical, and other technical requirements.

B. Raw material awaiting testing must be separately identified or segregated from already tested and approved material but can be released for initial production providing that identification and control can be maintained.

C. Material tested and approved must retain its identity until such time as its identity is necessarily obliterated by processing.

6.6 MANUFACTURING PROCESS CONTROL

The quality control program must assure that all basic production operations (i.e. purchasing, handling, machining, assembling, fabricating, processing, inspection, testing, etc.) of any type shall be described in comprehensive and complete written documented instructions. Such instructions shall provide the criteria for performing the work functions and they shall be compatible with acceptance criteria for workmanship. The instructions are intended to also serve for supervising, inspecting and managing work. The preparation and maintenance of and compliance with work instructions shall be monitored as a function of the quality control program.

6.7 IN-PROCESS INSPECTION AND EXAMINATION PROGRAM

A. The quality control program shall describe the fabrication operations, including inspections and examinations, sufficiently to permit a customer or designated inspector to determine at what stages specific inspections and examinations are to be performed, and to positively identify the current inspection status of the product. The manufacturer shall prepare, maintain and use comprehensive written procedures addressing the in-process and final inspection operations that are to be performed in the course of manufacture and testing. These procedures shall specify the dimensional checks, visual inspection, nondestructive tests, and other pertinent operations that are to be performed to determine that the product meets contractual specifications. The procedures shall specify the applicable acceptance standards and shall provide for a means to document that key operations have been performed and the results determined to be satisfactory.

B. The quality control program shall assure there is a system for final inspection and test of completed products. Such inspection and testing shall provide a measure of overall quality of the completed product. When modifications, repairs or replacements are required after final inspection or testing, there shall be re-inspection and testing of any characteristics affected.

C. The inspector representing the customer shall have access at all times, while work on the contract is being performed, to all parts of the manufacturer's plant that concern the manufacture of the product ordered. The manufacturer shall afford the inspector reasonable facilities to satisfy the inspector that the product is being furnished in accordance with the contract specifications. Inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be scheduled not to interfere unnecessarily with the operations of the manufacturer. This requirement also applies to all subcontractors and vendors.

6.8 MEASURING AND TEST EQUIPMENT CONTROL

The manufacturer shall have at his disposal gages and other measuring and testing equipment necessary to assure that materials and supplies conform to the technical requirements specified. A system of comprehensive written calibration procedures shall be maintained. In order to assure continuous accuracy, the procedures shall include a schedule for equipment calibration against certified measurement standards which have known valid relationships to National Reference Standards. Defective equipment must be repaired, replaced, or re-calibrated as appropriate to the technical requirements specified. This requirement also applies to all subcontractors or vendors.

6.9 MATERIAL NON-CONFORMANCE CONTROL

The manufacturer shall establish and maintain an effective and positive system for promptly detecting and correcting materials or conditions adverse to quality, including comprehensive written procedures for their identification, segregation, and disposition. All non-conforming materials shall be positively identified and segregated in a unique holding location to prevent unauthorized use, shipment, or the intermingling with acceptable conforming materials. Repair or rework of non-conforming materials shall be in compliance with comprehensive written procedures.

6.10 CORRECTIVE ACTION (SUPPLIES AND SERVICES)

Design, purchasing, manufacturing, inspection, testing or other operations which could result in, or have resulted in non-conforming supplies, services, facilities, technical data, standards or other elements of contract performance must be identified and changed as a result of the quality control program. Corrective action shall extend to the performance of all suppliers and vendors. Corrective action shall include as a minimum:

- a.) analysis of data and examination of product scrapped or reworked to determine extent or causes.
- b.) analysis of trends in processes or performance of work to prevent recurrence of non-conformances.
- c.) introduction of required improvements and corrections, initial review of the adequacy of such measures and the continued monitoring of the corrective action effectiveness.

6.11 WELDING

Unless otherwise specified by contractual agreement, the welding personnel and procedures shall be qualified in accordance with the applicable sections of Section IX of the ASME Boiler & Pressure Vessel Code or equivalent for all pressure containing welds.

6.12 HEAT TREATMENT

Unless otherwise specified by contractual agreement, heat treatment, when required, shall be performed in accordance with the ASME Boiler & Pressure Vessel Code requirements or equivalent or the recommendations of the material manufacturers.

6.13 PACKAGING, PRESERVATION, SHIPPING AND STORAGE

The manufacturer shall utilize standard commercial practices in packaging, preservation, shipping and storage to assure protection of the product during shipment, unless superseded by contractual agreement. These commercial practices shall be adequate to protect the quality of the products fabricated from deterioration to the point of final destination.

6.14 CUSTOMER QUALITY ASSURANCE AUDITS

Documents, procedures, and processes shall be comprehensively written and available for review by the customer specifying their implementation, or a third party inspection agency authorized to act in the customer's behalf.

6.15 RECORDS RETENTION

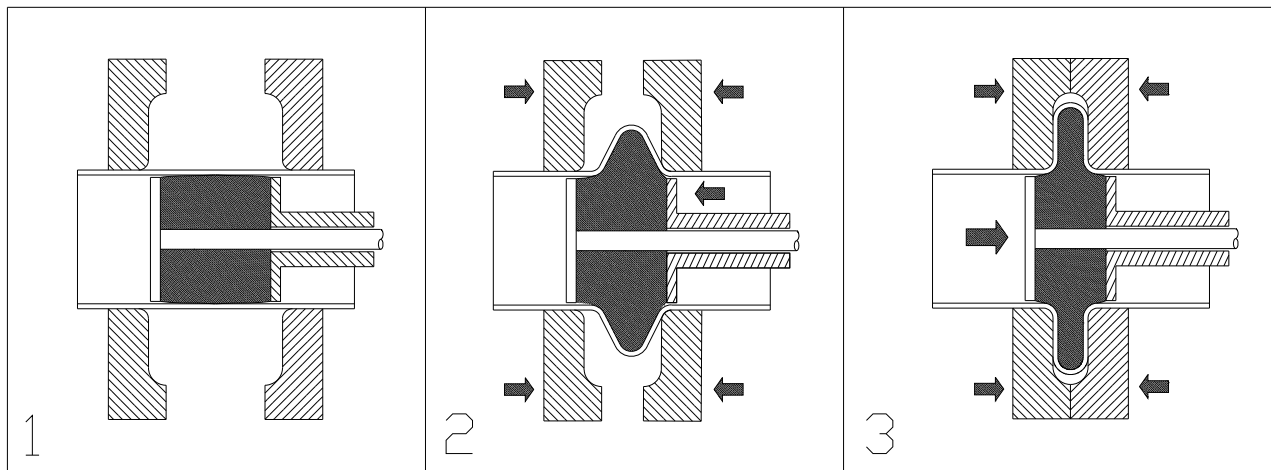
The manufacturer shall use and maintain all adequate records or data essential to the economical and effective operation of this quality control program. The records shall, as a minimum, indicate the nature and number of observations made, the number and type of deficiencies found, the quantities approved and rejected and the nature of the corrective actions taken. The quality control program shall assure the records are complete and reliable. Also, the records for monitoring work performance and for inspection and testing shall indicate the acceptability of work or products and the corrective action taken in connection with deficiencies. The quality control program shall provide for the analysis and use of these records as a basis for management review.

6.16 METHODS OF FORMING METAL BELLOWS

The following are examples of commonly used bellows forming methods. Only seamless tubes or longitudinally welded metal tubes are allowed for use with forming methods 6.16.1 thru 6.16.6.

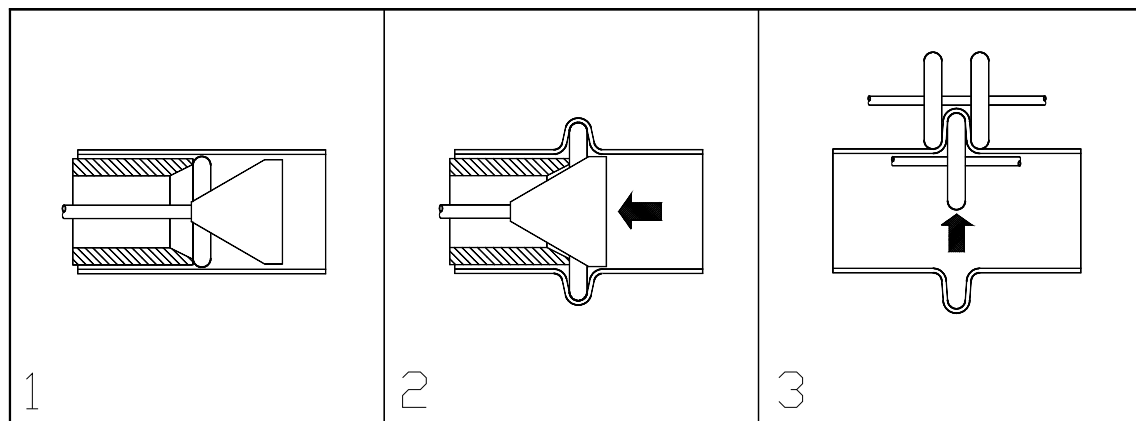
6.16.1 ELASTOMERIC FORMING

A tube is inserted over a mandrel containing a rubber torus. Axial force on the mandrel expands the torus, forming a bulge in the tube. The torus is then relaxed and the bulge is axially compressed into a convolution by external dies. Convolutions are formed one at a time. The tube is free to shorten as the convolution is formed.



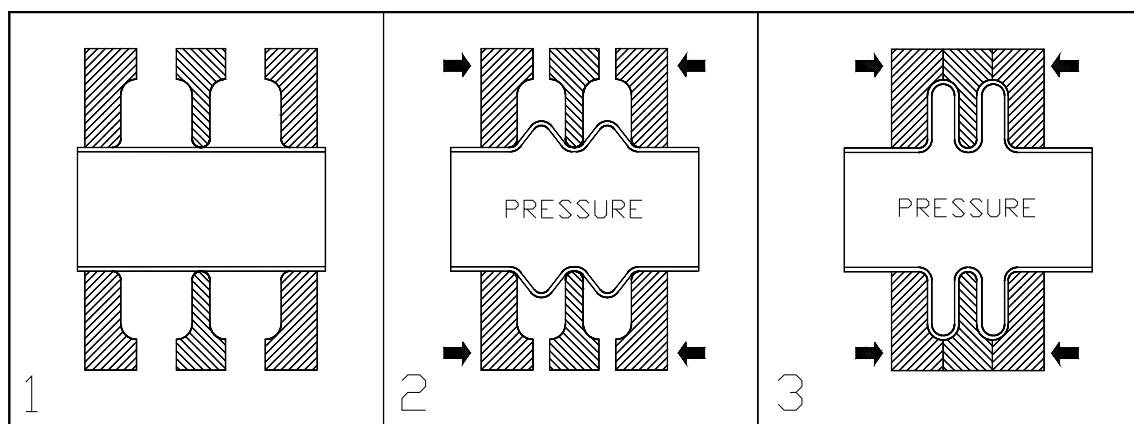
6.16.2 EXPANSION (EXPANDING MANDREL) FORMING

Individual convolutions are formed in a tube by an expanding internal mandrel. Flat spots are minimized by expanding the mandrel partially, and rotating the tube slightly. This process is repeated until an intermediate convolution height is achieved. Each convolution is subsequently sized by means of specially contoured inner and outer rollers.



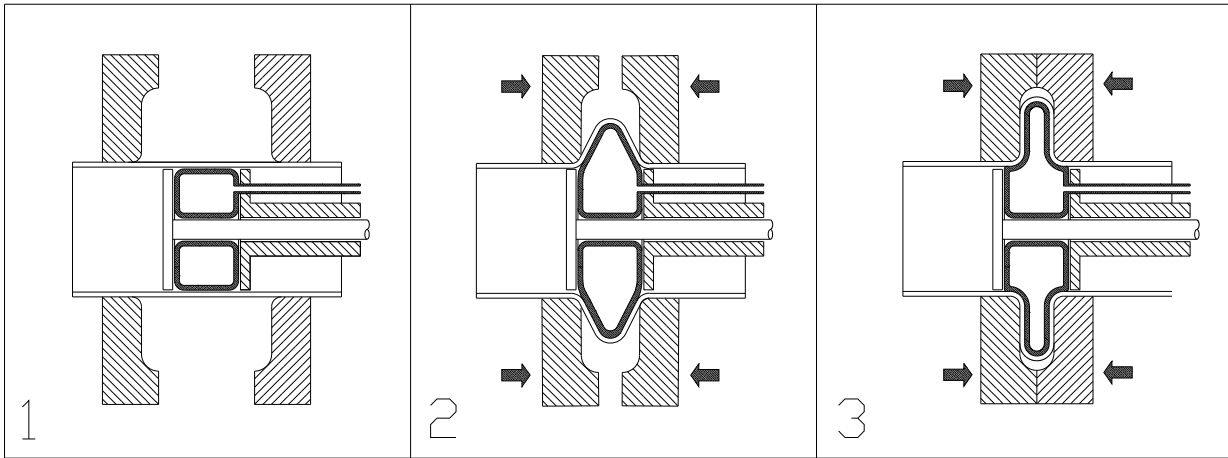
6.16.3 HYDRAULIC FORMING

A tube is placed in a hydraulic press or bellows forming machine. Circular external die rings of suitable contour are placed outside the tube at longitudinal intervals approximately equal to the developed length of the completed convolutions. The tube is filled with a medium such as water and pressurized until circumferential yielding occurs. This forming operation continues with a simultaneous circumferential yielding and controlled longitudinal shortening of the tube until the proper configuration is obtained. Individual or multiple convolutions may be formed by this method. Depending on the bellows configuration, several partial-forming steps with intermediate heat treatment may be required. Reinforced bellows may be formed by utilizing external reinforcing rings that act as part of the forming dies. After completion, when the dies are removed, the rings remain as an integral part of the bellows.

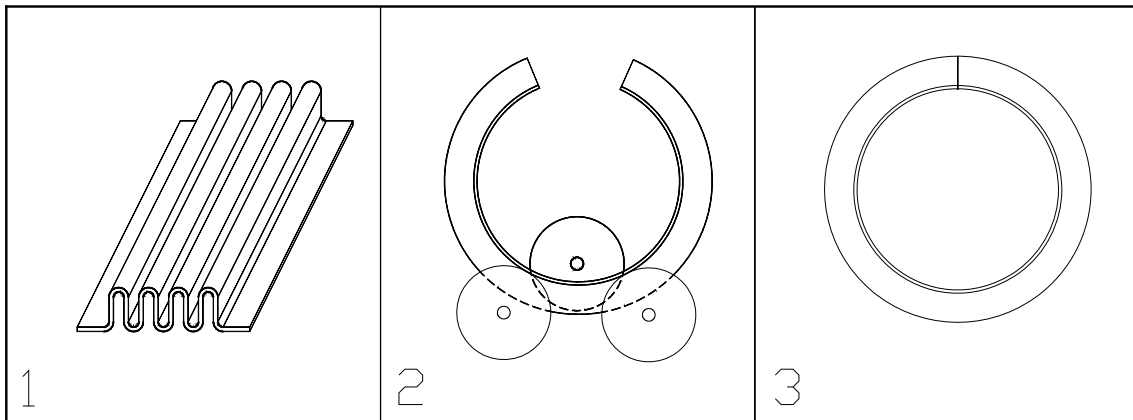


6.16.4 PNEUMATIC TUBE FORMING

This method is identical to “Elastomeric Forming” except that the initial bulge is formed by pressurizing a rubber “inner tube”.

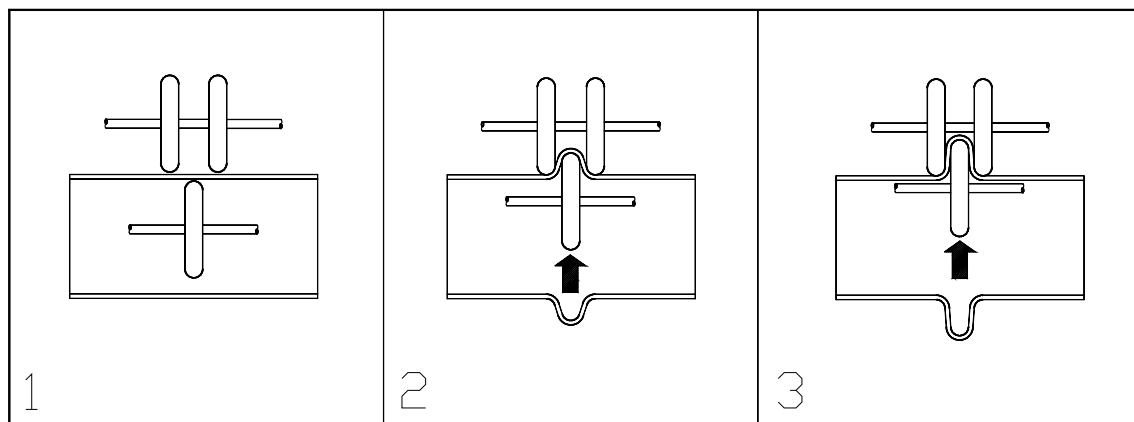
**6.16.5 ROLLED CONVOLUTED SHEET**

A flat sheet is mechanically convoluted by either the press-brake method or the roll forming method modified to produce straight sections. This pre-formed rail is then rolled into a tube. The bellows is completed by longitudinally welding the convoluted ends of the rail together.



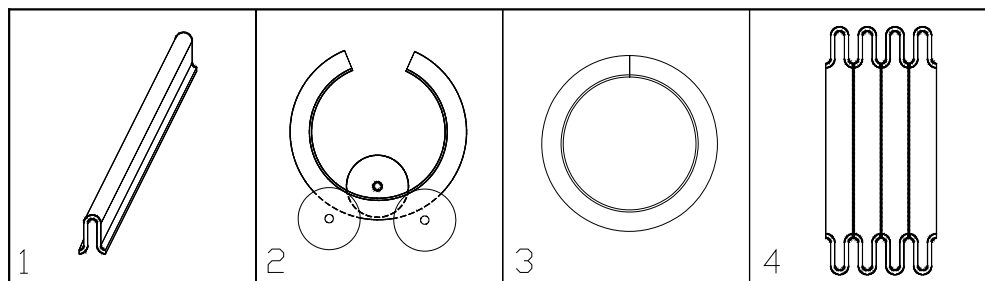
6.16.6 ROLL FORMING

A tube is placed in a forming machine and individual or multiple convolutions are formed by means of pressure exerted by forming wheels. Generally, the wheels are on both the inside and outside of the tube. Controlled longitudinal shortening of the bellows tube occurs during the forming operation. The tube may rotate about fixed-shaft forming wheels, or the tube may be fixed and the wheels rotated about the tube's circumference. The example below shows the fixed-shaft method.



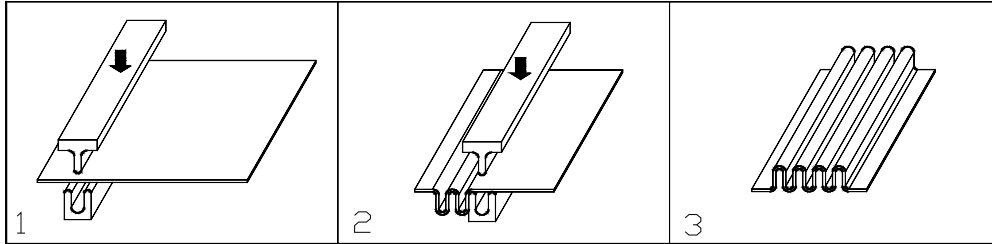
6.16.7 ROLLED RING

A flat sheet is formed into a single convolution and then rolled into a ring. The ring is completed by a longitudinal weld across the convolution. If more than one convolution is desired, the bellows is built up by a series of circumferential welds joining the convolutions together.



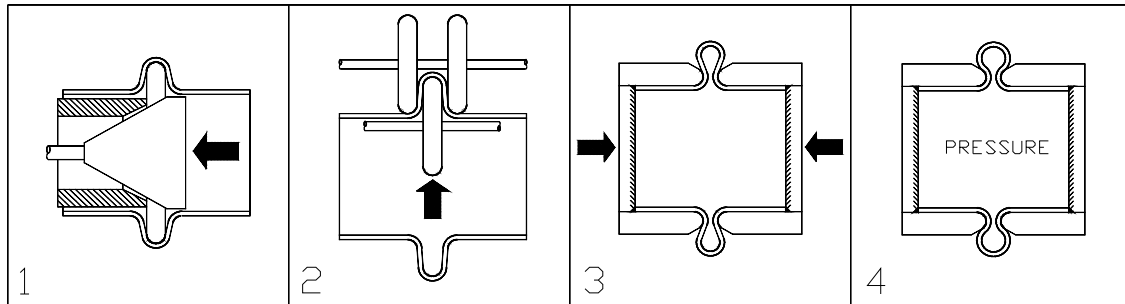
6.16.8 PRESS-BRAKE FORMING

A flat sheet is convoluted using a press-brake die to form the individual convolutions. This method is used primarily in the manufacture of bellows for rectangular Expansion Joints described in Section 5. Many convolution profiles can be achieved using this method. The most common styles are the “U” profile and “V” profiles shown in Figure 5.9. Material availability and press-break tooling limit the length of the rail. Longer lengths can be manufactured by splicing the rails together with longitudinal welds.



6.16.9 COMBINED FORMING

Some of the methods described in previous sections can be combined. One procedure for forming a toroidal bellows (Figure 4.15) combines two methods. A convolution is expansion formed with a convolution height greater than the final desired torus height. The convolution is located between forming rings similar to hydraulic forming. The rings are then pushed together and the toroid is hydraulically formed.



6.17 FABRICATION TOLERANCES

This Section covers standard manufacturing tolerances for fabricated assemblies containing Expansion Joints. If required, closer tolerances than those indicated may be obtained but must be subject to agreement between the purchaser and the manufacturer of the Expansion Joint.

Flanges for Round Expansion Joints
(Up to 96 in. (2400 mm) Nominal Diameter)

<i>Standard Flanges:</i> Flanges to standards such as ANSI B16.47, B16.5, MSS SP44, AWWA C207	Dimensions and tolerances conform to the standard.
<i>Non-Standard Machined Flanges:</i> Including plate flanges with standard drilling	Flanges to be faced and drilled. Drilling tolerance for bolt, circle and hole location same as standard. Minimum thickness to be specified
<i>Non-Standard Unmachined Flanges:</i> Rolled angle, rolled bar, flame cut plate flanges, etc.	All dimensions are nominal.

LENGTH TOLERANCE (Measured between working points):

± 1/8 in. up through 3 ft.	± 3 mm up through 1m
± 1/4 in. above 3 ft. through 12 ft.	± 6 mm above 1m up through 4m
± 3/8 in. over 12 ft.	± 10 mm above 4m

THIS SYMBOL DESIGNATES WORKING POINT: 

Flanges must be installed so that bolt holes straddle a common centerline within 1/16 in. (1.5mm)

NOTE: Good practice suggests that one mating flange in the piping system remain unwelded until the Expansion Joint has been located in position.

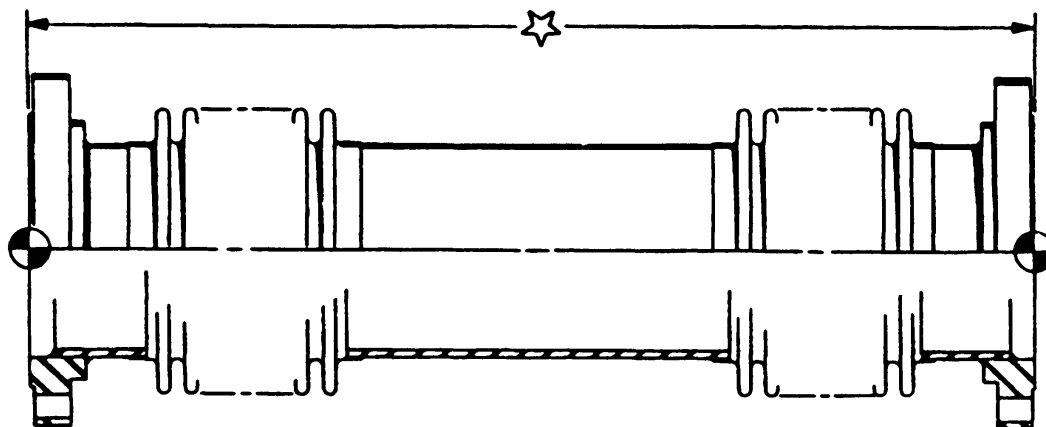
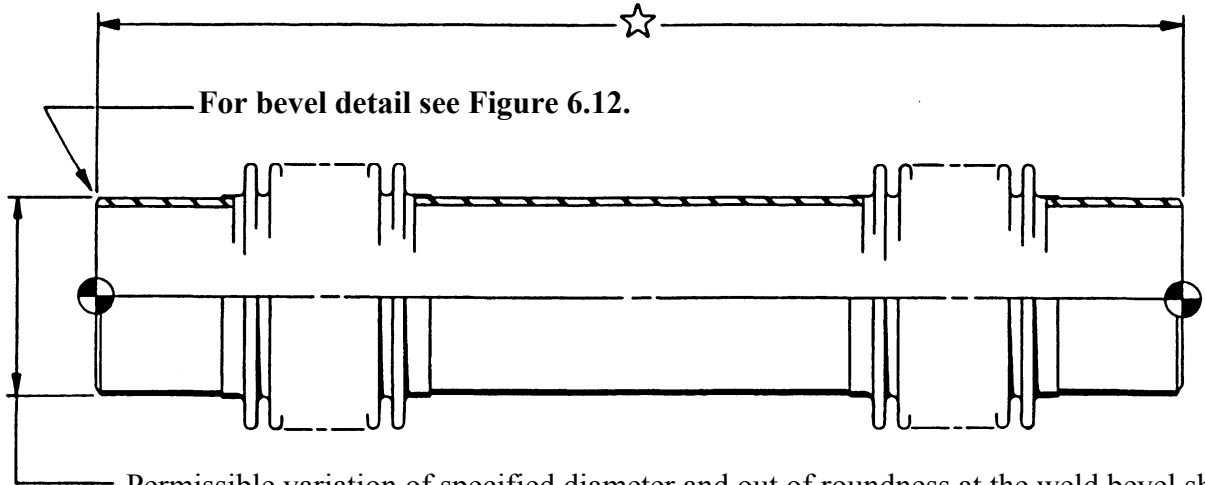
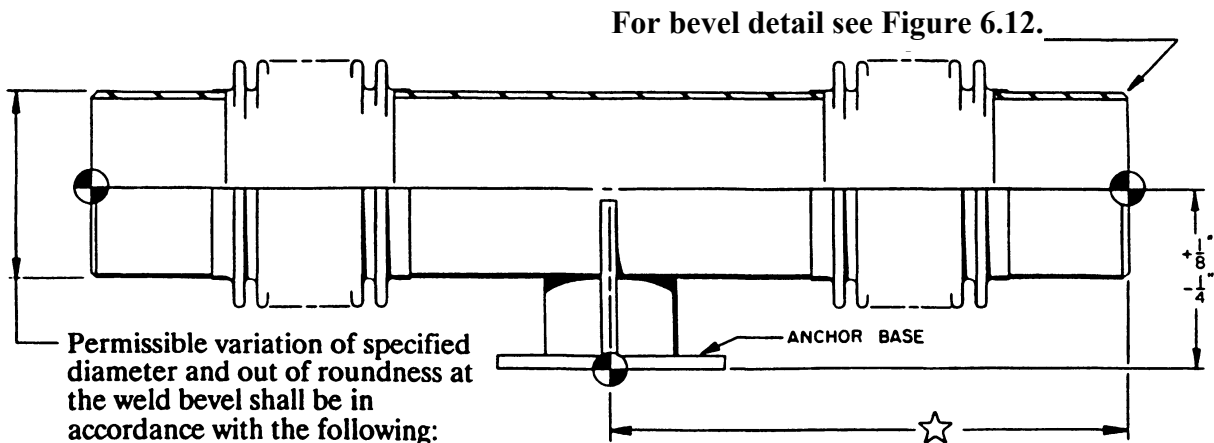


FIGURE 6.1



Permissible variation of specified diameter and out of roundness at the weld bevel shall be in accordance with the following:
 24 in. (600 mm) diameter or less in accordance with pipe specification.
 Over 24 in. (600 mm) diameter: Outside diameter 0.5% of the specified outside diameter based on circumferential measurement.
 Out-of-roundness: Difference between major and minor diameters not to exceed 1% of nominal diameter.

FIGURE 6.2



Permissible variation of specified diameter and out of roundness at the weld bevel shall be in accordance with the following:

24 in. (600 mm) diameter or less in accordance with pipe specification.
 Over 24 in. (600 mm) diameter: Outside diameter 0.5% of the specified outside diameter based on circumferential measurement.
 Out-of-roundness: Difference between major and minor diameters not to exceed 1% of nominal diameter.

FIGURE 6.3

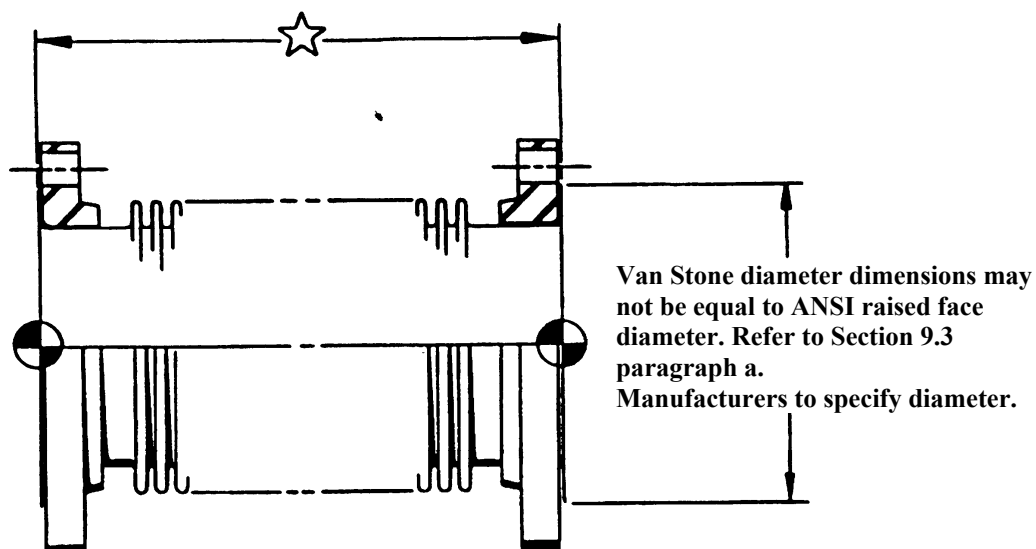


FIGURE 6.4

Flanges must be installed so that bolt holes straddle a common centerline within 1/16 in. (1.5 mm).

NOTE: Good practice suggests that one mating flange in the piping system remain unwelded until the Expansion Joint has been located in position.

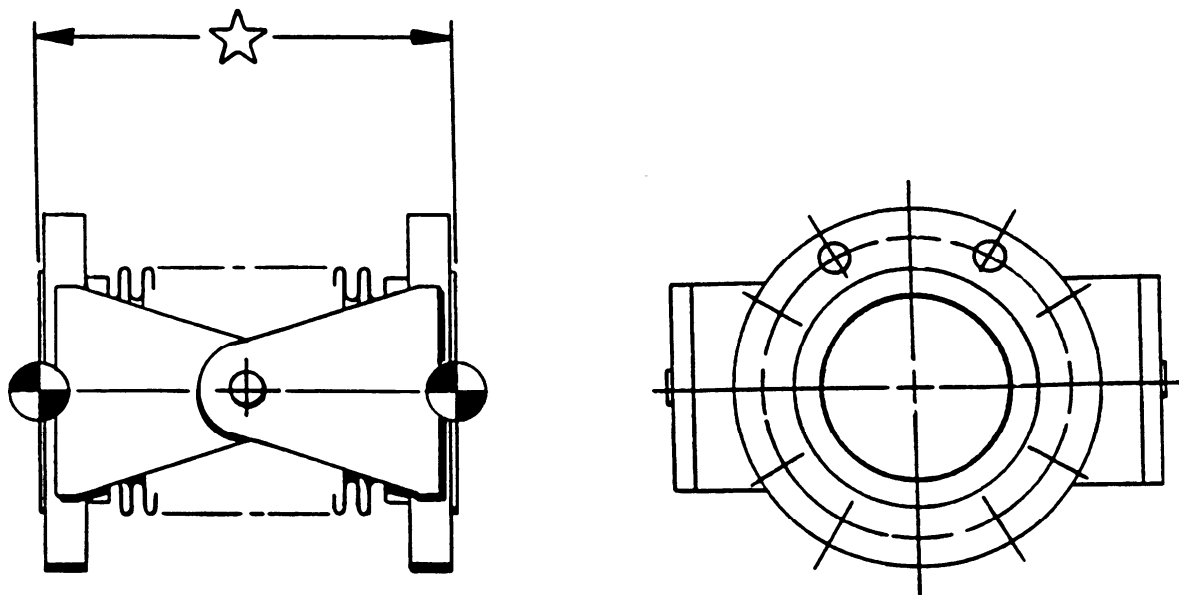
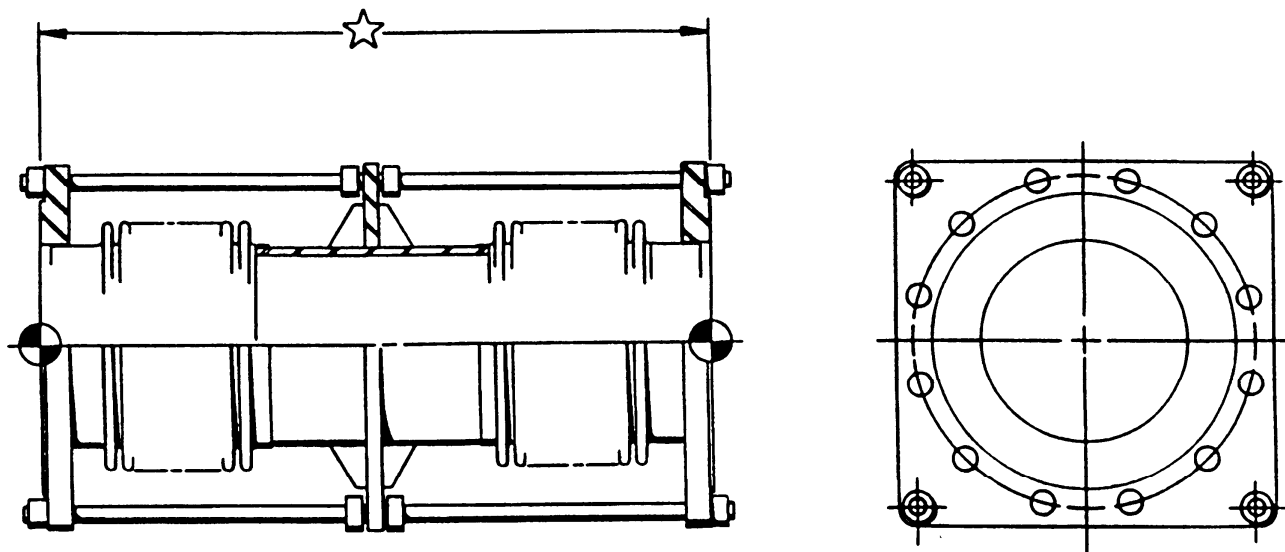


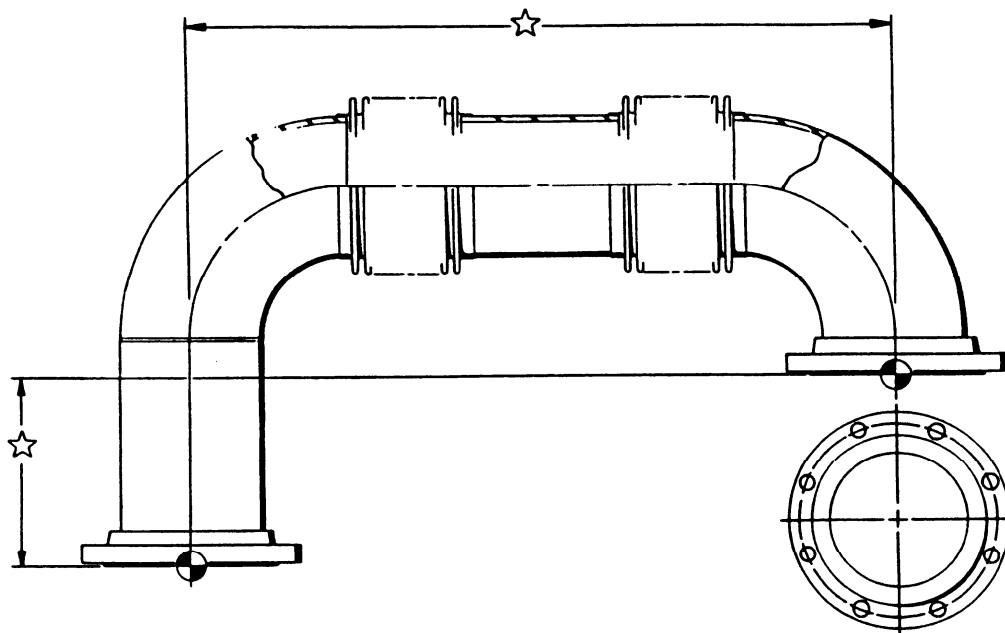
FIGURE 6.5



Flanges must be installed so that bolt holes straddle a common centerline within 1/16 in. (1.5 mm).

NOTE: Good practice suggests that one mating flange in the piping system remain unwelded until the Expansion Joint has been located in position.

FIGURE 6.6

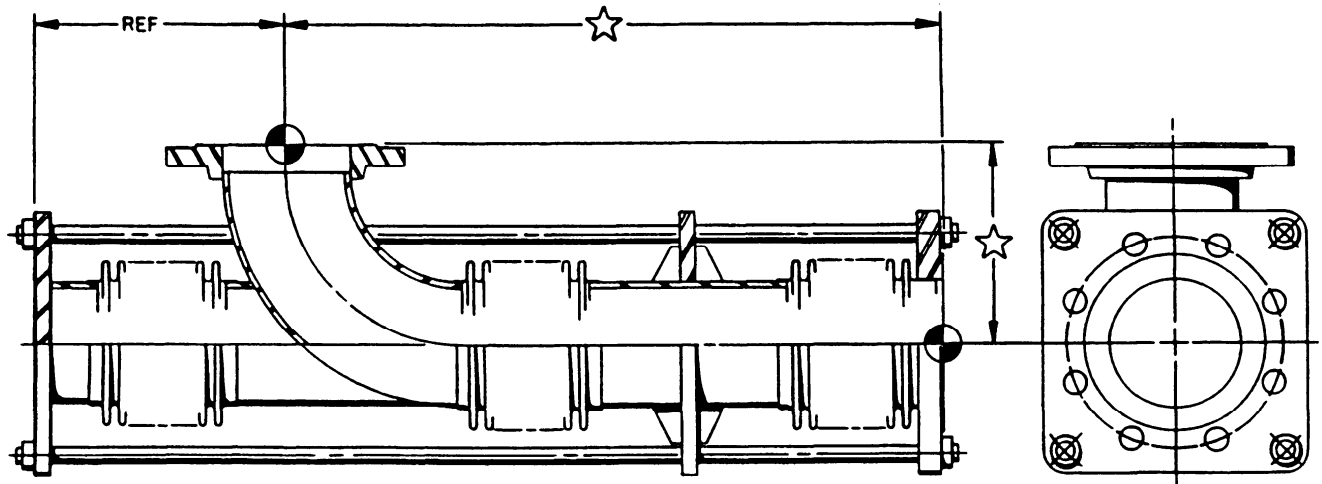


Flanges must be installed so that bolt holes straddle a common centerline within 1/16 in. (1.5 mm).

NOTE: Good practice suggests that one mating flange in the piping system remain unwelded until the Expansion Joint has been located in position.

FIGURE 6.7

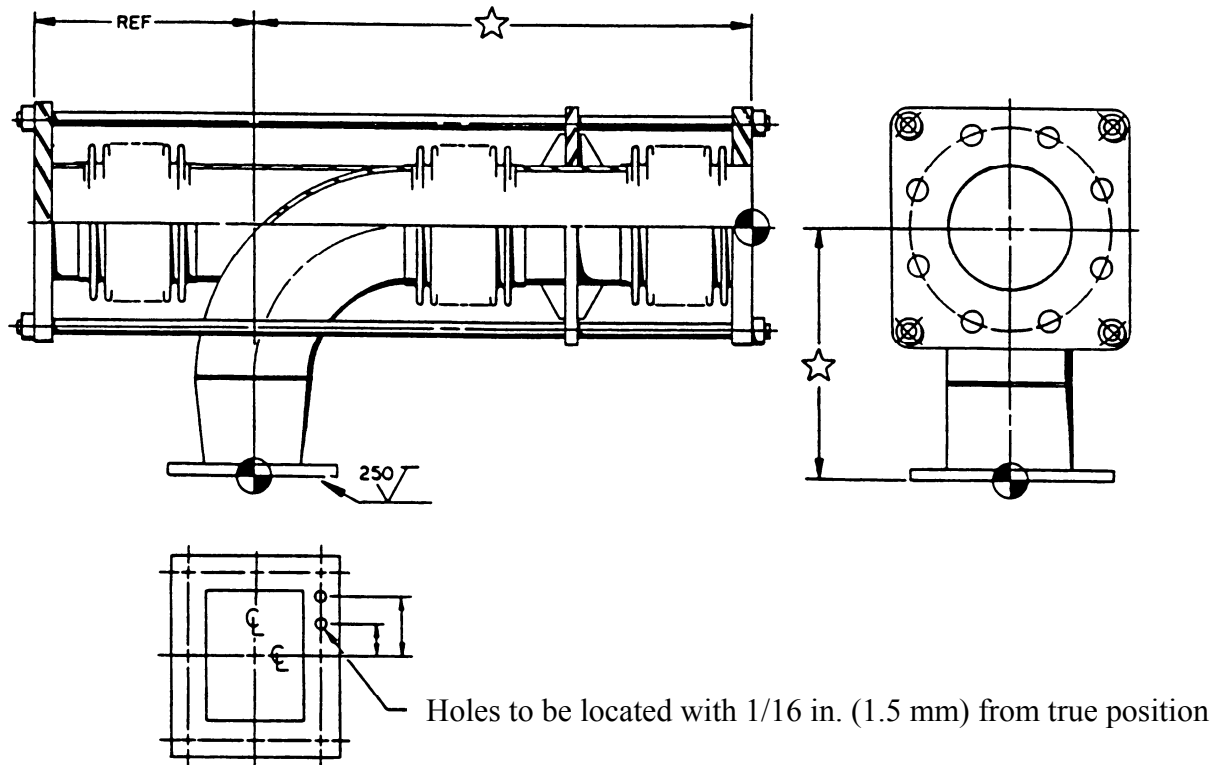
STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION



Flanges must be installed so that bolt holes straddle a common centerline within 1/16 in. (1.5 mm).

NOTE: Good practice suggests that one mating flange in the piping system remain unwelded until the Expansion Joint has been located in position.

FIGURE 6.8



Flange face at turbine connection to be flat within 1/16 in. (1.5 mm)

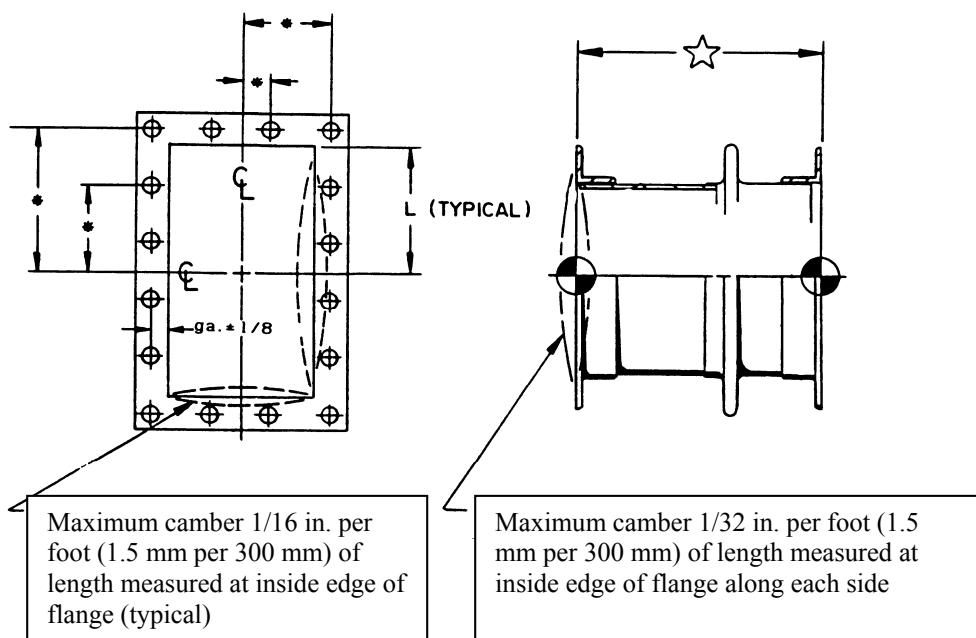
Flanges must be installed so that bolt holes straddle a common centerline within 1/16 in. (1.5 mm).

NOTE: Good practice suggests that one mating flange in the piping system remain unwelded until the Expansion Joint has been located in position.

NOTE: Design of the duct must provide for field fit-up connection to allow proper alignment of the Expansion Joint and duct, without producing unanticipated loadings in the system. Closer tolerances than those indicated shall be subject to agreement between the purchaser and Expansion Joint manufacturer.

FLANGED EXPANSION JOINT WITH MACHINED PLATE FLANGES
For Turbine Type Application Including Boiler Feed
Pump Turbine Exhaust

FIGURE 6.9



*Holes to be located within 1/8 in. (3 mm) of true position for L up to 12 ft (4 m) and within 3/16 in. (5 mm) of true position for L greater than 12 ft (4 m).

NOTE: Options for providing true hole locations:

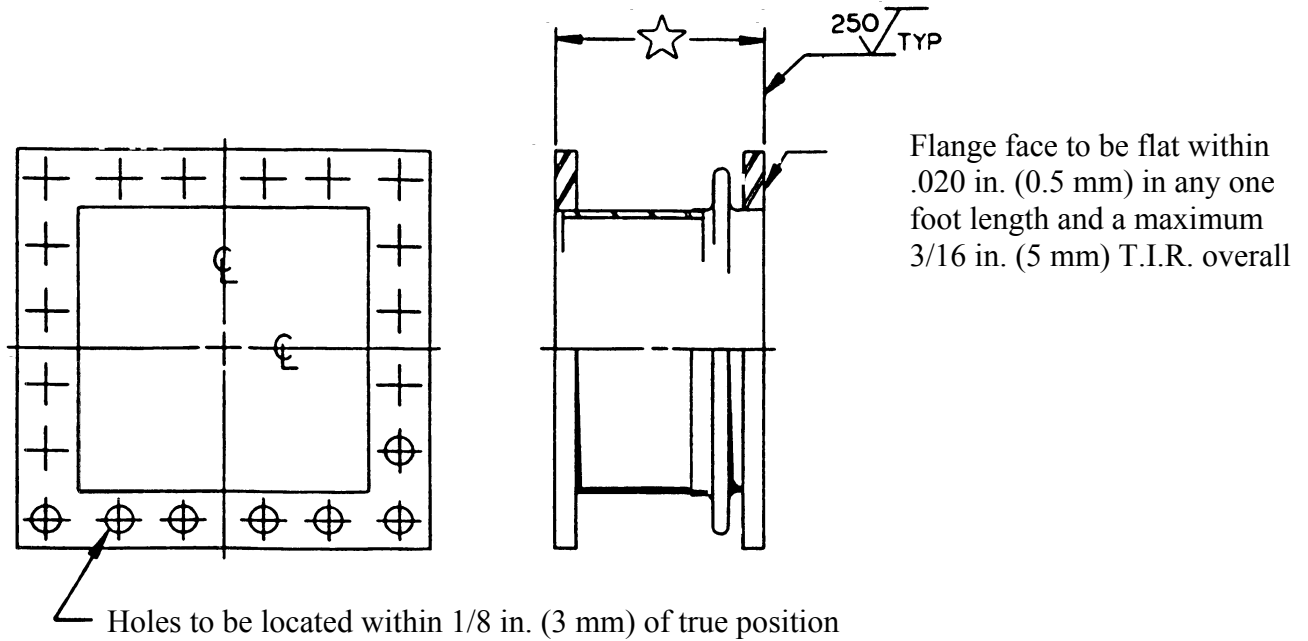
1. Purchaser may provide manufacturer with template having the desired hole size and pattern.
2. Purchaser may request blank flange or flanges with drilling to be made by constructor at installation.
3. Expansion Joint manufacturer may provide loose mating flanges.

Closer tolerances than those indicated shall be subject to agreement between the purchaser and Expansion Joint manufacturer.

RECTANGULAR EXPANSION JOINT

With Angle Type Flanges or 1/2 in. (13 mm) Maximum Thickness Plate Flanges
(All flange faces are mill finish)

FIGURE 6.10



Closer tolerances than those indicated shall be subject to agreement between the purchaser and Expansion Joint manufacturer.

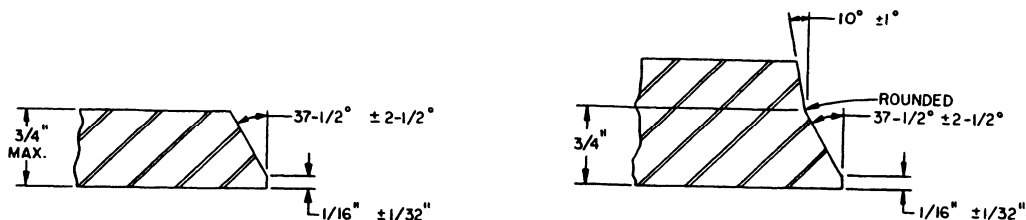
NOTE: Options for providing true hole locations:

1. Purchaser may provide manufacturer with template having the desired hole size and pattern.
2. Purchaser may request blank flange or flanges with drilling to be made by constructor at installation.
3. Expansion Joint manufacturer may provide loose mating flanges.

RECTANGULAR EXPANSION JOINT
With Plate Type Flanges Having Machined Faces

FIGURE 6.11

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION



End preparation for pipe wall thickness 3/16 in. to 3/4 in.

End preparation for pipe wall thickness above 3/4 in.

Note: Dimensions of weld end preparations produced by means other than machining (torch cut, hand ground, etc.) are nominal only.

MACHINE BUTT WELDING END PREPARATION (Ref. ANSI B16.25)

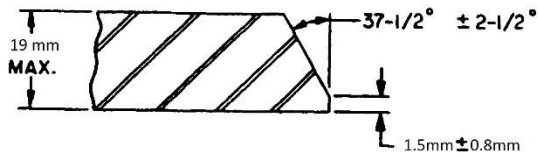
FIGURE 6.12

BELLOWS DIMENSION (in.)	MANUFACTURING TOLERANCE (in.)
Convolution Pitch (q)	
$\leq \frac{1}{2}$	$\pm \frac{1}{16}$
$> \frac{1}{2}$ to 1	$\pm \frac{1}{8}$
> 1 to $1\frac{1}{2}$	$\pm \frac{3}{16}$
$> 1\frac{1}{2}$ to 2	$\pm \frac{1}{4}$
> 2	$\pm \frac{5}{16}$
Convolution Height (w)	
$\leq \frac{1}{2}$	$\pm \frac{1}{32}$
$> \frac{1}{2}$ to 1	$\pm \frac{1}{16}$
> 1 to $1\frac{1}{2}$	$\pm \frac{3}{32}$
$> 1\frac{1}{2}$ to 2	$\pm \frac{1}{8}$
> 2 to $2\frac{1}{2}$	$\pm \frac{5}{32}$
$> 2\frac{1}{2}$ to 3	$\pm \frac{3}{16}$
> 3 to $3\frac{1}{2}$	$\pm \frac{7}{32}$
$> 3\frac{1}{2}$ to 4	$\pm \frac{1}{4}$
> 4	$\pm \frac{9}{32}$
Convolution Inside Diameter (Db)	
$\leq 8 \frac{5}{8}$	$\pm \frac{1}{16}$
$> 8 \frac{5}{8}$ to 24	$\pm \frac{1}{8}$
> 24 to 48	$\pm \frac{3}{16}$
> 48 to 60	$\pm \frac{1}{4}$
> 60	$\pm \frac{5}{16}$

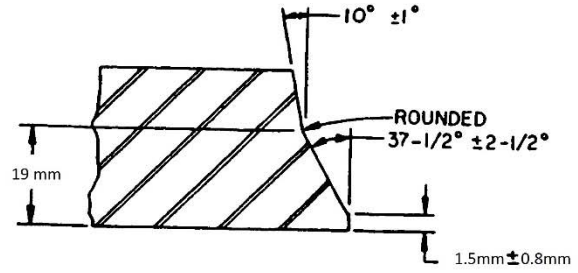
UNREINFORCED AND REINFORCED BELLOWS MANUFACTURING TOLERANCES

FIGURE 6.13

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION



End preparation for pipe wall thickness 5 mm to 19 mm



End preparation for pipe wall thickness above 19 mm

Note: Dimensions of weld end preparations produced by means other than machining (torch cut, hand ground, etc.) are nominal only.

MACHINE BUTT WELDING END PREPARATION (Ref. ANSI B16.25)

FIGURE 6.12M

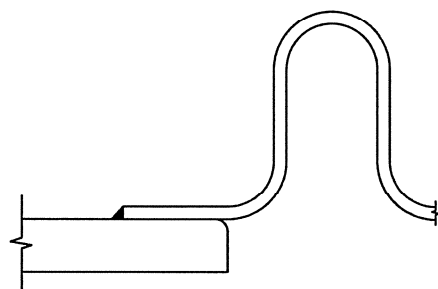
BELLOWS DIMENSION (mm)	MANUFACTURING TOLERANCE (mm)
Convolution Pitch (q)	
<= 13	± 1.5
> 13 to 25	± 3
> 25 to 38	± 5
> 38 to 50	± 6
> 50	± 8
Convolution Height (w)	
<= 13	± 1
> 13 to 25	± 1.5
> 25 to 38	± 2.5
> 38 to 50	± 3
> 50 to 64	± 4
> 64 to 75	± 5
> 75 to 89	± 5.5
> 89 to 100	± 6
> 100	± 7
Convolution Inside Diameter (Db)	
<= 220	± 1.5
> 200 to 600	± 3
> 600 to 1200	± 5
> 1200 to 1500	± 6
> 1500	± 8

UNREINFORCED AND REINFORCED BELLOWS MANUFACTURING TOLERANCES

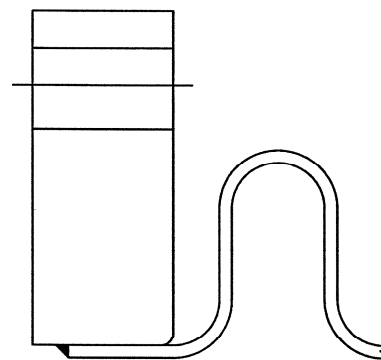
FIGURE 6.13M

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION

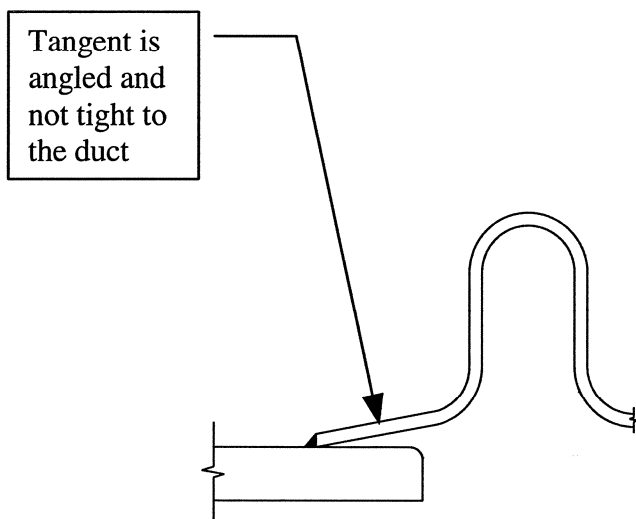
It is important that the fit of the bellows tangent be tight to the duct, flange, or other method of end attachment being used. Figure 6.14 shows two examples with the bellows tangent attached using a preferred method and a non-preferred method. **Hammering of the bellows tangent to achieve the non-preferred attachment is not acceptable.**



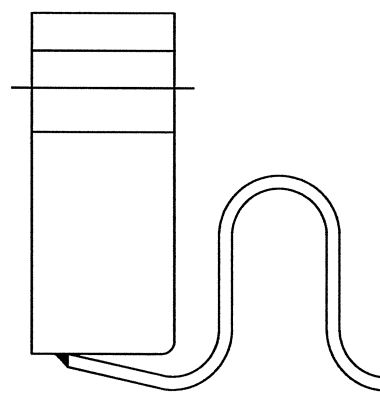
Weld end with preferred tangent attachment.



Flanged end with preferred tangent attachment.



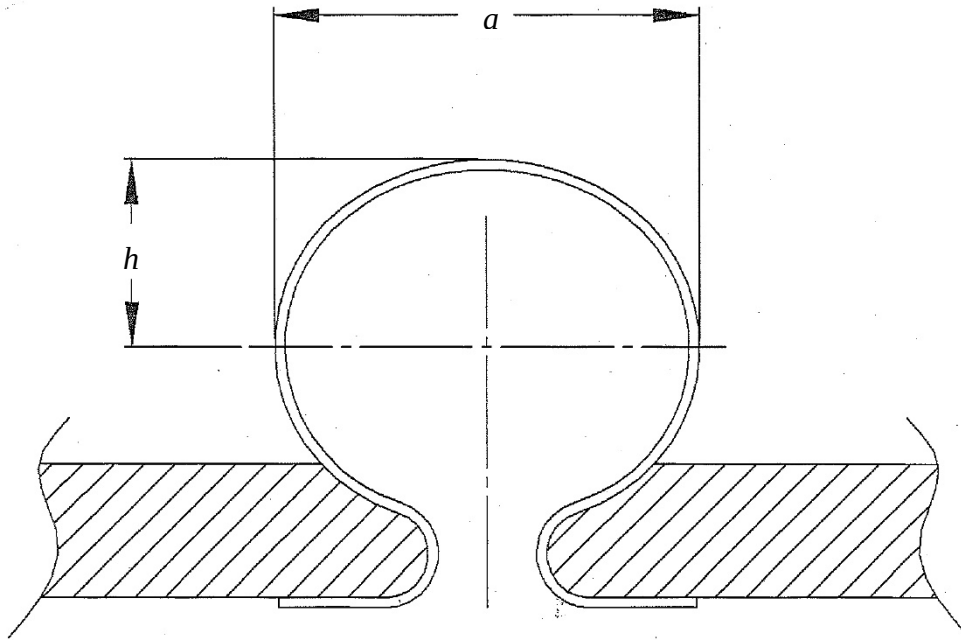
Weld end with acceptable but not preferred tangent



Flange with acceptable but not preferred tangent

TANGENT ATTACHMENTS

FIGURE 6.14

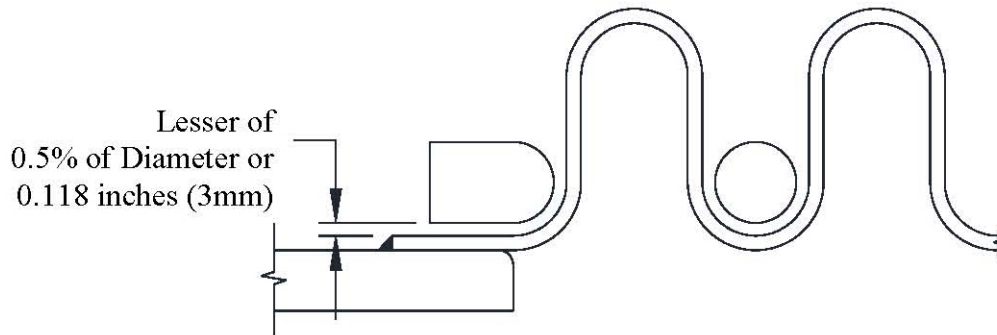


$$0.8 \leq \frac{a}{2h} \leq 1.2$$

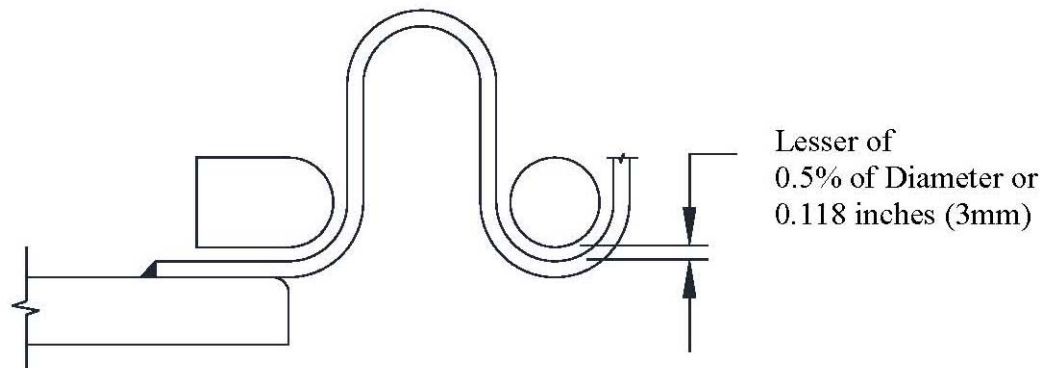
Toroidal Bellows Manufacturing Tolerances

Figure 6.15

It is important that the tangent reinforcement member, equalizing and reinforcing rings be tight to the bellows tangent or the root of the convolution. There is an acceptable radial gap which is the lesser of 0.5% of the diameter or 0.118 inches (3mm). Attention must be paid to dissimilar material growth rates of the bellows, band and reinforcing ring. If the band or reinforcing ring grow more than the bellows, they may become too loose and if they grow less than the bellows, they may become too tight.



Tangent Reinforcement



Equalizing or Reinforcing Ring

FIGURE 6.16

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SECTION 7 – EXAMINATION AND TESTING

To assure a purchaser that the product has been properly designed and manufactured requires some method of examination and/or testing of the product. It is not the intention of these standards to give detailed procedures for performing any examination or test, but rather to give a general description of some examinations and tests used to evaluate bellows Expansion Joints.

Any of the following examinations/tests may be performed on Expansion Joints when specified. It is primarily the responsibility of the purchaser to specify which methods will be required and the acceptance criteria. Unless otherwise specified, inspection methods, acceptance criteria and inspector qualification should be in accordance with the latest edition of the ASME/ANSI Piping Codes and the ASME Boiler and Pressure Vessel Codes.

7.1 NON-DESTRUCTIVE EXAMINATION

7.1.1 RADIOGRAPHIC EXAMINATION

Radiographic examination is based on the principle that extremely high frequency light waves, usually x-ray or from a radioactive source such as Cobalt 60, will penetrate solid materials and, when projected onto a photosensitive film, will reveal voids, areas of discontinuity, and lack of homogeneity. This examination is widely used in evaluating the soundness of welds and in general, is limited to evaluating butt welds of parts of substantially the same thickness and material. In the case of bellows, this is normally limited to the evaluation of longitudinal seam welds before forming.

Unless required by the purchaser, radiographic examination of the longitudinal seam of a bellows need not be specified. Examination of the longitudinal seam can be accomplished by some other means, such as liquid penetrant examination. If a radiographic examination is required on the longitudinal seam of a bellows then it should be performed before the bellows is convoluted. After the forming operation, it is usually not possible for the source or the film to be placed to yield a meaningful radiograph.

Radiographic examination of the bellows attachment weld should not be specified. Interpretation of such radiographs is impractical due to the weldment geometry, differences in thickness and penetrability. In view of the above, and recognition of the attachment weld as a seal weld, non-destructive examination of this weld should be accomplished by some other means such as liquid penetrant examination.

7.1.2 LIQUID PENETRANT EXAMINATION

Liquid penetrant examination consists of cleaning a surface, coating it with a dye, wiping the dye off and coating the surface with a developer which after sufficient time will draw the dye from the cracks, pin holes, and make them apparent to the observer. Liquid penetrant examination is limited in scope to detecting surface indications such as fine hairline cracks, pin holes and weld roll-over. With the thin material used in bellows, the probability of any defect remaining subsurface is unlikely. This examination is frequently used in evaluating bellows welds. The bellows base material may also be inspected by this method but shall be performed prior to convolution forming. The developer used in this procedure acts as a blotter; therefore, when rechecking a questionable indication it is absolutely essential to reclean that area and reapply dye and developer. Unless otherwise specified, examination procedures shall conform to the requirements of ASTM-E165.

7.1.3 FLUORESCENT PENETRANT EXAMINATION

Fluorescent penetrant examination is similar in purpose to the liquid penetrant examination but is accomplished by the use of a dye which contains a fluorescent material and developer. The parts being inspected are examined in subdued light under an ultraviolet light source. Parts must be thoroughly cleaned prior to testing. Scan the parts with the ultraviolet light before applying the fluorescent material since hydrocarbons, greases and oils, and lint may give misleading indications. Depending on the fluorescent material used in the execution of this examination, there are varying levels of sensitivity, and the purchaser must state the material to be used. Fluorescent penetrant examination is limited to determining the presence of surface defects. It would be a duplication to require both the liquid penetrant and fluorescent penetrant examination for the same components of an Expansion Joint.

7.1.4 MAGNETIC PARTICLE EXAMINATION

Magnetic particle examination consists of coating a surface with finely powdered iron and establishing a magnetic field in the material being examined. The presence of discontinuities or irregularities in the magnetic field, as indicated by the lines of powdered iron, will indicate surface and also subsurface defects, cracks, slag inclusions, and lack of weld penetration. This examination is limited to magnetic material and will not indicate deep subsurface defects. Although generally used for examination of welds, it is possible to examine base material if there is reason to suspect material defects such as laminated plate.

7.1.5 ULTRASONIC EXAMINATION

Ultrasonic examination uses high frequency sound waves to detect flaws, and is useful in determining thickness, depth, and exact location of defects. Interpretation of indications in sections of sharply varying thickness is difficult. The examination is not limited to any group of materials.

7.1.6 HALOGEN LEAK EXAMINATION

Halogen leak examination utilizes a probe of suitable design which selectively indicates the presence of halogen gases. This examination is more sensitive than a hydrostatic test or air jet leak examination but since it is done at low pressures, it can only determine the presence of a leak and can not validate the structural integrity of the item being examined. A halogen leak examination must be performed in a suitable area since many gases common to manufacturing plants will give indications. This examination is helpful in not only determining the presence of a leak, but also in locating that leak. The acceptance criteria is failure to detect leakage in excess of that specified by the purchaser.

7.1.7 MASS SPECTROMETER EXAMINATION

Mass spectrometer examination is an extremely sensitive means of determining the presence of a leak. The gas used is helium. The examination is more sensitive than would be required for common commercial installations and is normally specified where Expansion Joints are for lethal gas service, explosive environment service, or high vacuum service.

Mass spectrometer examinations are capable of detecting leakage rates to 10^{-10} standard cubic centimeters per second. The Expansion Joint may be examined for the sum total of leakage or with a probe to locate individual leaks. When the probe method is employed, sensitivity is limited to between 10^{-6} and 10^{-8} standard cubic centimeters per second. The acceptance criterion is the absence of leakage rates in excess of that specified by the purchaser.

7.1.8 AIR JET LEAK EXAMINATION

Air jet leak examination utilizes compressed air is directed through a nozzle on to a small area between two welded parts. A leak detector solution is applied on the opposite side of the welded connection which will bubble up if the compressed air is able to pass through the weld. This examination is useful on low pressure Expansion Joint bellows end connection welds where other forms of examination and testing are not practical.

7.2 NON-DESTRUCTIVE TESTING

7.2.1 PRESSURE TESTING

Hydrostatic and pneumatic are two types of pressure tests that can be performed on an Expansion Joint. Hydrostatic pressure testing involves filling the Expansion Joint with a liquid, usually potable water, while pneumatic pressure testing involves filling the Expansion Joint with air or other gas. After the Expansion Joint is filled it can then be pressurized to the required test pressure. Pneumatic pressure testing is hazardous and it is recommended that special precautions be taken.

Normally, the required test pressure is a multiple of the design pressure. Expansion Joints placed in high temperature service may require the pressure test be performed at an adjusted pressure. It is imperative that the test pressure does not produce a membrane stress in excess of yield strength or cause permanent deformation or instability (squirm) of the bellows at test temperature. It may be necessary to reduce the test pressure adjusted for temperature, to the maximum pressure that will not exceed yield or cause instability.

An Expansion Joint should not be subjected to a test in the field at a higher pressure than was used in the manufacturer's shop without the manufacturer's knowledge. All anchors and guides must be installed (See Section 2.10) and shipping devices removed prior to such testing. In the case of large Expansion Joints, additional supports may be required to support the weight of the water used during hydrostatic testing.

A pressure test is not only useful for detecting leaks but is also a way to test for bellows squirm, meridional yield and rupture. (See Sections 7.3.2 and 7.3.3)

7.3 DESTRUCTIVE TESTING

Destructive testing will render the Expansion Joint or at least the bellows unsuitable for installation in an operating system. These tests then must be performed on a prototype Expansion Joint. A prototype Expansion Joint is defined as one having the same pressure and temperature rating as production models, identical diameter, height, pitch, and general shape of the convolution, the thickness and type of bellows materials, bellows reinforcement, method of manufacture, and maximum movement per convolution. Since it is more practical to test an Expansion Joint under axial movement rather than combined movement it is acceptable to use equivalent maximum axial movement as calculated in Section 4.

7.3.1 FATIGUE LIFE TESTING

Fatigue life testing is a verification of the ability of a bellows to withstand a given number of flexing cycles. It is recommended that the bellows subjected to fatigue life testing be identified by the parameters in Table I. With all other shape factors remaining constant, cycle life will generally increase with diameter; for prototype testing, it may be acceptable to cycle test the smallest size Expansion Joint being furnished for a given series for identical service condition. It is also acceptable to cycle test at room temperature any Expansion Joint which will be furnished for operating temperatures up to the active creep range. For Expansion Joints operating above this range, consideration should be given to testing at elevated temperatures.

Fatigue testing may be performed at constant pressure or at varying pressure. This latter condition more closely approximates the service to which the Expansion Joint will be subjected.

When the system designer specifies the minimum number of cycles, this number should be consistent with the life of the system in which the Expansion Joint is to be installed. Excessive cycle life requirements will not necessarily ensure desired results.

7.3.2 SQUIRM TESTING

The objective of a squirm test is to determine the internal pressure which will cause a bellows to become unstable. Squirm is defined on the basis of a change in pitch of the bellows convolutions under internal pressure. Identification of the bellows should be established using the parameters in Table I. The following is a recommended test procedure:

The Expansion Joint should be placed in a suitable fixture with the bellows fixed in the straight position which will effectively seal the ends during pressurization and most importantly, will prevent any movement of the ends during testing.

The bellows may be tested with its axis in either the horizontal or vertical position. The testing medium shall be water for purposes of safety. If the expansion joint's operating condition is to be in the horizontal position and the bellows element is of an extremely flexible nature then a test in the horizontal position may be preferred. The convolutions of the bellows during testing should not be restrained by external means, unless such restraints form an integral part of the final assembly.

The test specimen should be instrumented, such that the resultant lateral deflection of the center one or two convolutions, and the change in pitch of all of the convolutions, can be accurately determined. The former may be obtained by vectorially adding the deflections measured by two mutually perpendicular dial gauges. Pitch measurements should be made in the plane of maximum convolution deflection.

Pressurize the specimen in steps without relieving the pressure between steps. Each interval should not exceed 10% of the final anticipated instability pressure. Smaller intervals are preferred as the pressure increases.

Instability of axially aligned bellows is generally characterized by a sudden acceleration of either the change in resultant lateral deflection and/or the change in convolution pitch. However, in the case where bellows are tested in the laterally offset or rotated position, no true stability condition may appear. Instead, movement of the convolutions will occur due to lateral pressure component being superimposed on the applied deflection. (See Section 4.12.1.6)

Squirm shall be considered to have occurred if under internal pressure an initially symmetrical bellows deforms resulting in lack of parallelism and/or uneven spacing of adjacent convolutions at any point on the circumference. This deformation shall be construed as unacceptable squirm when the convolution pitch under internal pressure to the convolution pitch before application of pressure exceeds 1.15 for unreinforced and 1.20 for reinforced bellows.

7.3.3 MERIDIONAL YIELD-RUPTURE TESTING

The objective of a meridional yield-rupture test is to determine the internal pressure which will cause yielding and rupture of a bellows. An accurate evaluation of the yield pressure is quite important since it is this value, rather than rupture, which usually provides the limiting criteria for establishing suitable operating pressures. The test specimen should have a minimum of three convolutions to minimize the effects of the end attachments. Identification of the bellows should be established using the parameters in Table I.

Place the Expansion Joint in any suitable fixture with the bellows fixed in the straight position which will effectively seal the ends during pressurization, and most importantly, will prevent any movement of the ends during testing. The fixture must also safely restrain the bellows when rupture occurs. The test medium should be limited to water as a safety precaution.

Pressurize the specimen in steps, returning to zero pressure after each step, up to at least twice the yield pressure. Thereafter, the specimen may be pressurized continuously until rupture occurs. The initial pressure intervals should not exceed 10% of the anticipated yield pressure. A constant holding time at pressure should be established for each step throughout the yield point determination.

As a minimum, the width or space between each convolution at the mean diameter, should be measured and recorded before and after each pressure step.

Instrumentation, such as a pressure-time recorder, strain gauges, etc., can also provide valuable information.

Both meridional yielding (bulging of the flat sides of the convolutions), and circumferential yielding of the bellows root diameter will be experienced in a test of this type. Although a plot of permanent deformation versus pressure will resemble that of a simple tensile test stress-strain curve, a well defined yield point does not usually appear. Thus, close visual observation of the test and a careful analysis of the data is necessary to accurately determine the yield pressure. Beyond the yield point, gross distortion, root collapse, and ultimate failure will occur.

TABLE I

RECOMMENDED IDENTIFICATION DATA REQUIRED FOR
BELLOWS SUBJECTED TO DESTRUCTIVE TESTS

	Fatigue Life	Squirm	Yield- Rupture
Inside Diameter	R	R	R
Bellows Pitch	R	R	R
Convolution Height	R	R	R
Convolution Width*	R	R	R
Bellows Material Thickness	R	R	R
Number of Plies	R	R	R
Material of Ply (Plies)	R	R	R
Reinforcing Ring Dimensions	D	D	R
Material of Reinforcing Rings	NR	NR	NR
Pressure	R	NR	NR
Pressure Range	R	R	R
Movement	R	R	R
Number of Convolutions	R	NR	NR
Number of Cycles to Failure	R	R	R
Post Form Heat Treat	R	R	R
Failure Definition	R	R	R
Test Temperature	R	R	R

R = REQUIRED, D = DESIRABLE, NR = NOT REQUIRED

*Distance between convolution sidewalls measured on outside for internal pressure tests or inside for external pressure tests.

SECTION 8 – SHIPPING AND INSTALLATION

Responsible manufacturers of Expansion Joints take every reasonable precaution, through stringent purchasing specifications, receiving inspection, reliable design standards, manufacturing methods, quality control procedures, and packaging specifications, to assure the user of the reliability he requires. The installer and the user have a responsibility with the manufacturer to handle, store, install, and apply these Expansion Joints in a way which will not impair the quality built into them. (See Section 3)

Some conditions of outside storage may be detrimental and, where possible, should be avoided. Where this cannot be accomplished, the Expansion Joint manufacturer should be so advised either through the specifications or purchase contract. Preferably, storage should be in a clean and dry area. Variations in weather conditions should not prove detrimental to bellows-type Expansion Joints. Care must be exercised to prevent mechanical damage such as that caused by stacking, bumping, or dropping.

Certain industrial and natural atmospheres can be detrimental to some bellows materials. If Expansion Joints are to be stored or installed in such atmospheric environments, the system designer should select materials compatible with these environments.

8.1 SHIPPING TAGS

Expansion Joints are shipped with tags which furnish the installer with instructions covering the installation of the particular Expansion Joint. These shipping tags should be left on the Expansion Joint until installation. If the project coordinator wishes duplicate instructions so he may properly plan his installation, these will be furnished on request.

8.2 SHIPPING DEVICES

All manufacturers should provide some means for maintaining the proper face-to-face dimension of an Expansion Joint during shipment and installation. Sometimes these consist of overall bars or angles welded to the flanges or nipples at the extremities of the Expansion Joint. At other times, they consist of washers bolted between equalizing rings, or they may take the form of wooden blocks between equalizing rings. Although such devices are adequate protection for the Expansion Joint during shipment, storage, and installation, they will not be sufficiently strong to protect the Expansion Joint or piping system if the line is hydrostatically tested prior to the installation of anchors and guides.

Changes in ambient temperature of a newly installed pipe line can, in long runs of pipe, result in considerable thermal expansion or contraction. Hydrostatic testing, particularly in warm weather, will cause an appreciable drop in pipe line temperatures. It is obvious from the foregoing that an Expansion Joint may be subject to considerable flexing before the system is placed in operation. Shipping devices must be removed before an Expansion Joint can function properly and must be removed before hydrostatic testing of the pipeline.

Shipping devices which must be removed from Expansion Joints manufactured by members of the Expansion Joint Manufacturers' Association, Inc., are usually painted yellow, or otherwise distinctively marked as an additional aid to the installers.

8.3 INSTALLATION

It is important that Expansion Joints be installed at the proper lengths as recommended by the manufacturer. They should never be extended or compressed to make up deficiencies in pipe length, or offset to accommodate piping which is not properly aligned unless such installation's tolerances have been specified by the system designer and anticipated by the Expansion Joint manufacturer. Do not neglect pre-compression or pre-extension of the Expansion Joint where it is required or as designated by the manufacturer. Generally, such instructions are included on the shipping tags and additional information is available in Section 2.

All Expansion Joints provided with internal sleeves should be provided with flow arrows or other suitable means of assisting the installer in properly orienting the Expansion Joint to flow direction. Correct installation of Expansion Joints with internal sleeves is most important and should be checked by the installer. (See Section 4.10)

In order to insure the proper functioning of any Expansion Joint, it is highly important that all pipelines in which the Expansion Joints are located be suitably anchored, guided, and supported. (See Sections 2.2 through 2.10)

Remember, a bellows is designed to absorb motion by flexing. The bellows is sufficiently thick to withstand the design pressure, but also sufficiently thin to withstand its cyclic movement. Optimum design will always require a bellows of thinner materials than virtually every other component of the piping system in which it is installed. The installer must recognize this and take all necessary measures to protect the bellows during installation. Avoid denting, weld spatter, arc strikes, or the possibility of allowing foreign matter to interfere with the proper flexing of the bellows. With reasonable care during storage, handling, and installation, the user will be assured of the reliability designed and built into the Expansion Joint.

8.4 GASKETS

When removable flanged sleeves are inserted in the Expansion Joint, an extra gasket is required between the face of the Expansion Joint and the back face of the flanged sleeve, i.e., two gaskets per Expansion Joint ordinarily, three gaskets if one flanged sleeve is used per Expansion Joint and four gaskets if a pair of telescoping flanged sleeves are used. Caution should be used with graphite impregnated gaskets in contact with stainless steel facings or sleeves at high temperature.

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8.5 RECOMMENDED INSTALLATION INSTRUCTIONS

Metal Bellows Expansion Joints have been designed to absorb a specified amount of movement by flexing of the thin-gauge convolutions. If proper care is not taken during installation, it may reduce the cycle life and the pressure capacity of the expansion joints which could result in an early failure of the bellows element or damage the piping system.

The following recommendations are included to avoid the most common errors that occur during installation. When in doubt about an installation procedure, contact the manufacturer for clarification before attempting to install the Expansion Joint.

DO'S	DON'T
Inspect for damage during shipment, i.e., dents, broken hardware, water marks on carton, etc. Store in clean dry area where it will not be exposed to heavy traffic or damaging environment. Use only designated lifting lugs. Make the piping systems fit the expansion joint. By stretching, compressing, or offsetting the joint to fit the piping, it may be overstressed when the system is in service. It is good practice to leave one flange loose until the expansion joint has been fitted into position. Make necessary adjustment of loose flange before welding. Install joint with arrow pointing in the direction of flow. Install single Van Stone liners pointing in the direction of flow. Be sure to install a gasket between the liner and Van Stone flange as well as between the mating flange and liner. With telescoping Van Stone liners, install the smallest I.D. liner pointing in the direction of flow. Remove all shipping devices after the installation is complete and before any pressure test of the fully installed system Remove any foreign material that may have become lodged between the convolutions. Refer to EJMA Standards for proper guide spacing and anchor recommendations.	Do not drop or strike carton. Do not remove shipping bars until installation is complete. Do not remove any moisture-absorbing dessicant bags or protective coatings until ready for installation. Do not use hanger lugs as lifting lugs without approval of manufacturer Do not use chains or any lifting device directly on the bellows or bellows cover. Do not allow weld splatter to hit unprotected bellows. Protect with wet chloride-free insulation. Do not use cleaning agents that contain chlorides Do not use steel wool or wire brushes on bellows. Do not force-rotate one end of an expansion joint for alignment of bolt holes. Ordinary bellows are not capable of absorbing torque. Do not hydrostatic pressure test or evacuate the system before installation of all guides and anchors. Pipe hangers are not adequate guides. Do not exceed a pressure test of 1 1/2 times the rated working pressure of the expansion joint. Do not use shipping bars to retain thrust if tested prior to installation.

The manufacturer's warranty may be void if improper installation procedures have been used.

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SECTION 9 – FEATURES, ACCESSORIES, AND MATERIALS

9.1 MULTI-PLY BELLOWS

A multi-ply bellows can be used in many applications. It is important to understand the functional characteristics of each type of construction. These Standards apply to bellows with no more than five plies.

9.1.1 MULTI-PLY CONSTRUCTION WITH THE SAME TOTAL THICKNESS AS A SINGLE PLY CONSTRUCTION

9.1.1.1 PRESSURE CAPACITY

The circumferential membrane (S_2) and meridional membrane (S_3) pressure stresses are unaffected since the total bellows thickness is the same as a single ply construction. The meridional stress due to pressure (S_4) will be higher for the multi-ply construction due to the thinner material per ply.

9.1.1.2 FATIGUE LIFE

An increase in fatigue life over that of a single ply construction will usually result since the meridional deflection stresses (S_5) and (S_6) are reduced due to the thinner material per ply.

9.1.1.3 SPRING FORCES

A decrease in the spring force will result since the spring rate will be lower due to the thinner material per ply.

9.1.1.4 BELLOWS STABILITY

Column stability is reduced due to the thinner material per ply. In-plane stability is also reduced.

9.1.2 MULTI-PLY CONSTRUCTION WITH THE SAME THICKNESS FOR EACH PLY AS A SINGLE PLY CONSTRUCTION

9.1.2.1 PRESSURE CAPACITY

The pressure capacity of the bellows is higher than a single ply construction. The circumferential membrane (S_2) and meridional membrane (S_3) pressure stresses are lower since the total bellows thickness is greater. The meridional bending stress due to pressure (S_4) will be lower for the multi-ply construction.

9.1.2.2 FATIGUE LIFE

The effect on fatigue life over that of a single ply construction will be minimal.

9.1.2.3 SPRING FORCES

An increase in the spring force will result since the spring rate will be higher due to the greater total material thickness.

9.1.2.4 BELLOWS STABILITY

In-plane and column stability are increased due to the greater total material thickness.

9.1.3 MULTI-PLY CONSTRUCTION WITH GREATER THICKNESS FOR EACH PLY THAN FOR SINGLE PLY CONSTRUCTION

9.1.3.1 PRESSURE CAPACITY

The pressure capacity of the bellows is higher than a single ply construction. The circumferential membrane (S_2) and meridional membrane (S_3) pressure stresses are lower since the total bellows thickness is greater. The meridional bending stress due to pressure (S_4) will be lower for the multi-ply construction.

9.1.3.2 FATIGUE LIFE

A decrease in fatigue life over that of a single ply construction will result since the meridional deflection stresses (S_5) and (S_6) are increased due to the thicker material per ply.

9.1.3.3 SPRING FORCES

An increase in the spring force will result since the spring rate will be higher due to the greater total material thickness.

9.1.3.4 BELLOWS STABILITY

In-plane and column stability are increased due to the greater total material thickness.

BELLOWS DESIGN CRITERIA	MULTIPLE PLY CONSTRUCTION FUNCTIONAL CHARACTERISTICS				
	tt=sp 9.1.1	tt/n = sp 9.1.2	tt/n > sp 9.1.3	tt > sp tt/n < sp	
Circumferential Membrane (S_2)	Same	Decreases	Decreases	Decreases	tt = total thickness sp = single ply construction n = number of plies
Meridional Bending (S_4)	Increases	Decreases	Decreases	Usually Decreases	
Fatigue Life	Usually Increases	Nominal Change	Decreases	Increases	
Spring Force	Decreases	Increases	Increases	Usually Increases	
In-plane Stability	Decreases	Increases	Increases	Usually Increases	
Column Stability	Decreases	Increases	Increases	Usually Increases	

FIGURE 9.1

Multi-Ply Response when Compared to an Equivalently Designed Single Ply Bellows

9.1.4 MULTIPLE MATERIAL USAGE

Corrosion considerations may indicate the desirability of different materials for the inner and outer bellows plies to suit the internal / external environment. In this manner the primary ply specified to resist corrosion can be supplemented by less costly additional plies.

9.1.5 REDUNDANT PLY CONSTRUCTION WITH THE SAME THICKNESS FOR EACH PLY AS A SINGLE PLY CONSTRUCTION

Redundant two ply bellows are used when it is desired to continue normal system operation if one ply should fail, until such time as a suitable replacement can be made.

9.1.5.1 PRESSURE CAPACITY

The pressure capacity of each bellows ply is the same as an equivalent single ply design since the redundant plies have been designed to withstand the system design pressure independently.

9.1.5.2 FATIGUE LIFE

The effect on fatigue life over that of a single ply construction will be nominal.

9.1.5.3 SPRING FORCES

An increase in the spring forces will result since the spring rate will be higher due to the greater total material thickness.

9.1.5.4 BELLOWS STABILITY

In-plane and column stability are increased due to the greater total material thickness.

9.1.5.5 MONITORED PLY BELLOWS

The annular space between plies can be monitored for leakage to detect a ply failure. This will serve as a warning of an impending problem, reducing the chances of a costly unscheduled shutdown.

9.2 TIE RODS, HINGES AND SIMILAR ACCESSORIES

In a piping system containing Expansion Joints, it is frequently impractical to use main anchors to absorb the pressure thrust or to transmit this force to the connected equipment. In such cases, the proper use of tied, hinged, or gimbal Expansion Joints can solve the problem. The use of such Expansion Joints requires that the tie rods, hinges or gimbals and their attachment to the piping be properly designed to absorb the imposed forces.

9.2.1 FORCES AND LOADS

The thrust absorbing members of an Expansion Joint are normally designed to restrain only the pressure thrust developed within the piping system and the force required to compress or extend the bellows due to thermal growth.

If other forces are to be considered in the Expansion Joint design, this fact, along with information regarding the magnitude and direction of these forces, must be provided to the Expansion Joint designer. The additional forces to be considered may include the following:

- a. Unsupported weight of connecting pipe and insulation between a pair of bellows.
- b. Weight of contained fluid under operation and/or test conditions.
- c. Wind, earthquake and/or impact loads.
- d. Torsion about the longitudinal axis.

The effects of temperature and flow conditions (transient and steady state) must be accounted for in conjunction with the above forces and loads.

9.2.2 METHODS OF ATTACHMENT

Tie rods, hinges or gimbals are attached to the pipe in two basic ways:

- a. By structures whose primary functions are to transmit the loads to the pipe.
- b. By direct attachment to pipe flanges in the piping run. In this method, the load is transmitted from the tie rods, hinges or gimbals to the connecting pipe through the flange bolts and mating flange.

9.2.3 DESIGN CONSIDERATIONS

9.2.3.1 TIE RODS, HINGES AND GIMBALS

The major design factors to be considered are:

a. TIE RODS Tensile and/or compressive forces due to pressure thrust and other longitudinally applied loads; the bending stresses resulting from connecting the tie rod to its attachment; the stress concentration effects in threaded areas. For general structural rigidity to withstand extraneous loads during handling and installation, it is recommended that minimum tie rod diameters as a function of the size of the expansion joint be in accordance with Figure 9.2.

b. HINGES Hinge plate tensile and/or compressive forces due to pressure thrust and other longitudinally applied loads; bending forces such as those resulting from weight loads or torsion applied about the longitudinal axis of the expansion joint; shear and bearing forces at the hinge pin hole. Evaluation of the shear, bearing and bending forces in the hinge pin itself is also required. The bending and shear effects in the hinge plates and pins are significantly different depending on whether the hinge design places the pins in single or double shear. A double shear arrangement is recommended for all but very low loads.

c. GIMBALS Bending and torsional effects in the gimbal ring due to pressure thrust and other longitudinally applied loads. Gimbal rings may be circular or square and may be evaluated using the concept of a ring under four point loading and torsionally unrestrained. The hinge plate and pin portions of the gimbal assembly, as well as the shear and bearing effects at the gimbal ring pin holes, may be evaluated similar to paragraph 9.2.3.1-b.

9.2.3.2 ATTACHMENTS TO PIPING

A variety of structural attachments may be used to connect tie rods and hinge plates to the piping portion of an expansion joint. These may be simple lugs, lugs with gussets or solid single or double plates (or rings) extending completely around the pipe circumference. See Figures 9.3 and 9.4 for typical examples. In all such arrangements the stress in the pipe must be evaluated as well as the stress in the structural member. In the case of lugs or lugs with gussets it may be necessary to evaluate local deflection of the pipe which could impose undesirable stresses in the bellows attachment weld and cylindrical tangent. In high temperature applications involving solid plate or ring structures, the effects due to differential thermal expansion should also be considered.

The published literature provides various methods for evaluating the structure and the pipe stresses individually rather than in combination. Also, most published work assumes that the loads on the pipe occur on infinitely long cylinders. This is not true when the structural attachment is located adjacent to the bellows and close to an open pipe end. It is, therefore, customary to perform the necessary evaluation by means of approximations based on published literature supplemented by empirical methods which experience has shown provide satisfactory results.

9.2.3.3 COMPONENT DESIGN STRESS LIMITS

Expansion joint load bearing component stress limits are required to comply with pressure vessel, piping, and structural codes and standards. The stress limits in Table II apply for load bearing component design.

TABLE II
Component Design Stress Limits

Component	Type of Stress	Stress Limit
Tie or Limit Rods (Pipe or Round Bar)	Tension	S
	Compression	S^*
Lugs (with or without Gussets)	Max. Membrane plus Bending	$K_s \times S$
	Max. Shear	$0.8 \times S$
	Average Shear	$0.6 \times S$
Hinge or Pantograph Linkage Pins	Max. Shear	$0.8 \times S$
	Average Shear	$0.6 \times S$
	Max. Membrane plus Bending	$K_s \times S$
	Average Bearing	$1.5 \times S$
Hinge and Clevis Plates (Single or Double Shear)	Tension or Compression	S
	Tension or Compression plus Max. Bending	$K_s \times S$
	Average Bearing (hole)	$1.5 \times S$
Gimbal Rings (Square or Round)	Max. Membrane plus Bending	$K_s \times S$
	Max. Shear (round only)	$0.8 \times S$
	Average Shear	$0.6 \times S$
	Average Bearing (hole)	$1.5 \times S$
Full Encirclement Rings (Fixed or Floating)	Max. Membrane plus Bending	$K_s \times S$
	Max. Shear	$0.8 \times S$
	Average Shear	$0.6 \times S$
Full Encirclement Plates (with or without Gussets)	Max. Membrane plus Bending	$K_s \times S$
	Max. Shear	$0.8 \times S$
	Average Shear	$0.6 \times S$
Trunnions (Round or Rectangular)	Max. Shear	$0.8 \times S$
	Average Shear	$0.6 \times S$
	Max. Membrane plus Bending	$K_s \times S$
Pipe Wall at Trunnions, Lugs, Gussets, Shear Pads, Shear Rings, etc.	General Membrane	S
	Local Membrane	$1.5 \times S$
	Local Membrane plus Bending	$3.0 \times S^{**}$
Pantograph Linkage	Tension	S
	Compression	S^*
	Tension or Compression plus Max. Bending	$K_s \times S$
	Average Bearing (hole)	$1.5 \times S$

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION

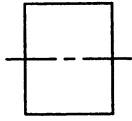
Gussets	Max. Membrane plus Bending	$K_s \times S$
	Max. Shear	$0.8 \times S$
	Average Shear	$0.6 \times S$
Shear Pads or Shear Rings	Average Bearing	$1.5 \times S$
Fillet Welds (Throat)	Shear	$0.80 \times S$
Groove Welds	Tension	$0.74 \times S$
	Shear	$0.60 \times S$
Butt Welds	Tension	$E \times S$
	Shear	$0.70 \times S$

Notes:

1. The stress limits are based on ASME Section VIII Div. 1 & 2, ASME B31.1, ASME B31.3, the AISC Manual of Steel Construction, and the ASME Criteria Document.
2. S is the basic allowable stress at the design temperature for the component from the applicable Code. If a detailed stress analysis is performed such as Finite Element Analysis, the basic allowable stress S may be used in place of the design stress intensity S_m . For welds, the basic allowable stress S is based on the weaker of the two materials joined.
3. K_s is the shape factor for the cross section (See Table III).
4. S^* is the lesser of S or the allowable stress for compression members from the AISC Manual of Steel Construction.
5. S^{**} is the average of the tabulated values of the basic allowable stress S for the highest and lowest temperatures during the operation cycle under consideration.
6. E is the weld joint efficiency/quality factor from the applicable Code.
7. At design temperatures in the creep range, additional considerations may be required.
8. Excessive deformation or local buckling may limit the loading on components.
9. Membrane stress is uniform and equal to the average stress across the section. Bending stress varies with location across the section.
10. For cyclic loading, additional fatigue evaluation may be required.
11. Bearing stress limits for holes are based with smooth machined surfaces.
12. Component stresses during the pressure test shall not exceed 1.5 times the applicable stress limit where the stress limit is based on the test temperature.

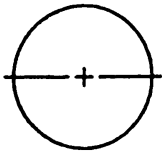
TABLE III
Shape Factors

Solid Rectangle



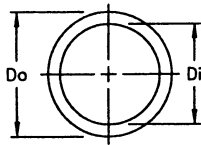
$$K_s = 1.5$$

Solid Cylinder



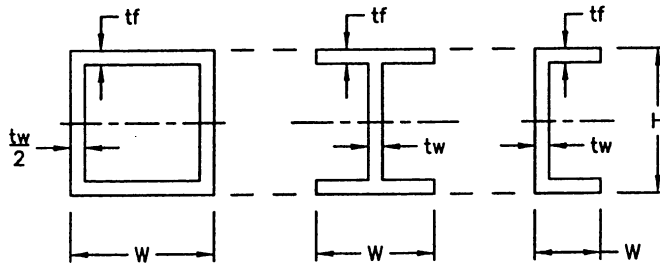
$$K_s = 1.7$$

Hollow Cylinder



$$K_s = \frac{1.7(D_o^4 - D_i^4)}{D_o^4 - D_i^4}$$

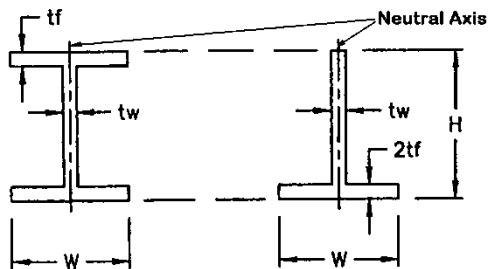
Hollow rectangle, Beam, Channel



$$d = H - 2t_f$$

$$K_s = \frac{1.5H \langle d^2 t_w + 4W t_f (d + t_f) \rangle}{WH^3 - d^3 (W - t_w)}$$

Beam, Tee



$$d = H - 2t_f$$

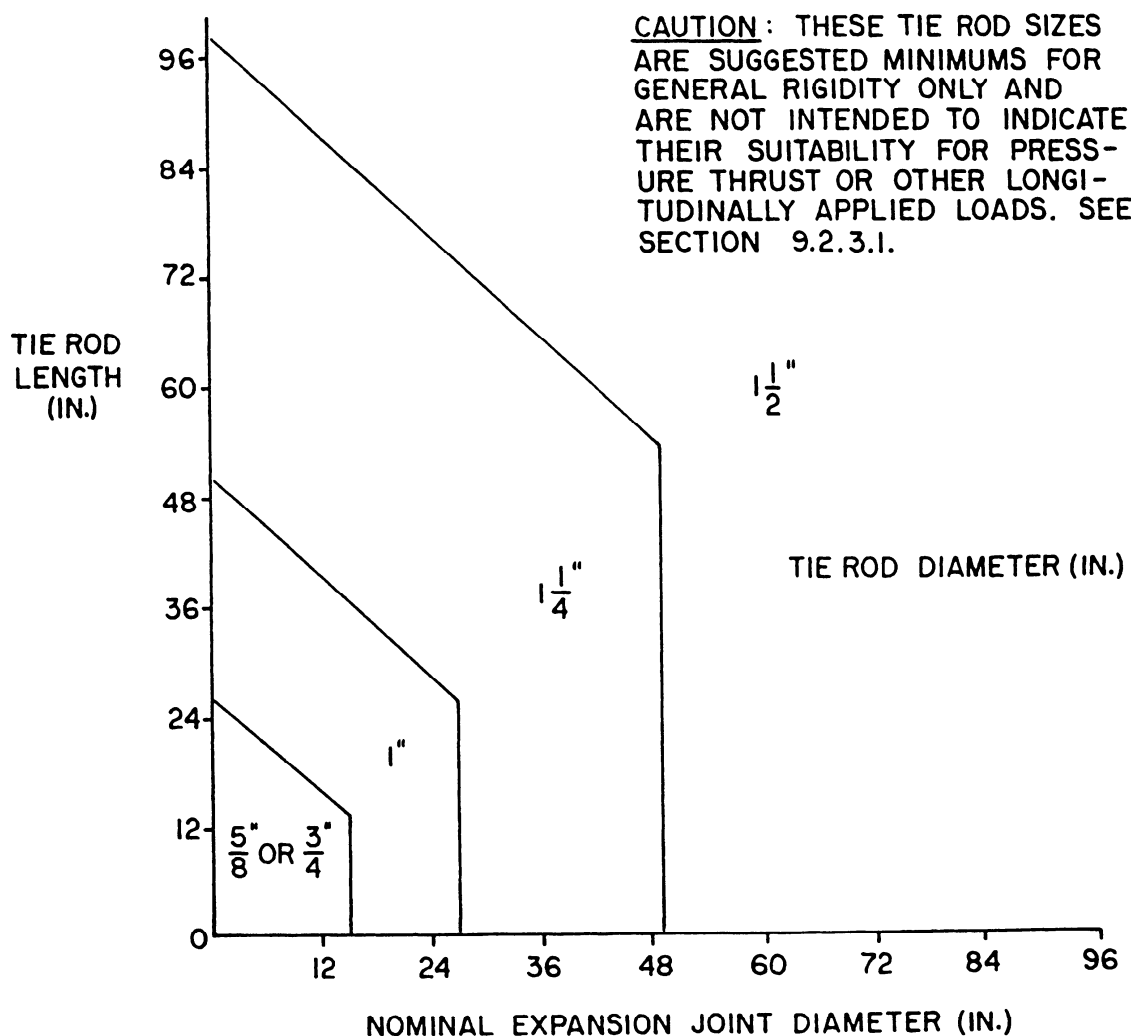
$$K_s = \frac{1.5W (2W^2 t_f + t_w^2 d)}{2W^3 t_f + t_w^3 d}$$

Channel, Tee



$$K_s = 1.5 \text{ or calculated value}$$

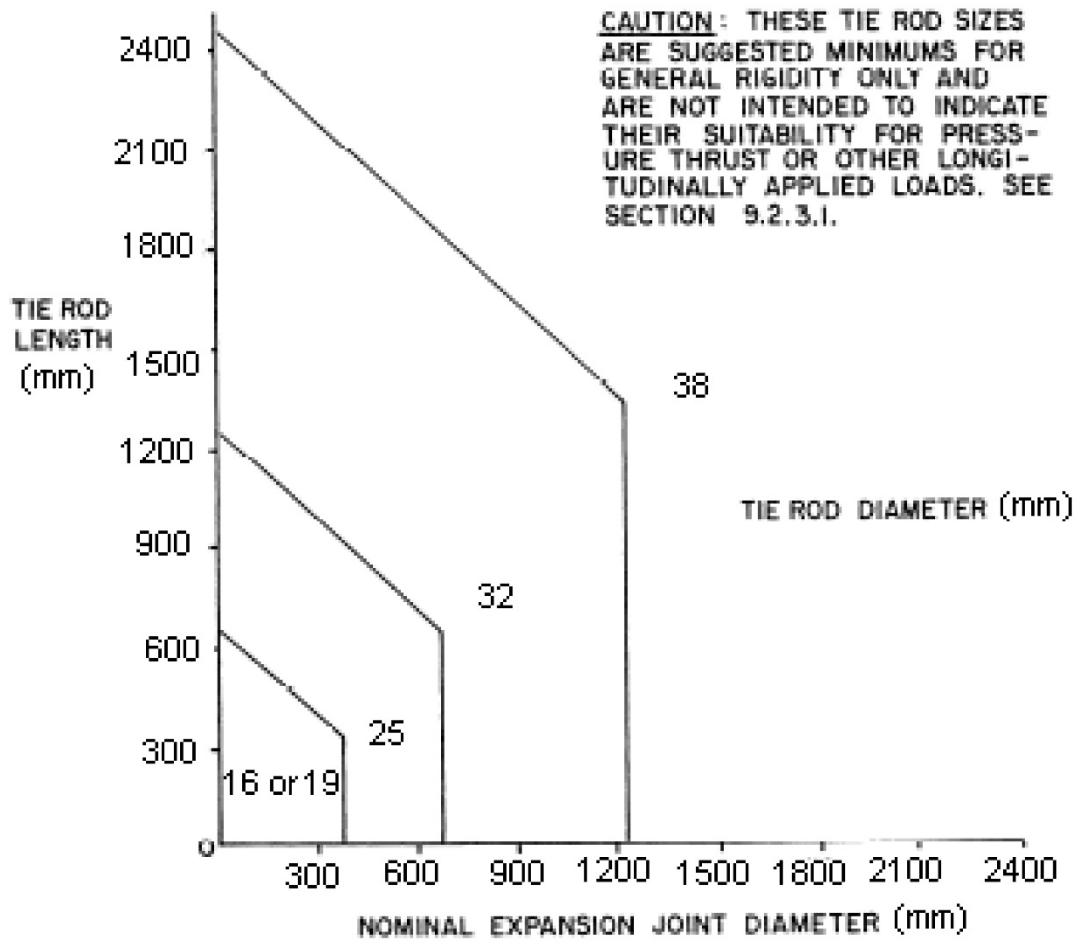
STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION



RECOMMENDED MINIMUM TIE ROD SIZE

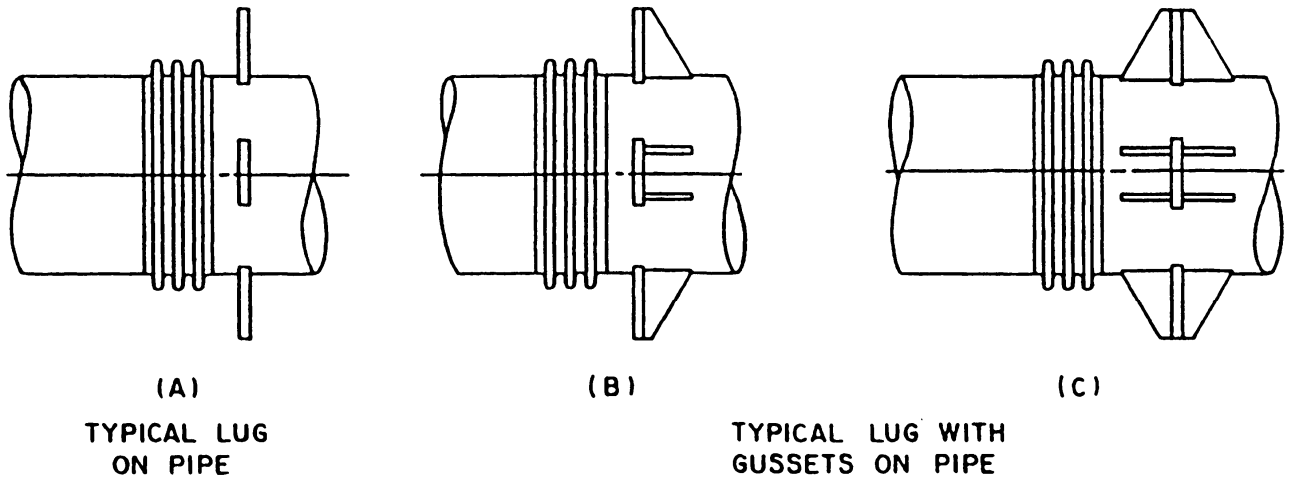
FIGURE 9.2

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION

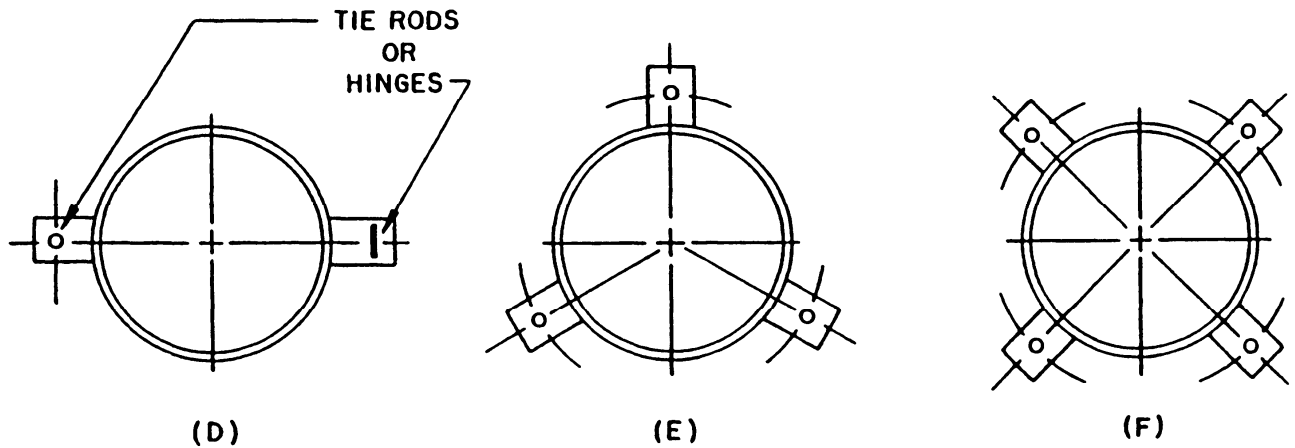


RECOMMENDED MINIMUM TIE ROD SIZE

FIGURE 9.2M

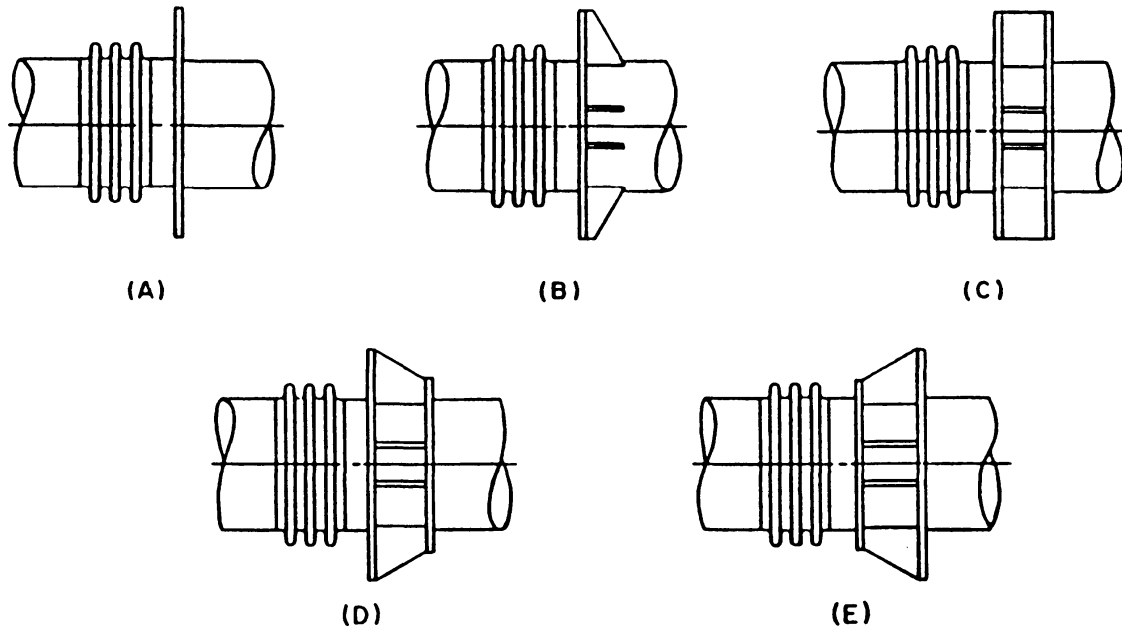


TYPICAL LUG ARRANGEMENTS

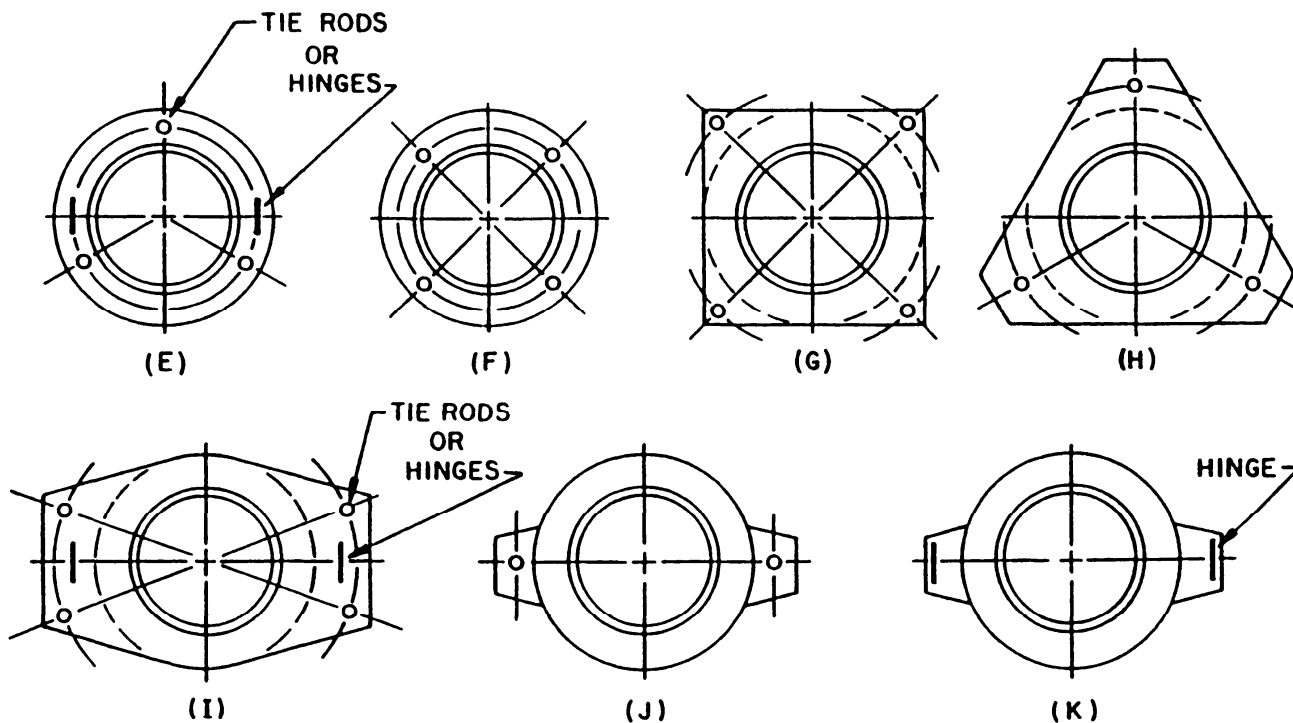


TIE ROD AND HINGE ATTACHMENTS

FIGURE 9.3



TYPICAL ARRANGEMENTS OF
SINGLE AND DOUBLE RINGS



TIE ROD AND HINGE ATTACHMENTS

FIGURE 9.4

9.2.3.4 REFERENCES

The following references are not all-inclusive but may provide information useful to the evaluation of tie rods, hinges, gimbals, and their attachment to the piping:

- 1 . Blake, Alexander: *Practical Stress Analysis in Engineering Design*, Mercel Dekker, Inc.
- 2 Blodgett, D.W.: *Design of Welded Structures*, James F. Lincoln Arc Welding Foundation.
- 3 Brownell, L.E. And E.H. Young: *Process Equipment Design*, John Wiley and Sons, Inc.
4. Roark, R.J. and W.C. Young: *Formulas for Stress and Strain*, McGraw-Hill Book Co.
5. Timoshenko, S.: *Strength of Materials*, D. Van Nostrand Co., Inc.
6. Wichman, K.R., A.G. Hopper, J.L. Merschon: *Local Stresses in Spherical and Cylindrical Shells, Pressure Vessels and Piping, Vol. 2*, American Society of Mechanical Engineers.
7. Wichman, K.R., Hopper, A.G., J.L. Merschon: *Local Stresses in Spherical and Cylindrical Shells Due to External Loadings, Bulletin 107*, Welding Research Council

9.3 FLANGES

Expansion Joints may be fitted with flanges. The choice of flanges will depend upon the specified service conditions, the flanges furnished on connected piping or equipment and the Expansion Joint manufacturer's standards. Expansion Joint flanges may conform to ASME/ANSI standard dimensions and drilling, but special facings and drilling are available to suit specific service conditions and applications.

There are three different types of construction used in the fabrication of Expansion Joints with flanged ends. The Expansion Joint manufacturer may use any one or all of the following methods, depending upon the particular application in question:

- a. Van Stoned Ends - The flanges are slipped over the tangents of the bellows and the bellows material is flared out or "Van Stoned" over the faces of the flanges. The bellows material prevents contact between the flanges and the medium flowing through the pipe. During installation, the Expansion Joint flanges can be rotated to match the bolt holes in the mating pipe line flanges. Although flat faced flanges are generally used for this type of construction, the Van Stoned portion of the bellows material overlapping the face of the flanges creates a condition which is, in effect, equivalent to a raised face.
- b. Flanges Welded to Pipe Nipples - The manufacturer welds the bellows to short pipe nipples and then welds the flanges to the other end of these pipe nipples. Since the flanges will not be free to rotate, it is sometimes desirable to ship the Expansion Joint with one flange unwelded to facilitate field installation.
- c. Flanges Welded to Bellows - The manufacturer welds the bellows directly to the flanges. This construction generally results in the shortest possible face-to-face dimension. The customer should give consideration to the type of pipe line flanges specified when ordering Expansion Joints, since the foregoing differences in Expansion Joint construction may have an effect on the type of mating flanges required. If flat faced flanges are specified, it is not advisable to specify Van Stoned Expansion Joints, since the Van Stoned portion of the bellows is actually equivalent to a raised face. Consequently, if flat faced flanges must be furnished, the Expansion Joints should be ordered with flanges welded on.

If raised face flanges are specified, Van Stoned Expansion Joints are generally considered satisfactory, providing proper gaskets are used. However, the face of the Van Stone is not a machined finish and it may not comply in every respect with the ASME/ANSI dimensions for a raised face. Consequently, if ASME/ANSI flanges must be furnished, or if a particularly fine machined surface is required for use with metallic and similar gasket materials, it may be necessary to specify Expansion Joints with flanges welded on.

Because of the wide variation in the type of flanges available, it is essential for the customer to specify his flange requirements completely and accurately when ordering Expansion Joints. Flange specifications for pipe sizes up to and including 24 in. (610mm) are clearly defined by the ASME/ANSI standards. In the case of pipe sizes larger than 24 in. (610mm), however, the ASME/ANSI standards are incomplete and are subject to frequent misinterpretation.

IN ORDER TO PREVENT CONFUSION WHEN SPECIFYING LARGE DIAMETER FLANGES, IT IS NECESSARY THAT THE CUSTOMER SPECIFY EITHER A FLANGE MANUFACTURER'S CATALOG AND PAGE NUMBER, OR THE ACTUAL MATERIAL, OUTSIDE DIAMETER, THICKNESS, DRILLING AND FACING FINISH REQUIRED.

9.4 CORROSION

Corrosion can significantly reduce the service life of an Expansion Joint. The design and operating characteristics of Expansion Joints are such that they may be subjected to corrosive attack under conditions which might not affect piping and fittings of similar materials.

Possible types of corrosion that may be experienced in expansion joint applications are as follows:

- Stress-corrosion, which is evidenced by a cracking of the material as the result of a combination of stress and a corrosive environment
- Intergranular-corrosion, which is characterized by a preferential attack along the grain boundaries in metals
- Pitting, which is a localized attack on metals; general corrosion or the gradual eating away of the metals in a system
- Impingement and corrosion erosion, associated with the impact of a liquid or gas medium on the surface of the material under attack
- Elevated temperature oxidation is another form of material degradation most commonly encountered in hot air and exhaust lines.

Occurrence of all types of corrosion depends upon the material type and condition, as well as its initial surface condition. Selection of the material type should be such that there is no possibility of corrosion occurring or that it is not affected by corrosion to an extent greater than 0.002 inches (0.508 mm) penetration per year.

The corrosion resistance of stainless steel depends on the formation of a thin, unbroken, chromic oxide surface, which will form slowly in the atmosphere on clean stainless steel. Particles of steel, such as welding spatter, will prevent the formation of this chromic oxide surface; therefore, to produce maximum general corrosion resistance, all scale should be removed by pickling. The adherence of welding spatter should be prevented both in the shop and during installation by covering the bellows or by using an anti-spatter compound. Although it is sometimes desirable to heat treat austenitic stainless steel piping components in order to improve their resistance to corrosion, this is not usually the case with bellows. Expansion Joint bellows are invariably used at movements producing high stresses, frequently within the plastic range; thus, any beneficial effect of removing residual stresses would be quickly nullified by operating stresses. The possible occurrence of stress corrosion in austenitic stainless steel bellows cannot be eliminated by heat treatment or by reducing the movement.

In the design of piping systems containing Expansion Joints, attention is given to the internal conditions and medium; but little if any, to the external conditions. This practice can lead to reduced service life, since external corrosion can be experienced where fumes or sprays may contact the bellows or in tunnel and manhole installations where water is allowed to collect.

Many corrosion problems encountered in the field can be reduced in magnitude, if not completely eliminated, by careful planning and design. The process engineer or designer must anticipate situations where corrosive attack might result from a certain design configuration or material selection and avoid such conditions wherever possible.

Since corrosion problems may be complex, it is often advisable to consult a qualified corrosion engineer.

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
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STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX A

STANDARD ROUND EXPANSION JOINT SPECIFICATION SHEET

Customer:		Date:	Page:
Project:		Prepared By:	
Item or Tag Number:			
Quantity			
Size			
Style or Type (single, universal, hinged, gimbal, etc.)			
End Connections	Thickness / Flange Rating		
	Material		
*Pressure	Design		
	Operating		
	Test		
*Temperature	Design		
	Operating		
	Installation		
Media	Media		
	Flow Velocity		
	Flow Direction		
Movements and Life Cycle	Installation	Axial Extension	
		Axial Compression	
		Lateral	
		Angular	
		Number of Cycles	
	Design	Axial Extension	
		Axial Compression	
		Lateral	
		Angular	
		Number of Cycles	
	Operating	Axial Extension	
		Axial Compression	
		Lateral	
		Angular	
		Number of Cycles	
Materials	Bellows		
	Liner		
	Cover		
Dimensions	Overall Length		
	Maximum O.D.		
	Minimum I.D.		
Spring Rates	Maximum Axial Spring Rate		
	Maximum Lateral Spring Rate		
	Maximum Angular Spring Rate		
Quality Assurance	Bellows Long. Seam Weld		
	Bellows Attachment Weld		
	Piping		
Required Code			
Applicable Codes and Standards: B31.1, B31.3, Sect 8 Div 1			



STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX A
STANDARD RECTANGULAR EXPANSION JOINT SPECIFICATION SHEET



Customer:		Date:	Page:
Project:		Prepared By:	
Item or Tag Number:			
Quantity			
Size (long side and short side)			
Orientation (horizontal / vertical / inclined)			
Style or Type			
Corner Type			
End Connections	Thickness/ Flange Size		
	Material		
Pressure	Design		
	Operating		
Temperature	Design		
	Operating		
	Installation		
Media	Media		
	Flow Velocity		
	Flow Direction		
Movements	Axial Extension		
	Axial Compression		
	Lateral (parallel to short side)		
	Lateral (parallel to long side)		
	Angular (parallel to short side)		
	Angular (parallel to long side)		
Materials	Bellows		
	Liner		
	Cover		
Dimensions	Overall Length		
Maximum Spring Rates	Axial		
	Lateral (parallel to short side)		
	Lateral (parallel to long side)		
	Angular (parallel to short side)		
	Angular (parallel to long side)		
Quality Assurance	Bellows Long. Seam Weld		
	Bellows Attachment Weld		
Required Code			
Applicable Codes and Standards: B31.1, B31.3, Sect 8 Div 1 or other			

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX B

KEY TO SYMBOLS USED



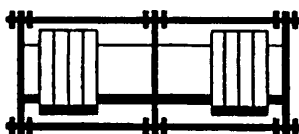
SINGLE EXPANSION JOINT



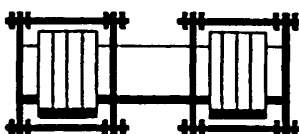
SINGLE EXPANSION JOINT
WITH TIE RODS



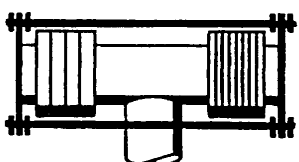
DOUBLE EXPANSION
JOINT WITH INTER-
MEDIATE ANCHOR



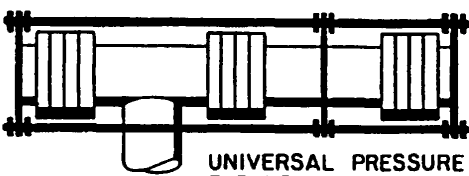
UNIVERSAL EXPANSION
JOINT WITH OVERALL
TIE RODS



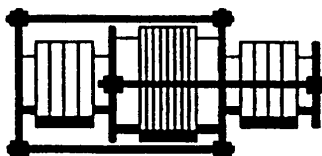
UNIVERSAL EXPANSION
JOINT WITH LIMIT OR
CONTROL RODS



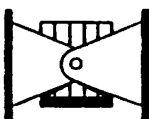
PRESSURE BALANCED
EXPANSION JOINT



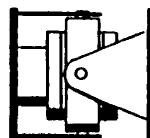
UNIVERSAL PRESSURE BALANCED
EXPANSION JOINT



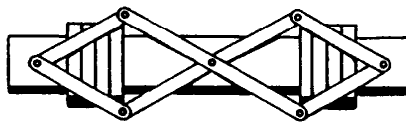
INLINE PRESSURE
BALANCED EXPANSION
JOINT



HINGED EXPANSION JOINT



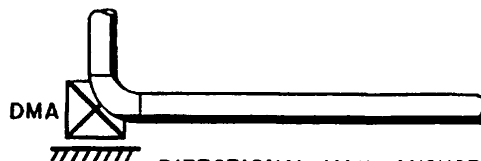
GIMBAL EXPANSION JOINT



UNIVERSAL PANTOGRAPH EXPANSION JOINT



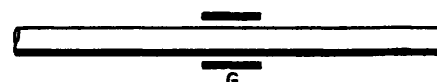
MAIN ANCHOR



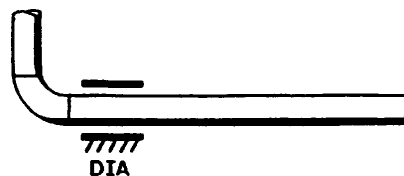
DIRECTIONAL MAIN ANCHOR



INTERMEDIATE ANCHOR



PIPE ALIGNMENT GUIDE



DIRECTIONAL INTERMEDIATE
ANCHOR WITH GUIDE

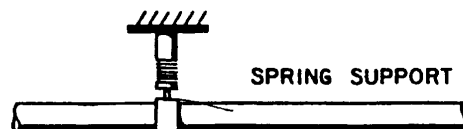


SIDE VIEW



END VIEW

PLANAR PIPE ALIGNMENT GUIDE



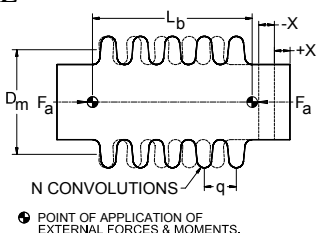
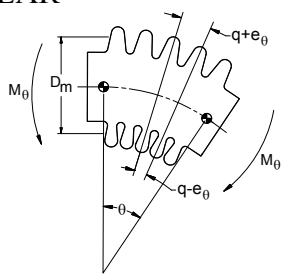
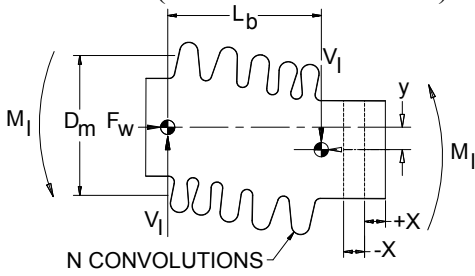
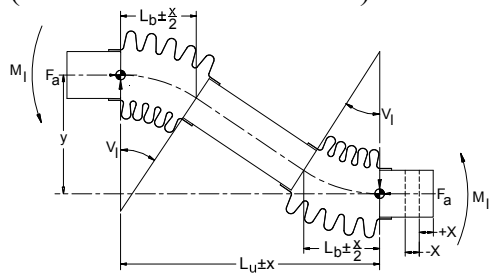
SPRING SUPPORT

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX B

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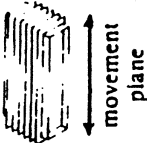
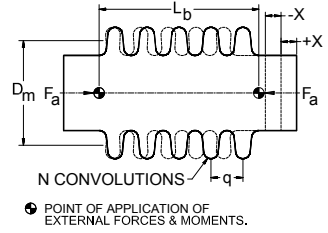
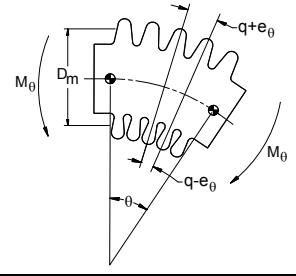
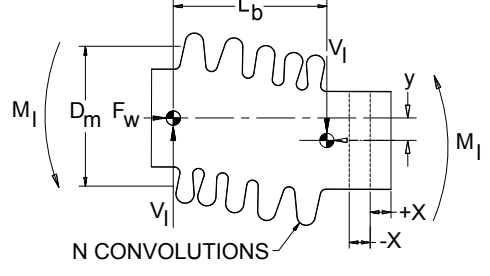
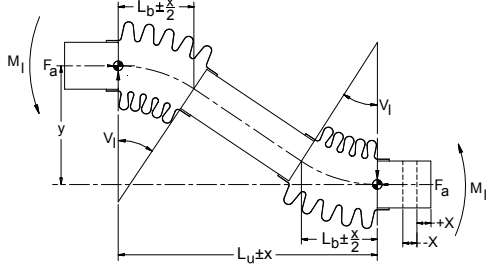
STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX C

CIRCULAR MOVEMENT, FORCE, AND MOMENT EQUATIONS

TYPE MOVEMENT	CIRCULAR BELLOWS
	
AXIAL  • POINT OF APPLICATION OF EXTERNAL FORCES & MOMENTS.	$e_x = \frac{x}{N}$ $F_a = f_w e_x$ $V_l = 0$ $M_l = 0$ $M_\theta = 0$
ANGULAR 	$e_\theta = \frac{\theta D_m}{2N}$ $F_a = 0$ $V_l = 0$ $M_\theta = \frac{f_w D_m e_\theta}{4}$ $M_l = 0$
LATERAL (SINGLE BELLOWS) 	$e_y = \frac{3D_m y}{N(L_b \pm x)}$ $V_l = \frac{f_w D_m e_y}{2(L_b \pm x)}$ $M_l = \frac{f_w D_m e_y}{4}$ $M_\theta = 0$
LATERAL AND AXIAL (UNIVERSAL BELLOWS) 	$e_x = \frac{x}{2N}$ $e_y = \frac{3D_m L^*/L_b (1 + L^*/L_b) y}{2N [1 + 3(L^*/L_b)^2] (L^* \pm x/2)}$ $F_a = f_w e_x$ $V_l = \frac{f_w D_m e_y}{2(L_u \pm x)}$ $M_l = \frac{f_w D_m e_y}{4}$ $M_\theta = 0$

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX C

RECTANGULAR MOVEMENT, FORCE, AND MOMENT EQUATIONS

TYPE MOVEMENT	RECTANGULAR BELLOWS	
	MOVEMENT PARALLEL WITH LONG SIDE	MOVEMENT PARALLEL WITH SHORT SIDE
	 $L_{ml} = \frac{L_l}{3} \left(\frac{3L_s + L_l}{L_s + L_l} \right)$	 $L_{ms} = \frac{L_s}{3} \left(\frac{3L_l + L_s}{L_s + L_l} \right)$
AXIAL  N CONVOLUTIONS • POINT OF APPLICATION OF EXTERNAL FORCES & MOMENTS.	$e_x = \frac{x}{N}$ $F_a = f_w e_x$ $V_{Ll} = 0$ $M_{Ll} = 0$ $M_{\theta l} = 0$	$e_x = \frac{x}{N}$ $F_a = f_w e_x$ $V_{Ls} = 0$ $M_{Ls} = 0$ $M_{\theta s} = 0$
ANGULAR 	$e_{\theta l} = \frac{\theta L_l}{2N}$ $F_a = 0$ $V_{Ll} = 0$ $M_{\theta l} = \frac{f_w L_{ml} e_{\theta l}}{4}$ $M_{Ll} = 0$	$e_{\theta s} = \frac{\theta L_s}{2N}$ $F_a = 0$ $V_{Ls} = 0$ $M_{\theta s} = \frac{f_w L_{ms} e_{\theta s}}{4}$ $M_{Ls} = 0$
LATERAL (SINGLE BELLOWS)  N CONVOLUTIONS	$e_{yl} = \frac{3L_l y_l}{N(L_b \pm x)}$ $V_{Ll} = f_w L_{ml} \frac{e_{yl}}{L_b}$ $M_{Ll} = \frac{f_w L_{ml} e_{yl}}{2}$ $M_{\theta l} = 0$	$e_{ys} = \frac{3L_l y_s}{N(L_b \pm x)}$ $V_{Ls} = f_w L_{ms} \frac{e_{ys}}{L_b}$ $M_{Ls} = \frac{f_w L_{ms} e_{ys}}{2}$ $M_{\theta s} = 0$
LATERAL AND AXIAL (UNIVERSAL BELLOWS) 	$e_x = \frac{x}{2N}$ $e_{yl} = \frac{3L_s L^*/L_b (1 + L^*/L_b) y_s}{2N \left[1 + 3(L^*/L_b)^2 \right] (L^* \pm x/2)}$ $F_a = f_w e_x, lbs.$ $V_{Ll} = \frac{f_w L_{ml} e_{yl}}{L_u}$ $M_{Ll} = \frac{f_w L_{ml} e_{yl}}{2}$ $M_{\theta l} = 0$	$e_x = \frac{x}{2N}$ $e_{ys} = \frac{3L_l L^*/L_b (1 + L^*/L_b) y_l}{2N \left[1 + 3(L^*/L_b)^2 \right] (L^* \pm x/2)}$ $F_a = f_w e_x$ $V_{Ls} = \frac{f_w L_{ms} e_{ys}}{L_u}$ $M_{Ls} = \frac{f_w L_{ml} e_{ys}}{2}$ $M_{\theta s} = 0$

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

Conversion Factors
Acceleration

<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Feet/Second/Second	3.048 E – 01	Meters/Second/Second

Angle

<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Degrees	1.745 E – 02	Radians

Area

<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Square Inches	6.452 E + 02	Square Millimeters
Square Feet	9.290 E + 04	Square Millimeters
Square Feet	9.290 E – 02	Square Meters

Density

<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Pounds/Cubic Foot	1.602 E + 01	Kilograms/Cubic Meter
Pounds/Cubic Inch	2.768 E + 04	Kilograms/Cubic Meter
Pounds/Cubic Inch	2.768 E + 01	Grams/Cubic Centimeter

Force

<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Pounds	4.536 E – 01	Kilograms
Pounds	4.448 E + 00	Newtons
Pounds	4.448 E + 05	Dynes

Length

<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Inches	2.540 E + 01	Millimeters
Inches	2.540 E – 02	Meters
Feet	3.048 E + 02	Millimeters
Feet	3.048 E – 01	Meters

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

Pressure, Modulus, Stress		
<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Pounds/Square Inch	6.895 E – 03	Newtons/Millimeter Squared
Pounds/Square Inch	7.031 E – 04	Kilograms/Millimeter Squared
Pounds/Square Inch	6.895 E + 03	Pascals
Pounds/Square Inch	6.895 E + 00	Kilopascals
Pounds/Square Inch	6.895 E – 03	Megapascals
Pounds/Square Inch	6.895 E – 02	Bar

Spring Constant		
<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Pounds/Inch	1.751 E – 01	Newtons/Millimeter
Pounds/Inch	1.751 E + 02	Newtons/Meter
Pounds/Inch	1.786 E – 02	Kilograms/ Millimeter

Temperature		
<i>Convert</i>	<i>To</i>	
Degrees Fahrenheit	Degrees Centigrade	Subtract 32 & divide by 1.8

Torque (Moment)		
<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Inch-Pounds	1.130 E + 02	Newton-Millimeters
Foot-Pounds	1.356 E + 03	Newton-Millimeters
Foot-Pounds	1.356 E + 00	Newton-Meters

Velocity		
<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Feet/Second	3.048 E + 02	Millimeters/ Second
Feet/Second	3.048 E – 01	Meters/Second

Volume		
<i>Multiply</i>	<i>By</i>	<i>To Obtain</i>
Cubic Inches	1.639 E + 04	Cubic Millimeters
Cubic Inches	1.639 E + 01	Cubic Centimeters
Cubic Feet	2.832 E + 07	Cubic Millimeters
Cubic Feet	2.832 E + 04	Cubic Centimeters
Cubic Feet	2.832 E – 02	Cubic Meters

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

Steam Pressure Table

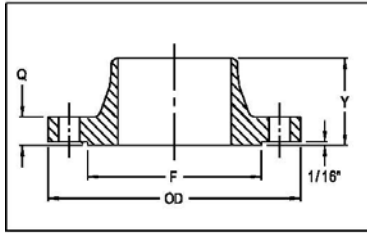
Temperature		Saturated Steam		Temperature		Saturated Steam	
F	C	Psig	MPa	F	C	Psig	MPa
212	100	0.00	0.000	460	238	451.60	3.114
220	104	2.49	0.017	480	249	550.80	3.798
240	116	10.27	0.071	500	260	665.30	4.587
260	127	20.72	0.143	520	271	796.70	5.493
280	138	34.48	0.238	540	282	946.80	6.528
300	149	52.28	0.360	560	293	1117.10	7.702
320	160	74.90	0.516	580	304	1309.60	9.029
340	171	103.23	0.712	600	316	1526.30	10.523
360	182	138.22	0.953	620	327	1769.70	12.202
380	193	180.90	1.247	640	338	2042.40	14.082
400	204	232.40	1.602	660	349	2347.30	16.184
420	216	293.80	2.026	680	360	2690.30	18.549
440	227	366.50	2.527	700	371	3075.30	21.203

LOW-PRESSURE CONVERSIONS

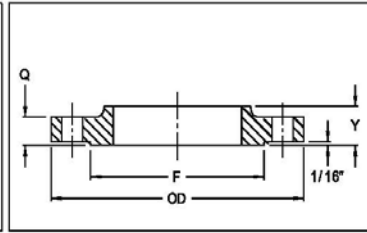
1 in. Mercury = 0.4912 psig
1 in. Mercury = 13.60 in. of water
1 in. Mercury = 0.03386 bar
1 in. Mercury = 3.3864 kPa

1 kPa = 0.145 psig
1 kPa = 0.01 bar
1 bar = 0.1 N/sq. mm
1 psig = 0.06895 bar

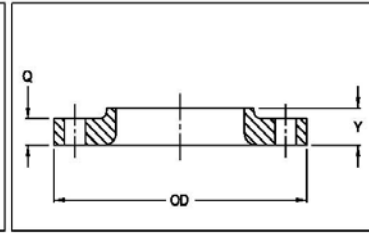
STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D



WELDING NECK



SLIP-ON



LAP JOINT

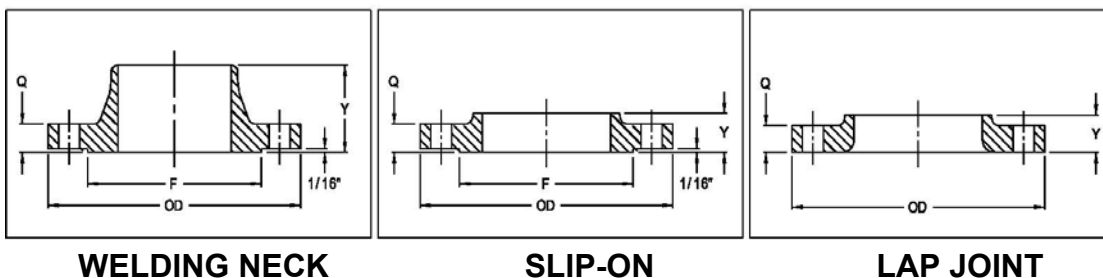
CLASS 150 FORGED FLANGE DIMENSIONS (ASME B16.5-2013) FOR REFERENCE ONLY

NOM PIPE SIZE	FLG DIA. (OD)	FLG THICK (Q)	RF DIA. (F)	HUB LENGTH (Y)			DRILLING		
				WN	SO	LJ	NO. OF HOLES	DIA. OF HOLES	BOLT CIRCLE DIA.
½ DN 15	3.50 (90)	0.44 (11.2)	1.38 (34.9)	1.88 (48)	0.62 (16)	0.62 (16)	4	5/8 (15.9)	2.38 (60.3)
¾ DN 20	3.88 (100)	0.50 (12.7)	1.69 (42.9)	2.06 (52)	0.62 (16)	0.62 (16)	4	5/8 (15.9)	2.75 (69.9)
1 DN 25	4.25 (110)	0.56 (14.3)	2.00 (50.8)	2.19 (56)	0.69 (17)	0.69 (17)	4	5/8 (15.9)	3.12 (79.4)
1¼ DN 32	4.62 (115)	0.62 (15.9)	2.50 (63.5)	2.25 (57)	0.81 (21)	0.81 (21)	4	5/8 (15.9)	3.50 (88.9)
1½ DN 40	5.00 (125)	0.69 (17.5)	2.88 (73.0)	2.44 (62)	0.88 (22)	0.88 (22)	4	5/8 (15.9)	3.88 (98.4)
2 DN 50	6.00 (150)	0.75 (19.1)	3.62 (92.1)	2.50 (64)	1.00 (25)	1.00 (25)	4	¾ (19.1)	4.75 (120.7)
2½ DN 65	7.00 (180)	0.88 (22.3)	4.12 (104.8)	2.75 (70)	1.13 (29)	1.13 (29)	4	¾ (19.1)	5.50 (139.7)
3 DN 80	7.50 (190)	0.94 (23.9)	5.00 (127.0)	2.75 (70)	1.19 (30)	1.19 (30)	4	¾ (19.1)	6.00 (152.4)
3½ DN 90	8.50 (215)	0.94 (23.9)	5.50 (139.7)	2.81 (71)	1.25 (32)	1.25 (32)	8	¾ (19.1)	7.00 (177.8)
4 DN 100	9.00 (230)	0.94 (23.9)	6.19 (157.2)	3.00 (76)	1.31 (33)	1.31 (33)	8	¾ (19.1)	7.50 (190.5)
5 DN 125	10.00 (255)	0.94 (23.9)	7.31 (185.7)	3.50 (89)	1.44 (37)	1.44 (36)	8	7/8 (22.2)	8.50 (215.9)
6 DN 150	11.00 (280)	1.00 (25.4)	8.50 (215.9)	3.50 (89)	1.56 (40)	1.56 (40)	8	7/8 (22.2)	9.50 (241.3)
8 DN 200	13.50 (345)	1.12 (28.6)	10.62 (269.9)	4.00 (102)	1.75 (44)	1.75 (44)	8	7/8 (22.2)	11.75 (298.5)
10 DN 250	16.00 (405)	1.19 (30.2)	12.75 (323.8)	4.00 (102)	1.94 (49)	1.94 (49)	12	1 (25.4)	14.25 (362.0)
12 DN 300	19.00 (485)	1.25 (31.8)	15.00 (381.0)	4.50 (114)	2.19 (56)	2.19 (56)	12	1 (25.4)	17.00 (431.8)
14 DN 350	21.00 (535)	1.38 (35.0)	16.25 (412.8)	5.00 (127)	2.25 (57)	3.12 (79)	12	1 1/8 (28.6)	18.75 (476.3)
16 DN 400	23.50 (595)	1.44 (36.6)	18.50 (469.9)	5.00 (127)	2.50 (64)	3.44 (87)	16	1 1/8 (28.6)	21.25 (539.8)
18 DN 450	25.00 (635)	1.56 (39.7)	21.00 (533.4)	5.50 (140)	2.69 (68)	3.81 (97)	16	1 1/4 (31.8)	22.75 (577.9)
20 DN 500	27.50 (700)	1.69 (42.9)	23.00 (584.2)	5.69 (144)	2.88 (73)	4.06 (103)	20	1 1/4 (31.8)	25.00 (635.0)
24 DN 600	32.00 (815)	1.88 (47.7)	27.25 (692.2)	6.00 (152)	3.25 (83)	4.38 (111)	20	1 3/8 (34.9)	29.50 (749.3)

Dimensions are in inches (metric dimensions are in parenthesis)

Reference Section 9.3

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D



CLASS 300 FORGED FLANGE DIMENSIONS (ASME B16.5-2013) FOR REFERENCE ONLY

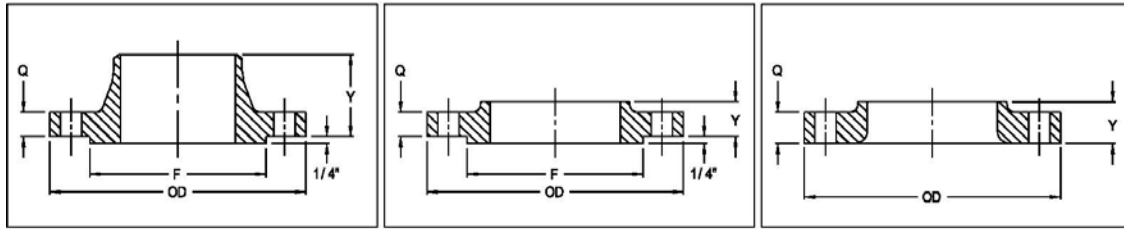
NOM PIPE SIZE	FLG DIA. (OD)	FLG THICK (Q)	RF DIA. (F)	HUB LENGTH (Y)			DRILLING		
				WN	SO	LJ	NO. OF HOLES	DIA. OF HOLES	BOLT CIRCLE DIA.
½	3.75	0.56	1.38	2.06	0.88	0.88	4	5/8	2.62
DN 15	(95)	(14.3)	(34.9)	(52)	(22)	(22)		(15.9)	(66.7)
¾	4.62	0.62	1.69	2.25	1.00	1.00	4	¾	3.25
DN 20	(115)	(15.9)	(42.9)	(57)	(25)	(25)		(19.1)	(82.6)
1	4.88	0.69	2.00	2.44	1.06	1.06	4	¾	3.50
DN 25	(125)	(17.5)	(50.8)	(62)	(27)	(27)		(19.1)	(88.9)
1¼	5.25	0.75	2.50	2.56	1.06	1.06	4	¾	3.88
DN 32	(135)	(19.1)	(63.5)	(65)	(27)	(27)		(19.1)	(98.4)
1½	6.12	0.81	2.88	2.69	1.19	1.19	4	7/8	4.50
DN 40	(155)	(20.7)	(73.0)	(68)	(30)	(30)		(22.2)	(114.3)
2	6.50	0.88	3.62	2.75	1.31	1.31	8	¾	5.00
DN 50	(165)	(22.3)	(92.1)	(70)	(33)	(33)		(19.1)	(127.0)
2½	7.50	1.00	4.12	3.00	1.50	1.50	8	7/8	5.88
DN 65	(190)	(25.4)	(104.8)	(76)	(38)	(38)		(22.2)	(149.2)
3	8.25	1.12	5.00	3.12	1.69	1.69	8	7/8	6.62
DN 80	(210)	(28.6)	(127.0)	(79)	(43)	(43)		(22.2)	(168.3)
3½	9.00	1.19	5.50	3.19	1.75	1.75	8	7/8	7.25
DN 90	(230)	(30.2)	(139.7)	(81)	(44)	(44)		(22.2)	(184.2)
4	10.00	1.25	6.19	3.38	1.88	1.88	8	7/8	7.88
DN 100	(255)	(31.8)	(157.2)	(86)	(48)	(48)		(22.2)	(200.0)
5	11.00	1.38	7.31	3.88	2.00	2.00	8	7/8	9.25
DN 125	(280)	(35.0)	(185.7)	(98)	(51)	(51)		(22.2)	(235.0)
6	12.50	1.44	8.50	3.88	2.06	2.06	12	7/8	10.62
DN 150	(320)	(36.6)	(215.9)	(98)	(52)	(52)		(22.2)	(269.9)
8	15.00	1.62	10.62	4.38	2.44	2.44	12	1	13.00
DN 200	(380)	(41.3)	(269.9)	(111)	(62)	(62)		(25.4)	(330.2)
10	17.50	1.88	12.75	4.62	2.62	3.75	16	1 1/8	15.25
DN 250	(445)	(47.7)	(323.8)	(117)	(67)	(95)		(28.6)	(387.4)
12	20.50	2.00	15.00	5.12	2.88	4.00	16	1 1/4	17.75
DN 300	(520)	(50.8)	(381.0)	(130)	(73)	(102)		(31.8)	(450.8)
14	23.00	2.12	16.25	5.62	3.00	4.38	20	1 1/4	20.25
DN 350	(585)	(54.0)	(412.8)	(143)	(76)	(111)		(31.8)	(514.4)
16	25.50	2.25	18.50	5.75	3.25	4.75	20	1 3/8	22.50
DN 400	(650)	(57.2)	(469.9)	(146)	(83)	(121)		(34.9)	(571.5)
18	28.00	2.38	21.00	6.25	3.50	5.12	24	1 3/8	24.75
DN 450	(710)	(60.4)	(533.4)	(159)	(89)	(130)		(34.9)	(628.6)
20	30.50	2.50	23.00	6.38	3.75	5.50	24	1 3/8	27.00
DN 500	(775)	(63.5)	(584.2)	(162)	(95)	(140)		(34.9)	(685.8)
24	36.00	2.75	27.25	6.62	4.19	6.00	24	1 5/8	32.00
DN 600	(915)	(69.9)	(692.2)	(168)	(106)	(152)		(41.3)	(812.8)

Dimensions are in inches (metric dimensions are in parenthesis)

Reference Section 9.3

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D



WELDING NECK

SLIP-ON

LAP JOINT

CLASS 600 FORGED FLANGE DIMENSIONS (ASME B16.5-2013) FOR REFERENCE ONLY

NOM PIPE SIZE	FLG DIA. (OD)	FLG THICK (Q)	RF DIA. (F)	HUB LENGTH (Y)			DRILLING		
				WN	SO	LJ	NO. OF HOLES	DIA. OF HOLES	BOLT CIRCLE DIA.
½ DN 15	3.75 (95)	0.56 (14.3)	1.38 (34.9)	2.06 (52)	0.88 (22)	0.88 (22)	4	5/8 (15.9)	2.62 (66.7)
¾ DN 20	4.62 (115)	0.62 (15.9)	1.69 (42.9)	2.25 (57)	1.00 (25)	1.00 (25)	4	¾ (19.1)	3.25 (82.6)
1 DN 25	4.88 (125)	0.69 (17.5)	2.00 (50.8)	2.44 (62)	1.06 (27)	1.06 (27)	4	¾ (19.1)	3.50 (88.9)
1¼ DN 32	5.25 (135)	0.81 (20.7)	2.50 (63.5)	2.62 (67)	1.12 (29)	1.12 (29)	4	¾ (19.1)	3.88 (98.4)
1½ DN 40	6.12 (155)	0.88 (22.3)	2.88 (73.0)	2.75 (70)	1.25 (32)	1.25 (32)	4	7/8 (22.2)	4.50 (114.3)
2 DN 50	6.50 (165)	1.00 (25.4)	3.62 (92.1)	2.88 (73)	1.44 (37)	1.44 (37)	8	¾ (19.1)	5.00 (127.0)
2½ DN 65	7.50 (190)	1.12 (28.6)	4.12 (104.8)	3.12 (79)	1.62 (41)	1.62 (41)	8	7/8 (22.2)	5.88 (149.2)
3 DN 80	8.25 (210)	1.25 (31.8)	5.00 (127.0)	3.25 (83)	1.81 (46)	1.81 (46)	8	7/8 (22.2)	6.62 (168.3)
3½ DN 90	9.00 (230)	1.38 (35.0)	5.50 (139.7)	3.38 (86)	1.94 (49)	1.94 (49)	8	1 (25.4)	7.25 (184.2)
4 DN 100	10.75 (275)	1.50 (38.1)	6.19 (157.2)	4.00 (102)	2.12 (54)	2.12 (54)	8	1 (25.4)	8.50 (215.9)
5 DN 125	13.00 (330)	1.75 (44.5)	7.31 (185.7)	4.50 (114)	2.38 (60)	2.38 (60)	8	1 1/8 (28.6)	10.50 (266.7)
6 DN 150	14.00 (355)	1.88 (47.7)	8.50 (215.9)	4.62 (117)	2.62 (67)	2.62 (67)	12	1 1/8 (28.6)	11.50 (292.1)
8 DN 200	16.50 (420)	2.19 (55.6)	10.62 (269.9)	5.25 (133)	3.00 (76)	3.00 (76)	12	1 1/4 (31.8)	13.75 (349.2)
10 DN 250	20.00 (510)	2.50 (63.5)	12.75 (323.8)	6.00 (152)	3.38 (86)	4.38 (111)	16	1 3/8 (34.9)	17.00 (431.8)
12 DN 300	22.00 (560)	2.62 (66.7)	15.00 (381.0)	6.12 (156)	3.62 (92)	4.62 (117)	20	1 3/8 (34.9)	19.25 (489.0)
14 DN 350	23.75 (605)	2.75 (69.9)	16.25 (412.8)	6.50 (165)	3.69 (94)	5.00 (127)	20	1 1/2 (38.1)	20.75 (527.0)
16 DN 400	27.00 (685)	3.00 (76.2)	18.50 (469.9)	7.00 (178)	4.19 (106)	5.50 (140)	20	1 5/8 (41.3)	23.75 (603.2)
18 DN 450	29.25 (745)	3.25 (82.6)	21.00 (533.4)	7.25 (184)	4.62 (117)	6.00 (152)	20	1 3/4 (44.4)	25.75 (654.0)
20 DN 500	32.00 (815)	3.50 (88.9)	23.00 (584.2)	7.50 (190)	5.00 (127)	6.50 (165)	24	1 3/4 (44.4)	28.50 (723.9)
24 DN 600	37.00 (940)	4.00 (101.6)	27.25 (692.2)	8.00 (203)	5.50 (140)	7.25 (184)	24	2 (50.8)	33.00 (838.2)

Dimensions are in inches (metric dimensions are in parenthesis)

Reference Section 9.3

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (In.)	Inside Dia. (In.)	Inside Area (In ²)	Sq. Ft. Outside Surface (per ft)	Weight per ft. (Lb.)	Wt. of water per ft. (Lb.)	Moment of Inertia (In ⁴)	Section Modulus (In ³)	Radius of Gyration (In.)
½ (0.840)	10S	0.083	0.674	0.357	0.220	0.67	0.154	0.014	0.034	0.269
	40	0.109	0.622	0.304	0.220	0.85	0.132	0.017	0.041	0.261
	40S	0.109	0.622	0.304	0.220	0.85	0.132	0.017	0.041	0.261
	80	0.147	0.546	0.234	0.220	1.09	0.101	0.020	0.048	0.250
	80S	0.147	0.546	0.234	0.220	1.09	0.101	0.020	0.048	0.250
	160	0.188	0.464	0.169	0.220	1.31	0.073	0.022	0.053	0.240
	XXS	0.294	0.252	0.050	0.220	1.72	0.022	0.024	0.058	0.219
¾ (1.050)	5S	0.065	0.920	0.665	0.275	0.68	0.288	0.025	0.047	0.349
	10S	0.083	0.884	0.614	0.275	0.86	0.266	0.030	0.057	0.343
	40	0.113	0.824	0.533	0.275	1.13	0.231	0.037	0.071	0.334
	40S	0.113	0.824	0.533	0.275	1.13	0.231	0.037	0.071	0.334
	80	0.154	0.742	0.432	0.275	1.48	0.187	0.045	0.085	0.321
	80S	0.154	0.742	0.432	0.275	1.48	0.187	0.045	0.085	0.321
	160	0.219	0.612	0.294	0.275	1.95	0.127	0.053	0.101	0.304
1 (1.315)	XXS	0.308	0.434	0.148	0.275	2.44	0.064	0.058	0.110	0.284
	5S	0.065	1.185	1.103	0.344	0.87	0.478	0.050	0.076	0.443
	10S	0.109	1.097	0.945	0.344	1.41	0.409	0.076	0.115	0.428
	40	0.133	1.049	0.864	0.344	1.68	0.374	0.087	0.133	0.421
	40S	0.133	1.049	0.864	0.344	1.68	0.374	0.087	0.133	0.421
	80	0.179	0.957	0.719	0.344	2.17	0.311	0.106	0.161	0.407
	80S	0.179	0.957	0.719	0.344	2.17	0.311	0.106	0.161	0.407
1¼ (1.660)	160	0.250	0.815	0.522	0.344	2.85	0.226	0.125	0.190	0.387
	XXS	0.358	0.599	0.282	0.344	3.66	0.122	0.140	0.214	0.361
	5S	0.065	1.530	1.839	0.435	1.11	0.796	0.104	0.125	0.564
	10S	0.109	1.442	1.633	0.435	1.81	0.707	0.161	0.193	0.550
	40	0.140	1.380	1.496	0.435	2.27	0.648	0.195	0.235	0.540
	40S	0.140	1.380	1.496	0.435	2.27	0.648	0.195	0.235	0.540
	80	0.191	1.278	1.283	0.435	3.00	0.555	0.242	0.291	0.524
1½ (1.900)	80S	0.191	1.278	1.283	0.435	3.00	0.555	0.242	0.291	0.524
	160	0.250	1.160	1.057	0.435	3.77	0.458	0.284	0.342	0.506
	XXS	0.382	0.896	0.631	0.435	5.22	0.273	0.341	0.411	0.472
	5S	0.065	1.770	2.461	0.497	1.28	1.065	0.158	0.166	0.649
	10S	0.109	1.682	2.222	0.497	2.09	0.962	0.247	0.260	0.634
	40	0.145	1.610	2.036	0.497	2.72	0.882	0.310	0.326	0.623
	40S	0.145	1.610	2.036	0.497	2.72	0.882	0.310	0.326	0.623
2 (2.375)	80	0.200	1.500	1.767	0.497	3.63	0.765	0.391	0.412	0.605
	80S	0.200	1.500	1.767	0.497	3.63	0.765	0.391	0.412	0.605
	160	0.281	1.338	1.406	0.497	4.86	0.609	0.483	0.508	0.581
	XXS	0.400	1.100	0.950	0.497	6.41	0.411	0.568	0.598	0.549
	5S	0.065	2.245	3.958	0.622	1.61	1.714	0.315	0.265	0.817
	10S	0.109	2.157	3.654	0.622	2.64	1.582	0.499	0.420	0.802
	40	0.154	2.067	3.356	0.622	3.66	1.453	0.666	0.561	0.787
2½ (2.875)	40S	0.154	2.067	3.356	0.622	3.66	1.453	0.666	0.561	0.787
	80	0.218	1.939	2.953	0.622	5.03	1.279	0.868	0.731	0.767
	80S	0.218	1.939	2.953	0.622	5.03	1.279	0.868	0.731	0.767
	160	0.344	1.687	2.235	0.622	7.47	0.968	1.165	0.981	0.728
	XXS	0.436	1.503	1.774	0.622	9.04	0.768	1.312	1.105	0.703
	5S	0.083	2.709	5.764	0.753	2.48	2.496	0.710	0.494	0.988
	10S	0.120	2.635	5.453	0.753	3.53	2.361	0.988	0.687	0.975
3 (3.500)	40	0.203	2.469	4.788	0.753	5.80	2.073	1.530	1.064	0.948
	40S	0.203	2.469	4.788	0.753	5.80	2.073	1.530	1.064	0.948
	80	0.276	2.323	4.238	0.753	7.67	1.835	1.925	1.339	0.924
	80S	0.276	2.323	4.238	0.753	7.67	1.835	1.925	1.339	0.924
	160	0.375	2.125	3.547	0.753	10.02	1.536	2.353	1.637	0.894
	XXS	0.552	1.771	2.463	0.753	13.71	1.067	2.872	1.998	0.844
	5S	0.083	3.334	8.730	0.916	3.03	3.780	1.301	0.744	1.209
3 (3.500)	10S	0.120	3.260	8.347	0.916	4.34	3.614	1.822	1.041	1.196
	40	0.216	3.068	7.393	0.916	7.58	3.201	3.018	1.725	1.164
	40S	0.216	3.068	7.393	0.916	7.58	3.201	3.018	1.725	1.164
	80	0.300	2.900	6.605	0.916	10.26	2.860	3.895	2.226	1.136
	80S	0.300	2.900	6.605	0.916	10.26	2.860	3.895	2.226	1.136
	160	0.438	2.624	5.408	0.916	14.34	2.342	5.040	2.880	1.094
	XXS	0.600	2.300	4.155	0.916	18.60	1.799	5.994	3.425	1.047

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (In.)	Inside Dia. (In.)	Inside Area (In ²)	Sq. Ft. Outside Surface (per ft)	Weight per ft. (Lb.)	Wt. of water per ft. (Lb.)	Moment of Inertia (In ⁴)	Section Modulus (In ³)	Radius of Gyration (In.)
3½ (4.000)	5S	0.083	3.834	11.55	1.047	3.48	5.00	1.960	0.980	1.385
	10S	0.120	3.760	11.10	1.047	4.98	4.81	2.756	1.378	1.373
	40	0.226	3.548	9.89	1.047	9.12	4.28	4.789	2.394	1.337
	40S	0.226	3.548	9.89	1.047	9.12	4.28	4.789	2.394	1.337
	80	0.318	3.364	8.89	1.047	12.52	3.85	6.282	3.141	1.307
	80S	0.318	3.364	8.89	1.047	12.52	3.85	6.282	3.141	1.307
4 (4.500)	5S	0.083	4.334	14.75	1.178	3.92	6.39	2.811	1.249	1.562
	10S	0.120	4.260	14.25	1.178	5.62	6.17	3.964	1.762	1.549
	40	0.237	4.026	12.73	1.178	10.80	5.51	7.234	3.215	1.510
	40S	0.237	4.026	12.73	1.178	10.80	5.51	7.234	3.215	1.510
	80	0.337	3.826	11.50	1.178	15.00	4.98	9.613	4.272	1.477
	80S	0.337	3.826	11.50	1.178	15.00	4.98	9.613	4.272	1.477
	120	0.438	3.624	10.31	1.178	19.02	4.47	11.665	5.184	1.445
	160	0.531	3.438	9.28	1.178	22.53	4.02	13.274	5.900	1.416
5 (5.563)	XXS	0.674	3.152	7.80	1.178	27.57	3.38	15.288	6.794	1.374
	5S	0.109	5.345	22.44	1.456	6.36	9.72	6.949	2.498	1.929
	10S	0.134	5.295	22.02	1.456	7.78	9.53	8.428	3.030	1.920
	40	0.258	5.047	20.01	1.456	14.63	8.66	15.166	5.452	1.878
	40S	0.258	5.047	20.01	1.456	14.63	8.66	15.166	5.452	1.878
	80	0.375	4.813	18.19	1.456	20.80	7.88	20.676	7.433	1.839
	80S	0.375	4.813	18.19	1.456	20.80	7.88	20.676	7.433	1.839
	120	0.500	4.563	16.35	1.456	27.06	7.08	25.738	9.253	1.799
6 (6.625)	160	0.625	4.313	14.61	1.456	32.99	6.33	30.034	10.798	1.760
	XXS	0.750	4.063	12.97	1.456	38.59	5.61	33.643	12.095	1.722
	5S	0.109	6.407	32.24	1.734	7.59	13.96	11.848	3.577	2.304
	10S	0.134	6.357	31.74	1.734	9.30	13.74	14.401	4.348	2.296
	40	0.280	6.065	28.89	1.734	18.99	12.51	28.149	8.498	2.246
	40S	0.280	6.065	28.89	1.734	18.99	12.51	28.149	8.498	2.246
	80	0.432	5.761	26.07	1.734	28.60	11.29	40.501	12.227	2.195
	80S	0.432	5.761	26.07	1.734	28.60	11.29	40.501	12.227	2.195
8 (8.625)	120	0.562	5.501	23.77	1.734	36.43	10.29	49.623	14.981	2.153
	160	0.719	5.187	21.13	1.734	45.39	9.15	59.043	17.824	2.104
	XXS	0.864	4.897	18.83	1.734	53.21	8.16	66.350	20.030	2.060
	5S	0.109	8.407	55.51	2.258	9.92	24.04	26.447	6.133	3.011
	10S	0.148	8.329	54.48	2.258	13.41	23.59	35.424	8.214	2.998
	20	0.250	8.125	51.85	2.258	22.38	22.45	57.737	13.388	2.963
	30	0.277	8.071	51.16	2.258	24.72	22.15	63.369	14.694	2.953
	40	0.322	7.981	50.03	2.258	28.58	21.66	72.508	16.813	2.938
10 (10.750)	40S	0.322	7.981	50.03	2.258	28.58	21.66	72.508	16.813	2.938
	60	0.406	7.813	47.94	2.258	35.67	20.76	88.759	20.582	2.910
	80	0.500	7.625	45.66	2.258	43.43	19.77	105.743	24.520	2.878
	80S	0.500	7.625	45.66	2.258	43.43	19.77	105.743	24.520	2.878
	100	0.594	7.437	43.44	2.258	51.00	18.81	121.517	28.178	2.848
	120	0.719	7.187	40.57	2.258	60.77	17.57	140.717	32.630	2.807
	140	0.812	7.001	38.50	2.258	67.82	16.67	153.761	35.655	2.778
	XXS	0.875	6.875	37.12	2.258	72.49	16.07	162.026	37.571	2.758
160	160	0.906	6.813	36.46	2.258	74.76	15.79	165.930	38.477	2.748
	5S	0.134	10.482	86.29	2.814	15.21	37.37	62.984	11.718	3.754
	10S	0.165	10.420	85.28	2.814	18.67	36.92	76.884	14.304	3.743
	20	0.250	10.250	82.52	2.814	28.06	35.73	113.743	21.162	3.714
	30	0.307	10.136	80.69	2.814	34.27	34.94	137.455	25.573	3.694
	40	0.365	10.020	78.85	2.814	40.52	34.14	160.776	29.912	3.674
	40S	0.365	10.020	78.85	2.814	40.52	34.14	160.776	29.912	3.674
	60	0.500	9.750	74.66	2.814	54.79	32.33	212.005	39.443	3.629
	80	0.594	9.562	71.81	2.814	64.49	31.09	245.250	45.628	3.597
	80S	0.500	9.750	74.66	2.814	54.79	32.33	212.005	39.443	3.629
	100	0.719	9.312	68.10	2.814	77.10	29.49	286.522	53.306	3.556
	120	0.844	9.062	64.50	2.814	89.38	27.93	324.601	60.391	3.515
	140	1.000	8.750	60.13	2.814	104.23	26.04	367.900	68.447	3.466
	160	1.125	8.500	56.75	2.814	115.75	24.57	399.410	74.309	3.427

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (In.)	Inside Dia. (In.)	Inside Area (In ²)	Sq. Ft. Outside Surface (per ft)	Weight per ft. (Lb.)	Wt. of water per ft. (Lb.)	Moment of Inertia (In ⁴)	Section Modulus (In ³)	Radius of Gyration (In.)
12 (12.750)	5S	0.156	12.438	121.5	3.34	21.00	52.6	122.4	19.20	4.45
	10S	0.180	12.390	120.6	3.34	24.19	52.2	140.5	22.03	4.45
	20	0.250	12.250	117.9	3.34	33.41	51.0	191.9	30.10	4.42
	30	0.330	12.090	114.8	3.34	43.81	49.7	248.5	38.98	4.39
	40S	0.375	12.000	113.1	3.34	49.61	49.0	279.4	43.83	4.38
	STD	0.375	12.000	113.1	3.34	49.61	49.0	279.4	43.83	4.38
	40	0.406	11.938	111.9	3.34	53.57	48.5	300.3	47.10	4.37
	80S	0.500	11.750	108.4	3.34	65.48	47.0	361.6	56.73	4.34
	60	0.562	11.626	106.2	3.34	73.22	46.0	400.5	62.83	4.31
	80	0.688	11.374	101.6	3.34	88.71	44.0	475.8	74.64	4.27
	100	0.844	11.062	96.1	3.34	107.42	41.6	562.3	88.21	4.22
	120	1.000	10.750	90.8	3.34	125.61	39.3	641.8	100.68	4.17
	140	1.125	10.500	86.6	3.34	139.81	37.5	700.7	109.92	4.13
	160	1.312	10.126	80.5	3.34	160.42	34.9	781.3	122.56	4.07
14 (14.000)	5S	0.156	13.688	147.2	3.67	23.09	63.7	162.6	23.23	4.90
	10S	0.188	13.624	145.8	3.67	27.76	63.1	194.6	27.80	4.88
	10	0.250	13.500	143.1	3.67	36.75	62.0	255.4	36.48	4.86
	20	0.312	13.376	140.5	3.67	45.65	60.8	314.5	44.92	4.84
	30	0.375	13.250	137.9	3.67	54.62	59.7	372.9	53.27	4.82
	40S	0.375	13.250	137.9	3.67	54.62	59.7	372.9	53.27	4.82
	40	0.438	13.124	135.3	3.67	63.50	58.6	429.6	61.37	4.80
	80S	0.500	13.000	132.7	3.67	72.16	57.5	483.9	69.13	4.78
	60	0.594	12.812	128.9	3.67	85.13	55.8	563.3	80.47	4.74
	80	0.750	12.500	122.7	3.67	106.23	53.1	687.5	98.21	4.69
	100	0.938	12.124	115.4	3.67	130.98	50.0	825.3	117.91	4.63
	120	1.094	11.812	109.6	3.67	150.93	47.4	930.4	132.92	4.58
	140	1.250	11.500	103.9	3.67	170.37	45.0	1027.5	146.78	4.53
	160	1.406	11.188	98.3	3.67	189.29	42.6	1116.9	159.56	4.48
16 (16.000)	5S	0.165	15.670	192.9	4.19	27.93	83.5	257.4	32.17	5.60
	10S	0.188	15.624	191.7	4.19	31.78	83.0	292.0	36.50	5.59
	10	0.250	15.500	188.7	4.19	42.09	81.7	383.8	47.97	5.57
	20	0.312	15.376	185.7	4.19	52.32	80.4	473.4	59.17	5.55
	30	0.375	15.250	182.7	4.19	62.64	79.1	562.2	70.28	5.53
	40S	0.375	15.250	182.7	4.19	62.64	79.1	562.2	70.28	5.53
	40	0.500	15.000	176.7	4.19	82.85	76.5	732.1	91.52	5.48
	80S	0.500	15.000	176.7	4.19	82.85	76.5	732.1	91.52	5.48
	60	0.656	14.688	169.4	4.19	107.60	73.4	932.6	116.57	5.43
	80	0.844	14.312	160.9	4.19	136.74	69.7	1157.7	144.72	5.37
	100	1.031	13.938	152.6	4.19	164.98	66.1	1364.8	170.60	5.31
	120	1.219	13.562	144.5	4.19	192.61	62.5	1556.8	194.60	5.24
	140	1.438	13.124	135.3	4.19	223.85	58.6	1761.2	220.15	5.17
	160	1.594	12.812	128.9	4.19	245.48	55.8	1894.8	236.86	5.13
18 (18.000)	5S	0.165	17.670	245.2	4.71	31.46	106.2	367.7	40.86	6.31
	10S	0.188	17.624	243.9	4.71	35.80	105.6	417.4	46.37	6.30
	10	0.250	17.500	240.5	4.71	47.44	104.1	549.3	61.03	6.28
	20	0.312	17.376	237.1	4.71	58.99	102.7	678.4	75.38	6.26
	30	0.438	17.124	230.3	4.71	82.23	99.7	932.5	103.61	6.21
	40S	0.375	17.250	233.7	4.71	70.65	101.2	806.8	89.65	6.23
	40	0.562	16.876	223.7	4.71	104.76	96.9	1171.8	130.20	6.17
	80S	0.500	17.000	227.0	4.71	93.54	98.3	1053.4	117.05	6.19
	60	0.750	16.500	213.8	4.71	138.30	92.6	1515.0	168.34	6.11
	80	0.938	16.124	204.2	4.71	171.08	88.4	1835.6	203.95	6.04
	100	1.156	15.688	193.3	4.71	208.15	83.7	2180.2	242.25	5.97
	120	1.375	15.250	182.7	4.71	244.37	79.1	2498.7	277.64	5.90
	140	1.562	14.876	173.8	4.71	274.48	75.3	2749.8	305.53	5.84
	160	1.781	14.438	163.7	4.71	308.79	70.9	3020.7	335.64	5.77

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (In.)	Inside Dia. (In.)	Inside Area (In ²)	Sq. Ft. Outside Surface (per ft)	Weight per ft. (Lb.)	Wt. of water per ft. (Lb.)	Moment of Inertia (In ⁴)	Section Modulus (In ³)	Radius of Gyration (In.)
20 (20.000)	5S	0.188	19.624	302.5	5.24	39.82	131.0	574.3	57.43	7.01
	10S	0.218	19.564	300.6	5.24	46.10	130.2	663.0	66.30	7.00
	10	0.250	19.500	298.6	5.24	52.78	129.3	756.6	75.66	6.98
	20	0.375	19.250	291.0	5.24	78.67	126.0	1113.8	111.38	6.94
	30	0.500	19.000	283.5	5.24	104.23	122.8	1457.2	145.72	6.90
	40S	0.375	19.250	291.0	5.24	78.67	126.0	1113.8	111.38	6.94
	40	0.594	18.812	277.9	5.24	123.23	120.4	1706.8	170.68	6.87
	80S	0.500	19.000	283.5	5.24	104.23	122.8	1457.2	145.72	6.90
	60	0.812	18.376	265.2	5.24	166.56	114.8	2257.3	225.73	6.79
	80	1.031	17.938	252.7	5.24	209.06	109.4	2772.3	277.23	6.72
	100	1.281	17.438	238.8	5.24	256.34	103.4	3315.9	331.59	6.63
	120	1.500	17.000	227.0	5.24	296.65	98.3	3755.1	375.51	6.56
	140	1.750	16.500	213.8	5.24	341.41	92.6	4216.7	421.67	6.48
	160	1.969	16.062	202.6	5.24	379.53	87.7	4588.0	458.80	6.41
22 (22.000)	10	0.250	21.500	363.1	5.76	58.13	157.2	1010.5	91.87	7.69
	20	0.375	21.250	354.7	5.76	86.69	153.6	1490.1	135.46	7.65
	30	0.500	21.000	346.4	5.76	114.92	150.0	1953.0	177.54	7.60
	60	0.875	20.250	322.1	5.76	197.60	139.5	3245.7	295.07	7.48
	80	1.125	19.750	306.4	5.76	251.05	132.7	4031.5	366.50	7.39
	100	1.375	19.250	291.0	5.76	303.16	126.0	4759.7	432.70	7.31
	120	1.625	18.750	276.1	5.76	353.94	119.6	5433.4	493.95	7.23
	140	1.875	18.250	261.6	5.76	403.38	113.3	6055.3	550.48	7.15
	160	2.125	17.750	247.4	5.76	451.49	107.1	6628.1	602.55	7.07
24 (24.000)	5S	0.218	23.564	436.1	6.28	55.42	188.8	1151.9	95.99	8.41
	10S	0.250	23.500	433.7	6.28	63.47	187.8	1315.7	109.64	8.40
	10	0.250	23.500	433.7	6.28	63.47	187.8	1315.7	109.64	8.40
	20	0.375	23.250	424.6	6.28	94.71	183.8	1942.8	161.90	8.35
	30	0.562	22.876	411.0	6.28	140.81	178.0	2843.9	236.99	8.29
	40S	0.375	23.250	424.6	6.28	94.71	183.8	1942.8	161.90	8.35
	40	0.688	22.624	402.0	6.28	171.45	174.1	3426.7	285.56	8.25
	80S	0.500	23.000	415.5	6.28	125.61	179.9	2550.0	212.50	8.31
	60	0.969	22.062	382.3	6.28	238.57	165.5	4658.0	388.17	8.15
	80	1.219	21.562	365.1	6.28	296.86	158.1	5677.2	473.10	8.07
	100	1.531	20.938	344.3	6.28	367.74	149.1	6853.5	571.12	7.96
	120	1.812	20.376	326.1	6.28	429.79	141.2	7826.6	652.21	7.87
	140	2.062	19.876	310.3	6.28	483.57	134.3	8627.2	718.94	7.79
	160	2.344	19.312	292.9	6.28	542.64	126.8	9460.7	788.39	7.70

Notes:

- Weights are given in pounds per linear foot (kilograms per meter) and are for carbon steel pipe with plain ends
- The different grades of stainless steel permit considerable variations in weight. The ferritic stainless steels may be about 5% less, and the austenitic stainless steels about 2% greater, than the values shown in this table, which are based on weights for carbon steel.

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (mm)	Inside Dia. (mm)	Inside Area (mm ²)	Sq. m. Outside Surface (per m)	Weight per m (kg)	Wt. of water per m (kg)	Moment of Inertia (cm ⁴)	Section Modulus (cm ³)	Radius of Gyration (mm)
DN 15 (21.3)	10S	2.11	17.12	230	0.067	1.00	0.230	0.60	0.56	6.84
	40	2.77	15.80	196	0.067	1.27	0.196	0.71	0.67	6.64
	40S	2.77	15.80	196	0.067	1.27	0.196	0.71	0.67	6.64
	80	3.73	13.87	151	0.067	1.62	0.151	0.84	0.78	6.36
	80S	3.73	13.87	151	0.067	1.62	0.151	0.84	0.78	6.36
	160	4.78	11.79	109	0.067	1.95	0.109	0.92	0.86	6.09
	XXS	7.47	6.40	32	0.067	2.55	0.032	1.01	0.95	5.57
DN 20 (26.7)	5S	1.65	23.37	429	0.084	1.02	0.428	1.02	0.76	8.87
	10S	2.11	22.45	396	0.084	1.28	0.395	1.24	0.93	8.72
	40	2.87	20.93	344	0.084	1.69	0.344	1.54	1.16	8.48
	40S	2.87	20.93	344	0.084	1.69	0.344	1.54	1.16	8.48
	80	3.91	18.85	279	0.084	2.20	0.279	1.86	1.40	8.17
	80S	3.91	18.85	279	0.084	2.20	0.279	1.86	1.40	8.17
	160	5.56	15.54	190	0.084	2.90	0.190	2.20	1.65	7.72
DN 25 (33.4)	XXS	7.82	11.02	95	0.084	3.64	0.095	2.41	1.81	7.22
	5S	1.65	30.10	712	0.105	1.29	0.711	2.08	1.25	11.24
	10S	2.77	27.86	610	0.105	2.09	0.609	3.15	1.89	10.88
	40	3.38	26.64	558	0.105	2.50	0.557	3.64	2.18	10.68
	40S	3.38	26.64	558	0.105	2.50	0.557	3.64	2.18	10.68
	80	4.55	24.31	464	0.105	3.24	0.464	4.40	2.63	10.33
	80S	4.55	24.31	464	0.105	3.24	0.464	4.40	2.63	10.33
DN 32 (42.2)	160	6.35	20.70	337	0.105	4.24	0.336	5.21	3.12	9.83
	XXS	9.09	15.21	182	0.105	5.45	0.182	5.85	3.50	9.18
	5S	1.65	38.86	1186	0.132	1.65	1.185	4.32	2.05	14.34
	10S	2.77	36.63	1054	0.132	2.69	1.052	6.68	3.17	13.96
	40	3.56	35.05	965	0.132	3.39	0.964	8.11	3.85	13.71
	40S	3.56	35.05	965	0.132	3.39	0.964	8.11	3.85	13.71
	80	4.85	32.46	828	0.132	4.47	0.827	10.07	4.78	13.30
DN 40 (48.3)	80S	4.85	32.46	828	0.132	4.47	0.827	10.07	4.78	13.30
	160	6.35	29.46	682	0.132	5.61	0.681	11.82	5.61	12.86
	XXS	9.70	22.76	407	0.132	7.77	0.406	14.20	6.74	11.98
	5S	1.65	44.96	1587	0.152	1.90	1.586	6.57	2.72	16.49
	10S	2.77	42.72	1434	0.152	3.11	1.432	10.28	4.26	16.12
	40	3.68	40.89	1313	0.152	4.05	1.312	12.90	5.35	15.82
	40S	3.68	40.89	1313	0.152	4.05	1.312	12.90	5.35	15.82
DN 50 (60.3)	80	5.08	38.10	1140	0.152	5.41	1.139	16.29	6.75	15.37
	80S	5.08	38.10	1140	0.152	5.41	1.139	16.29	6.75	15.37
	160	7.14	33.99	907	0.152	7.25	0.906	20.08	8.32	14.76
	XXS	10.15	27.94	613	0.152	9.55	0.612	23.64	9.80	13.94
	5S	1.65	57.02	2554	0.190	2.39	2.551	13.11	4.35	20.76
	10S	2.77	54.79	2358	0.190	3.93	2.355	20.78	6.89	20.38
	40	3.91	52.50	2165	0.190	5.44	2.162	27.72	9.19	20.00
DN 65 (73.0)	40S	3.91	52.50	2165	0.190	5.44	2.162	27.72	9.19	20.00
	80	5.54	49.25	1905	0.190	7.48	1.903	36.14	11.98	19.47
	80S	5.54	49.25	1905	0.190	7.48	1.903	36.14	11.98	19.47
	160	8.74	42.85	1442	0.190	11.11	1.440	48.47	16.07	18.50
	XXS	11.07	38.18	1145	0.190	13.44	1.143	54.59	18.10	17.85
	5S	2.11	68.81	3719	0.229	3.69	3.714	29.56	8.10	25.09
	10S	3.05	66.93	3518	0.229	5.26	3.514	41.10	11.26	24.77
DN 80 (88.9)	40	5.16	62.71	3089	0.229	8.63	3.085	63.68	17.44	24.07
	40S	5.16	62.71	3089	0.229	8.63	3.085	63.68	17.44	24.07
	80	7.01	59.00	2734	0.229	11.41	2.731	80.11	21.94	23.47
	80S	7.01	59.00	2734	0.229	11.41	2.731	80.11	21.94	23.47
	160	9.53	53.98	2288	0.229	14.92	2.285	97.95	26.83	22.70
	XXS	14.02	44.98	1589	0.229	20.39	1.587	119.52	32.73	21.44
	5S	2.11	84.68	5632	0.279	4.52	5.626	54.17	12.19	30.70
DN 80 (88.9)	10S	3.05	82.80	5385	0.279	6.46	5.379	75.86	17.07	30.38
	40	5.49	77.93	4769	0.279	11.29	4.764	125.62	28.26	29.56
	40S	5.49	77.93	4769	0.279	11.29	4.764	125.62	28.26	29.56
	80	7.62	73.66	4261	0.279	15.27	4.256	162.14	36.48	28.87
	80S	7.62	73.66	4261	0.279	15.27	4.256	162.14	36.48	28.87
	160	11.13	66.65	3489	0.279	21.35	3.485	209.80	47.20	27.78
	XXS	15.24	58.42	2680	0.279	27.68	2.677	249.49	56.13	26.60

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (mm)	Inside Dia. (mm)	Inside Area (mm ²)	Sq. m. Outside Surface (per m)	Weight per M (kg)	Wt. of water per M (kg)	Moment of Inertia (cm ⁴)	Section Modulus (cm ³)	Radius of Gyration (mm)
DN 90 (101.6)	5S	2.11	97.38	7448	0.319	5.18	7.439	81.59	16.06	35.19
	10S	3.05	95.50	7164	0.319	7.41	7.155	114.71	22.58	34.86
	40	5.74	90.12	6379	0.319	13.57	6.371	199.33	39.24	33.96
	40S	5.74	90.12	6379	0.319	13.57	6.371	199.33	39.24	33.96
	80	8.08	85.45	5734	0.319	18.64	5.727	261.47	51.47	33.19
	80S	8.08	85.45	5734	0.319	18.64	5.727	261.47	51.47	33.19
DN 100 (114.3)	5S	2.11	110.08	9518	0.359	5.84	9.506	116.98	20.47	39.68
	10S	3.05	108.20	9196	0.359	8.37	9.184	164.98	28.87	39.35
	40	6.02	102.26	8213	0.359	16.08	8.203	301.12	52.69	38.35
	40S	6.02	102.26	8213	0.359	16.08	8.203	301.12	52.69	38.35
	80	8.56	97.18	7417	0.359	22.32	7.408	400.12	70.01	37.51
	80S	8.56	97.18	7417	0.359	22.32	7.408	400.12	70.01	37.51
	120	11.13	92.05	6655	0.359	28.32	6.647	485.54	84.96	36.69
	160	13.49	87.33	5989	0.359	33.54	5.982	552.53	96.68	35.96
	XXS	17.12	80.06	5034	0.359	41.03	5.028	636.32	111.34	34.89
DN 125 (141.3)	5S	2.77	135.76	14476	0.444	9.46	14.46	289.24	40.94	48.99
	10S	3.40	134.49	14207	0.444	11.56	14.19	350.78	49.65	48.78
	40	6.55	128.19	12907	0.444	21.77	12.89	631.26	89.35	47.70
	40S	6.55	128.19	12907	0.444	21.77	12.89	631.26	89.35	47.70
	80	9.53	122.25	11738	0.444	30.97	11.72	860.61	121.81	46.72
	80S	9.53	122.25	11738	0.444	30.97	11.72	860.61	121.81	46.72
	120	12.70	115.90	10550	0.444	40.28	10.54	1071.3	151.64	45.69
	160	15.88	109.55	9426	0.444	49.12	9.414	1250.1	176.94	44.70
	XXS	19.05	103.20	8365	0.444	57.43	8.355	1400.4	198.21	43.75
DN 150 (168.3)	5S	2.77	162.74	20800	0.529	11.31	20.78	493.18	58.62	58.53
	10S	3.40	161.47	20477	0.529	13.83	20.45	599.42	71.24	58.31
	40	7.11	154.05	18639	0.529	28.26	18.62	1171.7	139.26	57.04
	40S	7.11	154.05	18639	0.529	28.26	18.62	1171.7	139.26	57.04
	80	10.97	146.33	16817	0.529	42.56	16.80	1685.8	200.36	55.76
	80S	10.97	146.33	16817	0.529	42.56	16.80	1685.8	200.36	55.76
	120	14.27	139.73	15334	0.529	54.21	15.31	2065.5	245.49	54.69
	160	18.26	131.75	13633	0.529	67.57	13.62	2457.6	292.09	53.44
	XXS	21.95	124.38	12151	0.529	79.22	12.14	2761.7	328.24	52.32
DN 200 (219.1)	5S	2.77	213.54	35813	0.688	14.78	35.77	1100.8	100.50	76.49
	10S	3.76	211.56	35152	0.688	19.97	35.11	1474.5	134.61	76.15
	20	6.35	206.38	33451	0.688	33.32	33.41	2403.2	219.40	75.25
	30	7.04	205.00	33008	0.688	36.82	32.97	2637.6	240.80	75.02
	40	8.18	202.72	32276	0.688	42.55	32.24	3018.0	275.52	74.63
	40S	8.18	202.72	32276	0.688	42.55	32.24	3018.0	275.52	74.63
	60	10.31	198.45	30931	0.688	53.09	30.89	3694.5	337.28	73.91
	80	12.70	193.68	29460	0.688	64.64	29.42	4401.4	401.82	73.11
	80S	12.70	193.68	29460	0.688	64.64	29.42	4401.4	401.82	73.11
	100	15.09	188.90	28026	0.688	75.92	27.99	5057.9	461.75	72.33
	120	18.26	182.55	26173	0.688	90.44	26.14	5857.1	534.72	71.30
	140	20.62	177.83	24836	0.688	100.93	24.81	6400.1	584.28	70.55
	XXS	22.23	174.63	23950	0.688	107.93	23.92	6744.1	615.69	70.05
	160	23.01	173.05	23520	0.688	111.27	23.49	6906.6	630.52	69.80
DN 250 (273.1)	5S	3.40	266.24	55673	0.858	22.61	55.61	2621.6	192.02	95.35
	10S	4.19	264.67	55017	0.858	27.78	54.95	3200.2	234.40	95.08
	20	6.35	260.35	53236	0.858	41.76	53.17	4734.4	346.78	94.33
	30	7.80	257.45	52059	0.858	51.01	52.00	5721.4	419.07	93.83
	40	9.27	254.51	50874	0.858	60.29	50.81	6692.0	490.17	93.33
	40S	9.27	254.51	50874	0.858	60.29	50.81	6692.0	490.17	93.33
	60	12.70	247.65	48169	0.858	81.53	48.11	8824.4	646.35	92.17
	80	15.09	242.88	46329	0.858	95.98	46.27	10208	747.71	91.37
	80S	12.70	247.65	48169	0.858	81.53	48.11	8824.4	646.35	92.17
	100	18.26	236.53	43938	0.858	114.71	43.89	11926	873.54	90.32
	120	21.44	230.18	41611	0.858	133.01	41.56	13511	989.64	89.29
	140	25.40	222.25	38795	0.858	155.10	38.75	15313	1121.64	88.03
	160	28.58	215.90	36610	0.858	172.27	36.57	16625	1217.71	87.03

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (mm)	Inside Dia. (mm)	Inside Area (mm ²)	Sq. m. Outside Surface (per m)	Weight per M (kg)	Wt. of water per M (kg)	Moment of Inertia (cm ⁴)	Section Modulus (cm ³)	Radius of Gyration (mm)
DN 300 (323.9)	5S	3.96	315.93	78390	1.017	31.24	78.29	5095.6	314.69	113.12
	10S	4.57	314.71	77786	1.017	35.98	77.69	5846.2	361.05	112.91
	20	6.35	311.15	76038	1.017	49.71	75.95	7986.4	493.22	112.29
	30	8.38	307.09	74065	1.017	65.19	73.97	10344	638.82	111.59
	40S	9.53	304.80	72966	1.017	73.86	72.88	11630	718.23	111.20
	STD	9.53	304.80	72966	1.017	73.86	72.88	11630	718.23	111.20
	40	10.31	303.23	72214	1.017	79.71	72.13	12499	771.90	110.93
	80S	12.70	298.45	69958	1.017	97.44	69.87	15053	929.60	110.11
	60	14.27	295.30	68489	1.017	108.93	68.41	16671	1029.56	109.58
	80	17.48	288.90	65552	1.017	132.05	65.47	19805	1223.07	108.51
	100	21.44	280.98	62005	1.017	159.87	61.93	23406	1445.48	107.20
	120	25.40	273.05	58557	1.017	186.92	58.49	26715	1649.84	105.91
	140	28.58	266.70	55865	1.017	208.08	55.80	29167	1801.25	104.90
	160	33.32	257.20	51956	1.017	238.69	51.89	32522	2008.43	103.40
DN 350 (355.6)	5S	3.96	347.68	94938	1.117	34.34	94.82	6768.2	380.66	124.35
	10S	4.78	346.05	94052	1.117	41.36	93.94	8100.6	455.60	124.06
	10	6.35	342.90	92348	1.117	54.69	92.24	10629	597.82	123.52
	20	7.92	339.75	90659	1.117	67.91	90.55	13089	736.17	122.97
	30	9.53	336.55	88959	1.117	81.33	88.85	15520	872.86	122.42
	40S	9.53	336.55	88959	1.117	81.33	88.85	15520	872.86	122.42
	40	11.13	333.35	87275	1.117	94.55	87.17	17882	1005.72	121.87
	80S	12.70	330.20	85634	1.117	107.40	85.53	20141	1132.78	121.33
	60	15.09	325.43	83175	1.117	126.72	83.07	23445	1318.60	120.52
	80	19.05	317.50	79173	1.117	158.11	79.08	28616	1609.44	119.19
	100	23.83	307.95	74482	1.117	194.98	74.39	34354	1932.16	117.62
	120	27.79	300.03	70698	1.117	224.66	70.61	38727	2178.11	116.33
	140	31.75	292.10	67012	1.117	253.58	66.93	42767	2405.32	115.06
	160	35.71	284.18	63426	1.117	281.72	63.35	46491	2614.77	113.81
DN 400 (406.4)	5S	4.19	398.02	124422	1.277	41.56	124.27	10713	527.19	142.23
	10S	4.78	396.85	123693	1.277	47.34	123.54	12153	598.09	142.02
	10	6.35	393.70	121737	1.277	62.65	121.59	15974	786.10	141.48
	20	7.92	390.55	119797	1.277	77.83	119.65	19703	969.65	140.93
	30	9.53	387.35	117842	1.277	93.27	117.70	23402	1151.67	140.38
	40S	9.53	387.35	117842	1.277	93.27	117.70	23402	1151.67	140.38
	40	12.70	381.00	114010	1.277	123.31	113.87	30474	1499.69	139.28
	80S	12.70	381.00	114010	1.277	123.31	113.87	30474	1499.69	139.28
	60	16.66	373.08	109316	1.277	160.13	109.18	38817	1910.28	137.94
	80	21.44	363.53	103791	1.277	203.54	103.67	48189	2371.52	136.33
	100	26.19	354.03	98437	1.277	245.57	98.32	56807	2795.62	134.76
	120	30.96	344.48	93198	1.277	286.66	93.08	64799	3188.92	133.21
	140	36.53	333.35	87275	1.277	333.21	87.17	73307	3607.63	131.42
	160	40.49	325.43	83175	1.277	365.38	83.07	78870	3881.40	130.18
DN 450 (457)	5S	4.19	448.82	158209	1.436	46.79	158.02	15306	669.53	160.19
	10S	4.78	447.65	157387	1.436	53.31	157.20	17372	759.94	159.99
	10	6.35	444.50	155180	1.436	70.57	154.99	22863	1000.12	159.44
	20	7.92	441.35	152988	1.436	87.71	152.80	28238	1235.26	158.89
	30	11.13	434.95	148583	1.436	122.38	148.40	38813	1697.85	157.78
	40S	9.53	438.15	150778	1.436	105.17	150.59	33583	1469.09	158.33
	40	14.27	428.65	144311	1.436	155.81	144.14	48774	2133.58	156.70
	80S	12.70	431.80	146439	1.436	139.16	146.26	43848	1918.10	157.24
	60	19.05	419.10	137952	1.436	205.75	137.78	63061	2758.55	155.08
	80	23.83	409.55	131736	1.436	254.57	131.58	76403	3342.23	153.47
	100	29.36	398.48	124708	1.436	309.64	124.56	90749	3969.79	151.64
	120	34.93	387.35	117842	1.436	363.58	117.70	104006	4549.68	149.83
	140	39.67	377.85	112132	1.436	408.28	112.00	114457	5006.85	148.30
	160	45.24	366.73	105627	1.436	459.39	105.50	125734	5500.15	146.55

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

PROPERTIES OF PIPE (ASME B36.10M-2004 / ASME B36.19M-2004) FOR REFERENCE ONLY

Nominal Pipe Size (OD)	Pipe Schedule	Wall Thickness (mm)	Inside Dia. (mm)	Inside Area (mm ²)	Sq. m. Outside Surface (per m)	Weight per M (kg)	Wt. of water per M (kg)	Moment of Inertia (cm ⁴)	Section Modulus (cm ³)	Radius of Gyration (mm)
DN 500 (508)	5S	4.78	498.45	195135	1.596	59.32	194.90	23905	941.15	177.95
	10S	5.54	496.93	193943	1.596	68.89	193.71	27595	1086.41	177.68
	10	6.35	495.30	192676	1.596	78.56	192.44	31493	1239.90	177.40
	20	9.53	488.95	187768	1.596	117.15	187.54	46358	1825.13	176.29
	30	12.70	482.60	182922	1.596	155.13	182.70	60655	2388.00	175.20
	40S	9.53	488.95	187768	1.596	117.15	187.54	46358	1825.13	176.29
	40	15.09	477.83	179320	1.596	183.43	179.10	71041	2796.90	174.38
	80S	12.70	482.60	182922	1.596	155.13	182.70	60655	2388.00	175.20
	60	20.62	466.75	171104	1.596	247.84	170.90	93958	3699.11	172.49
	80	26.19	455.63	163045	1.596	311.19	162.85	115394	4543.06	170.62
	100	32.54	442.93	154082	1.596	381.55	153.90	138018	5433.77	168.52
	120	38.10	431.80	146439	1.596	441.52	146.26	156301	6153.57	166.70
	140	44.45	419.10	137952	1.596	508.15	137.78	175514	6909.98	164.66
	160	50.01	407.98	130725	1.596	564.85	130.57	190969	7518.46	162.91
DN 550 (559)	10	6.35	546.10	234226	1.756	86.55	233.94	42062	1505.42	195.36
	20	9.53	539.75	228811	1.756	129.14	228.53	62021	2219.79	194.25
	30	12.70	533.40	223459	1.756	171.10	223.19	81289	2909.40	193.15
	60	22.23	514.35	207783	1.756	294.27	207.53	135099	4835.32	189.90
	80	28.58	501.65	197648	1.756	373.85	197.41	167804	6005.84	187.76
	100	34.93	488.95	187768	1.756	451.45	187.54	198116	7090.76	185.65
	120	41.28	476.25	178140	1.756	527.05	177.92	226157	8094.36	183.58
	140	47.63	463.55	168766	1.756	600.67	168.56	252042	9020.80	181.53
	160	53.98	450.85	159645	1.756	672.30	159.45	275884	9874.14	179.52
DN 600 (610)	5S	5.54	598.53	281357	1.915	82.58	281.02	47946	1573.01	213.61
	10S	6.35	596.90	279830	1.915	94.53	279.49	54763	1796.69	213.32
	10	6.35	596.90	279830	1.915	94.53	279.49	54763	1796.69	213.32
	20	9.53	590.55	273908	1.915	141.12	273.58	80866	2653.08	212.21
	30	14.27	581.05	265167	1.915	209.65	264.85	118374	3883.66	210.57
	40S	9.53	590.55	273908	1.915	141.12	273.58	80866	2653.08	212.21
	40	17.48	574.65	259357	1.915	255.43	259.04	142631	4679.50	209.47
	80S	12.70	584.20	268049	1.915	187.07	267.72	106140	3482.28	211.11
	60	24.61	560.38	246632	1.915	355.28	246.33	193883	6360.98	207.03
	80	30.96	547.68	235579	1.915	442.11	235.29	236305	7752.79	204.90
	100	38.89	531.83	222141	1.915	547.74	221.87	285265	9359.06	202.27
	120	46.02	517.55	210376	1.915	640.07	210.12	325769	10687.93	199.94
	140	52.37	504.85	200178	1.915	720.19	199.94	359095	11781.31	197.90
	160	59.54	490.53	188979	1.915	808.27	188.75	393786	12919.47	195.64

Notes:

- Weights are given in pounds per linear foot (kilograms per meter) and are for carbon steel pipe with plain ends
- The different grades of stainless steel permit considerable variations in weight. The ferritic stainless steels may be about 5% less, and the austenitic stainless steels about 2% greater, than the values shown in this table, which are based on weights for carbon steel.

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.
TENTH EDITION APPENDIX D

STANDARD AVAILABLE SHEET GAUGES

GAUGE	STAINLESS STEEL				NICKEL ALLOYS			
	THICKNESS		WEIGHT		THICKNESS		WEIGHT	
	(In.)	(mm)	(Lb/Sq.Ft.)	(Kg/Sq.M)	(In.)	(mm)	(Lb/Sq.Ft.)	(Kg/Sq.M)
7	0.1874	4.76	7.871	38.43	0.187	4.75	8.590	41.94
8	0.1650	4.19	6.930	33.84	-	-	-	-
10	0.1350	3.43	5.670	27.68	0.140	3.56	6.431	31.40
11	0.1200	3.05	5.040	24.61	0.125	3.18	5.742	28.04
12	0.1054	2.68	4.427	21.62	0.109	2.77	5.007	24.45
13	0.0900	2.29	3.780	18.46	0.093	2.36	4.272	20.86
14	0.0751	1.91	3.154	15.40	0.078	1.98	3.583	17.49
16	0.0595	1.51	2.499	12.20	0.062	1.57	2.848	13.91
18	0.0480	1.22	2.016	9.84	0.050	1.27	2.297	11.22
19	0.0420	1.07	1.764	8.61	0.043	1.09	1.975	9.64
20	0.0355	0.90	1.491	7.28	0.037	0.940	1.700	8.30
22	0.0293	0.74	1.231	6.01	0.031	0.787	1.424	6.95
24	0.0235	0.60	0.987	4.82	0.025	0.635	1.148	5.61
26	0.0178	0.45	0.748	3.65	0.018	0.457	0.827	4.04

(Source: Joseph T. Ryerson & Son, Inc.)

EUROPEAN COMPARABLE FOR COMMON EXPANSION JOINT MATERIALS

UNITED STATES	EUROPEAN COMMUNITY		
ASTM Std.	EN Std. / Remarks	Material Number	Steel Name
A240 304	EN 10028-7 / (Flat products – stainless steels)	1.4301	X5CrNi18-10
A240 304 L	EN 10028-7 / (Flat products – stainless steels)	1.4306	X2CrNi19-11
A240 316	EN 10028-7 / (Flat products – stainless steels)	1.4401	X5CrNiMo17-12-2
A240 316 L	EN 10028-7 / (Flat products – stainless steels)	1.4404	X2CrNiMo17-12-2
A240 321	EN 10028-7 / (Flat products – stainless steels)	1.4541	X6CrNiTi18-10
A105 CS	EN 10028-2 / Flat products made of steels for pressure purposes – Part 2: Non-alloy and alloy steels with specified elevated temperature properties		
A182 F304	EN 10222-5 / (steel forgings – SS etc.)	1.4301	X5CrNi18-10
A182 F316	EN 10222-5 / (steel forgings – SS etc.)	1.4401	X5CrNiMo17-12-2
A182 F11	EN 10222-2 / (steel forgings – steels for elevated temperatures)	1.7335	13CrMo4-5
A182 F12	EN 10222-2 / (steel forgings – steels for elevated temperatures)	1.7335	13CrMo4-5
A53-B (smls)	no EN Std. available		
A106-B	EN 10216-2 / Seamless steels tubes for pressure purposes – Part 2: Non-alloy and alloy steel tubes with specified elevated temperature properties		
A312 304	EN 10217-2 / (welded steel tubes)	1.4301	X5CrNi18-10
A312 316	EN 10217-2 / (welded steel tubes)	1.4401	X5CrNiMo17-12-2
A335 P11	EN 10216-2 / (seamless steel tubes)	1.7335	13CrMo4-5
A335 P12	EN 10216-2 / (seamless steel tubes)	1.7335	13CrMo4-5

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D

TABLE IV

Thermal Expansion of Pipe in Inches per 100 Feet

Temp. Degrees F.	Carbon C-Mo. 3Cr-Mo Steels	5CR-Mo through 9Cr-Mo Steels	Austenitic Stainless Steels 18Cr-8Ni	310 SS 25 Cr- 20Ni	Alloy 400	Cu-30Ni	Copper	Nickel 200	Alloy 800, 825	Alloy 600, 625,691	Aluminum	Temp Degrees F.
-325	-2.37	-2.22	-3.85	...	-2.62	-3.15	...	...	...	...	-4.68	-325
-300	-2.24	-2.10	-3.63	...	-2.50	-2.87	...	-2.44	...	...	-4.46	-300
-275	-2.11	-1.98	-3.41	...	-2.38	-2.70	...	-2.35	...	...	-4.21	-275
-250	-1.98	-1.86	-3.19	...	-2.26	-2.53	...	-2.25	...	-2.30	-3.97	-250
-225	-1.85	-1.74	-2.96	...	-2.14	-2.36	...	-2.13	...	-2.17	-3.71	-225
-200	-1.71	-1.62	-2.73	...	-2.02	-2.19	...	-2.01	...	-2.04	-3.44	-200
-175	-1.58	-1.50	-2.50	...	-1.90	-2.12	...	-1.83	...	-1.87	-3.16	-175
-150	-1.45	-1.37	-2.27	...	-1.79	-1.95	...	-1.65	...	-1.70	-2.88	-150
-125	-1.30	-1.23	-2.01	...	-1.59	-1.74	...	-1.47	...	-1.54	-2.57	-125
-100	-1.15	-1.08	-1.75	...	-1.38	-1.53	-1.83	-1.29	...	-1.37	-2.27	-100
-75	-1.00	-0.94	-1.50	...	-1.18	-1.33	-1.57	-1.11	...	-1.17	-1.97	-75
-50	-0.84	-0.79	-1.24	...	-0.98	-1.13	-1.31	-0.93	...	-0.97	-1.67	-50
-25	-0.68	-0.63	-0.98	...	-0.77	-0.89	-1.05	-0.75	...	-0.76	-1.32	-25
0	-0.49	-0.46	-0.72	...	-0.57	-0.66	-0.79	-0.56	...	-0.56	-0.97	0
25	-0.32	-0.30	-0.46	...	-0.37	-0.42	-0.51	-0.36	...	-0.36	-0.63	25
50	-0.14	-0.13	-0.21	...	-0.20	-0.19	-0.22	-0.16	...	-0.16	-0.28	50
70	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	70
100	0.23	0.22	0.34	0.32	0.28	0.31	0.34	0.25	0.28	0.26	0.46	100
125	0.42	0.40	0.62	0.58	0.52	0.56	0.62	0.47	0.52	0.48	0.85	125
150	0.61	0.58	0.90	0.84	0.75	0.82	0.90	0.69	0.76	0.70	1.23	150
175	0.80	0.76	1.18	1.10	0.99	1.07	1.18	0.92	0.99	0.92	1.62	175
200	0.99	0.94	1.46	1.37	1.22	1.33	1.48	1.15	1.23	1.15	2.00	200
225	1.21	1.13	1.75	1.64	1.46	1.59	1.77	1.38	1.49	1.38	2.41	225
250	1.40	1.33	2.03	1.91	1.71	1.86	2.05	1.61	1.76	1.61	2.83	250
275	1.61	1.52	2.32	2.18	1.96	2.13	2.34	1.85	2.03	1.85	3.24	275
300	1.82	1.71	2.61	2.45	2.21	2.40	2.62	2.08	2.30	2.09	3.67	300
325	2.04	1.90	2.90	2.72	2.44	2.68	2.91	2.32	2.59	2.32	4.09	325
350	2.26	2.10	3.20	2.99	2.68	2.96	3.19	2.56	2.88	2.56	4.52	350
375	2.48	2.30	3.50	3.26	2.91	3.24	3.48	2.80	3.18	2.80	4.95	375
400	2.70	2.50	3.80	3.53	3.25	3.52	3.88	3.05	3.48	3.05	5.39	400
425	2.93	2.72	4.10	3.80	3.52	...	4.17	3.30	3.76	3.29	5.83	425
450	3.16	2.93	4.41	4.07	3.79	...	4.47	3.55	4.04	3.53	6.28	450
475	3.39	3.14	4.71	4.34	4.06	...	4.76	3.80	4.31	3.78	6.72	475
500	3.62	3.35	5.01	4.61	4.33	...	5.06	4.05	4.59	4.02	7.17	500
525	3.86	3.58	5.31	4.88	4.61	...	5.35	4.31	4.87	4.27	7.63	525
550	4.11	3.80	5.62	5.15	4.90	...	5.64	4.56	5.16	4.52	8.10	550
575	4.35	4.02	5.93	5.42	5.18	...	...	4.83	5.44	4.77	8.56	575
600	4.60	4.24	6.24	5.69	5.46	...	...	5.09	5.72	5.02	9.03	600
625	4.86	4.47	6.55	5.96	5.75	...	...	5.35	6.01	5.27	...	625
650	5.11	4.69	6.87	6.23	6.05	...	...	5.62	6.30	5.53	...	650

STANDARDS OF THE EXPANSION JOINT MANUFACTURERS ASSOCIATION, INC.

TENTH EDITION APPENDIX D

TABLE IV (continued) Thermal Expansion of Pipe in Inches per 100 Feet

Temp. Degrees F.	Carbon C-Mo. 3Cr-Mo Steels	5CR-Mo through 9Cr-Mo Steels	Austenitic Stainless Steels 18Cr-8Ni	310 SS 25 Cr- 20Ni	Alloy 400	Cu-30Ni	Copper	Nickel 200	Alloy 800, 825	Alloy 600, 625,691	Aluminum	Temp Degrees F.
675	5.37	4.92	7.18	6.50	6.34	...	...	5.89	6.58	5.79	...	675
700	5.63	5.14	7.50	6.77	6.64	...	...	6.16	6.88	6.05	...	700
725	5.90	5.38	7.82	7.04	6.94	...	...	6.44	7.17	6.31	...	725
750	6.16	5.62	8.15	7.31	7.25	...	...	6.71	7.47	6.57	...	750
775	6.43	5.86	8.47	7.58	7.55	...	...	6.99	7.76	6.84	...	775
800	6.70	6.10	8.80	7.85	7.85	...	...	7.27	8.06	7.10	...	800
825	6.97	6.34	9.13	8.15	8.16	...	...	7.54	8.35	7.38	...	825
850	7.25	6.59	9.46	8.45	8.48	...	...	7.82	8.66	7.67	...	850
875	7.53	6.83	9.79	8.75	8.80	...	...	8.09	8.95	7.95	...	875
900	7.81	7.07	10.12	9.05	9.12	...	...	8.37	9.26	8.23	...	900
925	8.08	7.31	10.46	9.35	9.44	...	...	8.64	9.56	8.52	...	925
950	8.35	7.56	10.80	9.65	9.77	...	...	8.92	9.87	8.80	...	950
975	8.62	7.81	11.14	9.95	10.09	...	...	9.20	10.18	9.09	...	975
1000	8.89	8.06	11.48	10.25	10.42	...	...	9.49	10.49	9.37	...	1000
1025	9.17	8.30	11.82	10.55	10.75	...	...	9.77	10.80	9.66	...	1025
1050	9.46	8.55	12.16	10.85	11.09	...	...	10.05	11.11	9.94	...	1050
1075	9.75	8.80	12.50	11.15	11.43	...	...	10.34	11.42	10.23	...	1075
1100	10.04	9.05	12.84	11.45	11.77	...	...	10.63	11.74	10.51	...	1100
1125	10.31	9.28	13.18	11.78	12.11	...	...	10.92	12.05	10.80	...	1125
1150	10.57	9.52	13.52	12.11	12.47	...	...	11.21	12.38	11.09	...	1150
1175	10.83	9.76	13.86	12.44	12.81	...	...	11.50	12.69	11.37	...	1175
1200	11.10	10.00	14.20	12.77	13.15	...	...	11.80	13.02	11.66	...	1200
1225	11.38	10.26	14.54	13.10	13.50	...	...	12.09	13.36	11.98	...	1225
1250	11.66	10.53	14.88	13.43	13.86	...	...	12.39	13.71	12.29	...	1250
1275	11.94	10.79	15.22	13.76	14.22	...	...	12.69	14.04	12.61	...	1275
1300	12.22	11.06	15.56	14.09	14.58	...	...	12.99	14.39	12.93	...	1300
1325	12.50	11.30	15.90	14.39	14.94	...	...	13.29	14.74	13.25	...	1325
1350	12.78	11.55	16.24	14.69	15.30	...	...	13.59	15.10	13.56	...	1350
1375	13.06	11.80	16.58	14.99	15.66	...	...	13.90	15.44	13.88	...	1375
1400	13.34	12.05	16.92	15.29	16.02	...	...	14.20	15.80	14.20	...	1400
1425	...	...	17.30	...	...	...	...	14.51	16.16	14.51	...	1425
1450	...	...	17.69	...	...	...	...	14.82	16.53	14.83	...	1450
1475	...	...	18.08	...	...	...	...	15.13	16.88	15.14	...	1475
1500	...	...	18.47	...	...	...	...	15.44	17.25	15.45	...	1500
1525	...	...	...	...	...	...	...	15.76	17.61	15.77	...	1525
1550	...	...	...	...	...	...	...	16.07	17.98	16.08	...	1550
1575	...	...	...	...	...	...	...	16.39	18.35	16.40	...	1575
1600	...	...	...	...	...	...	...	16.71	18.73	16.71	...	1600

Notes: 1. Table shows expansion resulting from change in temperature from 70° F to indicated temperature

2. This table is for information only and it is not to be implied that materials are suitable for all the temperature ranges shown.

3. The thermal expansion values in this table may be interpolated to determine values for intermediate temperatures.

TABLE IV**Thermal Expansion of Pipe in mm per m**

Temp. °C	C-Mo, 3 Cr-Mo Steels	5Cr- to 9Cr-Mo Steels	Austenitic SS 18 Cr-8 Ni	310 SS 25 Cr-20 Ni	Alloy 400	Temp. °C
-200	-1.97	-1.84	-3.19	-	-2.17	-200
-175	-1.79	-1.68	-2.87	-	-2.02	-175
-150	-1.60	-1.51	-2.54	-	-1.84	-150
-125	-1.40	-1.32	-2.20	-	-1.63	-125
-100	-1.19	-1.12	-1.84	-	-1.40	-100
-75	-0.96	-0.91	-1.48	-	-1.14	-75
-50	-0.72	-0.68	-1.10	-	-0.87	-50
-25	-0.47	-0.45	-0.72	-	-0.57	-25
0	-0.21	-0.20	-0.32	-0.32	-0.26	0
20	0.00	0.00	0.00	0.00	0.00	20
50	0.33	0.32	0.49	0.48	0.40	50
75	0.62	0.59	0.91	0.87	0.76	75
100	0.92	0.87	1.34	1.27	1.12	100
125	1.23	1.16	1.77	1.67	1.49	125
150	1.55	1.45	2.21	2.07	1.87	150
175	1.88	1.76	2.65	2.47	2.26	175
200	2.22	2.06	3.10	2.88	2.65	200
225	2.56	2.38	3.55	3.28	3.05	225
250	2.91	2.70	4.01	3.68	3.46	250
275	3.27	3.02	4.48	4.09	3.88	275
300	3.64	3.35	4.94	4.50	4.31	300
325	4.01	3.69	5.42	4.91	4.74	325
350	4.39	4.03	5.89	5.32	5.17	350
375	4.77	4.37	6.37	5.74	5.62	375
400	5.17	4.72	6.85	6.16	6.07	400
425	5.56	5.07	7.34	6.58	6.53	425
450	5.96	5.43	7.83	7.00	6.99	450
475	6.37	5.78	8.32	7.43	7.46	475
500	6.78	6.15	8.82	7.87	7.94	500
525	7.19	6.51	9.32	8.31	8.43	525
550	7.60	6.88	9.82	8.75	8.92	550
575	8.02	7.25	10.32	9.20	9.42	575
600	8.44	7.62	10.83	9.66	9.93	600
625	8.86	8.00	11.34	10.13	10.45	625
650	9.28	8.38	11.86	10.60	10.97	650
675	9.70	8.77	12.38	11.09	11.49	675
700	10.12	9.15	12.90	11.58	12.03	700
725	10.55	9.54	13.43	12.09	12.56	725
750	10.96	9.94	13.96	12.60	13.10	750
775	-	-	14.50	-	-	775
800	-	-	15.04	-	-	800

Notes: 1. Table shows expansion resulting from change in temperature from 21° C to indicated temperature

2. This table is for information only and it is not to be implied that materials are suitable for all the temperature ranges shown.

3. The thermal expansion values in this table may be interpolated to determine values for intermediate temperatures.

TABLE IV (continued) Thermal Expansion of Pipe in mm per m

Temp. °C	Cu-30Ni	Copper	Nickel 200	Alloy 800, 825	Alloy 600, 625, 691	Aluminium	Temp. °C
-200	-2.52	-	-	-	-	-3.90	-200
-175	-2.31	-	-1.99	-	-	-3.55	-175
-150	-2.08	-	-1.80	-	-1.88	-3.17	-150
-125	-1.83	-	-1.58	-	-1.64	-2.78	-125
-100	-1.56	-	-1.34	-	-1.39	-2.36	-100
-75	-1.27	-1.54	-1.09	-	-1.12	-1.91	-75
-50	-0.96	-1.15	-0.82	-	-0.85	-1.44	-50
-25	-0.64	-0.75	-0.54	-	-0.55	-0.95	-25
0	-0.29	-0.33	-0.25	-0.27	-0.25	-0.43	0
20	0.00	0.00	0.00	0.00	0.00	0.00	20
50	0.45	0.51	0.38	0.42	0.39	0.67	50
75	0.84	0.93	0.71	0.79	0.72	1.26	75
100	1.24	1.36	1.05	1.17	1.06	1.86	100
125	1.64	1.80	1.40	1.56	1.40	2.48	125
150	2.05	2.23	1.76	1.96	1.76	3.12	150
175	2.46	2.67	2.12	2.37	2.12	3.76	175
200	2.85	3.12	2.49	2.79	2.48	4.42	200
225	-	3.57	2.87	3.22	2.85	5.09	225
250	-	4.04	3.25	3.65	3.23	5.76	250
275	-	4.52	3.63	4.08	3.60	6.44	275
300	-	-	4.03	4.52	3.99	7.10	300
325	-	-	4.42	4.95	4.37	7.77	325
350	-	-	4.82	5.39	4.76	-	350
375	-	-	5.22	5.84	5.15	-	375
400	-	-	5.63	6.28	5.55	-	400
425	-	-	6.04	6.73	5.95	-	425
450	-	-	6.45	7.17	6.35	-	450
475	-	-	6.87	7.62	6.75	-	475
500	-	-	7.29	8.07	7.16	-	500
525	-	-	7.71	8.52	7.58	-	525
550	-	-	8.13	8.98	7.99	-	550
575	-	-	8.56	9.44	8.42	-	575
600	-	-	8.99	9.90	8.84	-	600
625	-	-	9.42	10.37	9.28	-	625
650	-	-	9.86	10.85	9.72	-	650
675	-	-	10.30	11.33	10.16	-	675
700	-	-	10.74	11.83	10.62	-	700
725	-	-	11.19	12.33	11.08	-	725
750	-	-	11.65	12.85	11.55	-	750
775	-	-	12.11	13.39	12.03	-	775
800	-	-	12.57	13.94	12.52	-	800
825	-	-	13.05	14.51	13.02	-	825
850	-	-	13.53	15.10	13.54	-	850
875	-	-	14.01	15.72	14.07	-	875
900	-	-	14.51	16.37	14.61	-	900

See Notes on page D-18.

TABLE V
Moduli of Elasticity of Commonly Used Bellows Material – psi
(Multiply tabulated value by 10⁶)

Temp. Degrees F	Carbon Steel C ≤ 0.3%	Carbon Steel C > 3%	Alloy 800	Alloy 825	Aluminum	Alloy 400	Aust. Stainless Steel	Alloy 200	Alloy 600	Alloy 625
-325	31.4	31.2	30.5	30.0	11.3	27.8	30.3	32.2	33.3	32.2
-200	30.8	30.6	29.9	29.3	11.1	27.2	29.7	31.4	32.5	31.4
-100	30.3	30.1	29.3	28.8	10.9	26.8	29.2	30.9	31.9	30.9
70	29.4	29.2	28.5	28.0	10.6	26.0	28.3	30.0	31.0	30.0
200	28.8	28.6	27.9	27.4	10.4	25.5	27.5	29.4	30.3	29.4
300	28.3	28.1	27.5	27.0	10.2	25.1	27.0	28.9	29.9	28.9
400	27.9	27.7	27.1	26.6	9.5	24.7	26.4	28.5	29.4	28.5
500	27.3	27.1	26.7	26.2	8.5	24.3	25.9	28.1	29.0	28.1
600	26.5	26.4	26.2	25.8	-	23.9	25.3	27.6	28.6	27.6
700	25.5	25.3	25.8	25.4	-	23.6	24.8	27.2	28.1	27.2
800	24.2	24.0	25.4	24.9	-	23.1	24.1	26.7	27.6	26.7
900	22.5	22.3	24.9	24.4	-	22.7	23.5	26.2	27.1	26.2
1000	20.4	20.2	24.4	23.9	-	22.2	22.8	25.7	26.5	25.7
1100	18.0	17.9	23.8	23.4	-	21.7	22.0	25.1	25.9	25.1
1200	-	15.4	23.2	22.8	-	21.2	21.2	24.5	25.3	24.5
1300	-	-	22.6	22.2	-	20.6	20.3	23.8	24.6	23.8
1400	-	-	21.9	21.6	-	20.0	19.2	23.1	23.9	23.1

Notes: This table is for information only. It is not to be implied that materials are suitable for all temperature ranges shown. Data on Alloy 600, 625 and Alloy 800 and 825 are from Special Metals, Inc. Balance of data from ASME Section VIII – Div. 1, ASME B31.1, and ASME B31.3.

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TABLE V
Moduli of Elasticity of Commonly Used Bellows Material – N/mm²
(Multiply tabulated value by 10⁵)

Temp. °C	C-Steel C ≤ 0.3%	C-Steel C > 0.3%	Alloy 800	Alloy 825	Alumi- nium	Alloy 400	Austenitic stainless Steel	Alloy 200	Alloy 600	Alloy 625
-200	2.17	2.16	2.10	2.07	0.78	1.92	2.09	2.22	2.30	2.22
-150	2.14	2.12	2.07	2.04	0.77	1.89	2.06	2.19	2.26	2.19
-100	2.11	2.09	2.04	2.01	0.76	1.86	2.03	2.15	2.22	2.15
0	2.04	2.02	1.98	1.94	0.74	1.80	1.96	2.08	2.15	2.08
20	2.03	2.01	1.97	1.93	0.73	1.79	1.95	2.07	2.14	2.07
100	1.98	1.98	1.92	1.89	0.72	1.75	1.89	2.02	2.09	2.02
150	1.96	1.95	1.89	1.87	0.70	1.73	1.86	2.00	2.06	2.00
200	1.93	1.92	1.87	1.84	0.67	1.71	1.82	1.97	2.04	1.97
250	1.89	1.88	1.84	1.82	0.61	1.68	1.79	1.94	2.01	1.94
300	1.85	1.84	1.81	1.79	-	1.66	1.75	1.92	1.98	1.92
350	1.79	1.78	1.79	1.76	-	1.63	1.72	1.89	1.95	1.89
400	1.72	1.70	1.76	1.74	-	1.60	1.68	1.86	1.92	1.86
450	1.62	1.61	1.73	1.71	-	1.58	1.64	1.83	1.89	1.83
500	1.51	1.50	1.70	1.68	-	1.55	1.60	1.80	1.86	1.80
550	1.38	1.37	1.67	1.64	-	1.52	1.56	1.76	1.82	1.76
600	1.22	1.22	1.64	1.61	-	1.48	1.51	1.73	1.79	1.73
650	-	1.04	1.60	1.57	-	1.45	1.46	1.69	1.75	1.69
700	-	-	1.57	1.54	-	1.42	1.40	1.64	1.70	1.64
750	-	-	1.53	1.49	-	1.38	1.34	1.60	1.66	1.60

Notes: This table is for information only. It is not to be implied that materials are suitable for all temperature ranges shown. Data on Alloy 600, 625 and Alloy 800 and 825 are from Special Metals, Inc. Balance of data from ASME Section VIII – Div. 1, ASME B31.1, and ASME B31.3.

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Preparation of Technical Inquiries

Introduction

The EJMA Technical Committee will consider written requests for interpretations and revisions of the EJMA Standards. The Committee's activities in this regard are limited strictly to interpretations of the Standards. EJMA does not approve, certify, rate, or endorse any item, construction, proprietary device, or activity. EJMA does not act as a consultant on specific engineering problems or on general application or understanding of the EJMA Standards. Inquiries requiring such consideration will be returned.

Requirements

Inquiries shall be limited strictly to interpretations of the Standards or to the consideration of revisions to the present Standards on the basis of new data or technology. Inquiries shall meet the following requirements:

(a) Scope. Involve a single subject or closely related subjects in the scope of the Standard. An inquiry letter concerning unrelated subjects will be returned.

(b) Background. State the purpose of the inquiry, which may be either to obtain an interpretation of the Standard or to propose consideration of a revision to the present Standard. Provide concisely the information needed for the Committee's understanding of the inquiry, being sure to include reference to the applicable Standard Section, Edition, Addenda, paragraphs, figures, and tables. If sketches are provided, they shall be limited to the scope of the inquiry.

(c) Inquiry Structure

(1) Proposed Question(s). The inquiry shall be stated in a condensed and precise question format, omitting superfluous background information and where appropriate, composed in such a way that "yes" or "no" (perhaps with provisos) would be an acceptable reply. The inquiry statement should be technically and editorially correct

(2) Proposed Reply(ies). Provide a proposed reply stating what it is believed that the Standard requires. If in the inquirer's opinion, a revision to the Standards is needed, recommended wording shall be provided in addition to information justifying the change.

Submittal

Inquiries should be submitted in typewritten form; however, legible handwritten inquiries will be considered. They shall include the name and return address of the inquirer and be emailed, mailed, or faxed to the following address:

EJMA Technical Inquiries
25 North Broadway
Tarrytown, NY 10591 USA
Fax: 1-914-332-1541
E-mail: inquiries@ejma.org

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BELLOWS FATIGUE TEST REQUIREMENTS

F-1 INTRODUCTION

The Expansion Joint Manufacturers Association has adopted the following minimum requirements for bellows fatigue testing. It is the intention that the test results will accurately represent performance of typical production bellows. The results may be used to prepare fatigue curves for use with bellows intended for service below the active creep temperature range.

F-2 TEST SPECIMENS

F-2.1 MANUFACTURING METHODS

The bellows used for fatigue testing shall be representative of the bellows manufactured for normal production purposes. The same shearing, tube rolling, welding, planishing, convolution forming, re-rolling, final sizing, and thermal treatment methods shall be employed for the test specimens.

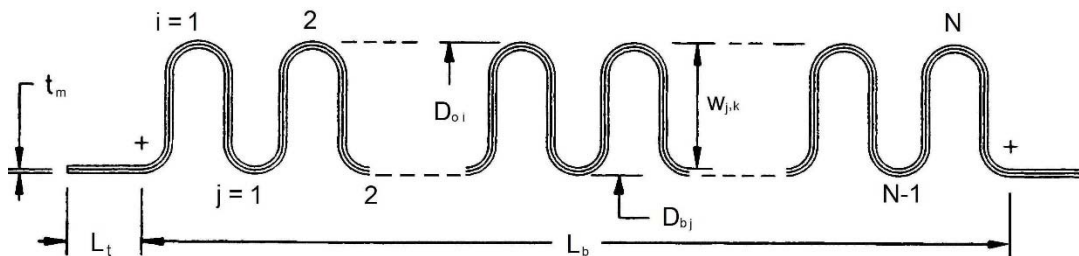
The detailed steps of manufacturing shall be recorded for each test specimen.

The finished test specimens shall have the same typical variations in dimensions, surface finish, and condition of cold work as normal production bellows.

Multi-ply bellows shall have provisions for a leak path through the outer plies.

F-2.2 DIMENSIONAL MEASUREMENTS

Test specimen as-built dimensions shall be measured and recorded. The nomenclature is as follows:



where

D_{oi} = Outside diameter of convolution crest, i (1 to N), as determined by circumferential measurement.

D_{bj} = Inside diameter of convolution root, j (1 to $N-1$), as determined by circumferential measurement.

$w_{j,k}$ = Convolution height, j (1 to $N-1$), at k (1 to 4) locations equally spaced around the circumference

t_m = Bellows material thickness at the tangent, m (1 to n)

L_b = Bellows convoluted length

L_t = Bellows tangent length

N = Number of convolutions in the bellows

n = Number of bellows material plies

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The tolerances for measurement accuracy of each dimension are as follows:

$$D_{oi} \pm 0.005 \text{ in. (0.125 mm)}$$

$$D_{bj} \pm 0.005 \text{ in. (0.125 mm)}$$

$$w_{j,k} \pm 0.005 \text{ in. (0.125 mm)}$$

$$t_m \pm 0.001 \text{ in. (0.25 mm)}$$

$$L_b \pm 0.031 \text{ in. (0.8 mm)}$$

$$L_t \pm 0.063 \text{ in. (1.6 mm)}$$

F-2.3 DIMENSIONAL REQUIREMENTS

Test specimens shall meet the following requirements:

- a. Min. convolution inside diameter (D_b) = 6.63 in. (168 mm)
- b. Max. bellows convoluted length (L_b) = $2D_b$
- c. Min. convolution height (w) = L_b / N
- d. Min. number of convolutions (N) = 3
- e. Min. bellows tangent length (L_t) = $\sqrt{(D_b)(t_m) / 2}$

F-2.4 BELLOWS MATERIALS

Test specimens shall be manufactured from typical production quality material. Any special treatments or finishing of the bellows material must be recorded.

F-2.5 BELLOWS ATTACHMENTS

The bellows shall be attached to the test apparatus in a manner that duplicates normal production bellows attachments.

F-2.6 BELLOWS HEAT TREATMENT

If heat treatment is performed on test specimens, the following information shall be recorded:

- a. Atmosphere
- b. Heating rate
- c. Holding temperature
- d. Holding time
- e. Cooling rate

F-3 TEST APPARATUS

F-3.1 GENERAL REQUIREMENTS

The test apparatus shall be constructed and controlled so that the test specimens can be rigidly held in position and cycled repeatedly with the specified movement. Bellows failure shall be defined as a leak through the material which causes a reduction of the internal pressure below a set minimum limit and/or allows for penetrating liquid to become visible on the outer surface of the bellows.

F-3.2 CYCLE COUNTERS

The apparatus shall provide for reliable cycle counters which record the total number of cycles to failure for each test bellows.

F-3.3 TEST MEDIA

Testing shall be completed using an internal pressurizing media and/or an internal penetrating liquid.

A liquid or gas media may be used to pressurize the test specimens internally during the fatigue test. The test media shall not be detrimental to the bellows material. The pressure may be constant or variable during the test. The apparatus shall be constructed so that the loss of media through a leak will result in a rapid reduction in pressure. The reduction in pressure below a set minimum limit may be used to identify the presence of a leak. Controls shall assure that a reduction in pressure below the set minimum limit results in a recording of the total number of movement cycles shown on the bellows cycle counters. The pressure during the movement cycles shall be constant but may differ from as low as possible up to the allowable pressure but, shall in no case cause bellows instability or convolution deformation during the test.

An internal penetrating liquid in contact with the inside surfaces of the test specimens may be used during the fatigue test. The liquid shall not be detrimental to the bellows material. When a leak develops during the test, the penetrating liquid shall rapidly become visible on the outer surface of the bellows and the total number of movement cycles shall be recorded.

F-3.4 TRAVEL SPEED

The apparatus shall control the motion to be smooth over the length of travel. The travel speed shall not exceed 60 in./min (1.5 m/min).

F-3.5 BELLWS MOVEMENT

The bellows test specimens shall be cycled with axial movement only. One cycle is defined as movement through the full movement range (amplitude) and return to the starting position.

The bellows movements for the test specimens shall be selected to produce cycles to failure that cover the desired range for the fatigue curve. The movements shall not be excessive and shall not cause detrimental convolution deformation.

The bellows movement range shall be measured and recorded at the beginning and end of the test. The tolerance for the measured movement range is +/- 0.5%.

F-4 FATIGUE CURVES

F-4.1 CALCULATIONS

Total stress range (S_t) calculations for each bellows shall be in accordance with the EJMA performance equations using the measured movement range and the room temperature Modulus of Elasticity for the material. The stresses due to test media pressure shall be included in the calculation for total stress range.

The variables in the performance equations shall be found using the test specimen measurements and the following equations:

- a. Bellows Outside Diameter (D_o)

$$D_o = \frac{1}{N} \sum_{i=1}^N D_{oi}$$

- b. Bellows Inside Diameter (D_b)

$$D_b = \frac{1}{(N-1)} \sum_{j=1}^{N-1} D_{bj}$$

- c. Convolution Height (w)

$$w = \frac{1}{4(N-1)} \sum_{j=1}^{N-1} \sum_{k=1}^4 w_{jk} \quad \text{for } D_b < 10.75 \text{ in. (273 mm)}$$

$$= (D_o - D_b - 2nt) / 2 \quad \text{for } D_b \geq 10.75 \text{ in. (273 mm)}$$

- d. Bellows Nominal Thickness of One Ply (t)

$$t = \frac{1}{n} \sum_{m=1}^n t_m$$

- e. Convolution Pitch (q)

$$q = L_b / N$$

- f. Mean Diameter of the Bellows (D_m)

$$D_m = D_b + w + nt$$

F-5 TEST DOCUMENTATION

The following documents are required:

- Certified mill test reports for the bellows material
- Heat treat charts (where applicable)
- Photographs of the test
- Test log sheets and records
- Final report of the results

BELLOWS HIGH TEMPERATURE CYCLE LIFE

G-1 INTRODUCTION

The Expansion Joint Manufacturers Association has adopted the following minimum requirements for the evaluation of bellows cycle life at high temperatures. The cycle life evaluation is based on high temperature test results. The empirical method is taken from Bellows High Temperature Cycle Life -1995, ASME PVP Vol. 301, pages 129 -138.

High temperature cycle tests shall include the effects of all conditions necessary to validate the correlation between the calculations and the finished product including material type, material condition (annealed or as-formed), and convolution profile. It is the intention that the required test results accurately represent the performance of typical production bellows.

G-2 TEST SPECIMENS

The test specimens shall meet the requirement of Section F-2 of the Standards.

G-3 TEST APPARATUS

The test apparatus shall meet the requirements of Section F-3 of the Standards.

G-3.1 TEMPERATURE CONTROL

The test apparatus shall be constructed so that the bellows metal temperature can be maintained within +/- 10 degrees F of the set test temperature.

G-3.2 HOLD TIME CONTROL

The test apparatus shall be constructed so that the time between deflection cycles can be maintained within +/- 1 percent of the set hold time.

G-4 TEST REQUIREMENTS

A total of four (4) bellows specimens labeled 1 through 4 are required for each separate test. The specimens shall meet the following requirements:

- a. No. 1 and 2 shall have the same design.
- b. No. 3 and 4 shall have the same design.
- c. No. 1 and 2 shall differ in calculated total stress range (S_t) from No. 3 and 4 by a factor of at least 2.0.
- d. No. 1 and 3 shall differ in hold time at temperature between cycles (H_t) from No. 2 and 4 by a factor of at least 100.

One cycle is defined as movement from the starting position to the final position, holding at the final position, and then returning to the starting position.

G-5 CYCLE LIFE CALCULATION PROCEDURE

1. Perform the cycle tests and record the results.
2. Calculate the total stress range (S_t) for each specimen in accordance with Section F-4.1 of the Standards. Label the results corresponding to each specimen and tabulate the results using the nomenclature as follows:

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Specimen No.	Total Stress Range (S_t)	Hold Time (H_t)	Cycles to Failure (N_c)
1	S_{t1}	H_{t1}	N_{c1}
2	S_{t2}	H_{t2}	N_{c2}
3	S_{t3}	H_{t1}	N_{c3}
4	S_{t4}	H_{t2}	N_{c4}

3. Calculate the mean stress ranges as follows:

$$S_{t12} = (S_{t1} + S_{t2}) / 2 \quad (G-1)$$

$$S_{t34} = (S_{t3} + S_{t4}) / 2 \quad (G-2)$$

4. Find the intermediate values as follows:

$$A = \frac{\log(N_{c1} / N_{c2})}{\log(H_{t2} / H_{t1})} \quad (G-3)$$

$$B = \frac{\log(N_{c3} / N_{c4})}{\log(H_{t2} / H_{t1})} \quad (G-4)$$

5. Find the constants as follows:

$$a = \frac{\log \left[\frac{N_{c4} H_{t2}^{B-A}}{N_{c2}} \right]}{\log(S_{t12} / S_{t34})} \quad (G-5)$$

$$b = N_{c4} H_{t2}^B S_{t34}^a \quad (G-6)$$

$$c = \frac{B - A}{\log(S_{t12} / S_{t34})} \quad (G-7)$$

$$d = A + c \log S_{t12} \quad (G-8)$$

6. Find the average cycles to failure for any total stress range and hold time as follows:

$$N_c = b S_t^{-a+c \log H_t} H_t^{-d} \quad (G-9)$$

This equation is applicable for the tested bellows material and material condition up to the set test temperature.

G-6 BELLOWS HOLD TIME

The hold time at temperature between cycles (H_t) can be determined based on historical records and planned operating schedules. Hold times may vary between cycles.

G-7 BELLOWS CYCLE LIFE REQUIREMENTS

The calculated cycle life may be evaluated as follows:

$$N_c \geq \frac{\text{Total Operating Life at Temperature (hours)}}{H_t} \quad (\text{G-10})$$

If the hold times vary between cycles, the calculated cycle life may be evaluated using the following:

$$\sum_{i=1}^n N_{ci} H_{ti} \geq \text{Total Operating Life at Temperature (hours)} \quad (\text{G-11})$$

where n is the total number of different hold times and N_{ci} is the calculated number of cycles at hold time H_{ti} .

An overly conservative estimate of the cycle life requirement can result in an increased number of convolutions and a bellows more prone to instability.

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ANGULAR ROTATION ABOUT ONE END

H-1 INTRODUCTION

An expansion joint absorbs pure angular rotation by extending uniformly on one side and compressing uniformly on the other (See Figure 4.3). Pure angular rotation occurs when the expansion joint bends with a constant radius about a center point. However, when the expansion joint bends about one end, the radius of curvature is not constant and the convolution movement is not uniform (See Figure H1). For this special case, the expansion joint can be modeled as an elastic beam having one end fixed and the other end simply supported with a concentrated end moment. The bellows can be treated as elastic beams.

H-2 FORCE, MOMENT AND MOVEMENT CALCULATION (SINGLE)

$$M_{\theta} = \frac{f_w D_m^2 \theta}{2N} \quad (H-1)$$

$$V_i = \frac{3 f_w D_m^2 \theta}{4N(L_b \pm x)} \quad (H-2)$$

$$e_{\theta} = \frac{2\theta D_m}{N} \quad (H-3)$$

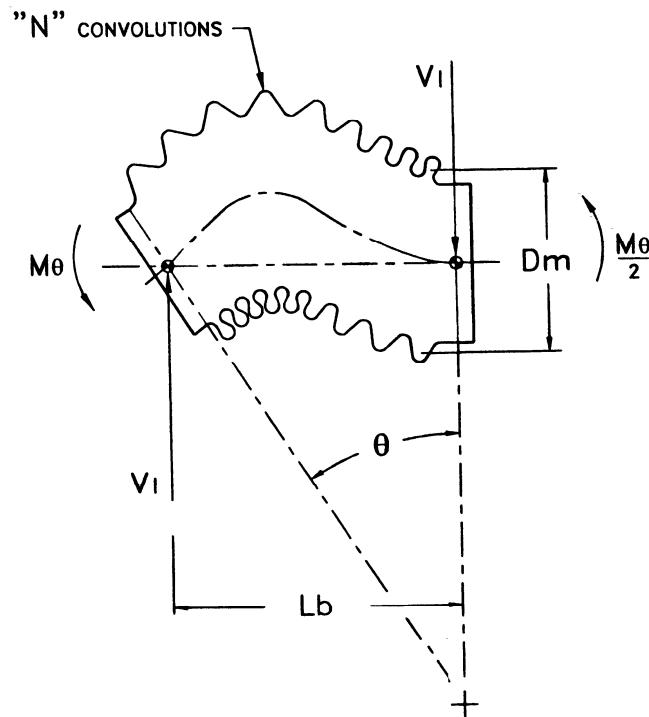


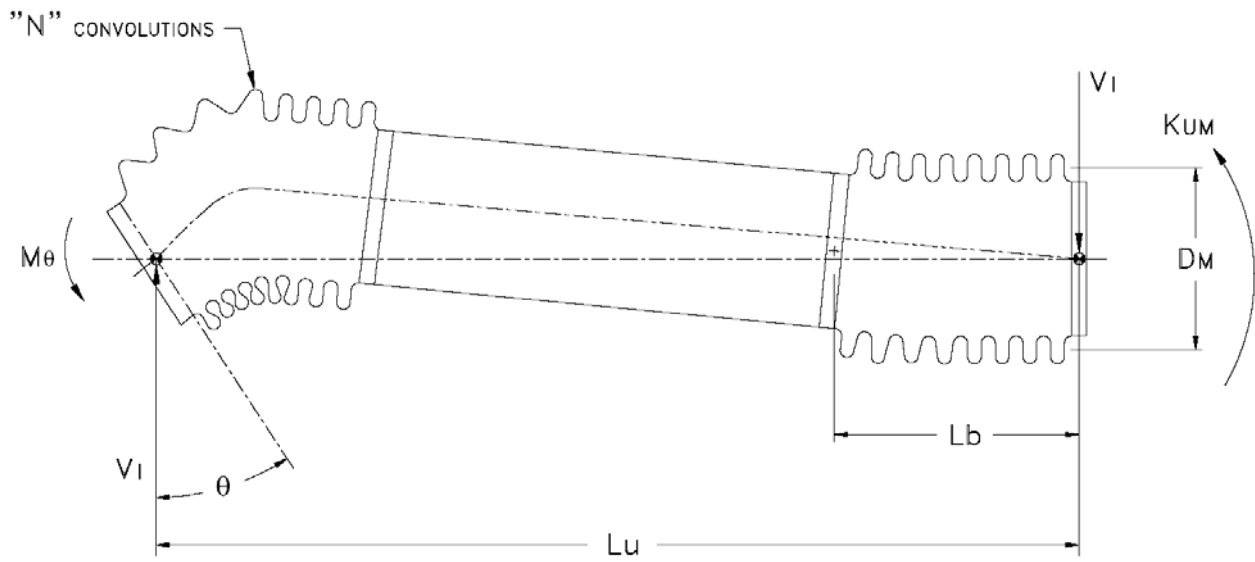
FIGURE H1

H-3 FORCE, MOMENT, AND MOVEMENT CALCULATION (UNIVERSAL)

$$M_{\theta} = \frac{f_w D_m^2 \theta}{4N K_{u\theta}} \quad (\text{H-4})$$

$$V_l = \frac{3 f_w D_m^2 \theta K_{uv}}{8N (L_b \pm x)} \quad (\text{H-5})$$

$$e_{\theta} = \frac{\theta D_m}{N K_{u\theta}} \quad (\text{H-6})$$

**FIGURE H2**

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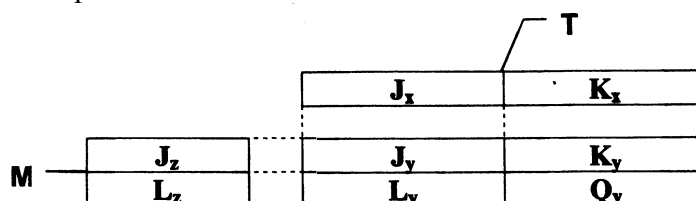
TABULATED VALUES FOR C_p , C_f , C_d , B_1 , B_2 , AND B_3

I-1 INTRODUCTION

Tables I1, I2, I3, and I4 contain tabulated values taken from the figures indicated for C_p , C_f , C_d , B_1 , B_2 , and B_3 . A method for interpolating between tabulated values is also included.

I-2 INTERPOLATION BETWEEN TABULATED VALUES

The following equations can be used as a guide for linear interpolation between the tabulated values for C_p , C_f and C_d . The boxes below represent the tables in Appendix I. They are used to organize data for two dimensional interpolation.



$$T = \frac{2r_m}{w} \quad (I-1)$$

$$M = \frac{1.82r_m}{\sqrt{D_m t_p}} \quad (I-2)$$

J_x and K_x are the values in the table that surround T . J_z and L_z are the values in the table that surround M . J_y , K_y , L_y and Q_y are the tabular values found at the intersection of the J_x , K_x , J_z and L_z values.

$$A = \left(\frac{T - J_x}{K_x - J_x} \right) (K_y - J_y) + J_y \quad (I-3)$$

$$B = \left(\frac{T - J_x}{K_x - J_x} \right) (Q_y - L_y) + L_y \quad (I-4)$$

$$C_p, C_f, C_d = \left(\frac{M - J_z}{L_z - J_z} \right) (B - A) + A \quad (I-5)$$

Example: Given $T = 0.63$ and $M = 2.3$, find C_p :

From Table I1, the following chart can be completed:

		0.63	
		0.60	0.65
2.3	2.00	0.323	0.316
	2.50	0.272	0.260

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$$T = \frac{2r_m}{w} = 0.63$$

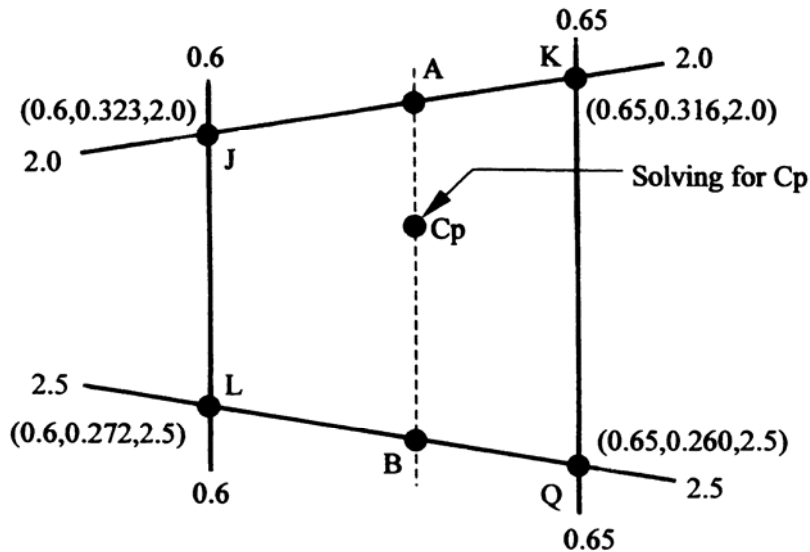
$$M = \frac{1.82r_m}{\sqrt{D_m t_p}} = 2.3$$

$$A = \left(\frac{0.63 - 0.60}{0.65 - 0.60} \right) (0.316 - 0.323) + 0.323 = 0.3188$$

$$B = \left(\frac{0.63 - 0.60}{0.65 - 0.60} \right) (0.260 - 0.272) + 0.272 = 0.2648$$

$$C_p = \left(\frac{2.30 - 2.00}{2.50 - 2.00} \right) (0.2648 - 0.3188) + 0.3188 = 0.2864$$

Figure I1 presents the method of interpolation in graphical form.



Example taken from tabulated data for C_p

FIGURE I1 METHOD OF LINEAR INTERPOLATION

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$\frac{1.82 r_m}{\sqrt{D_m t_p}}$	$2r_m/w$																				
	0	0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55	0.6	0.65	0.7	0.75	0.8	0.85	0.9	0.95	1
0.2	1.000	0.976	0.946	0.912	0.876	0.84	0.803	0.767	0.733	0.702	0.674	0.649	0.627	0.61	0.596	0.585	0.577	0.571	0.566	0.560	0.550
0.4	1.000	0.962	0.926	0.890	0.856	0.823	0.790	0.755	0.720	0.691	0.665	0.642	0.622	0.606	0.593	0.583	0.576	0.571	0.566	0.560	0.550
0.6	0.980	0.910	0.870	0.840	0.816	0.784	0.753	0.722	0.696	0.670	0.646	0.624	0.605	0.590	0.585	0.577	0.569	0.566	0.558	0.550	0.543
0.8	0.950	0.842	0.770	0.722	0.700	0.680	0.662	0.64	0.627	0.610	0.593	0.585	0.579	0.574	0.569	0.563	0.557	0.553	0.546	0.540	0.533
1.0	0.950	0.841	0.744	0.657	0.592	0.559	0.536	0.541	0.548	0.551	0.551	0.550	0.547	0.544	0.540	0.536	0.531	0.526	0.521	0.515	0.510
1.2	0.950	0.841	0.744	0.657	0.579	0.518	0.501	0.502	0.503	0.503	0.503	0.502	0.500	0.497	0.494	0.491	0.488	0.485	0.482	0.479	0.476
1.4	0.950	0.84	0.744	0.651	0.564	0.495	0.462	0.46	0.458	0.455	0.453	0.450	0.447	0.444	0.442	0.439	0.437	0.435	0.433	0.432	0.431
1.6	0.950	0.841	0.731	0.632	0.549	0.481	0.432	0.426	0.42	0.414	0.408	0.403	0.398	0.394	0.391	0.388	0.385	0.384	0.382	0.381	0.38
2.0	0.950	0.841	0.731	0.632	0.549	0.481	0.421	0.388	0.369	0.354	0.342	0.332	0.323	0.316	0.309	0.304	0.299	0.296	0.294	0.293	0.292
2.5	0.950	0.840	0.732	0.630	0.550	0.480	0.421	0.367	0.332	0.315	0.300	0.285	0.272	0.260	0.251	0.242	0.235	0.230	0.224	0.219	0.215
3.0	0.950	0.840	0.732	0.630	0.550	0.480	0.421	0.367	0.328	0.299	0.275	0.258	0.242	0.228	0.215	0.203	0.195	0.188	0.180	0.175	0.171
3.5	0.950	0.840	0.732	0.630	0.550	0.480	0.421	0.367	0.322	0.287	0.262	0.241	0.222	0.208	0.194	0.182	0.171	0.161	0.152	0.146	0.140
4.0	0.950	0.840	0.732	0.630	0.550	0.480	0.421	0.367	0.312	0.275	0.248	0.225	0.205	0.190	0.176	0.163	0.152	0.142	0.134	0.126	0.119

Table I1 Tabulated Values for C_p (From Figure 4.16)

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$\frac{1.82r_m}{\sqrt{D_m t_p}}$	$\frac{2r_m}{w}$																					
	0.0	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	0.95	1.0	
0.2	1.000	1.116	1.211	1.297	1.376	1.451	1.524	1.597	1.669	1.740	1.812	1.882	1.952	2.020	2.087	2.153	2.217	2.282	2.349	2.421	2.501	
0.4	1.000	1.094	1.174	1.248	1.319	1.386	1.452	1.517	1.582	1.646	1.710	1.775	1.841	1.908	1.975	2.045	2.116	2.189	2.265	2.345	2.430	
0.6	1.000	1.092	1.163	1.225	1.281	1.336	1.392	1.449	1.508	1.568	1.630	1.692	1.753	1.813	1.871	1.929	1.987	2.049	2.119	2.201	2.305	
0.8	1.000	1.066	1.122	1.171	1.217	1.260	1.300	1.340	1.380	1.422	1.465	1.511	1.560	1.611	1.665	1.721	1.779	1.838	1.896	1.951	2.002	
1.0	1.000	1.026	1.052	1.077	1.100	1.124	1.147	1.171	1.195	1.220	1.246	1.271	1.298	1.325	1.353	1.382	1.415	1.451	1.492	1.541	1.600	
1.2	1.000	1.002	1.000	0.995	0.989	0.983	0.979	0.975	0.975	0.976	0.980	0.987	0.996	1.008	1.022	1.038	1.056	1.076	1.099	1.125	1.154	
1.4	1.000	0.983	0.962	0.938	0.915	0.892	0.870	0.851	0.834	0.820	0.809	0.799	0.792	0.787	0.783	0.780	0.779	0.780	0.781	0.785	0.792	
1.6	1.000	0.972	0.937	0.899	0.860	0.821	0.784	0.750	0.719	0.691	0.667	0.646	0.627	0.611	0.598	0.586	0.576	0.569	0.563	0.560	0.561	
2.0	1.000	0.948	0.892	0.836	0.782	0.730	0.681	0.636	0.595	0.557	0.523	0.492	0.464	0.439	0.416	0.394	0.373	0.354	0.336	0.319	0.303	
2.5	1.000	0.930	0.867	0.800	0.730	0.665	0.610	0.560	0.510	0.470	0.430	0.392	0.360	0.330	0.300	0.275	0.253	0.230	0.206	0.188	0.170	
3.0	1.000	0.920	0.850	0.780	0.705	0.640	0.580	0.525	0.470	0.425	0.380	0.342	0.300	0.271	0.242	0.212	0.188	0.167	0.146	0.130	0.115	
3.5	1.000	0.900	0.830	0.750	0.680	0.610	0.550	0.495	0.445	0.395	0.350	0.303	0.270	0.233	0.200	0.174	0.150	0.130	0.112	0.092	0.081	
4.0	1.000	0.900	0.820	0.735	0.655	0.590	0.525	0.470	0.420	0.370	0.325	0.285	0.252	0.213	0.182	0.152	0.130	0.109	0.090	0.074	0.061	

Table I2 Tabulated Values for C_f (From Figure 4.17)

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$1.82r_m$ $\sqrt{Dm_p}$	$\frac{2r_m}{w}$																				
	0.0	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	0.95	1.0
0.2	1.000	1.061	1.128	1.198	1.269	1.340	1.411	1.480	1.547	1.614	1.679	1.743	1.807	1.872	1.937	2.003	2.070	2.138	2.206	2.274	2.341
0.4	1.000	1.066	1.137	1.209	1.282	1.354	1.426	1.496	1.565	1.633	1.700	1.766	1.832	1.897	1.963	2.029	2.096	2.164	2.234	2.305	2.378
0.6	1.000	1.105	1.195	1.277	1.352	1.424	1.492	1.559	1.626	1.691	1.757	1.822	1.886	1.950	2.014	2.077	2.141	2.206	2.273	2.344	2.422
0.8	1.000	1.079	1.171	1.271	1.374	1.476	1.575	1.667	1.753	1.832	1.905	1.973	2.037	2.099	2.160	2.221	2.283	2.345	2.407	2.467	2.521
1.0	1.000	1.057	1.128	1.208	1.294	1.384	1.476	1.571	1.667	1.766	1.866	1.969	2.075	2.182	2.291	2.399	2.505	2.603	2.690	2.758	2.800
1.2	1.000	1.037	1.080	1.130	1.185	1.246	1.311	1.381	1.457	1.539	1.628	1.725	1.830	1.943	2.066	2.197	2.336	2.483	2.634	2.789	2.943
1.4	1.000	1.016	1.039	1.067	1.099	1.135	1.175	1.220	1.269	1.324	1.385	1.452	1.529	1.614	1.710	1.819	1.941	2.080	2.236	2.412	2.611
1.6	1.000	1.006	1.015	1.025	1.037	1.052	1.070	1.091	1.116	1.145	1.181	1.223	1.273	1.333	1.402	1.484	1.578	1.688	1.813	1.957	2.121
2.0	1.000	0.992	0.984	0.974	0.966	0.958	0.952	0.947	0.945	0.946	0.950	0.958	0.970	0.988	1.011	1.042	1.081	1.130	1.191	1.267	1.359
2.5	1.000	0.980	0.960	0.935	0.915	0.895	0.875	0.840	0.833	0.825	0.815	0.800	0.790	0.785	0.780	0.780	0.785	0.795	0.815	0.845	0.890
3.0	1.000	0.970	0.945	0.910	0.885	0.855	0.825	0.800	0.775	0.750	0.730	0.710	0.688	0.670	0.657	0.642	0.635	0.628	0.625	0.630	0.640
3.5	1.000	0.965	0.930	0.890	0.860	0.825	0.790	0.760	0.730	0.700	0.670	0.645	0.620	0.597	0.575	0.555	0.538	0.522	0.510	0.502	0.500
4.0	1.000	0.955	0.910	0.870	0.830	0.790	0.755	0.720	0.685	0.655	0.625	0.595	0.567	0.538	0.510	0.489	0.470	0.452	0.438	0.428	0.420

Table I3 Tabulated Values for C_d (From Figure 4.18)

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$\frac{6.61r^2}{D_m t_p}$	B₁	B₂	B₃
0	1.0	1.0	1.0
1	1.1	1.0	1.1
2	1.4	1.0	1.3
3	2.0	1.0	1.5
4	2.8	1.0	1.9
5	3.6	1.0	2.3
6	4.6	1.1	2.8
7	5.7	1.2	3.3
8	6.8	1.4	3.8
9	8.0	1.5	4.4
10	9.2	1.6	4.9
11	10.6	1.7	5.4
12	12.0	1.8	5.9
13	13.2	2.0	6.4
14	14.7	2.1	6.9
15	16.0	2.2	7.4
16	17.4	2.3	7.9
17	18.9	2.4	8.5
18	20.3	2.6	9.0
19	21.9	2.7	9.5
20	23.3	2.8	10.0

Table I4 Tabulated Values for B_1 , B_2 , B_3 (From Figure 4.19)

EXAMPLES

Examples for calculating the forces and moments acting on various points of typical piping systems due to the spring and pressure forces of metal bellows Expansion Joints are presented below.

ASSUMPTIONS:

1. The piping system and Expansion Joints are properly supported and guided.
2. The weight of the piping system and the fluid being conveyed is carried by properly designed supports and hangers and is, therefore, not included.
3. Friction forces caused by guides, supports, and other hardware extraneous to the piping are zero.
4. The origin of the pipe system is located at the point under consideration.
5. Forces and moments due to pipe flexibility are neglected.

GENERAL EQUATIONS:

All examples presented depict systems where static equilibrium exists.

$$\sum F_{x,y,z} = 0 \text{ and } \sum M_{x,y,z} = 0$$

For the coordinate system shown in Figure J1, the general moment equations, employing the "Right Hand Rule," are:

$$M_x = F_z Y - F_y Z \quad (J-1)$$

$$M_y = F_x Z - F_z X \quad (J-2)$$

$$M_z = F_y X - F_x Y \quad (J-3)$$

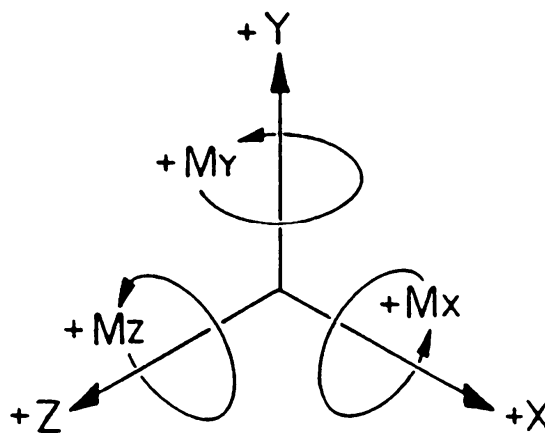


FIGURE J1

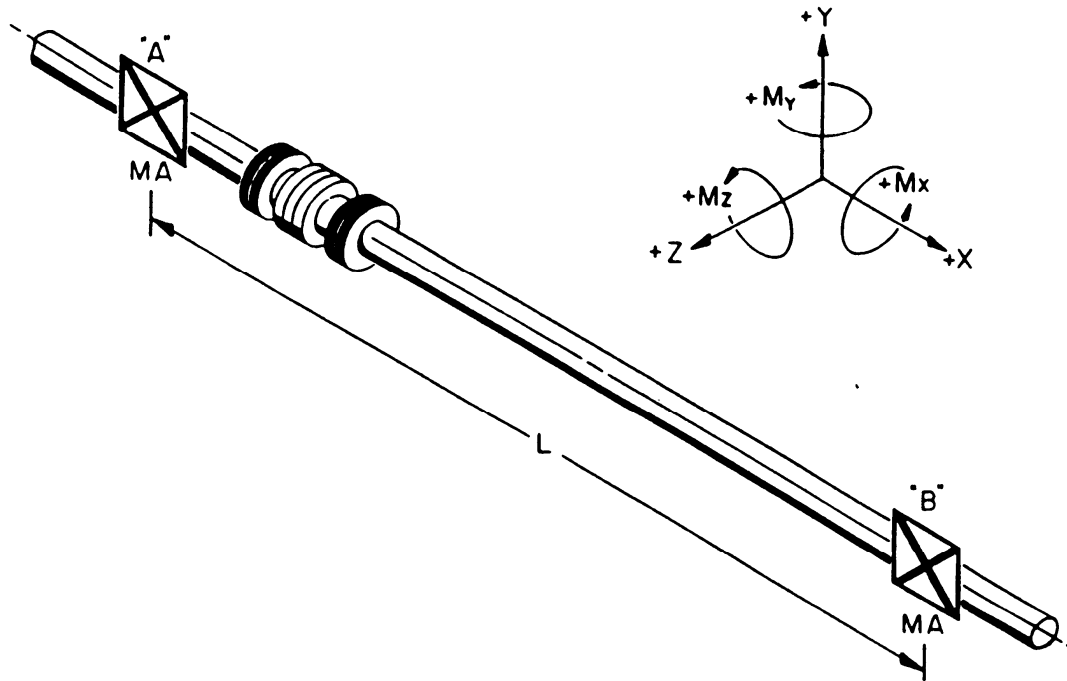
Example 1: Single Expansion Joint subjected to axial movement.

FIGURE J2

A straight run of 24 in. diameter carbon steel pipe 60 feet long and anchored at each end, (reference Figure J2), is to operate at 150 psig at 500°F. A single bellows Expansion Joint is utilized to absorb the thermal growth of the pipe. Thermal growth is calculated to be $60/100 \times 3.62 = 2.17$ in. What are the forces acting on the anchors?

Data provided by the Expansion Joint manufacturer:

$$A_e = 510.7 \text{ in}^2$$

$$f_w = 36840 \text{ lbs./in. per convolution}$$

$$N = 12$$

SOLUTION:

Forces acting on Main Anchor "A"

$$F_x = -F_s - F_a$$

$$F_x = -76605 - 6668$$

$$F_x = -83273 \text{ lbs.}$$

Where:

 F_s = the static thrust due to internal pressure to the Expansion Joint (lbs)= $A_e P_d$ Equation (2-2) Section 2.10.1.2.1

$$= (510.7)(150)$$

$$= 76605 \text{ lbs}$$

 e_x = axial movement per convolution

$$= \frac{x}{N} \text{ Equation (4-1) Section 4.1}$$

$$= \frac{2.17}{12}$$

$$= 0.181 \text{ in.}$$

 F_a = the force required to deflect the Expansion Joint.= $f_w e_x$ Equation (4-15) Section 4.6.1

$$= (36840)(0.181)$$

$$= 6668 \text{ lbs.}$$

Forces acting on Main Anchor "B"

$$F_x = F_s + F_a$$

$$F_x = 76605 + 6668$$

$$F_x = 83273 \text{ lbs.}$$

NOTE: Because the pipe system is linear with no bending, F_y , $F_z = 0$, and M_x , M_y , and $M_z = 0$ at Main Anchors "A" and "B".

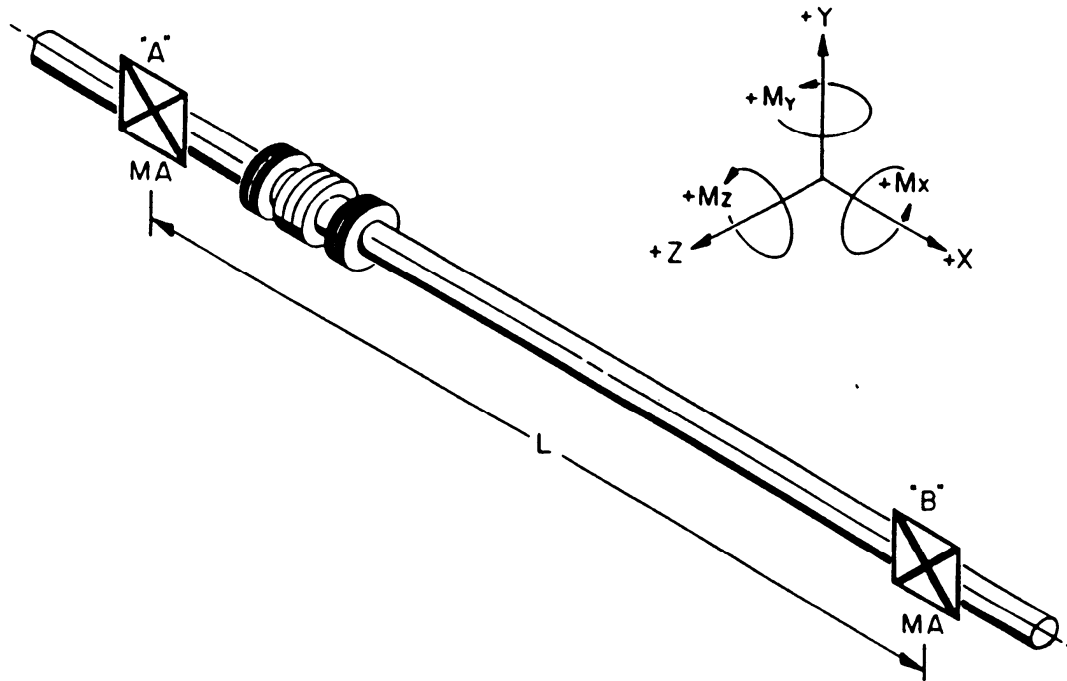
Example 1M: Single Expansion Joint subjected to axial movement.

FIGURE J2

A straight run of 610 mm diameter carbon steel pipe 18.28 m long and anchored at each end, (reference Figure J2), is to operate at 1 MPa at 260°C. A single bellows Expansion Joint is utilized to absorb the thermal growth of the pipe. Thermal growth is calculated to be $18.3 \times 3.02 = 55.2\text{mm}$. What are the forces acting on the anchors?

Data provided by the Expansion Joint manufacturer:

$$A_e = 329,500 \text{ mm}^2$$

$$f_w = 6,451 \text{ N/mm per convolution}$$

$$N = 12$$

SOLUTION:

Forces acting on Main Anchor "A"

$$F_x = -F_s - F_a$$

$$F_x = -340 - 30$$

$$F_x = -370 \text{ kN}$$

Where:

 F_s = the static thrust due to internal pressure to the Expansion Joint

$$= A_e P_d \text{ Equation (2-2) Section 2.10.1.2.1}$$

$$= 329,500 \times 1$$

$$= 330 \text{ kN}$$

 e_x = axial movement per convolution

$$= \frac{x}{N} \text{ Equation (4-1) Section 4.1}$$

$$= \frac{55.2}{12}$$

$$= 4.6 \text{ mm}$$

 F_a = the force required to deflect the Expansion Joint.

$$= f_w e_x \text{ Equation (4-15) Section 4.6.1}$$

$$= 6,451 \times 4.6$$

$$= 30 \text{ kN}$$

Forces acting on Main Anchor "B"

$$F_x = F_s + F_a$$

$$F_x = (340 + 30)$$

$$F_x = (370 \text{ kN})$$

NOTE: Because the pipe system is linear with no bending, F_y , $F_z = 0$, and M_x , M_y , and $M_z = 0$ at Main Anchors "A" and "B".

Example 2: Single Expansion Joint subjected to axial and lateral movement (similar to Figure 2.10, Section 2.4).

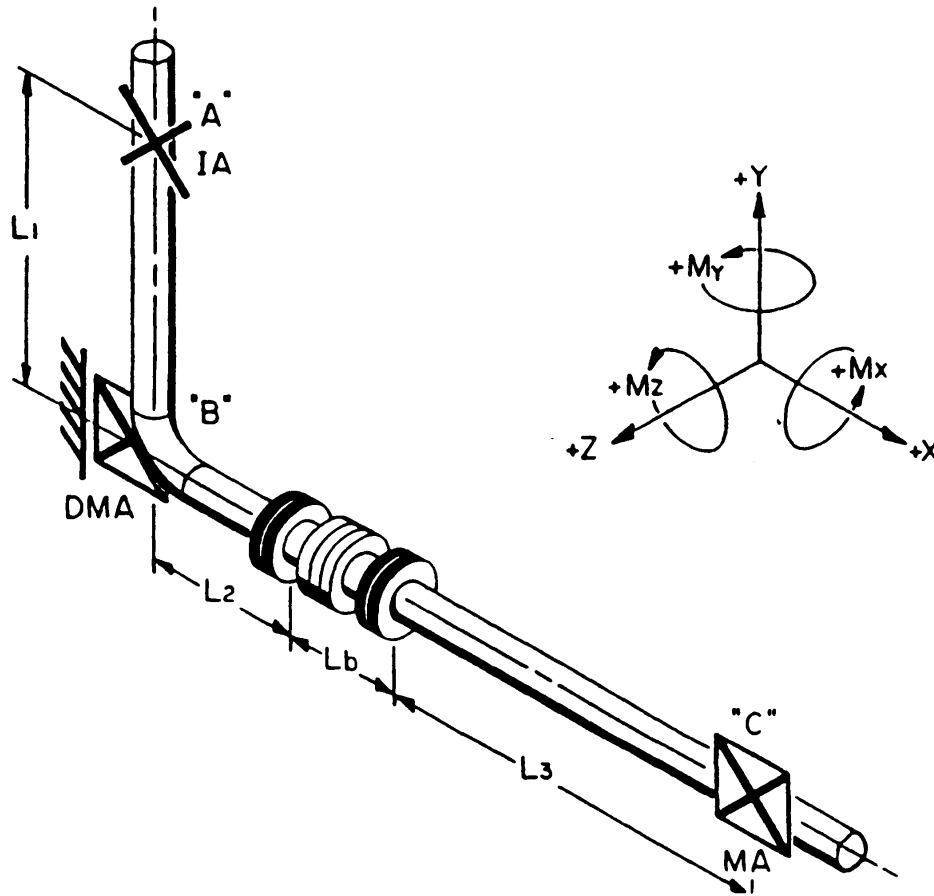


FIGURE J3

A single bellows Expansion Joint is placed in a 24 in. diameter carbon steel pipeline that runs between a main anchor and an intermediate anchor, and has one 90° elbow (reference Figure J3). The line is to operate at 125 psig and 400°F. The thermal growth that the Expansion Joint is to absorb is calculated to be 0.405 in. axially, and 0.216 in. laterally. Pipe lengths are: $L_1 = 8$ ft, $L_2 = 2$ ft, $L_b = 1$ ft, $L_3 = 12$ ft. What are the forces and moments acting at points "A," "B," and "C"?

Data provided by the Expansion Joint manufacturer:

$$A_e = 510.7 \text{ in.}^2$$

$$D_m = 25.50 \text{ in.}$$

$$f_w = 36840 \text{ lbs./in. per convolution}$$

$$L_b = 12 \text{ in.}$$

$$N = 12$$

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SOLUTION:

Equivalent axial movement per convolution.

$$\begin{aligned}
 e_x &= \frac{x}{N} \\
 &= \frac{0.405}{12} \\
 &= 0.034 \text{ in.} \\
 e_y &= \frac{3D_m y}{N(L_b - x)} \text{ Equation (4-5) Section 4.1} \\
 &= \frac{(3)(25.5)(0.216)}{12(12 - 0.405)} \\
 &= 0.119 \text{ in.}
 \end{aligned}$$

Calculation of F_s , F , V

$$\begin{aligned}
 F_s &= A_e P_d \\
 &= (510.7)(125) \\
 &= 63838 \text{ lbs.} \\
 F_a &= (f_w)(e_x) \\
 &= (36840)(.034) \\
 &= 1253 \text{ lbs.} \\
 V_l &= \frac{f_w D_m e_y}{2L_b} \text{ Equation (4-18) Section 4.6.1} \\
 &= \frac{(36840)(25.5)(.119)}{(2)(12)} \\
 &= 4658 \text{ lbs.}
 \end{aligned}$$

Forces and moments acting on intermediate anchor, IA, "A"

$F_x = 0$ (forces in X direction restrained by directional main anchor, DMA, "B")

$F_y = 4658 \text{ lbs.}$

$F_z = 0$ (no forces exist in Z direction)

$$\begin{aligned}
 M_x &= F_z Y - F_y Z \\
 &= 0
 \end{aligned}$$

$$\begin{aligned}
 M_y &= F_x Z - F_z X \\
 &= 0
 \end{aligned}$$

$$\begin{aligned}
 M_z &= F_y X - F_x Y \\
 &= (4658)(2.5) - 0 \\
 &= 11645 \text{ ft.lbs.}
 \end{aligned}$$

Where $Z = 0$

$$Y = -L_1 = -8 \text{ ft.}$$

$$X = L_2 + \frac{L_b}{2} = 2.5 \text{ ft.}$$

Forces and moments acting on directional main anchor, DMA, "B"

$$\begin{aligned}
 F_x &= -F_s - F_a \\
 &= -63838 - 1253 \\
 &= -65091 \text{ lbs.}
 \end{aligned}$$

$$F_y = 0 \text{ (DMA does not support in Y direction)}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$\begin{aligned}
 M_x &= F_z Y - F_y Z \\
 &= 0
 \end{aligned}$$

$$\begin{aligned}
 \text{Where } Y &= Z = 0 \\
 X &= L_2 + \frac{L_b}{2} = 2.5\text{ft.}
 \end{aligned}$$

$$\begin{aligned}
 M_y &= F_x Z - F_z X \\
 &= 0
 \end{aligned}$$

$$\begin{aligned}
 M_z &= F_y X - F_x Y \\
 &= 0
 \end{aligned}$$

Forces and moments acting on main anchor, MA, "C"

$$\begin{aligned}
 F_x &= F_s + F_a \\
 &= 63838 + 1253 \\
 &= 65091 \text{ lbs.}
 \end{aligned}$$

$$F_y = -4658 \text{ lbs.}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$\begin{aligned}
 M_x &= F_z Y - F_y Z \\
 &= 0
 \end{aligned}$$

$$\text{Where } Y = Z = 0$$

$$X = -\left(L_3 + \frac{L_b}{2}\right) = -12.5\text{ft.}$$

$$\begin{aligned}
 M_y &= F_x Z - F_z X \\
 &= 0
 \end{aligned}$$

$$\begin{aligned}
 M_z &= F_y X - F_x Y \\
 &= (-4658)(-12.5) - 0 \\
 &= 58225 \text{ ft.lbs.}
 \end{aligned}$$

Example 2M: Single Expansion Joint subjected to axial and lateral movement (similar to Figure 2.10, Section 2.4).

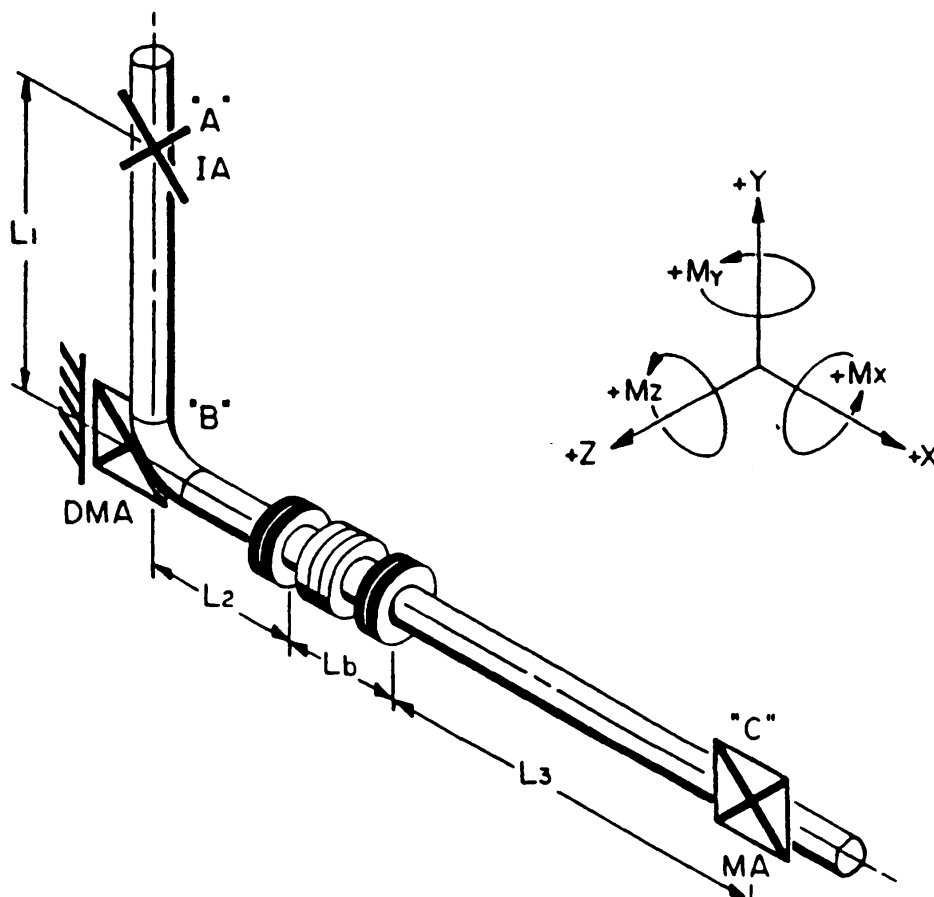


FIGURE J3

A single bellows Expansion Joint is placed in a 610 mm diameter carbon steel pipeline that runs between a main anchor and an intermediate anchor, and has one 90° elbow (reference Figure J3). The line is to operate at 0.86 MPa and 204°C. The thermal growth that the Expansion Joint is to absorb is calculated to be 10.3 mm axially, and 5.5 mm laterally. Pipe lengths are: $L_1 = 2.4$ m, $L_2 = 0.61$ m, $L_b = 0.30$ m, $L_3 = 3.66$ m. What are the forces and moments acting at points "A," "B," and "C"?

Data provided by the Expansion Joint manufacturer:

$$A_e = 329,500 \text{ mm}^2$$

$$D_m = 648 \text{ mm}$$

$$f_w = 6,451 \text{ N/mm per convolution}$$

$$L_b = 305 \text{ mm}$$

$$N = 12$$

SOLUTION:

Equivalent axial movement per convolution.

$$\begin{aligned}
 e_x &= \frac{x}{N} \\
 &= \frac{10.3}{12} \\
 &= 0.86 \text{ mm} \\
 e_y &= \frac{3D_m y}{N(L_b - x)} \text{ Equation (4-5) Section 4.1} \\
 &= \frac{3 \times 648 \times 5.5}{12 \times (305 - 10.3)} \\
 &= 3.02 \text{ mm}
 \end{aligned}$$

Calculation of F_s , F , V

$$\begin{aligned}
 F_s &= A_e P_d \\
 &= 329,500 \times 0.86 \\
 &= 284 \text{ kN} \\
 F_a &= (f_w)(e_x) \\
 &= 6,451 \times 0.86 \\
 &= 506 \text{ kN}
 \end{aligned}$$

$$\begin{aligned}
 V_l &= \frac{f_w D_m e_y}{2L_b} \text{ Equation (4-18) Section 4.6.1} \\
 &= \frac{6,451 \times 648 \times 3.02}{2 \times 305} \\
 &= 20.7 \text{ kN}
 \end{aligned}$$

Forces and moments acting on intermediate anchor, IA, "A"

$F_x = 0$ (forces in X direction restrained by directional main anchor, DMA, "B")

$$F_y = 20.7 \text{ kN}$$

$F_z = 0$ (no forces exist in Z direction)

Where $Z = 0$

$$M_x = F_z Y - F_y Z$$

$$Y = -L_1 = -2.4 \text{ m}$$

$$= 0$$

$$M_y = F_x Z - F_z X$$

$$X = L_2 + \frac{L_b}{2}$$

$$= 0$$

$$= .762 \text{ m}$$

$$M_z = F_y X - F_x Y$$

$$= 20.7 \times 0.762 - 0$$

$$= 15.8 \text{ Nm}$$

Forces and moments acting on directional main anchor, DMA, "B"

$$\begin{aligned} F_x &= -F_s - F_a \\ &= -284 - 5.6 \\ &= -289.6 \text{ kN} \end{aligned}$$

$$F_y = 0 \text{ (DMA does not support in Y direction)}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$\begin{aligned} M_x &= F_z Y - F_y Z \\ &= 0 \end{aligned}$$

$$\begin{aligned} \text{Where } Y &= Z = 0 \\ X &= L_2 + \frac{L_b}{2} \\ &= 0.762 \text{ m} \end{aligned}$$

$$\begin{aligned} M_y &= F_x Z - F_z X \\ &= 0 \end{aligned}$$

$$\begin{aligned} M_z &= F_y X - F_x Y \\ &= 0 \end{aligned}$$

Forces and moments acting on main anchor, MA, "C"

$$\begin{aligned} F_x &= F_s + F_a \\ &= 284 + 5.6 \\ &= 289.6 \text{ kN} \end{aligned}$$

$$F_y = -20.7 \text{ kN}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$\text{Where } Y = Z = 0$$

$$\begin{aligned} M_x &= F_z Y - F_y Z \\ &= 0 \end{aligned}$$

$$X = -\left(L_3 + \frac{L_b}{2}\right)$$

$$\begin{aligned} M_y &= F_x Z - F_z X \\ &= 0 \end{aligned}$$

$$= -3.81 \text{ m}$$

$$\begin{aligned} M_z &= F_y X - F_x Y \\ &= (-20.7) \times (-3.81) \\ &= 78.9 \text{ Nm} \end{aligned}$$

Example 3: Single Expansion Joint with tie rods subjected to axial and lateral movement (similar to Figure 2.11, Section 2.4)

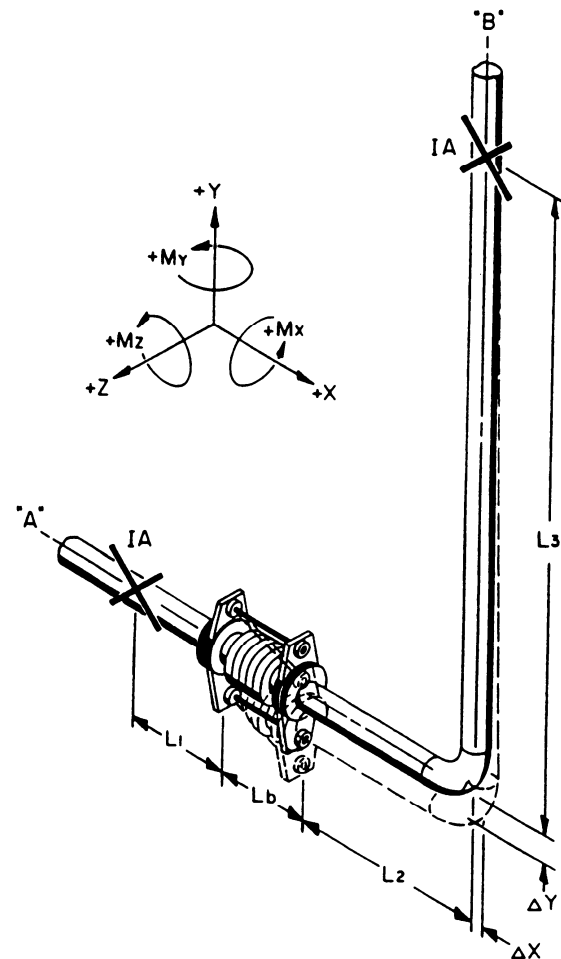


FIGURE J4

A tied single Expansion Joint is placed in a carbon steel 24 in. diameter pipe line that runs between two intermediate anchors and has a 90° bend. (Refer to Figure J4.) The line is to operate at 135 psig and 550° F. The pipe lengths of the system are $L_1 = 2$ ft., $L_b = 2$ ft., $L_2 = 3$ ft., $L_3 = 24$ ft. It is assumed that the tie rods are the same temperature and material as the pipe. The calculated thermal growth is .287 in. for the horizontal run and .984 in. for the vertical run of pipe. What are the bellows forces and moments on the intermediate anchors, "A" and "B"? Note: Forces and moments due to flexure of piping are not presented in these calculations.

Data provided by Expansion Joint manufacturer:

$$\begin{aligned} D_m &= 25.50 \text{ in.} \\ f_w &= 36840 \text{ lbs./in. per conv.} \\ N &= 18 \\ L_b &= 24 \text{ in.} \end{aligned}$$

SOLUTION:

Equivalent axial movement per convolution

$$\begin{aligned}e_x &= \frac{x}{N} \\&= 0 \text{ (tie rods prevent axial displacement)} \\e_y &= \frac{3D_m y}{N(L_b - x)} \\&= \frac{(3)(25.5)(.984)}{(18)(24 - 0)} \\&= .174 \text{ in.}\end{aligned}$$

Calculation for V_l

NOTE: for tied Expansion Joints F_s and $F_a = 0$

$$\begin{aligned}V_l &= \frac{f_w D_m e_y}{2L_b} \\&= \frac{(36840)(25.50)(.174)}{(2)(24)} \\&= 3405 \text{ lbs.}\end{aligned}$$

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Forces and moments acting on intermediate anchor, IA, "A"

$$F_x = 0 \text{ (note assumption 2 and 5)}$$

$$F_y = -V_l$$

$$= -3405 \text{ lbs.}$$

$$F_z = 0 \text{ (no forces exist in Z direction)} \quad \text{Where } Y = Z = 0$$

$$X = L_1 + \frac{L_b}{2} = 3 \text{ ft.}$$

$$M_x = F_z Y - F_y Z$$

$$= 0$$

$$M_y = F_x Z - F_z X$$

$$= 0$$

$$M_z = F_y X - F_x Y$$

$$= (-3405)(3) - 0$$

$$= -10215 \text{ ft.lbs.}$$

Forces and moments acting on intermediate anchor IA, "B"

$$F_x = 0$$

$$\text{Where } Z = 0$$

$$F_y = V_l$$

$$Y = -L_3 = -24 \text{ ft.}$$

$$= 3405 \text{ lbs.}$$

$$X = -L_2 - \frac{L_b}{2} = -4 \text{ ft.}$$

$$F_z = 0$$

$$M_x = 0$$

$$M_y = 0$$

$$M_z = F_y X - F_x Y$$

$$= (3405)(-4) - 0$$

$$= -13620 \text{ ft.lbs.}$$

Example 3M: Single Expansion Joint with tie rods subjected to axial and lateral movement (similar to Figure 2.11, Section 2.4)

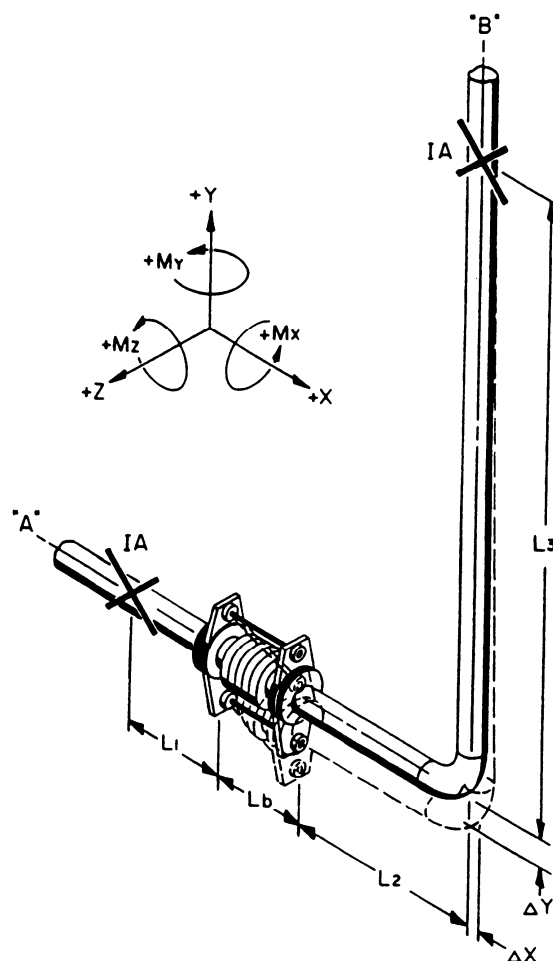


FIGURE J4

A tied single Expansion Joint is placed in a carbon steel 610 mm diameter pipe line that runs between two intermediate anchors and has a 90° bend. (Refer to Figure J4.) The line is to operate at 0.93 MPa and 288°C. The pipe lengths of the system are $L_1 = 0.61$ m, $L_b = 0.61$ m, $L_2 = 0.91$ m, $L_3 = 7.32$ m. It is assumed that the tie rods are the same temperature and material as the pipe. The calculated thermal growth is 7.29 mm for the horizontal run and 25 mm for the vertical run of pipe. What are the bellows forces and moments on the intermediate anchors, "A" and "B"? Note: Forces and moments due to flexure of piping are not presented in these calculations.

Data provided by Expansion Joint manufacturer:

$$\begin{aligned} D_m &= 648 \text{ mm} \\ f_w &= 6,451 \text{ N/mm per conv.} \\ N &= 18 \\ L_b &= 610 \text{ mm} \end{aligned}$$

SOLUTION:

Equivalent axial movement per convolution

$$\begin{aligned}e_x &= \frac{x}{N} \\&= 0 \text{ (tie rods prevent axial displacement)} \\e_y &= \frac{3D_m y}{N(L_b - x)} \\&= \frac{3 \times 648 \times 25}{18 \times (610 - 0)} \\&= 4.42 \text{ mm}\end{aligned}$$

Calculation for V_l

NOTE: for tied Expansion Joints F_s and $F_a = 0$

$$\begin{aligned}V_l &= \frac{f_w D_m e_y}{2L_b} \\&= \frac{6,451 \times 648 \times 4.42}{2 \times 610} \\&= 15.14 \text{ kN}\end{aligned}$$

Forces and moments acting on intermediate anchor, IA, "A"

$$F_x = 0 \text{ (note assumption 2 and 5)}$$

$$F_y = -V_l$$

$$= -15.14 \text{ kN}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$\text{Where } Y = Z = 0$$

$$X = L_1 + \frac{L_b}{2} = 0.91 \text{ m}$$

$$M_x = F_z Y - F_y Z$$

$$= 0$$

$$M_y = F_x Z - F_z X$$

$$= 0$$

$$M_z = F_y X - F_x Y$$

$$= -15.14 \times 0.91 - 0$$

$$= -13,800 \text{ Nm}$$

Forces and moments acting on intermediate anchor IA, "B"

$$F_x = 0$$

$$\text{Where } Z = 0$$

$$F_y = V_l$$

$$Y = -L_3 = -7.32 \text{ m}$$

$$= 15.14 \text{ kN}$$

$$X = -L_2 - \frac{L_b}{2} = -1.22 \text{ m}$$

$$F_z = 0$$

$$M_x = 0$$

$$M_y = 0$$

$$M_z = F_y X - F_x Y$$

$$= 15.14 \times (-1.22) - 0$$

$$= -18,470 \text{ Nm}$$

Example 4: Tied universal Expansion Joint subjected to lateral movement in two planes (similar to Figure 2.14, Section 2.5).

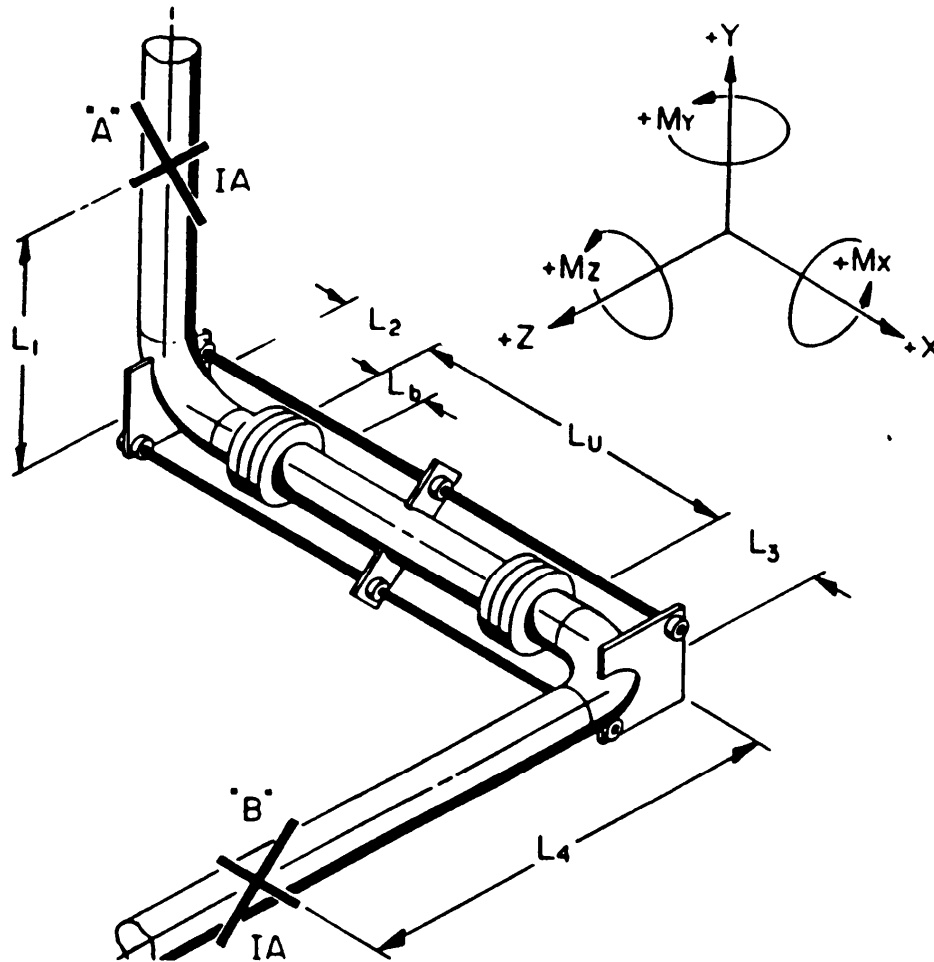


FIGURE J5

A 24 in. diameter carbon steel pipe line runs between two intermediate anchors A and B, as shown in Figure J5. The line operates at 100 psig, and 350° F. The pipe lengths are $L_1 = 35$ ft., $L_2 = 3.5$ ft., $L_u = 4$ ft., $L_3 = 3.5$ ft., $L_4 = 65$ ft., and $L_b = 1$ ft. The calculated thermal growth for each of these lengths is $\Delta L_1 = .788$ in., $\Delta L_2 = .079$ in., $\Delta L_u = .09$ in., $\Delta L_3 = .079$ in., $\Delta L_4 = 1.463$ in. and $\Delta L_b = .023$ in. What are the forces and moments at the intermediate anchors, IA, "A" and "B"?

Data provided by Expansion Joint manufacturer:

$$D_m = 25.50 \text{ in.}$$

$$f_w = 36840 \text{ lbs./in. per convolution}$$

$$L_b = 12 \text{ in.}$$

$$N = 12$$

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SOLUTION:

Equivalent axial movement per convolution, Equation (4-2) and (4-6), Section 4.1.

$$\begin{aligned} e_x &= \frac{x}{2N} \\ &= \frac{.248}{(2)(12)} \\ &= .010 \text{ in.} \end{aligned}$$

Where thermal growth x
between tie rod plates is:

$$\begin{aligned} x &= \Delta L_2 + \Delta L_u + \Delta L_3 \\ &= .079 + .090 + .079 \\ &= .248 \text{ in} \end{aligned}$$

$$\begin{aligned} (e_y)_y &= \frac{3 D_m}{2 N L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^* y_y}{(L^* - x/2)} \\ &= \frac{(3)(25.5)}{(2)(12)(12)} \cdot \frac{(1) + (3)}{(1) + (3)(3)^2} \cdot \frac{36(.788)}{(36 - .248/2)} \\ &= (.2656)(.14286)(1.005)(.788) \\ &= .030 \text{ in} \end{aligned}$$

$$\begin{aligned} \text{Where } L^* &= L_u - L_b = 48 - 12 \\ &= 36 \text{ in} \\ \text{and } L^*/L_b &= 36/12 = 3 \\ y_y &= .788 \text{ in} \end{aligned}$$

$$\begin{aligned} (e_y)_z &= \frac{3 D_m}{2 N L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^* y_z}{(L^* - x/2)} \\ &= \frac{(3)(25.5)}{(2)(12)(12)} \cdot \frac{(1) + (3)}{(1) + (3)(3)^2} \cdot \frac{36(1.463)}{(36 - .248/2)} \\ &= (.2656)(.14286)(1.005)(1.463) \\ &= .056 \text{ in} \end{aligned}$$

$$\text{Where } y_z = 1.463 \text{ in}$$

NOTE: First subscript applies to direction related to the bellows axis, second subscript to system coordinate axis.

Calculation for V_l , Equation (4-19), Section 4.6.1.

$$V_l = \frac{f_w D_m e_y}{2L_u}$$

$$(V_l)_y = \frac{(36840)(25.50)(.030)}{(2)(48)}$$

$$= 294 \text{ lbs.}$$

$$(V_l)_z = \frac{(36840)(25.50)(.056)}{(2)(48)}$$

$$= 548 \text{ lbs.}$$

NOTE First subscript refers to bellows
: lateral movement, second subscript
to system co-ordinate axis.

Forces and moments acting on intermediate anchor IA, "A"

$$F_y = -(V_l)_y \quad \text{where } X = L_2 + L_u / 2 = 5.5 \text{ ft.}$$

$$= -294 \text{ lbs.} \quad Y = -35 \text{ ft.}$$

$$F_x = 0 \text{ (Force due to axial bellows movements, } e_x \text{ is restrained by tie rods)} \quad Z = 0$$

$$F_z = (V_l)_z$$

$$= 548 \text{ lbs.}$$

$$M_x = F_z Y - F_y Z$$

$$= (548)(-35) - 0$$

$$= -19180 \text{ ft.lbs.}$$

$$M_y = F_x Z - F_z X$$

$$= 0 - (548)(5.5)$$

$$= -3014 \text{ ft. lbs.}$$

$$M_z = F_y X - F_x Y$$

$$= -294(5.5) - 0$$

$$= -1617 \text{ ft.lbs.}$$

Forces and moments acting on intermediate anchor IA, "B"

$$F_y = 294 \text{ lbs.}$$

$$\text{where } X = -(L_3 + L_u / 2) = -5.5 \text{ ft.}$$

$$F_x = 0$$

$$Y = 0$$

$$F_z = 548 \text{ lbs.}$$

$$Z = -65 \text{ ft.}$$

$$\begin{aligned} M_x &= F_z Y - F_y Z \\ &= 0 - 294(-65) \\ &= 19,110 \text{ ft.lbs.} \end{aligned}$$

$$\begin{aligned} M_y &= F_x Z - F_z X \\ &= 0 - (548)(-5.5) \\ &= 3014 \text{ ft. lbs.} \end{aligned}$$

$$\begin{aligned} M_z &= F_y X - F_x Y \\ &= (294)(-5.5) - 0 \\ &= -1617 \text{ ft. lbs.} \end{aligned}$$

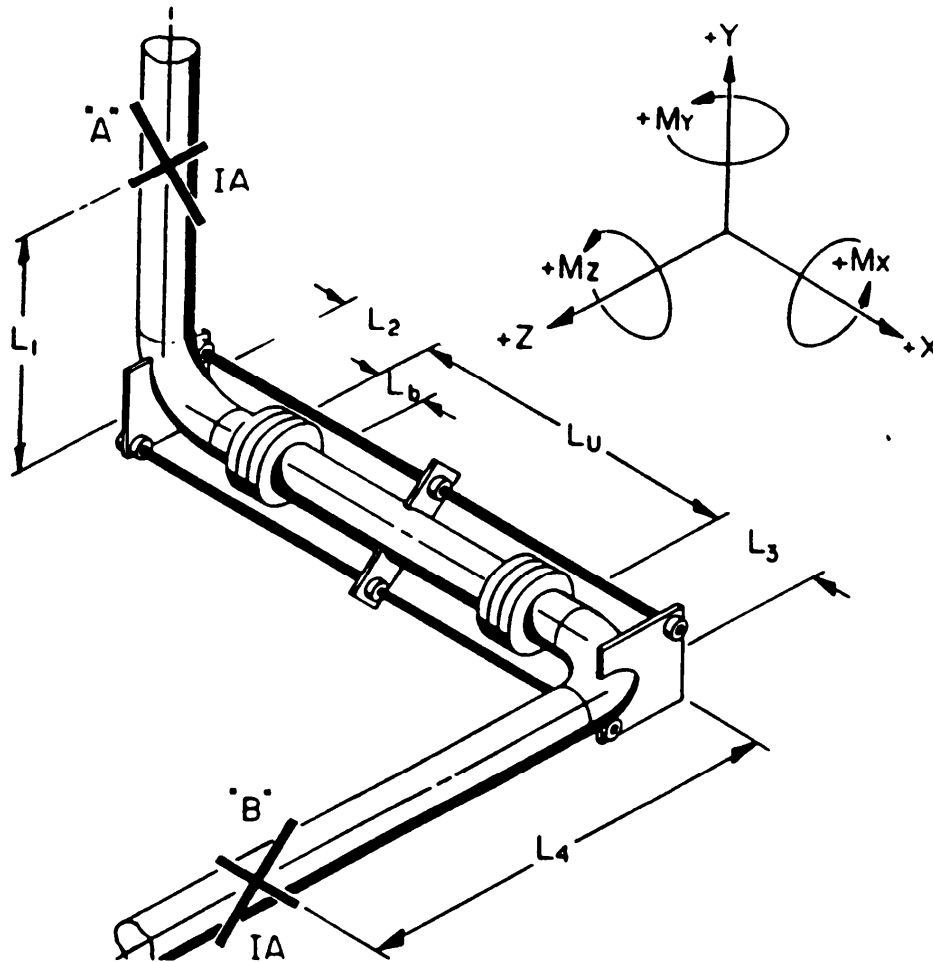
Example 4M: Tied universal Expansion Joint subjected to lateral movement in two planes (similar to Figure 2.14, Section 2.5).

FIGURE J5

A610 mm diameter carbon steel pipe line runs between two intermediate anchors A and B, as shown in Figure J5. The line operates at 0.69 MPa, and 177°C. The pipe lengths are $L_1 = 10.7$ m, $L_2 = 1.07$ m, $L_u = 1.22$ m, $L_3 = 1.07$ m, $L_4 = 19.8$ m, and $L_b = 0.305$ m. The calculated thermal growth for each of these lengths is $\Delta L_1 = 20$ mm, $\Delta L_2 = 2$ mm, $\Delta L_u = 2.3$ mm, $\Delta L_3 = 2$ mm, $\Delta L_4 = 37.2$ mm, and $\Delta L_b = 0.58$ mm. What are the forces and moments at the intermediate anchors, IA, "A" and "B"?

Data provided by Expansion Joint manufacturer:

$$D_m = 648 \text{ mm}$$

$$f_w = 6,451 \text{ N/mm per convolution}$$

$$L_b = 305 \text{ mm}$$

$$N = 12$$

SOLUTION:

Equivalent axial movement per convolution, Equation (4-2) and (4-6), Section 4.1.

$$\begin{aligned} e_x &= \frac{x}{2N} \\ &= \frac{6.3}{(2)(12)} \\ &= 0.26 \text{ mm} \end{aligned}$$

Where thermal growth x
between tie rod plates is:

$$\begin{aligned} x &= \Delta L_2 + \Delta L_u + \Delta L_3 \\ &= 2.01 + 2.29 + 2.01 \\ &= 6.3 \text{ mm} \end{aligned}$$

$$\begin{aligned} (e_y)_y &= \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^* y_y}{(L^* - x/2)} \\ &= \frac{3 \times 648}{2 \times 12 \times 305} \cdot \frac{1 + 3}{1 + 3 \times 3^2} \cdot \frac{914 \times 20}{914 - 6.3/2} \\ &= 0.2656 \times 0.14286 \times 1.003 \times 20 \\ &= 0.672 \text{ mm} \end{aligned}$$

$$\begin{aligned} \text{Where } L^* &= L_u - L_b = 1219 - 305 \\ &= 914 \text{ mm} \end{aligned}$$

$$\begin{aligned} \text{and } L^*/L_b &= 36/12 = 3 \\ y_y &= 20 \text{ mm} \end{aligned}$$

$$\begin{aligned} (e_y)_z &= \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^* y_z}{(L^* - x/2)} \\ &= \frac{(3)(648)}{(2)(12)(305)} \cdot \frac{(1) + (914/305)}{(1) + (3)(914/305)^2} \cdot \frac{914(1.463)}{(914 - 6.3/2)} \\ &= (.2656)(.14286)(1.003)(37.16) \\ &= 1.42 \text{ mm} \end{aligned}$$

$$\text{Where } y_z = 37.16 \text{ mm}$$

NOTE: First subscript applies to direction related to the bellows axis, second subscript to system coordinate axis.

Calculation for V_l , Equation (4-18, 4-19), Section 4.6.1.

$$V_l = \frac{f_w D_m e_y}{2L_u}$$

$$(V_l)_y = \frac{6451 \times 648 \times 0.762}{2 \times 1219}$$

$$= 1307 \text{ N}$$

$$(V_l)_z = \frac{6451 \times 648 \times 1.42}{2 \times 121}$$

$$= 2438 \text{ N}$$

NOTE First subscript refers to bellows
: lateral movement, second subscript
to system co-ordinate axis.

Forces and moments acting on intermediate anchor IA, "A"

$$F_y = -(V_l)_y \quad \text{where } X = L_2 + L_u / 2 = 1677 \text{ mm}$$

$$= -1307 \text{ N} \quad Y = -10668 \text{ mm}$$

$$F_x = 0 \text{ (Force due to axial bellows movements, } e_x \text{ is restrained by tie rods)} \quad Z = 0$$

$$F_z = (V_l)_z$$

$$= 2438 \text{ N}$$

$$M_x = F_z Y - F_y Z$$

$$= 2438 \times (-10668) - 0$$

$$= -26,009 \text{ Nm}$$

$$M_y = F_x Z - F_z X$$

$$= 0 - 2438 \times 1677$$

$$= -4,089 \text{ Nm}$$

$$M_z = F_y X - F_x Y$$

$$= -1307 \times 1677 - 0$$

$$= -2,192 \text{ Nm}$$

Forces and moments acting on intermediate anchor IA, "B"

$$F_y = 1307 \text{ N}$$

$$\text{where } X = -(L_3 + L_u / 2) = -1,677 \text{ mm}$$

$$F_x = 0$$

$$Y = 0$$

$$F_z = 2438 \text{ N}$$

$$Z = -19,812 \text{ mm}$$

$$\begin{aligned} M_x &= F_z Y - F_y Z \\ &= 0 - 1307 \times (-19812) \\ &= 25,894 \text{ Nm} \end{aligned}$$

$$\begin{aligned} M_y &= F_x Z - F_z X \\ &= 0 - 2438 \times (-1677) \\ &= 4,089 \text{ Nm} \end{aligned}$$

$$\begin{aligned} M_z &= F_y X - F_x Y \\ &= 1307 \times (-1677) - 0 \\ &= -2,192 \text{ Nm} \end{aligned}$$

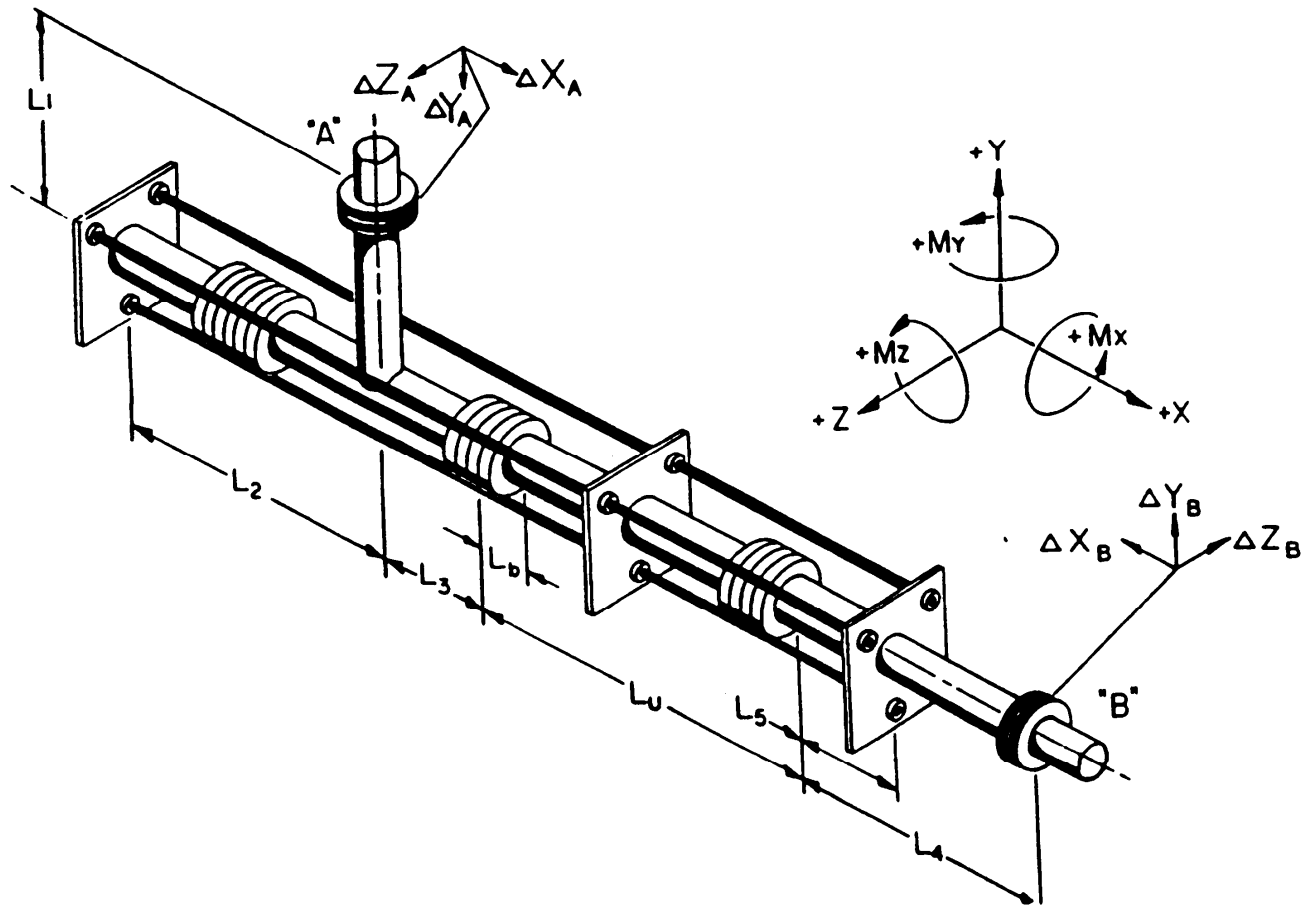
Example 5: Universal pressure balanced Expansion Joint located between two pieces of equipment with movements at end points.

FIGURE J6

A 66 in. diameter turbine exhaust duct system, shown in Figure J6, is fabricated of steel and operates at full vacuum and 250° F. Movements at the turbine exhaust outlet flange and condenser inlet are determined to be (for the direction shown):

	Axis	X	Y	Z
Point A	TURBINE	0.07 in.	0.12 in.	0
Point B	CONDENSER	0.26 in.	0.18 in.	0.12 in.

(Directions shown in Figure J6)

Thermal growth calculations for the 66 in. diameter piping are:

$L_1 = 10$ ft.	$\Delta L_1 = .140$ in.
$L_3 = 6$ ft.	$\Delta L_3 = .084$ in.
$L_u = 5$ ft.	$\Delta L_u = .070$ in.
$L_4 = 28$ ft.	$\Delta L_4 = .392$ in.
$L_b = 1$ ft.	$\Delta L_b = .014$ in.

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$$\begin{array}{ll} L_2 = 3 \text{ ft.} & \Delta L_2 = .042 \text{ in.} \\ L_5 = 1 \text{ ft. } 8.6 \text{ in.} & \Delta L_5 = .024 \text{ in.} \end{array}$$

Determine the forces and moments due to the bellows stiffness at the condenser and turbine connections.

Data provided by the Expansion Joint manufacturer:

$$\begin{array}{l} D_m = 68.00 \text{ in.} \\ f_w = 35425 \text{ lbs./in. per convolution} \\ N_f = 6 \text{ (Number of convolutions in one flow bellows)} \\ N_b = 6 \text{ (Number of convolutions in balancing bellows)} \end{array}$$

SOLUTION:

Calculate the total movement the Expansion Joint must absorb.

Flow bellows:

$$\begin{aligned} x &= \Delta L_3 + \Delta L_u + \Delta L_4 + \Delta X_A + \Delta X_B \\ &= .084 + .070 + .392 + .07 + .260 \\ &= .876 \text{ in.} \end{aligned}$$

$$\begin{aligned} y_y &= \Delta L_1 + \Delta Y_A + \Delta Y_B \\ &= .140 + 0.12 + 0.18 \\ &= .440 \text{ in.} \end{aligned}$$

$$\begin{aligned} y_z &= \Delta Z_A + \Delta Z_B \\ &= 0 + .12 \\ &= .12 \text{ in.} \end{aligned}$$

Balancing bellows:

$$\begin{aligned} x &= (\Delta L_4 - \Delta L_5) - \Delta L_2 + \Delta X_A + \Delta X_B \\ &= .392 - .024 - .042 + .07 + .260 \\ &= .656 \text{ in.} \end{aligned}$$

Calculate equivalent movement per convolution.

$$e_x = \frac{x}{2N_f} \quad (\text{Flow Bellows})$$

$$= \frac{.876}{(2)(6)}$$

$$= .073 \text{ in}$$

$$e_x = \frac{x}{2N_f} \quad (\text{Balancing Bellows})$$

$$= \frac{.656}{6}$$

$$= .109 \text{ in}$$

$$(e_y)_y = \frac{3D_m}{2N_f L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_y$$

$$= \frac{(3)(68)}{(2)(6)(12)} \cdot \frac{(1+4)}{[1+(3)(4)^2]} \cdot \frac{(48)}{(48-.876/2)} (.44)$$

$$= (1.417)(.102)(1.007)(.44)$$

$$= .064 \text{ in}$$

$$\text{Where } L^* = L_u - L_b$$

$$= 60 - 12$$

$$= 48 \text{ in.}$$

$$L^*/L_b = 48/12$$

$$= 4$$

$$\text{and } y_y = .440 \text{ in.}$$

$$(e_y)_z = \frac{3D_m}{2N_f L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_z$$

$$= \frac{(3)(68)}{(2)(6)(12)} \cdot \frac{(1+4)}{[1+(3)(4)^2]} \cdot \frac{(48)}{(48-.876/2)} (.12)$$

$$= (1.417)(.102)(1.007)(.12)$$

$$= .017 \text{ in}$$

$$\text{Where } y_z = .12 \text{ in.}$$

Calculation of F_x , $(V_l)_y$ and $(V_l)_z$, Equations (4-15) and (4-19) Section 4.6.1.

$$F_x = fw(e_{x \text{ flow}} + e_{x \text{ balancing}})$$

$$= 35423(.073 + .109)$$

$$= 6447 \text{ lbs.}$$

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$$\begin{aligned}(V_l)_y &= \frac{f_w D_m (e_y)_y}{2L_u} \\ &= \frac{(35425)(68)(.064)}{(2)(60)} \\ &= 1285 \text{ lbs.}\end{aligned}$$

$$\begin{aligned}(V_l)_z &= \frac{f_w D_m (e_y)_z}{2L_u} \\ &= \frac{(35425)(68)(.017)}{(2)(60)} \\ &= 341 \text{ lbs.}\end{aligned}$$

Forces and moments acting on turbine flange "A"

$$F_x = -6447 \text{ lbs.}$$

$$F_y = (V_l)_y = 1285 \text{ lbs.}$$

$$F_z = -(V_l)_z = -341 \text{ lbs.}$$

$$M_x = F_z Y - F_y Z$$

$$\begin{aligned}&= (-341)(-10) - 0 \\ &= 3410 \text{ ft. lbs.}\end{aligned}$$

$$M_y = F_x Z - F_z X$$

$$\begin{aligned}&= 0 - (-341)(8.5) \\ &= 2899 \text{ ft. lbs.}\end{aligned}$$

$$M_z = F_y X - F_x Y$$

$$\begin{aligned}&= (1285)(8.5) - (-6447)(-10) \\ &= 10923 - 64470 \\ &= -53547 \text{ ft. lbs.}\end{aligned}$$

$$\text{where: } X = L_3 + \frac{L_u}{2} = 8.5 \text{ ft}$$

$$Y = L_1 = -10 \text{ ft.}$$

$$Z = 0$$

Forces and moments acting on condenser connection "B"

$$F_x = 6447 \text{ lbs.}$$

$$-F_y = -(V_l)_y = -1285 \text{ lbs.}$$

$$F_z = (V_l)_z = 341 \text{ lbs.}$$

$$M_x = F_z Y - F_y Z$$

$$= 341(0) - (-1285)(0)$$

$$= 0$$

$$M_y = F_x Z - F_z X$$

$$= 0 - (341)(-30.5)$$

$$= 10400 \text{ ft. lbs.}$$

$$M_z = F_y X - F_x Y$$

$$= (-1285)(-30.5) - 0$$

$$= 39193 \text{ ft. lbs.}$$

$$\text{where: } X = -\left(\frac{L_u}{2} + L_4\right)$$

$$= -30.5 \text{ ft.}$$

$$Y = Z = 0$$

Example 5M: Universal pressure balanced Expansion Joint located between two pieces of equipment with movements at end points.

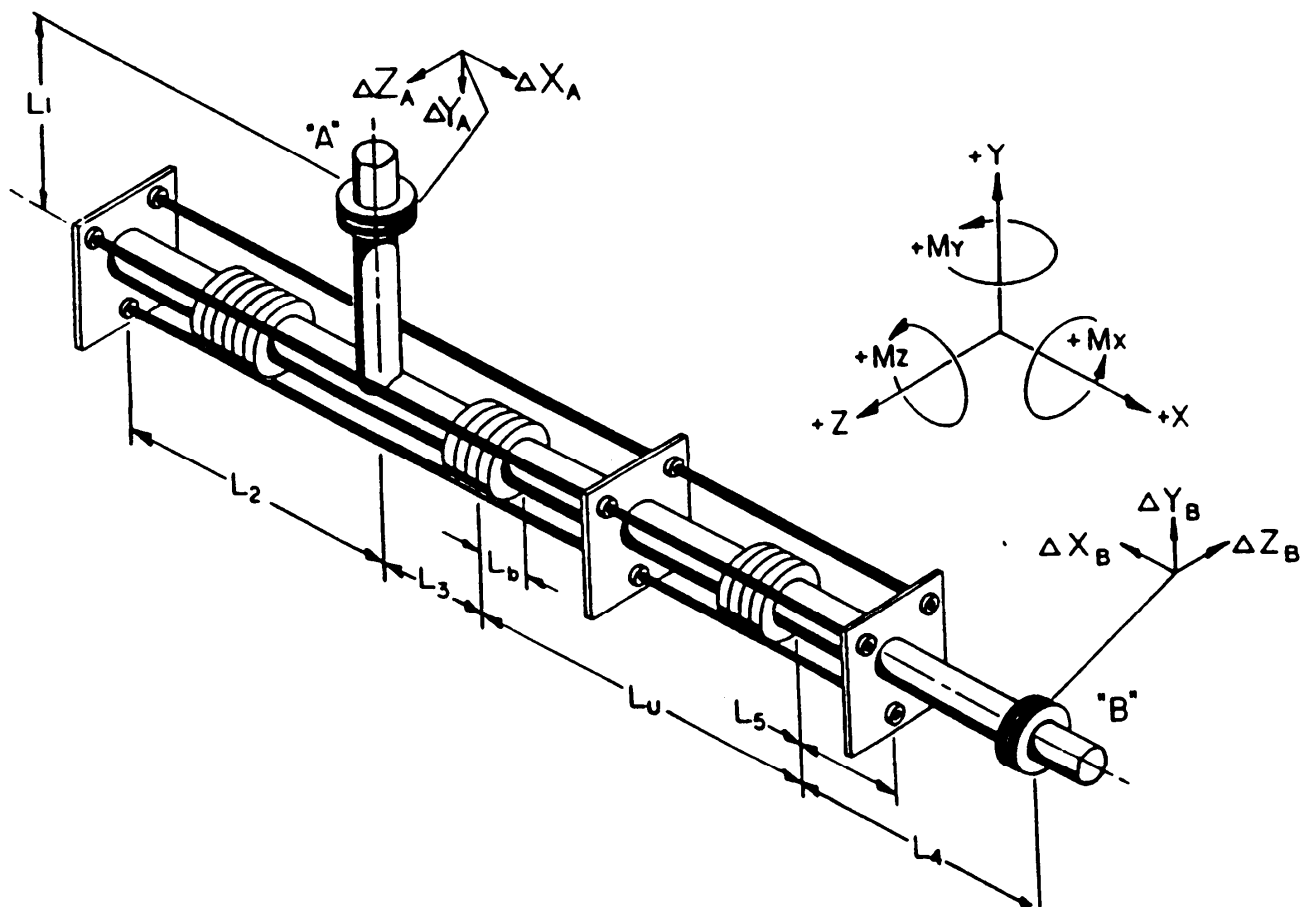


FIGURE J6

A 1676 mm diameter turbine exhaust duct system, shown in Figure J6, is fabricated of steel and operates at full vacuum and 121° C. Movements at the turbine exhaust outlet flange and condenser inlet are determined to be (for the direction shown):

	Axis	X	Y	Z
Point A	TURBINE	1.78 mm	3.05 mm	0
Point B	CONDENSER	6.60 mm	4.57 mm	3.05 mm

(Directions shown in Figure J6)

Thermal growth calculations for the 1676 mm diameter piping are:

$L_1 = 3048 \text{ mm}$	$\Delta L_1 = 3.56 \text{ mm}$
$L_3 = 1829 \text{ mm}$	$\Delta L_3 = 2.13 \text{ mm}$
$L_u = 1524 \text{ mm}$	$\Delta L_u = 1.78 \text{ mm}$
$L_4 = 8534 \text{ mm}$	$\Delta L_4 = 9.96 \text{ mm}$
$L_b = 305 \text{ mm}$	$\Delta L_b = 0.356 \text{ mm}$

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$$L_2 = 914 \text{ mm}$$

$$\Delta L_2 = 1.07 \text{ mm}$$

$$L_5 = 523 \text{ mm}$$

$$\Delta L_5 = 0.610 \text{ mm}$$

Determine the forces and moments due to the bellows stiffness at the condenser and turbine connections.

Data provided by the Expansion Joint manufacturer:

$$D_m = 1727 \text{ mm}$$

$$f_w = 6203 \text{ N/mm per convolution}$$

$$N_f = 6 \text{ (Number of convolutions in one flow bellows)}$$

$$N_b = 6 \text{ (Number of convolutions in balancing bellows)}$$

SOLUTION:

Calculate the total movement the Expansion Joint must absorb.

Flow bellows:

$$x = \Delta L_3 + \Delta L_u + \Delta L_4 + \Delta X_A + \Delta X_B$$

$$= 2.13 + 1.78 + 9.96 + 1.78 + 6.60$$

$$= 22.25 \text{ mm}$$

$$y_y = \Delta L_1 + \Delta Y_A + \Delta Y_B$$

$$= 3.56 + 3.05 + 4.57$$

$$= 11.18 \text{ mm}$$

$$y_z = \Delta Z_A + \Delta Z_B$$

$$= 0 + 3.05$$

$$= 3.05 \text{ mm}$$

Balancing bellows:

$$x = (\Delta L_4 - \Delta L_5) - \Delta L_2 + \Delta X_A + \Delta X_B$$

$$= 9.96 - 0.610 - 1.07 + 1.78 + 6.60$$

$$= 16.66 \text{ mm}$$

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Calculate equivalent movement per convolution.

$$\begin{aligned} e_x &= \frac{x}{2N_f} \quad (\text{Flow Bellows}) \\ &= \frac{22.25}{(2)(6)} \\ &= 1.85 \text{ mm} \end{aligned}$$

$$\begin{aligned} e_x &= \frac{x}{2N_f} \quad (\text{Balancing Bellows}) \\ &= \frac{16.66}{6} \\ &= 2.77 \text{ mm} \end{aligned}$$

$$\begin{aligned} (e_y)_y &= \frac{3D_m}{2N_f L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_y \\ &= \frac{3 \times 1727}{2 \times 6 \times 305} \cdot \frac{1 + 4}{1 + 3 \times 4^2} \cdot \frac{1219}{1219 - 22.25/2} \cdot 44 \\ &= 1.417 \times 0.102 \times 1.009 \times 11.2 \\ &= 1.63 \text{ mm} \end{aligned}$$

Where

$$\begin{aligned} L^* &= L_u - L_b \\ &= 1524 - 305 \\ &= 1219 \text{ mm} \end{aligned}$$

$$L^*/L_b = 48/12 = 4$$

and

$$y_y = 11.2 \text{ mm}$$

$$\begin{aligned} (e_y)_z &= \frac{3D_m}{2N_f L_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_z \\ &= \frac{3 \times 1727}{2 \times 6 \times 305} \cdot \frac{1 + 4}{1 + 3 \times 4^2} \cdot \frac{1219}{1219 - 22.25/2} \cdot 3.05 \\ &= 1.417 \times 0.102 \times 1.007 \times 3.05 \\ &= 0.44 \text{ mm} \end{aligned}$$

Where

$$y_z = 3.05 \text{ mm}$$

Calculation of F_x , $(V_l)_y$ and $(V_l)_z$, Equations (4-15) and (4-19) Section 4.6.1.

$$\begin{aligned} F_x &= fw(e_{x \text{ flow}} + e_{x \text{ balancing}}) \\ &= 6203 \times (1.85 + 2.77) \\ &= 287 \text{ kN} \end{aligned}$$

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$$\begin{aligned}(V_l)_y &= \frac{f_w D_m (e_y)_y}{2L_u} \\ &= \frac{6203 \times 1727 \times 1.63}{2 \times 1524} \\ &= 5.715 \text{ kN}\end{aligned}$$

$$\begin{aligned}(V_l)_z &= \frac{f_w D_m (e_y)_z}{2L_u} \\ &= \frac{6203 \times 1727 \times 0.432}{2 \times 1524} \\ &= 1.518 \text{ kN}\end{aligned}$$

Forces and moments acting on turbine flange "A"

$$F_x = -28.66 \text{ kN}$$

$$F_y = (V_l)_y = 5.716 \text{ kN}$$

$$F_z = -(V_l)_z = -1.518 \text{ kN}$$

$$\begin{aligned}M_x &= F_z Y - F_y Z \\ &= (-1,518) \times (-3048) - 0 \\ &= 4627 \text{ Nm}\end{aligned}$$

Where:

$$X = L_3 + \frac{L_u}{2} = 2591 \text{ mm}$$

$$\begin{aligned}M_y &= F_x Z - F_z X \\ &= 0 - (-1,518) \times (2591) \\ &= 3933 \text{ Nm}\end{aligned}$$

$$Y = L_1 = -3048 \text{ mm}$$

$$Z = 0$$

$$\begin{aligned}M_z &= F_y X - F_x Y \\ &= 5,729 \times 2591 - (-28,66) \times (-3048) \\ &= 14,844 - 87,350 \\ &= -72,506 \text{ Nm}\end{aligned}$$

Forces and moments acting on condenser connection "B"

$$F_x = 28,676 \text{ kN}$$

$$-F_y = -(V_l)_y = -5.716 \text{ kN}$$

$$F_z = (V_l)_z = 1.516 \text{ kN}$$

$$\begin{aligned} M_x &= F_z Y - F_y Z \\ &= 1516 \times 0 - (-5716) \times 0 \\ &= 0 \end{aligned}$$

$$\begin{aligned} \text{where: } X &= -\left(\frac{L_u}{2} + L_4\right) \\ &= -9296 \text{ mm} \\ Y &= Z = 0 \end{aligned}$$

$$\begin{aligned} M_y &= F_x Z - F_z X \\ &= 0 - 1.516 \times (-9296) \\ &= 14,093 \text{ Nm} \end{aligned}$$

$$\begin{aligned} M_z &= F_y X - F_x Y \\ &= (-5.716) \times (-9296) - 0 \\ &= 53,136 \text{ Nm} \end{aligned}$$

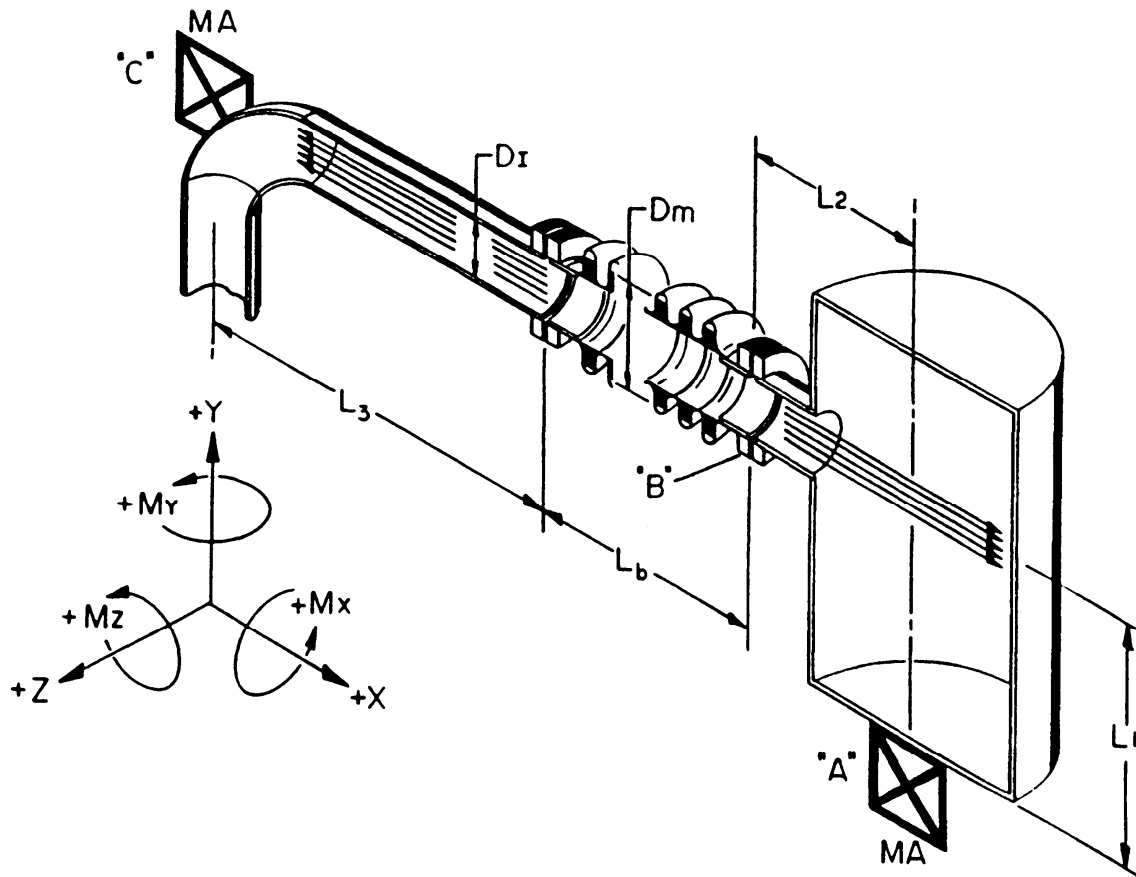
Example 6: Single Expansion Joint, attached to vessel nozzle, subjected to axial and lateral movement.

FIGURE J7

A large vertical vessel which operates at 150 psig at 500° F is equipped with a 24 in. diameter outlet line as shown in Figure J7. The outlet line contains a single bellows Expansion Joint which is designed to absorb the thermal growth of the vessel and pipe line. The lengths and calculated thermal growths for the all-carbon steel system are as follows:

$L_1 = 15 \text{ ft.}$	$\Delta L_1 = .543 \text{ in.}$
$L_2 = 6 \text{ ft.}$	$\Delta L_2 = .217 \text{ in.}$
$L_b = 1 \text{ ft.}$	$\Delta L_b = .036 \text{ in.}$
$L_3 = 14 \text{ ft.}$	$\Delta L_3 = .507 \text{ in.}$

Determine the forces and moments acting on the nozzle flange "B"

Data provided by the Expansion Joint manufacturer:

$$D_m = 25.50 \text{ in.}$$

$$D_l = 23.25 \text{ in.}$$

$$f_w = 36840 \text{ lbs./in. per convolution}$$

$$N = 12$$

SOLUTION:

Calculate the equivalent movements per convolution:

$$\begin{aligned} e_x &= \frac{x}{N} \\ &= \frac{.760}{12} \\ &= .063 \text{ in.} \end{aligned}$$

$$\begin{aligned} \text{where: } x &= \Delta L_2 + \Delta L_b + L_3 \\ &= .217 + .036 + .507 \\ &= .760 \text{ in.} \end{aligned}$$

$$\begin{aligned} e_y &= \frac{3D_m y}{N(L_b - x)} \\ &= \frac{(3)(25.50)(.543)}{(12)(12 - .76)} \\ &= .308 \text{ in.} \end{aligned}$$

$$\begin{aligned} y &= \Delta L_1 = .543 \text{ in.} \\ L_b &= 12 \text{ in.} \end{aligned}$$

Calculate F_a , F_s , F_p and V .

$$F_a = (f_w)(e_x)$$

$$= (36840)(.063)$$

$$= 2321 \text{ lbs.}$$

$$F_s = (A_e)(P_d)$$

$$= (510.7)(150)$$

$$= 76605 \text{ lbs.}$$

$$F_p = (A_p)(P_d)$$

$$= (424.6)(150)$$

$$= 63690 \text{ lbs.}$$

$$V_l = \frac{f_w D_m e_y}{2L_b}$$

$$= \frac{(36840)(25.50)(.308)}{(2)(12)}$$

$$= 12056 \text{ lbs.}$$

where:

$$A_e = \frac{\pi}{4}(D_m^2)$$

$$= \frac{\pi}{4}(25.5)^2$$

$$= 510.7 \text{ in.}^2$$

$$P_d = 150 \text{ psig}$$

$$A_p = \frac{\pi}{4}(D_l^2)$$

$$= \frac{\pi}{4}(23.25)^2$$

$$= 424.6 \text{ in.}^2$$

$$L_b = 12 \text{ in.}$$

Forces and moments acting on vessel anchor "A"

$$F_x = F_a + F_s = 2321 + 76605$$

$$= 78926 \text{ lbs.}$$

$$F_y = -V_l = -12056 \text{ lbs.}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$M_x = F_z Y - F_y Z = 0$$

$$M_y = F_x Z - F_z X = 0$$

$$M_z = F_y X - F_x Y$$

$$= (-12056)(-6.5) - (78926)(15)$$

$$= -1,105,526 \text{ ft. lbs.}$$

$$X = -(L_2 + L_b / 2)$$

$$= -6.5 \text{ ft.}$$

where:

$$Y = L_1 = 15 \text{ ft.}$$

$$Z = 0$$

Forces and moments acting on nozzle flange face "B"

$$\begin{aligned}
 F_x &= F_a + F_s - F_p \\
 &= 2321 + 76605 - 63690 \\
 &= 15236 \text{ lbs.} \\
 F_y &= -12056 \text{ lbs.} \\
 F_z &= 0
 \end{aligned}$$

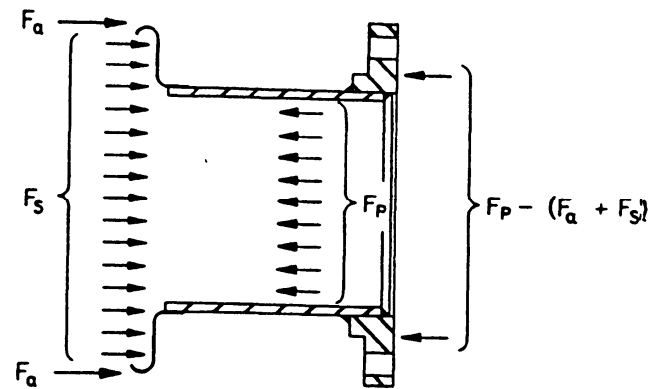


FIGURE J8

$$\begin{aligned}
 M_x &= F_z Y - F_y Z = 0 \\
 M_y &= F_x Z - F_z X = 0 \\
 M_z &= F_y X - F_x Y \\
 &= (-12056)(-0.5) - 0 \\
 &= 6028 \text{ ft. lbs.}
 \end{aligned}$$

where: $X = L_b / 2 = -0.5 \text{ ft.}$

$$Y = Z = 0$$

Forces and moments acting on main anchor "C"

$$\begin{aligned}
 F_x &= -F_a - F_s \\
 &= -2321 - 76605 \\
 &= -78926 \text{ lbs.} \\
 F_y &= 12056 \text{ lbs.} \\
 F_z &= 0 \\
 M_x &= F_z Y - F_y Z = 0 \\
 M_y &= F_x Z - F_z X = 0 \\
 M_z &= F_y X - F_x Y \\
 &= (12056)(14.5) - 0 \\
 &= 174812 \text{ ft. lbs.}
 \end{aligned}$$

where: $X = L_3 + L_b / 2 = 14.5 \text{ ft.}$

$$Y = Z = 0$$

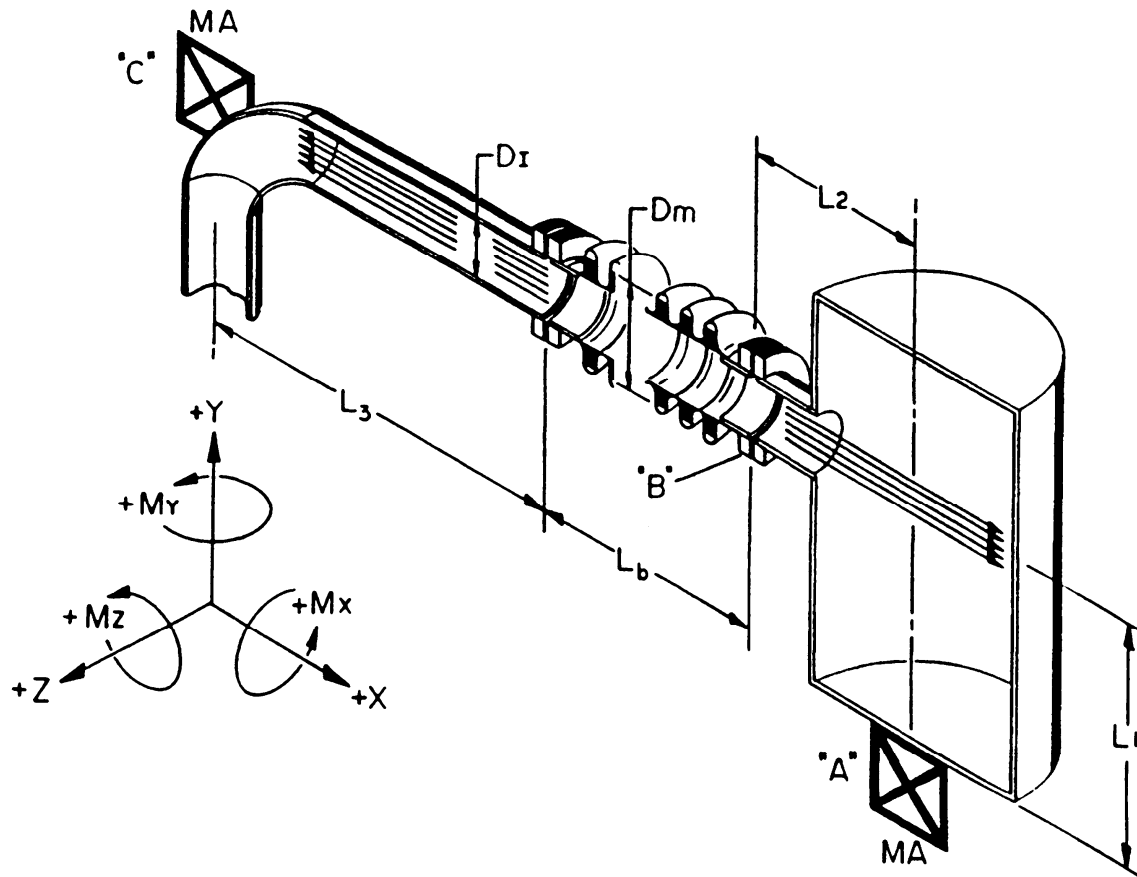
Example 6M: Single Expansion Joint, attached to vessel nozzle, subjected to axial and lateral movement.

FIGURE J7

A large vertical vessel which operates at 1 MPa at 260°C is equipped with a 610 mm diameter outlet line as shown in Figure J7. The outlet line contains a single bellows Expansion Joint which is designed to absorb the thermal growth of the vessel and pipe line. The lengths and calculated thermal growths for the all-carbon steel system are as follows:

$L_1 = 4.572 \text{ m}$	$\Delta L_1 = 13.79 \text{ mm}$
$L_2 = 1.829 \text{ m}$	$\Delta L_2 = 5.51 \text{ mm}$
$L_b = 0.305 \text{ m}$	$\Delta L_b = 0.914 \text{ mm}$
$L_3 = 4.267 \text{ m}$	$\Delta L_3 = 12.88 \text{ mm}$

Determine the forces and moments acting on the nozzle flange "B"

Data provided by the Expansion Joint manufacturer:

$$D_m = 648 \text{ mm}$$

$$D_l = 591 \text{ mm}$$

$$f_w = 6451 \text{ N/mm per convolution}$$

$$N = 12$$

SOLUTION:

Calculate the equivalent movements per convolution:

$$\begin{aligned} e_x &= \frac{x}{N} \\ &= \frac{19.30}{12} \\ &= .1.6 \text{ mm} \end{aligned}$$

$$\begin{aligned} \text{where: } x &= \Delta L_2 + \Delta L_b + L_3 \\ &= 5.51 + 0.914 + 12.88 \\ &= 19.3 \text{ mm} \end{aligned}$$

$$\begin{aligned} e_y &= \frac{3D_m y}{N(L_b - x)} \\ &= \frac{3 \times 648 \times 13.8}{12 \times (305 - 19.3)} \\ &= 7.82 \text{ m} \end{aligned}$$

$$\begin{aligned} y &= \Delta L_1 = 13.8 \text{ mm} \\ L_b &= 305 \text{ mm} \end{aligned}$$

Calculate F_a , F_s , F_p and V .

$$\begin{aligned} F_a &= (f_w)(e_x) \\ &= 6451 \times 1.6 \\ &= 10.3 \text{ kN} \end{aligned}$$

$$\begin{aligned} F_s &= (A_e)(P_d) \\ &= 329,792 \times 1.034 \\ &= 341 \text{ kN} \end{aligned}$$

$$\begin{aligned} F_p &= (A_p)(P_d) \\ &= 274,325 \times 1.034 \\ &= 283.7 \text{ kN} \end{aligned}$$

$$\begin{aligned} V_l &= \frac{f_w D_m e_y}{2L_b} \\ &= \frac{6451 \times 648 \times 7.82}{2 \times 305} \\ &= 53.6 \text{ kN} \end{aligned}$$

where:

$$\begin{aligned} A_e &= \frac{\pi}{4} (D_m^2) \\ &= \frac{\pi}{4} (648)^2 \\ &= 329,792 \text{ mm}^2 \\ P_d &= 1.03 \text{ MPa} \\ A_p &= \frac{\pi}{4} (D_l^2) \\ &= \frac{\pi}{4} (591)^2 \\ &= 274,325 \text{ mm}^2 \\ L_b &= 305 \text{ mm} \end{aligned}$$

Forces and moments acting on vessel anchor "A"

$$\begin{aligned} F_x &= F_a + F_s = 10.3 + 341 \\ &= 351 \text{ kN} \end{aligned}$$

$$F_y = -V_l = -53.6 \text{ kN}$$

$$F_z = 0 \text{ (no forces exist in Z direction)}$$

$$M_x = F_z Y - F_y Z = 0$$

$$M_y = F_x Z - F_z X = 0$$

$$\begin{aligned} M_z &= F_y X - F_x Y \\ &= (-53.6) \times (-1982) - 351 \times 4572 \\ &= -1,499,000 \text{ Nm} \end{aligned}$$

$$\begin{aligned} X &= -(L_2 + L_b / 2) \\ &= -1982 \text{ mm} \\ \text{where: } Y &= L_1 = 4572 \text{ mm} \\ Z &= 0 \end{aligned}$$

Forces and moments acting on nozzle flange face "B"

$$\begin{aligned}
 F_x &= F_a + F_s - F_p \\
 &= 10.3 + 341 - 283.7 \\
 &= 67.8 \text{ kN} \\
 F_y &= -53.6 \text{ kN} \\
 F_z &= 0
 \end{aligned}$$

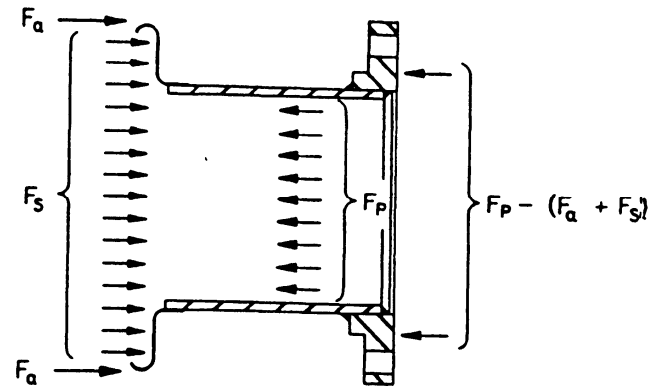


FIGURE J8

$$\begin{aligned}
 M_x &= F_z Y - F_y Z = 0 \\
 M_y &= F_x Z - F_z X = 0 \\
 M_z &= F_y X - F_x Y \\
 &= (-53.6) \times (-152.4) - 0 \\
 &= 8,170 \text{ Nm}
 \end{aligned}$$

where: $X = L_b / 2 = -152.4 \text{ mm}$

$$Y = Z = 0$$

Forces and moments acting on main anchor "C"

$$\begin{aligned}
 F_x &= -F_a - F_s \\
 &= -10.3 - 341 \\
 &= -351 \text{ kN} \\
 F_y &= 53.6 \text{ kN} \\
 F_z &= 0 \\
 M_x &= F_z Y - F_y Z = 0 \\
 M_y &= F_x Z - F_z X = 0 \\
 M_z &= F_y X - F_x Y \\
 &= 53.6 \times 4,420 - 0 \\
 &= 237,000 \text{ Nm}
 \end{aligned}$$

where: $X = L_3 + L_b / 2$

$$= 4,420 \text{ mm}$$

$$Y = Z = 0$$

Example 7: Calculation of Angular Rotation in a 3 Hinge Piping System

A 24" diameter stainless steel line runs between intermediate anchors ① and ② as shown in Figure 2.27. Three hinge expansion joints, located at points "A," "B" and "C" are used to absorb the thermal expansion of the piping system. The line operates at 100 psig and 1000° F. Calculate the angular movements of each hinge expansion joint.

The following information is known about the system:

$L_1 = 53.75$ in.	$L_6 = 42$ in.	$\delta = 0.00957$ in./in.
$\theta = 45$ deg.	$L_7 = 72$ in.	Unit Expansion Derived
	$L_8 = 42$ in.	from Table IV.
$L_4 = 96$ in.	$L_9 = 120$ in.	
$L_5 = 60$ in.	$L_{10} = 60$ in.	

SOLUTION:**MOVEMENT CALCULATIONS**

$L_2 = (L_1)(\sin \theta)$	$= 38$ in.	$L_3 = (L_1)(\cos \theta)$	$= 38$ in.
$L_{13} = (L_2) + (L_8) - (L_{10})$	$= 20$ in.	$L_{11} = (L_3) + (L_4) + (L_9)$	$= 254$ in.
$L_{14} = (L_{13})(\delta)$	$= 0.19$ in.	$L_{12} = (L_{11})(\delta)$	$= 2.43$ in.

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$$A^1 E^1 = L_6 + L_7 - L_{14} = 113.81 \text{ in.}$$

$$E^1 C^1 = L_5 - L_{12} = 57.57 \text{ in.}$$

$$A^1 C^1 = [(A^1 E^1)^2 + (E^1 C^1)^2]^{1/2} = 127.54 \text{ in.}$$

$$A^1 = \tan^{-1} \left[\frac{(E^1 C^1)}{(A^1 E^1)} \right] =$$

$$C^1 = 90^\circ - A^1 =$$

$$A^1 D^1 = (L_5)(1 + \delta) = 60.57 \text{ in.}$$

$$D^1 B^1 = (L_6)(1 + \delta) = 42.40 \text{ in.}$$

$$A^1 B^1 = [(A^1 D^1)^2 + (D^1 B^1)^2]^{1/2} = 73.94 \text{ in.}$$

$$A^1 = \tan^{-1} \left[\frac{(D^1 B^1)}{(A^1 D^1)} \right] =$$

$$B^1 = 90^\circ - A^1 =$$

$$B^1 C^1 = (L_7)(1 + \delta) = 72.69 \text{ in.}$$

$$B^1 = \cos^{-1} \left[\frac{(A^1 B^1)^2 + (B^1 C^1)^2 - (A^1 C^1)^2}{(2)(A^1 B^1)(B^1 C^1)} \right] =$$

$$C^1 = \cos^{-1} \left[\frac{(A^1 C^1)^2 + (B^1 C^1)^2 - (A^1 B^1)^2}{(2)(A^1 C^1)(B^1 C^1)} \right] =$$

$$A^1 = 180^\circ - B^1 - C^1 =$$

$$\Sigma =$$

Angles		
A^1	B^1	C^1
26.83		
		63.17
34.99		
	55.01	
	120.87	
		29.84
29.29		
91.11	175.88	93.01

CALCULATED ANGULAR MOVEMENTS

$$\Delta A = \Sigma A^1 - 90^\circ = 1.11 \text{ deg.}$$

$$\Delta B = 180^\circ - \Sigma B^1 = 4.12 \text{ deg.}$$

$$\Delta C = \Sigma C^1 - 90^\circ = 3.01 \text{ deg.}$$

$$\Delta B = \Delta A + \Delta C \text{ (Check)} = 4.12 \text{ deg.}$$

Example 7M: Calculation of Angular Rotation in a 3 Hinge Piping System

A 610 mm diameter stainless steel line runs between intermediate anchors ① and ② as shown in Figure 2.27. Three hinge expansion joints, located at points "A," "B" and "C" are used to absorb the thermal expansion of the piping system. The line operates at 0.69 MPa and 538°C. Calculate the angular movements of each hinge expansion joint.

The following information is known about the system:

$$L_1 = 1365.3 \text{ mm}$$

$$\theta = 45 \text{ deg.}$$

$$L_4 = 2438.4 \text{ mm}$$

$$L_5 = 1524 \text{ mm} \quad L_{10} = 1524 \text{ mm}$$

$$L_6 = 1066.8 \text{ mm}$$

$$L_7 = 1829 \text{ mm}$$

$$L_8 = 1066.8 \text{ mm}$$

$$L_9 = 3048 \text{ mm}$$

$$\delta = .00957 \text{ mm/mm}$$

Unit Expansion Derived
from Table IV.

SOLUTION:**MOVEMENT CALCULATIONS**

$$L_2 = (L_1)(\sin \theta) = 965 \text{ mm}$$

$$L_{13} = L_2 + L_8 - L_{10} = 508 \text{ mm}$$

$$L_{14} = (L_{13})(\delta) = 4.83 \text{ mm}$$

$$L_3 = (L_1)(\cos \theta) = 965 \text{ mm}$$

$$L_{11} = L_3 + L_4 + L_9 = 6451 \text{ mm}$$

$$L_{12} = (L_{11})(\delta) = 61.7 \text{ mm}$$

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$$\begin{aligned}
 A^1 E^1 &= L_6 + L_7 - L_{14} &= 2890.8 \text{ mm} \\
 E^1 C^1 &= L_5 - L_{12} &= 1462.3 \text{ mm} \\
 A^1 C^1 &= \left[(A^1 E^1)^2 + (E^1 C^1)^2 \right]^{1/2} &= 3239.5 \text{ mm} \\
 A^1 &= \tan^{-1} \left[\frac{(E^1 C^1)}{(A^1 E^1)} \right] &= 0.4683 \text{ rad.} \\
 C^1 &= 90^\circ - A^1 &= \\
 A^1 D^1 &= (L_5)(1 + \delta) &= 1538.6 \text{ mm} \\
 D^1 B^1 &= (L_6)(1 + \delta) &= 1077 \text{ mm} \\
 A^1 B^1 &= \left[(A^1 D^1)^2 + (D^1 B^1)^2 \right]^{1/2} &= 1878 \text{ mm} \\
 A^1 &= \tan^{-1} \left[\frac{(D^1 B^1)}{(A^1 D^1)} \right] &= 0.7000 \text{ rad.} \\
 B^1 &= 90^\circ - A^1 &= \\
 B^1 C^1 &= (L_7)(1 + \delta) &= 1846 \text{ mm} \\
 B^1 &= \cos^{-1} \left[\frac{(A^1 B^1)^2 + (B^1 C^1)^2 - (A^1 C^1)^2}{(2)(A^1 B^1)(B^1 C^1)} \right] &= -0.513 \text{ rad.} \\
 C^1 &= \cos^{-1} \left[\frac{(A^1 C^1)^2 + (B^1 C^1)^2 - (A^1 B^1)^2}{(2)(A^1 C^1)(B^1 C^1)} \right] &= 0.8674 \text{ rad.} \\
 A^1 &= 180^\circ - B^1 - C^1 &=
 \end{aligned}$$

$\Sigma =$

CALCULATED ANGULAR MOVEMENTS

$$\begin{aligned}
 \Delta A &= \Sigma A^1 - 90^\circ &= 1.11 \text{ deg.} \\
 \Delta B &= 180^\circ - \Sigma B^1 &= 4.12 \text{ deg.} \\
 \Delta C &= \Sigma C^1 - 90^\circ &= 3.01 \text{ deg.} \\
 \Delta B &= \Delta A + \Delta C \text{ (Check)} &= 4.12 \text{ deg.}
 \end{aligned}$$

Angles in deg.		
A ¹	B ¹	C ¹
26.83		
		63.17
34.99		
	55.01	
	120.87	
		29.84
29.29		
91.11	175.88	93.01

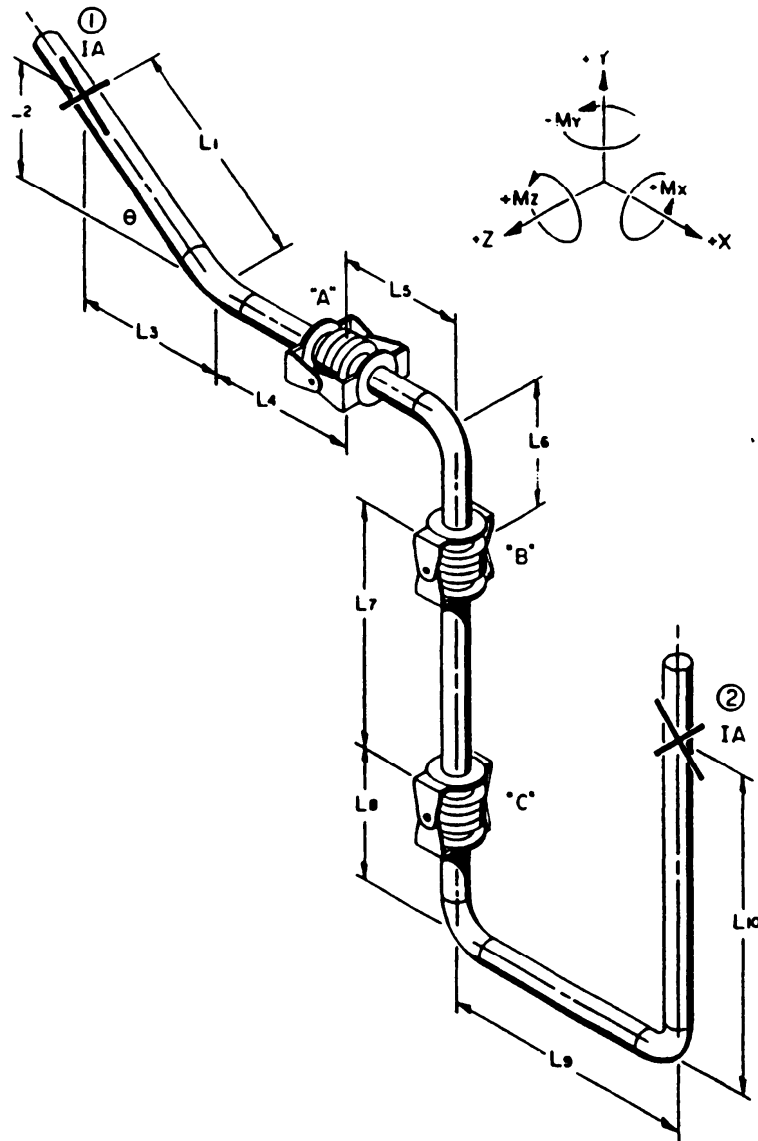
Example 8: Three (3) hinge Expansion Joint system

FIGURE J9

A 24 in. diameter stainless steel line runs between intermediate anchors ① and ② as shown in Figure J9. Three hinge Expansion Joints, located at points "A", "B" and "C" are used to absorb the thermal expansion of the piping. The line operates at 100 psig and 1000° F. Calculate the resultant forces and moments on each anchor.

The following information is known about the system:

$$\theta = 45 \text{ deg.}$$

$$L_1 = 53.75 \text{ in.}$$

$$L_2 = 38 \text{ in.}$$

$$L_3 = 38 \text{ in.}$$

$$L_4 = 96 \text{ in.}$$

$$L_5 = 60 \text{ in.}$$

$$L_6 = 42 \text{ in.}$$

$$L_7 = 72 \text{ in.}$$

$$L_8 = 42 \text{ in.}$$

$$L_9 = 120 \text{ in.}$$

$$L_{10} = 60 \text{ in.}$$

Data provided by the Expansion Joint manufacturer.

$$D_m = 25.50 \text{ in.}$$

$$f_w = 36840 \text{ lbs./in. per convolution}$$

$$N = 12$$

SOLUTION:

In Example 7 the angular movement for each Expansion Joint was calculated to be:

$$\Delta A = \theta_A = 1.11 \text{ deg.} = 0.019 \text{ radians}$$

$$\Delta B = \theta_B = 4.12 \text{ deg.} = 0.072 \text{ radians}$$

$$\Delta C = \theta_C = 3.01 \text{ deg.} = 0.053 \text{ radians}$$

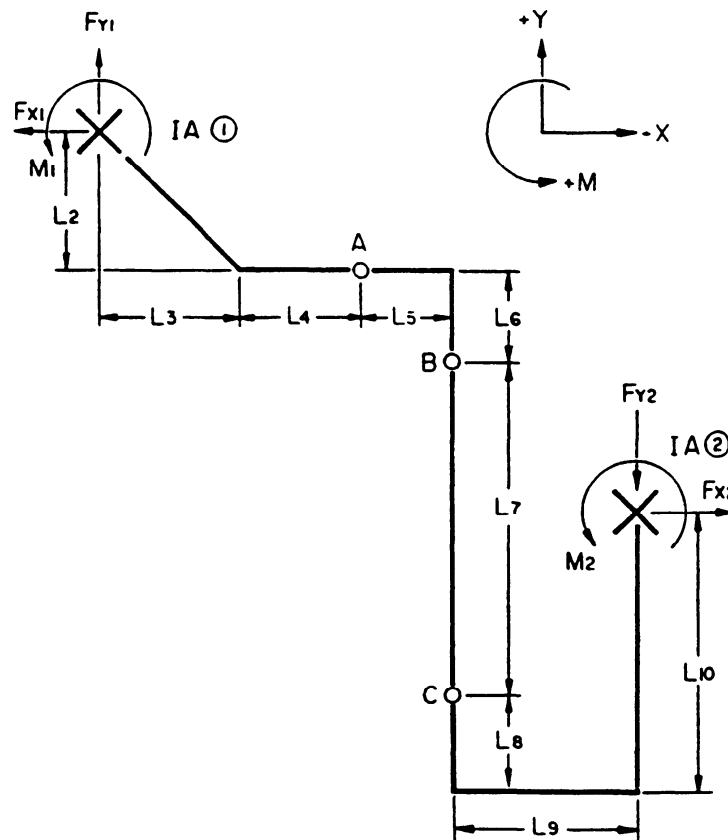


FIGURE J10

Calculate the equivalent axial movement per convolution for each hinge Expansion Joint, Equation (4-3) Section 4.1.

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$$e_{\theta} = \frac{\theta D_m}{2N}$$

$$e_{\theta A} = \frac{(0.019)(25.50)}{(2)(12)} = 0.020 \text{ in.}$$

$$e_{\theta B} = \frac{(0.072)(25.50)}{(2)(12)} = 0.077 \text{ in.}$$

$$e_{\theta C} = \frac{(0.053)(25.50)}{(2)(12)} = 0.056 \text{ in.}$$

Calculate the moments at each hinge Expansion Joint, Equation (4-17) Section 4.6.1.

$$M_{\theta} = \frac{f_w D_m e_{\theta}}{4}$$

$$M_{\theta A} = \frac{(36840)(25.50)(0.020)}{4} = 4697 \text{ in. lbs.}$$

$$M_{\theta B} = \frac{(36840)(25.50)(0.077)}{4} = 18084 \text{ in. lbs.}$$

$$M_{\theta C} = \frac{(36840)(25.50)(0.056)}{4} = 13152 \text{ in. lbs.}$$

Calculate the forces F_x and F_y acting on IA ① and IA ②

$$F_{x2} = \frac{M_{\theta B} + M_{\theta C}}{L_7} = \frac{(18084) + (13152)}{(72)}$$

$$= 434 \text{ lbs.}$$

$$F_{x1} = -434 \text{ lbs.}$$

$$F_{y1} = \frac{M_{\theta A} + M_{\theta B} + (F_{x2})(L_6)}{L_5} = \frac{(4697) + (18084) + (434)(42)}{(60)}$$

$$= 683 \text{ lbs.}$$

$$F_{y2} = -683 \text{ lbs.}$$

Calculate the moments acting on IA ① and IA ②

$$M_1 = M_{\theta A} + F_{y1}(L_3 + L_4) - F_{x2}(L_2)$$

$$= 4697 + (683)(38 + 96) - (434)(38)$$

$$= 79727 \text{ in. lbs.}$$

$$M_2 = F_{x2}(L_{10} - L_8) + F_{y1}(L_9) - M_{\theta C}$$

$$= (434)(60 - 42) + (683)(120) - 13152$$

$$= 76620 \text{ in. lbs.}$$

Example 8M: Three (3) hinge Expansion Joint system

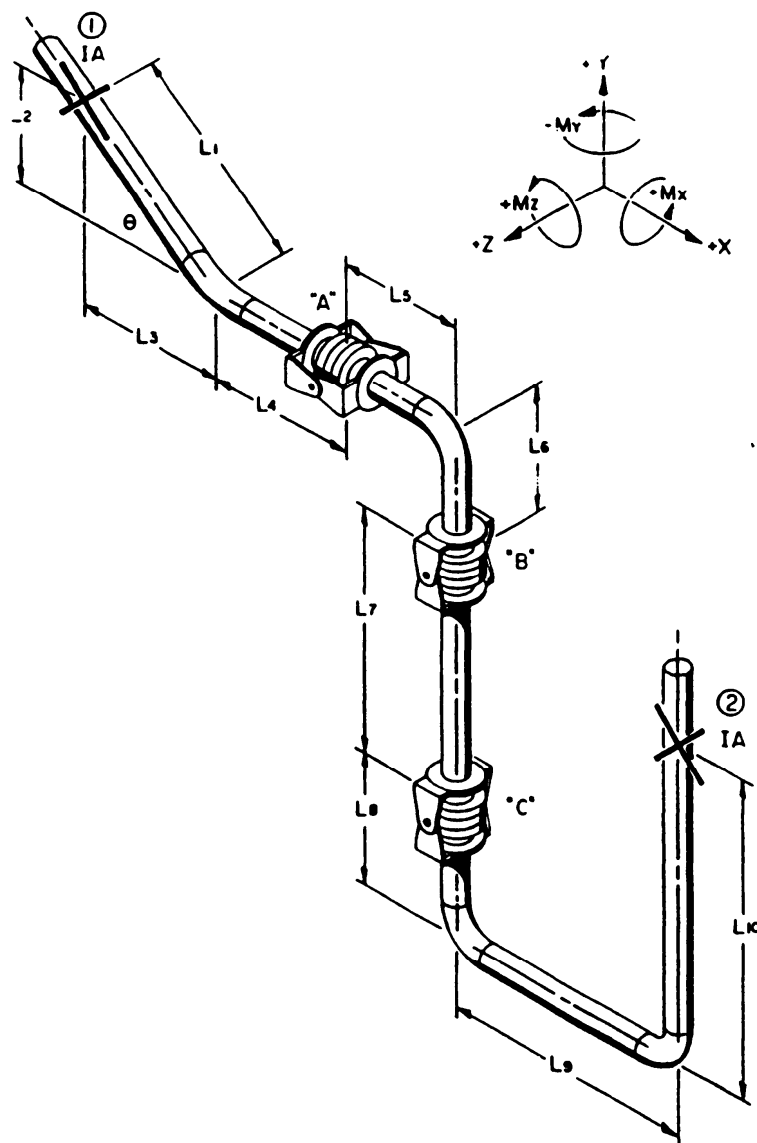


FIGURE J9

A 610 mm diameter stainless steel line runs between intermediate anchors ① and ② as shown in Figure J9. Three hinge Expansion Joints, located at points "A", "B" and "C" are used to absorb the thermal expansion of the piping. The line operates at 0.69 MPa and 538°C. Calculate the resultant forces and moments on each anchor.

The following information is known about the system:

$$\theta = 45 \text{ deg.}$$

$$L_1 = 1365.3 \text{ mm}$$

$$L_2 = 965 \text{ mm}$$

$$L_3 = 965 \text{ mm}$$

$$L_4 = 2438.4 \text{ mm}$$

$$L_5 = 1524 \text{ mm}$$

$$L_6 = 1066.8 \text{ mm}$$

$$L_7 = 1829 \text{ mm}$$

$$L_8 = 1066.8 \text{ mm}$$

$$L_9 = 3048 \text{ mm}$$

$$L_{10} = 1524 \text{ mm}$$

Data provided by the Expansion Joint manufacturer.

$$D_m = 648 \text{ mm}$$

$$f_w = 6,451 \text{ N/mm per convolution}$$

$$N = 12$$

SOLUTION:

In Example 7 the angular movement for each Expansion Joint was calculated to be:

$$\Delta A = \theta_A = 1.11 \text{ deg.} = 0.019 \text{ radians}$$

$$\Delta B = \theta_B = 4.12 \text{ deg.} = 0.072 \text{ radians}$$

$$\Delta C = \theta_C = 3.01 \text{ deg.} = 0.053 \text{ radians}$$

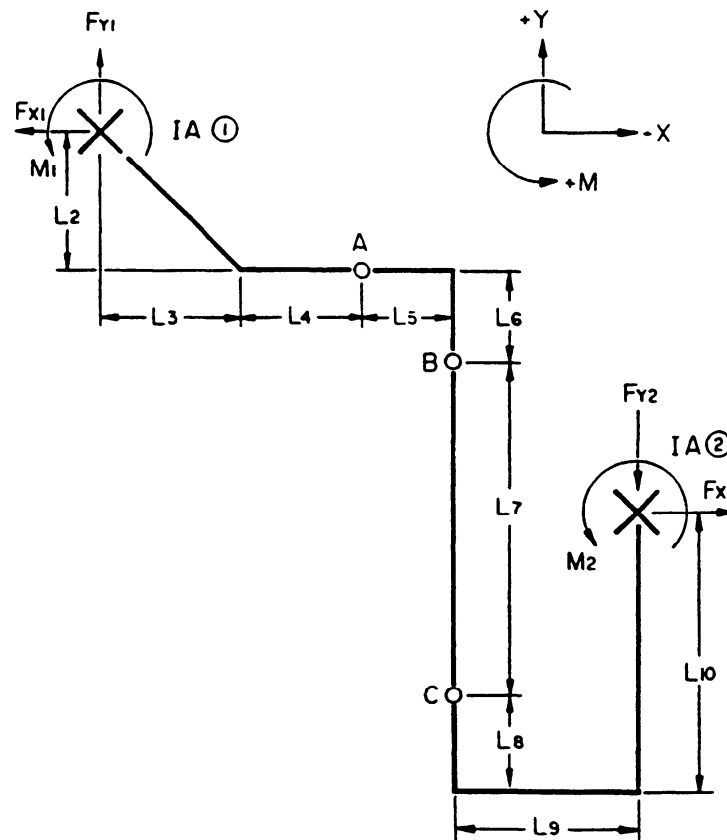


FIGURE J10

Calculate the equivalent axial movement per convolution for each hinge Expansion Joint, Equation (4-3) Section 4.1.

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$$e_{\theta} = \frac{\theta D_m}{2N}$$

$$e_{\theta A} = \frac{0.019 \times 648}{2 \times 12} = 0.508 \text{ mm}$$

$$e_{\theta B} = \frac{0.072 \times 648}{2 \times 12} = 1.956 \text{ mm}$$

$$e_{\theta C} = \frac{0.053 \times 648}{2 \times 12} = 1.422 \text{ mm}$$

Calculate the moments at each hinge Expansion Joint, Equation (4-16) Section 4.6.1.

$$M_{\theta} = \frac{f_w D_m e_{\theta}}{4}$$

$$M_{\theta A} = \frac{6,451 \times 648 \times 0.508}{4} = 531 \text{ Nm}$$

$$M_{\theta B} = \frac{6,451 \times 648 \times 1.956}{4} = 2,043 \text{ Nm}$$

$$M_{\theta C} = \frac{6,451 \times 648 \times 1.422}{4} = 1,486 \text{ Nm}$$

Calculate the forces F_x and F_y acting on IA ① and IA ②

$$F_{x2} = \frac{M_{\theta B} + M_{\theta C}}{L_7} = \frac{2,043 + 1,486}{1,829}$$

$$= 1.93 \text{ kN}$$

$$F_{x1} = -1.93 \text{ kN}$$

$$F_{y1} = \frac{M_{\theta A} + M_{\theta B} + (F_{x2})(L_6)}{L_5} = \frac{531 + 2,043 + 1,93 \times 1066.8}{1,524}$$

$$= 3.04 \text{ kN}$$

$$F_{y2} = -3.04 \text{ kN}$$

Calculate the moments acting on IA ① and IA ②

$$M_1 = M_{\theta A} + F_{y1}(L_3 + L_4) - F_{x2}(L_2)$$

$$= 531 + 3.04 \times (965 + 2438.4) - 1.93 \times 965$$

$$= 9,010 \text{ Nm}$$

$$M_2 = F_{x2}(L_{10} - L_8) + F_{y1}(L_9) - M_{\theta C}$$

$$= 1.93 \times (1524 - 1066.8) + 3.04 \times 3048 - 1486$$

$$= 8,660 \text{ Nm}$$

Example 9: Bellows Equivalent Movement per Convolution

Case 1: Assume a 28 inch diameter universal Expansion Joint is to be installed in the neutral position (no cold spring) and is to be subjected to the following two sets of operating deflections:

Condition 1 $x = 1$ in. compression

$y = 1.50$ in.

$\theta = 0$ radians

Condition 2 $x = 0.5$ in. extension

$y = 0.5$ in. on opposite side of neutral centerline from Condition 1

$\theta = 0$ radians

Bellows data: $D_m = 29.5$ in.

e_c (rated) = .50 in.

e_e (rated) = .25 in.

$q = 2$ in.

SOLUTION:

Let $N = 5$ and $L_u = 36$ in.

$$L_b = Nq = (5)(2) = 10, L_u / 2L_b = 36 / 20 = 1.80$$

$$L^* = L_u - L_b = 36 - 10 = 26$$

$$L^* / L_b = (26) / (10) = 2.6$$

$$C_\theta = 1 \text{ (for Universal Bellows)}$$

$$K_\theta = 1 \text{ (for Universal Bellows)}$$

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Condition 1

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)^3} y$$

$$= \frac{(3)(29,5)}{(2)(5)(10)} \cdot \frac{1 + (2.6)}{1 + (3)(2.6)^2} \cdot \frac{(26)}{(26 - 1/2)} (1.5)$$

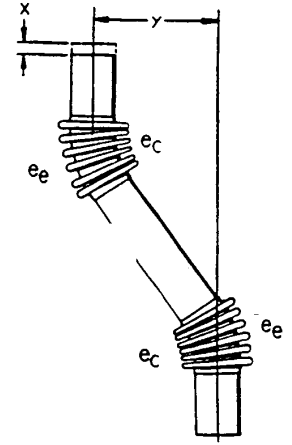
$$= (.885)(.169)(1.02)(1.5) = .23 \text{ in.}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{1}{(2)(5)} = 0.100 \text{ in. compression}$$

$$e_x = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .23 + 0 + .10 = .33 \\ 0 + .10 = .10 \end{Bmatrix} = .33 \text{ in.} < .50 \text{ in. (rated)} = e$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .23 + 0 - .10 = .13 \\ 0 - .10 = -.10 \end{Bmatrix} = .13 \text{ in.} < .25 \text{ in. (rated)}$$



Condition 2

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)^3} y$$

$$= \frac{(3)(29,5)}{(2)(5)(10)} \cdot \frac{1 + (2.6)}{1 + (3)(2.6)^2} \cdot \frac{(26)}{(26 + .5/2)} (.5)$$

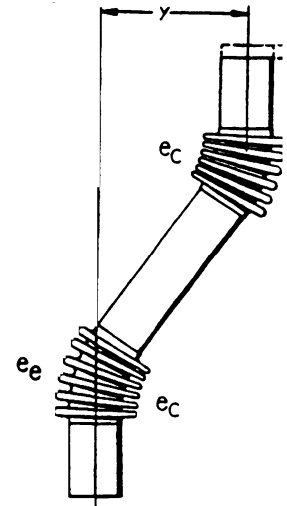
$$= (.885)(.1692)(.9905)(.5) = .074 \text{ in.}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{0.50}{(2)(5)} = 0.050 \text{ in. compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .074 + 0 - .05 = .024 \\ 0 - .05 = -.05 \end{Bmatrix} = .024 \text{ in.} < .50 \text{ in. (rated)}$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .074 + 0 + .05 = .124 \\ 0 + .05 = .05 \end{Bmatrix} = .124 \text{ in.} < .25 \text{ in. (rated)} = e$$



Case 2: Assume the same 28 in. diameter universal Expansion Joint except that it is to be installed with 0.50 in. lateral cold spring and 0.25 in. axial pre-compression and is to be subjected to the following operating deflection:

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Cold spring/Preset $x = .25$ in. compression

$y = .50$ in.

$\theta = 0$ radians

Operating

$x = 1$ in. compression

$y = 1.50$ in. in direction opposite to direction of lateral cold spring

(1 in. from the neutral position)

$\theta = 0$ radians

SOLUTION:

Cold Spring/Preset

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y$$

$$= \frac{(3)(29,5)}{(2)(5)(10)} \cdot \frac{1 + (2.6)}{1 + (3)(2.6)^2} \cdot \frac{(26)}{(26 - .25/2)} (.5) = (.885)(.1692)(1.005)(.5) = .075 \text{ in.}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{0.25}{(2)(5)} = 0.025 \text{ in. compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .075 + 0 - .025 = .10 \\ 0 + .025 = .025 \end{Bmatrix} = .10 \text{ in.} < .50 \text{ in. (rated)}$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .075 + 0 + .025 = .10 \\ 0 + .025 = .025 \end{Bmatrix} = .10 \text{ in.} < .25 \text{ in. (rated)}$$

Operating

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y$$

$$= \frac{(3)(29,5)}{(2)(5)(10)} \cdot \frac{1 + (2.6)}{1 + (3)(2.6)^2} \cdot \frac{(26)}{(26 - .5/2)} (1.0)$$

$$= (.885)(.1692)(1.01)(1.0) = .151 \text{ in.}$$

$$e_\theta = 0$$

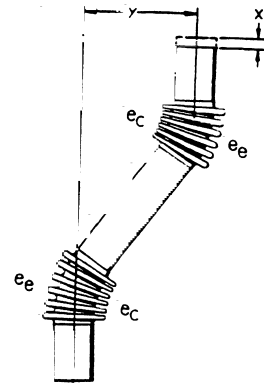
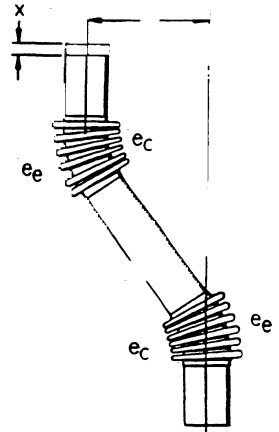
$$e_x = \frac{x}{2N} = \frac{1}{(2)(5)} = 0.100 \text{ in. compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .151 + 0 + .10 = .251 \\ 0 + .10 = .10 \end{Bmatrix} = .251 \text{ in.} < .50 \text{ in. (rated)}$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} .151 + 0 - .10 = .051 \\ 0 - .10 = -.10 \end{Bmatrix} = .051 \text{ in.} < .25 \text{ in. (rated)}$$

$$e_c(\text{cold spring/preset}) + e_e(\text{operating}) = .10 + .051 = .151$$

$$e_e(\text{cold spring/preset}) + e_c(\text{operating}) = .05 + .251 = .301$$



Case 3: Assume a 24 inch diameter single unreinforced Expansion Joint is to be installed with 0.5 in. axial pre-extension and is to be subjected to the following operating deflections:

Preset $x = 0.5$ in. extension
 $y = 0$
 $\checkmark = 0$ radians

Operating $x = 1$ in. compression
 $y = 0$
 $\checkmark = 0.0873$ radians

Bellows data: $D_m = 25.5$ in.
 $N = 12$
 $q = 1$ in.
 $f_{iu} = 12562$ lb/in
 $P = 40$ psi
 e_c (rated) = 0.250 in.
 e_e (rated) = 0.125 in.

SOLUTION:

$$L_b = Nq = (12)(1) = 12 \text{ in.}$$

Preset

$$e_x = \frac{x}{N} = \frac{0.5}{12} = 0.0417 \text{ in. extension}$$

$$e_y = \frac{3D_m y}{N(L_b + x)} = \frac{(3)(25.5)(0)}{(12)(12 + 0.5)} = 0 \text{ in.}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{(0)(25.5)}{(2)(12)} = 0 \text{ in.}$$

$$K_{\theta l} = 1.0$$

$$e_{yp} = 0$$

$$R_\theta = 1.0$$

$$K_\theta = 1.0$$

$$C_\theta = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 - 0.0417 = -0.0417 \\ 0 - 0.0417 = -0.0417 \end{array} \right\} = -0.0417 \text{ in.} < 0.250 \text{ in. (rated)}$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 + 0.0417 = 0.0417 \\ 0 + 0.0417 = 0.0417 \end{array} \right\} = 0.0417 \text{ in.} < 0.125 \text{ in. (rated)}$$

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Operating

$$e_x = \frac{x}{N} = \frac{1}{12} = 0.0833 \text{ in. compression}$$

$$e_y = \frac{3D_my}{N(L_b - x)} = \frac{(3)(25.5)(0)}{(12)(12 - 1)} = 0 \text{ in.}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{(0.0873)(25.5)}{(2)(12)} = 0.0928 \text{ in.}$$

$$K_{\theta l} = 1.0$$

$$e_{yp} = \frac{\pi D_m K_{\theta l} P \sin(\theta / 2)(L_b - x)}{4f_{iu}} = \frac{\pi(25.5)(1.0)(40) \sin(0.0873 / 2)(12 - 1)}{4(12562)} = 0.0306$$

$$R_\theta = \frac{1.18N^2(q - e_x)^2}{\pi^2 D_m K_{\theta l} \sin(\theta / 2)(L_b - x)} = \frac{1.18(12)^2(1 - 0.0833)^2}{\pi^2(25.5)(1.0) \sin(0.0873 / 2)(12 - 1)} = 1.1820$$

$$K_\theta = \frac{e_\theta + e_{yp}}{e_\theta} = \frac{0.0928 + 0.0306}{0.0928} = 1.3297$$

$$C_\theta = \text{Lesser of } R_\theta \text{ or } 1.0 = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0.0928 + 0.0833 = 0.1761 \\ 0.1234 + 0.0833 = 0.2067 \end{array} \right\} = 0.2067 \text{ in.} < 0.250 \text{ in. (rated)}$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0.0928 - 0.0833 = 0.0095 \\ 0.1234 - 0.0833 = 0.0401 \end{array} \right\} = 0.0401 \text{ in.} < 0.125 \text{ in. (rated)}$$

$$e_c(\text{preset}) + e_e(\text{operating}) = -0.0417 + 0.0401 = -0.0016 \text{ in.}$$

$$e_e(\text{preset}) + e_c(\text{operating}) = 0.0417 + 0.2067 = 0.2484 \text{ in.} = e$$

Case 4: Assume a 24 inch diameter single unreinforced Expansion Joint is to be installed with 0.5 in. axial pre-extension and is to be subjected to the following operating deflections:

Preset $x = 0.5$ in. extension
 $y = 0$ in.
 $\checkmark = 0$ radians

Operating $x = 1$ in. compression
 $y = 0.06$ in.
 $\checkmark = 0.0873$ radians

Bellows data: $D_m = 25.5$ in.
 $N = 12$
 $q = 1$ in.
 $f_{iu} = 12562$ lb/in
 $P = 40$ psi
 e_c (rated) = 0.250 in.
 e_e (rated) = 0.125 in.

SOLUTION:

$$L_b = Nq = (12)(1) = 12 \text{ in.}$$

Preset

$$e_x = \frac{x}{N} = \frac{0.5}{12} = 0.0417 \text{ in. extension}$$

$$e_y = \frac{3D_m y}{N(L_b + x)} = \frac{(3)(25.5)(0)}{(12)(12 + 0.5)} = 0 \text{ in.}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{(0)(25.5)}{(2)(12)} = 0 \text{ in.}$$

$$K_{\theta l} = 1.0$$

$$e_{yp} = 0$$

$$R_\theta = 1.0$$

$$K_\theta = 1.0$$

$$C_\theta = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 - 0.0417 = -0.0417 \\ 0 - 0.0417 = -0.0417 \end{array} \right\} = -0.0417 \text{ in.} < 0.250 \text{ in. (rated)}$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 + 0.0417 = 0.0417 \\ 0 + 0.0417 = 0.0417 \end{array} \right\} = 0.0417 \text{ in.} < 0.125 \text{ in. (rated)}$$

Operating

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$$e_x = \frac{x}{N} = \frac{1}{12} = 0.0833 \text{ in. compression}$$

$$e_y = \frac{3D_m y}{N(L_b - x)} = \frac{(3)(25.5)(0.06)}{(12)(12 - 1)} = 0.0348 \text{ in.}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{(0.0873)(25.5)}{(2)(12)} = 0.0928 \text{ in.}$$

$$K_{\theta l} = 1 + 0.024y(L_b / D_m)^{1.33} = 1 + (0.024)(0.06)(12 / 25.5)^{1.33} = 1.0053$$

$$e_{yp} = \frac{\pi D_m K_{\theta l} P \sin(\theta / 2)(L_b - x)}{4f_{iu}} = \frac{\pi(25.5)(1.0053)(40) \sin(0.0873 / 2)(12 - 1)}{4(12562)} = 0.0308$$

$$R_\theta = \frac{1.18N^2(q - e_x)^2}{\pi^2 D_m K_{\theta l} \sin(\theta / 2)(L_b - x)} = \frac{1.18(12)^2(1 - 0.0833)^2}{\pi^2(25.5)(1.0053) \sin(0.0873 / 2)(12 - 1)} = 1.1758$$

$$K_\theta = \frac{e_\theta + e_{yp}}{e_\theta} = \frac{0.0928 + 0.0308}{0.0928} = 1.3319$$

$$C_\theta = \text{Lesser of } R_\theta \text{ or } 1.0 = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0.0348 + 0.0928 + 0.0833 = 0.2109 \\ 0.1236 + 0.0833 = 0.2069 \end{array} \right\} = 0.2109 \text{ in.} < 0.250 \text{ in. (rated)}$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0.0348 + 0.0928 - 0.0833 = 0.0443 \\ 0.1236 - 0.0833 = 0.0403 \end{array} \right\} = 0.0443 \text{ in.} < 0.125 \text{ in. (rated)}$$

$$e_c(\text{preset}) + e_e(\text{operating}) = -0.0417 + 0.0443 = 0.0026 \text{ in.}$$

$$e_e(\text{preset}) + e_c(\text{operating}) = 0.0417 + 0.2109 = 0.2526 \text{ in.} = e$$

Example 9M: BELLOWS EQUIVALENT MOVEMENT PER CONVOLUTION

Case 1: Assume a 553 mm diameter universal Expansion Joint is to be installed in the neutral position (no cold spring) and is to be subjected to the following two sets of operating deflections:

$$\begin{aligned} \text{Condition 1} \quad x &= 25.4 \text{ mm compression} \\ y &= 38.1 \text{ mm} \\ \theta &= 0 \text{ radians} \end{aligned}$$

$$\begin{aligned} \text{Condition 2} \quad x &= 12.7 \text{ mm extension} \\ y &= 12.7 \text{ mm on opposite side of neutral centerline from Condition 1} \end{aligned}$$

$$\theta = 0 \text{ radians}$$

$$\text{Bellows data: } D_m = 749 \text{ mm}$$

$$e_c (\text{rated}) = 12.7 \text{ mm}$$

$$e_e (\text{rated}) = 6.35 \text{ mm}$$

$$q = 50.8 \text{ mm}$$

SOLUTION:

$$\text{Let } N = 5 \text{ and } L_u = 914 \text{ mm}$$

$$L_b = Nq = 5 \times 50.8 = 254 \text{ mm}, L_u / 2L_b = 914 / 508 = 1.8$$

$$L^* = L_u - L_b = 914 - 254 = 660 \text{ mm}$$

$$L^* / L_b = 660 / 254 = 2.6$$

$$C_\theta = 1 \text{ (for Universal Bellows)}$$

$$K_\theta = 1 \text{ (for Universal Bellows)}$$

Condition 1

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y$$

$$= \frac{3 \times 749}{2 \times 5 \times 254} \cdot \frac{1 + 2.6}{1 + 3 \times 2.6^2} \cdot \frac{660}{(660 - 25.4/2)} \cdot 38.1$$

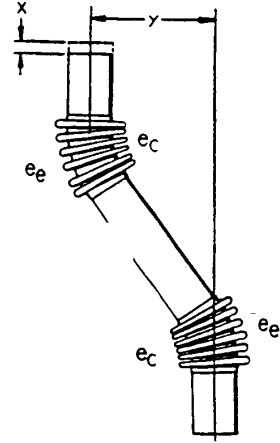
$$= 0.885 \times 0.169 \times 1.02 \times 38.1 = 5.84 \text{ mm}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{25.4}{2 \times 5} = 2.54 \text{ mm compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 5.84 + 0 + 2.54 = 8.38 \\ 0 + 2.54 = 2.54 \end{Bmatrix} = 8.38 \text{ mm} < 12.7 \text{ mm (rated)} = e$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 5.84 + 0 - 2.54 = 3.3 \\ 0 - 2.54 = -2.54 \end{Bmatrix} = 3.3 \text{ mm} < 6.35 \text{ mm (rated)}$$



Condition 2

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y$$

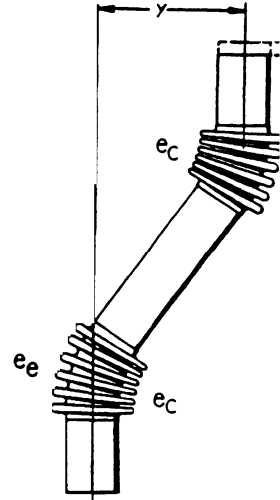
$$= 0.885 \times 0.1692 \times 0.9905 \times 12.7 = 1.88 \text{ mm}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{12.7}{2 \times 5} = 1.27 \text{ mm compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 1.88 + 0 - 1.27 = 0.61 \\ 0 - 1.27 = -1.27 \end{Bmatrix} = 0.61 \text{ mm} < 12.7 \text{ mm (rated)}$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 1.88 + 0 + 1.27 = 3.15 \\ 0 + 1.27 = 1.27 \end{Bmatrix} = 3.15 \text{ mm} < 6.35 \text{ mm (rated)} = e$$



Case 2: Assume the same 553 mm diameter universal Expansion Joint except that it is to be installed with 12.7 mm lateral cold spring and 6.35 mm axial pre-compression and is to be subjected to the following operating deflection:

Cold spring/Preset $x = 6.35$ mm compression
 $y = 12.7$ mm
 $\theta = 0$ radians

Operating $x = 25.4$ mm compression
 $y = 38.1$ mm in direction opposite to direction of lateral cold spring
(25.4 mm from the neutral position)
 $\theta = 0$ radians

SOLUTION:**Cold Spring/Preset**

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y$$

$$= \frac{3 \times 749}{2 \times 5 \times 254} \cdot \frac{1 + 2.6}{1 + 3 \times 2.6^2} \cdot \frac{660}{660 - 6.35/2} \cdot 12.7 = 0.885 \times 0.1692 \times 1.005 \times 12.7 = 1.91 \text{ mm}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{6.35}{2 \times 5} = 0.635 \text{ mm compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 1.905 + 0 + 0.635 = 2.54 \\ 0 + 0.635 = 0.635 \end{Bmatrix} = 2.54 \text{ mm} < 12.7 \text{ mm (rated)}$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 1.905 + 0 - 0.635 = 1.27 \\ 0 - 0.635 = -0.635 \end{Bmatrix} = 1.27 \text{ mm} < 6.35 \text{ mm (rated)}$$

Operating

$$e_y = \frac{3D_m}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y$$

$$= \frac{(3)(29,5)}{(2)(5)(10)} \cdot \frac{1 + (2.6)}{1 + (3)(2.6)^2} \cdot \frac{(26)}{(26 - .5/2)} (1.0) = 0.885 \times 0.1692 \times 1.01 \times 25.4 = 3.84 \text{ mm}$$

$$e_\theta = 0$$

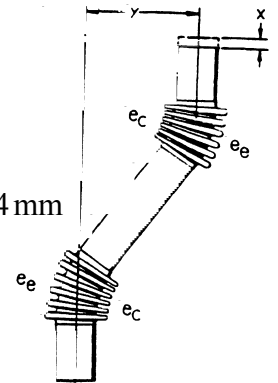
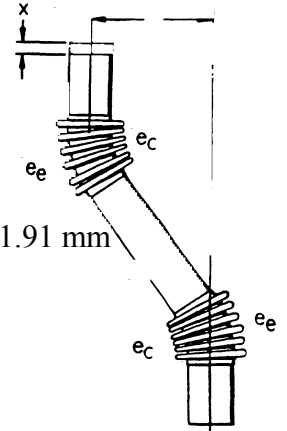
$$e_x = \frac{x}{2N} = \frac{25.4}{2 \times 5} = 2.54 \text{ mm compression}$$

$$e_c = \text{MAX} \begin{Bmatrix} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 3.84 + 0 + 2.54 = 6.38 \\ 0 + 2.54 = +2.54 \end{Bmatrix} = 6.38 \text{ mm} < 12.7 \text{ mm (rated)}$$

$$e_e = \text{MAX} \begin{Bmatrix} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{Bmatrix} = \text{MAX} \begin{Bmatrix} 3.84 + 0 - 2.54 = 1.30 \\ 0 - 2.54 = -2.54 \end{Bmatrix} = 1.30 \text{ mm} < 6.35 \text{ mm (rated)}$$

$$e_c(\text{cold spring/preset}) + e_e(\text{operating}) = 2.54 + 1.3 = 3.84 \text{ mm}$$

$$e_e(\text{cold spring/preset}) + e_c(\text{operating}) = 1.27 + 6.38 = 7.65 \text{ mm} = e$$



Case 3: Assume a 610 mm diameter single unreinforced Expansion Joint is to be installed with 12.7 mm axial pre-extension and is to be subjected to the following operating deflections:

Preset $x = 12.7$ mm extension
 $y = 0$
 $\theta = 0$ radians

Operating $x = 25.4$ mm compression
 $y = 0$
 $\theta = 0.0873$ radians

Bellows data: $D_m = 648$ mm
 $N = 12$
 $q = 25.4$ mm
 $f_{iu} = 2200$ N/mm
 $P = 0.276$ MPa
 e_c (rated) = 6.35 mm
 e_e (rated) = 3.18 mm

SOLUTION:

$$L_b = Nq = 12 \times 25.4 = 305 \text{ mm}$$

Preset

$$e_x = \frac{x}{N} = \frac{12.7}{12} = 1.06 \text{ mm}$$

$$e_y = \frac{3D_m y}{N(L_b + x)} = \frac{3 \times 648 \times 0}{12 \times (305 + 12.7)} = 0 \text{ mm}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{0 \times 648}{2 \times 305} = 0 \text{ mm}$$

$$K_{\theta l} = 1.0; e_{yp} = 0; R_\theta = 1.0; K_\theta = 1.0; C_\theta = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 - 1.06 = -1.06 \\ 0 - 1.06 = -1.06 \end{array} \right\} = -1.06 \text{ mm} < 6.35 \text{ mm (rated)}$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 + 1.06 = 1.06 \\ 0 + 1.06 = 1.06 \end{array} \right\} = 1.06 \text{ mm} < 3.18 \text{ mm (rated)}$$

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Operating

$$e_x = \frac{x}{N} = \frac{25.4}{12} = 2.12 \text{ mm compression}$$

$$e_y = \frac{3D_my}{N(L_b - x)} = \frac{3 \times 648 \times 0}{12 \times (305 - 25.4)} = 0 \text{ mm}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{0.0873 \times 648}{2 \times 12} = 2.36 \text{ mm}$$

$$K_{\theta l} = 1.0$$

$$e_{yp} = \frac{\pi D_m K_{\theta l} P \sin(\theta / 2)(L_b - x)}{4 f_{iu}} = \frac{\pi \times 648 \times 1.0 \times 0.276 \times \sin(.0873 / 2) \times (305 - 25.4)}{4 \times 2200} = 0.78 \text{ mm}$$

$$R_\theta = \frac{1.18 N^2 (q - e_x)^2}{\pi^2 D_m K_{\theta l} \sin(\theta / 2)(L_b - x)} = \frac{1.18 \times 12^2 \times (25.4 - 2.12)^2}{\pi^2 (648)(1.0) \sin(.0873 / 2)(305 - 25.4)} = 1.182$$

$$K_\theta = \frac{e_\theta + e_{yp}}{e_\theta} = \frac{2.36 + 0.78}{2.36} = 1.33$$

$$C_\theta = \text{MIN}(R_\theta; 1.0) = \text{MIN}(1.182; 1.0) = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 2.36 + 2.12 = 4.48 \\ 3.13 + 2.12 = 5.25 \end{array} \right\} = 5.25 \text{ mm} < 6.35 \text{ mm}$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 2.36 - 2.12 = 0.24 \\ 3.13 - 2.12 = 1.01 \end{array} \right\} = 1.01 \text{ mm} < 3.175 \text{ mm}$$

$$e_c \text{ (preset)} + e_e \text{ (operating)} = -1.06 + 1.02 = -0.04 \text{ mm}$$

$$e_e \text{ (preset)} + e_c \text{ (operating)} = 1.06 + 5.25 = 6.31 \text{ mm} = e$$

Case 4: Assume a 24 inch diameter single unreinforced Expansion Joint is to be installed with 0.5 in. axial pre-extension and is to be subjected to the following operating deflections:

Preset $x = 12.7$ mm extension
 $y = 0$ in.
 $\theta = 0$ radians

Operating $x = 25.4$ mm compression
 $y = 1.52$ mm
 $\theta = 0.0873$ radians

Bellows data: $D_m = 648$ mm
 $N = 12$
 $q = 25.4$ mm
 $f_{iu} = 2200$ N/mm
 $P = 0.276$ MPa
 e_c (rated) = 6.35 mm
 e_e (rated) = 3.175 mm

SOLUTION:

$$L_b = Nq = 12 \times 25.4 = 305 \text{ mm}$$

Preset

$$e_x = \frac{x}{N} = \frac{12.7}{12} = 1.06 \text{ mm extension}$$

$$e_y = \frac{3D_my}{N(L_b + x)} = \frac{3 \times 648 \times 0}{12 \times (305 + 12.7)} = 0 \text{ mm}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{0 \times 648}{2 \times 305} = 0 \text{ mm}$$

$$K_{\theta l} = 1.0; e_{yp} = 0; R_\theta = 1.0; K_\theta = 1.0; C_\theta = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 - 1.06 = -1.06 \\ 0 - 1.06 = -1.06 \end{array} \right\} = -1.06 \text{ mm} < 6.35 \text{ mm (rated)}$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0 + 0 + 1.06 = 1.06 \\ 0 + 1.06 = 1.06 \end{array} \right\} = 1.06 \text{ mm} < 3.18 \text{ mm (rated)}$$

Operating

$$e_x = \frac{x}{N} = \frac{25.4}{12} = 2.12 \text{ mm compression}$$

$$e_y = \frac{3D_m y}{N(L_b - x)} = \frac{3 \times 648 \times 1.524}{12 \times (305 - 25.4)} = 0.88 \text{ mm}$$

$$e_\theta = \frac{\theta D_m}{2N} = \frac{0.0873 \times 648}{2 \times 12} = 2.36 \text{ mm}$$

$$K_{\theta l} = 1 + .024(L_b / D_m)^{1.33} = 1 + 0.0095 \times 1.25 \times (305 / 648)^{1.33} = 1.005$$

$$e_{yp} = \frac{\pi D_m K_{\theta l} P \sin(\theta / 2)(L_b - x)}{4 f_{iu}} = \frac{\pi \times 648 \times 1.005 \times 0.276 \times \sin(.0873 / 2) \times (305 - 25.4)}{4 \times 2200} = 0.782 \text{ mm}$$

$$R_\theta = \frac{1.18 N^2 (q - e_x)^2}{\pi^2 D_m K_{\theta l} \sin(\theta / 2)(L_b - x)} = \frac{1.18 \times 12^2 \times (25.4 - 2.12)^2}{\pi^2 (648)(1.005) \sin(.0873 / 2)(305 - 25.4)} = 1.175$$

$$K_\theta = \frac{e_\theta + e_{yp}}{e_\theta} = \frac{2.36 + 0.782}{2.36} = 1.33$$

$$C_\theta = \text{MIN}(R_\theta; 1.0) = \text{MIN}(1.175; 1.0) = 1.0$$

$$e_c = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta + |e_x| \\ e_\theta K_\theta + |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0.88 + 2.36 + 2.12 = 5.36 \\ 3.13 + 2.12 = 5.25 \end{array} \right\} = 5.36 \text{ mm} < 6.35 \text{ mm}$$

$$e_e = \text{MAX} \left\{ \begin{array}{l} e_y + e_\theta - |e_x| \\ e_\theta K_\theta - |e_x| \end{array} \right\} = \text{MAX} \left\{ \begin{array}{l} 0.88 + 2.36 - 2.12 = 1.12 \\ 3.13 - 2.12 = 1.01 \end{array} \right\} = 1.12 \text{ mm} < 3.175 \text{ mm}$$

$$e_c (\text{preset}) + e_e (\text{operating}) = -1.06 + 1.125 = -0.065 \text{ mm}$$

$$e_e (\text{preset}) + e_c (\text{operating}) = 1.06 + 5.36 = 6.42 \text{ mm} = e$$

Example 10: Rectangular Expansion Joint Movements

Case 1: Assume a 108 x 66 in. rectangular universal Expansion Joint is to be installed in the neutral position (no cold spring) and is to be subjected to the following two (2) sets of operating deflections:

Condition 1

$$x = 1 \text{ in. compression}$$

$$y_l = .25 \text{ in.}$$

$$y_s = .45 \text{ in.}$$

$$\theta = 0$$

Condition 2

$$x = .5 \text{ in. compression}$$

$$y_l = .38 \text{ in.}$$

$$y_s = .25 \text{ in.}$$

$$\theta_l = .0175 \text{ radians (1 degree per bellows element)}$$

SOLUTION:*Bellows Data:*

$$\text{Convolution Height} = 6 \text{ in.}$$

$$L_l = 108 \text{ in.} + 6 \text{ in.} = 114 \text{ in.}$$

$$L_s = 66 \text{ in.} + 6 \text{ in.} = 72 \text{ in.}$$

$$e_c(\text{rated}) = .75 \text{ in.}$$

$$e_e(\text{rated}) = .50 \text{ in.}$$

$$q = 2.50 \text{ in.}$$

$$N = 4$$

$$L_u = 50 \text{ in.}$$

$$L_b = N_q = 4 \times 2.50 = 10 \text{ in.}$$

$$L^* = L_u - L_b = 50 - 10 = 40$$

$$L^*/L_b = 40/10 = 4$$

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Condition 1

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_l = \frac{(3)(114)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 - 1.0/2)} (.25)$$

$$= (4.275)(.102)(1.013)(.25) = .11 \text{ in.}$$

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_s = \frac{(3)(72)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 - 1.0/2)} (.45)$$

$$= (2.7)(.102)(1.013)(.45) = .126 \text{ in.}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{1}{(2)(4)} = .125 \text{ in. compression}$$

$$e_c = e_{yl} + e_{ys} + e_\theta + |e_x| = .11 + .126 + 0 + .125 = .361 \text{ in.} < .75 \text{ in. (rated)} = e$$

$$e_e = e_{yl} + e_{ys} + e_\theta - |e_x| = .11 + .126 + 0 - .125 = .111 \text{ in.} < .50 \text{ in. (rated)}$$

Condition 2

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_l = \frac{(3)(114)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 - .50/2)} (.38)$$

$$= (4.275)(.102)(1.006)(.38) = .167 \text{ in.}$$

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_s = \frac{(3)(72)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 - .50/2)} (.25)$$

$$= (2.7)(.102)(1.006)(.25) = .069 \text{ in.}$$

$$e_{\theta l} = \frac{\theta_l L_l}{2N} = \frac{(.0175)(114)}{(2)(4)} = 0.249 \text{ in.}$$

$$e_{\theta s} = 0$$

$$e_x = \frac{x}{2N} = \frac{.5}{(2)(4)} = 0.062 \text{ in. extension}$$

$$e_c = e_{yl} + e_{ys} + e_\theta + |e_x| = .167 + .069 + .126 + 0 + .062 = .424 \text{ in.} < .75 \text{ in. (rated)} = e$$

$$e_e = e_{yl} + e_{ys} + e_\theta - |e_x| = .167 + .069 + .126 + 0 - .062 = .30 \text{ in.} < .50 \text{ in. (rated)}$$

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Case 2: Assume the same 108 x 66 in. rectangular universal Expansion Joint except that it is to be installed with a .25 in. lateral cold spring (long side) and .5 in. axial pre-extension and is to be subjected to the following operating deflection:

Cold Spring $x = .50$ in. extension

$$y_l = .25 \text{ in.}$$

$$\theta = 0$$

Operating $x = 1$ in. compression from pre-extended position (.50 in. compression from neutral

$y_l = .25$ in. from neutral in direction opposite to lateral cold spring.

$$y_s = .45 \text{ in.}$$

$$\theta = 0$$

$$L_1 = 108 \text{ in.} + 6 \text{ in. convolution height} = 114 \text{ in.}$$

SOLUTION:

Cold Spring

$$\begin{aligned} e_{yl} &= \frac{3L_l}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* + x/2)} y_l \\ &= \frac{(3)(114)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 + .50/2)} (.25) \\ &= (4.275)(.102)(.9938)(.25) = .108 \text{ in.} \end{aligned}$$

$$e_{ys} = 0$$

$$e_{\theta} = 0$$

$$e_x = \frac{x}{2N} = \frac{.50}{(2)(4)} = .062 \text{ in. extension}$$

$$e_c = +e_{yl} + e_{ys} + e_{\theta} + |e_x| = .108 + 0 + 0 + .062 = .17 \text{ in.} < .75 \text{ in. (rated)}$$

$$e_e = +e_{yl} + e_{ys} + e_{\theta} - |e_x| = .108 + 0 + 0 - .062 = .046 \text{ in.} < .5 \text{ in. (rated)}$$

Operating

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_l = \frac{(3)(114)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 - 1.0/2)} (.25)$$

$$= (4.275)(.102)(1.013)(.25) = .11 \text{ in.}$$

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1 + L^*/L_b}{1 + 3(L^*/L_b)^2} \cdot \frac{L^*}{(L^* - x/2)} y_s = \frac{(3)(72)}{(2)(4)(10)} \cdot \frac{1 + (4)}{1 + (3)(4)^2} \cdot \frac{(40)}{(40 - 1.0/2)} (.45)$$

$$= (2.7)(.102)(1.013)(.45) = .126 \text{ in.}$$

$$e_\theta = 0$$

$$e_x = \frac{x}{2N} = \frac{.50}{(2)(4)} = .062 \text{ in. compression}$$

$$e_c = e_{yl} + e_{ys} + e_\theta + |e_x| = .11 + .126 + 0 + .062 = .298 \text{ in.} < .75 \text{ in. (rated)}$$

$$e_e = e_{yl} + e_{ys} + e_\theta - |e_x| = .11 + .126 + 0 - .062 = .174 \text{ in.} < .50 \text{ in. (rated)}$$

$$e_c(\text{cold spring}) + e_e(\text{operating}) = .17 + .174 = .344 \text{ in.}$$

$$e_e(\text{cold spring}) + e_c(\text{operating}) = .046 + .298 = .344 \text{ in.} = e$$

Example 10M: Rectangular Expansion Joint Movements

Case 1: Assume a 2773 x 1676 mm rectangular universal Expansion Joint is to be installed in the neutral position (no cold spring) and is to be subjected to the following two (2) sets of operating deflections:

Condition 1

$$x = 25.4 \text{ mm compression}$$

$$y_l = 6.35 \text{ mm}$$

$$y_s = 11.43 \text{ mm}$$

$$\theta = 0$$

Condition 2

$$x = 12.7 \text{ mm compression}$$

$$y_l = 9.65 \text{ mm}$$

$$y_s = 6.35 \text{ mm}$$

$$\theta_l = .0175 \text{ radians (1 degree) per bellows element}$$

SOLUTION:*Bellows Data:*

$$\text{Convolution Height} = 152.4 \text{ mm}$$

$$L_l = 2773 \text{ mm} + 152.4 \text{ mm} = 2896 \text{ mm [actually 2925.4]}$$

$$L_s = 1676 \text{ mm} + 152.4 \text{ mm} = 1829 \text{ mm}$$

$$e_c (\text{rated}) = 19.05 \text{ mm}$$

$$e_e (\text{rated}) = 12.7 \text{ mm}$$

$$q = 63.5 \text{ mm}$$

$$N = 4$$

$$L_u = 1270 \text{ mm}$$

$$L_b = Nq = 4 \times 63.5 = 254 \text{ mm}$$

$$L^* = L_u - L_b = 1270 - 254 = 1016 \text{ mm}$$

$$L^*/L_b = 1016/254 = 4$$

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Condition 1

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*-x/2)} y_l = \frac{3 \times 2896}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016-25.4/2} \cdot 6.35$$

$$= 4.275 \times 0.102 \times 1.013 \times 6.35 = 2.8 \text{ mm}$$

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*-x/2)} y_s = \frac{3 \times 1829}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016-25.4/2} \cdot 11.43$$

$$= 2.7 \times 0.102 \times 1.013 \times 11.43 = 3.2 \text{ mm}$$

$$e_{\theta} = 0$$

$$e_x = \frac{x}{2N} = \frac{25.4}{2 \times 4} = 3.175 \text{ mm compression}$$

$$e_c = e_{yl} + e_{ys} + e_{\theta} + |e_x| = 2.8 + 3.2 + 0 + 3.175 = 9.175 \text{ mm} < 19.05 \text{ (rated)} = e$$

$$e_e = e_{yl} + e_{ys} + e_{\theta} - |e_x| = 2.8 + 3.2 + 0 - 3.175 = 2.83 \text{ mm} < 12.7 \text{ (rated)} = e$$

Condition 2

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*-x/2)} y_l = \frac{3 \times 2896}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016-12.7/2} \cdot 9.56$$

$$= 4.275 \times 0.102 \times 1.006 \times 9.56 = 4.2 \text{ mm}$$

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*-x/2)} y_s = \frac{3 \times 1829}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016-12.7/2} \cdot 11.43$$

$$= 2.7 \times 0.102 \times 1.006 \times 6.35 = 1.76 \text{ mm}$$

$$e_{\theta l} = \frac{\theta_l L_l}{4N} = \frac{0.0175 \times 2,896}{4 \times 4} = 3.17 \text{ mm}$$

$$e_{\theta s} = 0$$

$$e_x = \frac{x}{2N} = \frac{12.7}{2 \times 4} = 1.6 \text{ mm extension}$$

$$e_c = e_{yl} + e_{ys} + e_{\theta l} + e_{\theta s} + |e_x| = 4.2 + 1.76 + 3.17 + 0 + 1.6 = 10.7 \text{ mm} < 19 \text{ mm (rated)} = e$$

$$e_e = e_{yl} + e_{ys} + e_{\theta} - |e_x| = 4.2 + 1.76 + 3.17 + 0 - 1.6 = 7.6 \text{ mm} < 12.7 \text{ mm (rated)}$$

Case 2: Assume the same 2773×1676 mm rectangular universal Expansion Joint except that it is to be installed with a 6.35 mm lateral cold spring (long side) and 12.7 mm axial pre-extension and is to be subjected to the following operating deflection:

Cold Spring

$$x = 12.7 \text{ mm extension}$$

$$y_l = 6.35 \text{ mm}$$

$$\theta = 0$$

Operating

$$x = 25.4 \text{ mm compression from pre-extended position (12.7 mm compression from neutral)}$$

$$y_l = 6.35 \text{ mm from neutral in direction opposite to lateral cold spring}$$

$$y_s = 11.4 \text{ mm}$$

$$\theta = 0$$

$$L_l = 2743 \text{ mm} + 152 \text{ mm convolution height} = 2895 \text{ mm}$$

SOLUTION:

Cold Spring

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*+x/2)} y_l = \frac{3 \times 2,895}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016+12.7/2} \cdot 6.35$$

$$= 4.275 \times 0.102 \times 0.9938 \times 6.35 = 2.74 \text{ mm}$$

$$e_{ys} = 0$$

$$e_{\theta} = 0$$

$$e_x = \frac{x}{2N} = \frac{12.7}{2 \times 4} = 1.58 \text{ mm extension}$$

$$e_c = +e_{yl} + e_{ys} + e_{\theta} + |e_x| = 2.74 + 0 + 0 + 1.58 = 4.32 \text{ mm} < 19.05 \text{ mm (rated)}$$

$$e_e = +e_{yl} + e_{ys} + e_{\theta} - |e_x| = 2.74 + 0 + 0 - 1.58 = 1.16 \text{ mm} < 12.7 \text{ mm (rated)}$$

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Operating

$$e_{yl} = \frac{3L_l}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*-x/2)} y_l = \frac{3 \times 2,895}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016-25.4/2} \cdot 6.35$$

$$= 4.275 \times 0.102 \times 1.013 \times 6.35 = 2.79 \text{ mm}$$

$$e_{ys} = \frac{3L_s}{2NL_b} \cdot \frac{1+L^*/L_b}{1+3(L^*/L_b)^2} \cdot \frac{L^*}{(L^*-x/2)} y_s = \frac{3 \times 1,829}{2 \times 4 \times 254} \cdot \frac{1+4}{1+3 \times 4^2} \cdot \frac{1016}{1016-25.4/2} \cdot 11.4$$

$$= 2.7 \times 0.102 \times 1.013 \times 11.4 = 3.2 \text{ mm}$$

$$e_{\theta} = 0$$

$$e_x = \frac{x}{2N} = \frac{12.7}{2 \times 4} = 1.59 \text{ mm compression}$$

$$e_c = e_{yl} + e_{ys} + e_{\theta} + |e_x| = 2.79 + 3.2 + 0 + 1.59 = 7.58 \text{ mm} < 19.05 \text{ mm (rated)}$$

$$e_e = e_{yl} + e_{ys} + e_{\theta} - |e_x| = 2.79 + 3.2 + 0 - 1.59 = 4.4 \text{ mm} < 12.7 \text{ mm (rated)}$$

$$e_c(\text{cold spring}) + e_e(\text{operating}) = 4.32 + 4.4 = 8.7 \text{ mm}$$

$$e_e(\text{cold spring}) + e_c(\text{operating}) = 1.16 + 7.58 = 8.7 \text{ mm} = e$$

Example 11: Sample Calculation for a Straight Run of Pipe Containing an Axial Expansion Joint (See Table IV)

Given a 150 foot length of carbon steel pipe operating between 25° F. and 375° F.

Expansion at 375° F = 2.48 in./100 ft.

Expansion at 25° F = -0.32 in./100 ft.

SOLUTION:

The difference = 2.80 in./100 ft.

The change in length for 150 feet would be $(2.80) (150/100) = 4.20$ in.

Although bellows Expansion Joints can be designed to absorb both axial compression and axial extension, for purposes of a sample calculation, it is assumed that the Expansion Joint can absorb only axial compression.

The above sample calculation would be complete if the Expansion Joint were being installed at 25° F. Frequently, an Expansion Joint is installed at a temperature higher than the minimum design temperature of the piping system. The piping will contract in such a case and the Expansion Joint will be extended beyond its installed length. It is obvious in the sample calculation above that if the Expansion Joint was installed at 70° F, the pipeline would contract a total of $0.32 \times 150/100$ inches or 0.48 inches and would expand $2.48 \times 150/100$ inches or 3.72 inches from this installation temperature. Since the Expansion Joint selected is rated for axial compression only, it must be pre-compressed prior to installation in order to provide for extension when the pipeline contracts from 70° F to the minimum design temperature of 25° F.

The amount of pre-compression equals:

$$\frac{(\text{Rated Movement}) \times (\text{Coefficient at } T_{\text{inst.}} - \text{Coefficient at } T_{\text{min.}})}{\text{Coefficient at } T_{\text{max.}} - \text{Coefficient at } T_{\text{min.}}} \quad (\text{J-4})$$

Assuming the selected Expansion Joint is rated for 5 inches axial compression, and substituting numerical values from the above example in this formula, we have:

$$\frac{(5) \times [0 - (-0.32)]}{[2.48 - (-0.32)]} = 0.57 \text{ inches of pre-compression}$$

This leaves 4.43 inches for compression from the installed position. Thus, the use of an Expansion Joint rated for 5 inches axial compression, installed pre-compressed 0.57 inches will provide a means for absorbing the contraction of the pipe from the installation temperature to the minimum temperature (0.48 in.) as well as the expansion of the pipe from the installation temperature to the maximum temperature (3.72 in.).

No allowance has been made in this sample calculation for conditions a, b, or c in Section 2.1.

It may be necessary to field pre-compress Expansion Joints where information on thermal expansion coefficients is not available. When this occurs pre-compression may be approximated by the following formula:

$$\frac{(\text{Rated Movement}) \times (T_{\text{inst.}} - T_{\text{min.}})}{(T_{\text{max.}} - T_{\text{min.}})} \quad (\text{J-5})$$

Example 11M: Sample Calculation for a Straight Run of Pipe Containing an Axial Expansion Joint (See Table IV)

Given a 45.7 m length of carbon steel pipe operating between -4°C and 191°C.

Expansion at 191°C = 2.07 mm/m

Expansion at -4°C = -0.266 mm/m

SOLUTION:

The difference = 2.336 mm/m

The change in length for 45.7 m would be $2.336 \times 45.7 = 106.8$ mm.

Although bellows Expansion Joints can be designed to absorb both axial compression and axial extension, for purposes of a sample calculation, it is assumed that the Expansion Joint can absorb only axial compression.

The above sample calculation would be complete if the Expansion Joint were being installed at -4°C. Frequently, an Expansion Joint is installed at a temperature higher than the minimum design temperature of the piping system. The piping will contract in such a case and the Expansion Joint will be extended beyond its installed length. It is obvious in the sample calculation above that if the Expansion Joint was installed at 21°C, the pipeline would contract a total of 0.266×45.7 m or 12.2 mm and would expand $2.07 \times 45.7/30.5 = 3.1$ mm or 94.5 mm from this installation temperature. Since the Expansion Joint selected is rated for axial compression only, it must be pre-compressed prior to installation in order to provide for extension when the pipeline contracts from 21°C to the minimum design temperature of -4°C.

The amount of pre-compression equals:

$$\frac{(\text{Rated Movement}) \times (\text{Coefficient at } T_{\text{inst.}} - \text{Coefficient at } T_{\text{min.}})}{\text{Coefficient at } T_{\text{max.}} - \text{Coefficient at } T_{\text{min.}}} \quad (\text{J-4})$$

Assuming the selected Expansion Joint is rated for 127 mm axial compression, and substituting numerical values from the above example in this formula, we have:

$$\frac{127 \times [0 - (-0.266)]}{2.07 - (-0.266)} = 14.5 \text{ mm of pre-compression.}$$

This leaves 112.5 mm for compression from the installed position. Thus, the use of an Expansion Joint rated for 127 mm axial compression, installed pre-compressed 14.5 mm will provide a means for absorbing the contraction of the pipe from the installation temperature to the minimum temperature (12.2 mm) as well as the expansion of the pipe from the installation temperature to the maximum temperature (94.5 mm).

No allowance has been made in this sample calculation for conditions a, b, or c in Section 2.1.

It may be necessary to field pre-compress Expansion Joints where information on thermal expansion coefficients is not available. When this occurs pre-compression may be approximated by the following formula:

$$\frac{(\text{Rated Movement}) \times (T_{\text{inst.}} - T_{\text{min.}})}{(T_{\text{max.}} - T_{\text{min.}})} \quad (\text{J-5})$$

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