Preliminary of Piping and Pipeline Engineering

Fundamental

The seven fundamental areas of competence in the mechanical engineering discipline are (1) materials. (2) design, (3) construction, (4) inspection, (5) testing, (6) maintenance, and (7) operations.

In each of the seven fundamental areas, the responsible engineer must make a series of decisions to achieve the optimum and most cost-effective operation. The following checklists summarize the key decision points related to piping and pipeline activities.

Piping Design

1. *Internal Pressure*

1.1 *Pressure Design of Piping*

1.1.1 *Thin wall approximation*

Consider a straight section of pipe filled with a pressurized liquid or gas. The internal pressure generates three principal stresses in the pipe wall : as illustrated in Figure 1-1: a hoop stress σ_r. When the ratio of the pipe diameter to its wall thickness D/t is greater than 20 the pipe may be considered to thin wall. In this case, the hoop stress is nearly constant through the wall thickness and equal to

$$\sigma_h = \frac{PD}{2t}$$

P = design pressure, psi
D = outside pipe diameter, in
t = pipe wall thickness, in

The longitudinal stress is also constant through the wall and equal to half the hoop stress

$$\sigma_1 = \frac{PD}{4t}$$

The radial stress varies through the wall, from P at the inner surface of the pipe to zero on the outer surface.

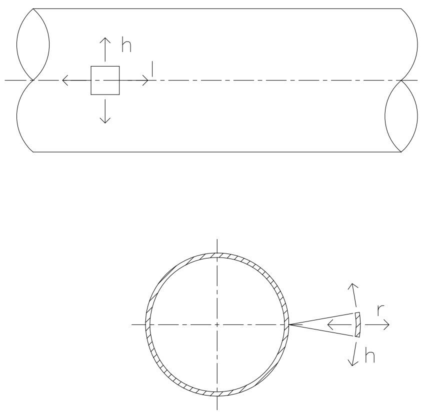


Figure 1-1 Hoop (h), Longitudinal (1) and Radial ® Stress Directions

1.1.2 *Pipeline design equation*

For oil and gas pipelines, the thickness of the pipe wall is obtained by writing that the hoop stress, which is the largest stress in the pipe, must be limited to a certain allowable stress S. Using the thin wall approximation, this condition corresponds to

$$\frac{PD}{2t} < S$$

P = internal design pressure, psi
D = pipe outer diamenter, in
t = pipe wall thickness, in
S = allowable stress, psi

For hazardous liquid pipelines (hydrocarbon, carbon dioxide, etc.) the allowable stress is set at [ASME B31.4]

$$S = 0.72\ S_Y E$$

0.72 = design factor
E = longitudinal weld joint factor, Table 1-1
S_y = specified minimum yield strength,psi, Table 1-2

For gas pipelines, the allowable stress is [ASME B31.8]

$$S = S_Y \text{ F E T}$$

P = design pressure, psi
D = nominal outside diameter, in
S_Y = specified minimum yield stress,psi, Table 1-2 (commonly referred to as SMYS in the pipeline industry)
F = design factor, Table 1-3
E = weld joint factor, Table 1-1
T = temperature derating factor, Table 1-4

There are two equally plausible versions of the origin of 0.72 S_Y. The first explanation is that 0.72 S_Y goes back to the early days of fabrication of steel line pipe. In the mill, the pipe was tested to a hydrostatic pressure causing a hoop stress PD/(2t) of 90% S_y = 72% S_Y. The second explanation is that the 90% S_Y hydrostatic test was reduced by 12.5% for fabrication tolerance on under thickness, then further divided by 1.1 to compensate for the 110% overpressure transient allowance (as was the common practice for water pipelines), which leads to 90% S_Y x 0.875 / 1.1 = 0.72 S_Y.

The weld quality or joint efficiency factor E is a factor introduced to account for the quality of the longitudinal or spiral seam in a pipe. It is a function of the reliability and quality of fabrication and the extent of inspections performed in the pipe mill. An electric resistant welded pipe is judges to have a superior seam quality, and its weld joint efficiency factor is assigned the maximum valve 1.0. On the other hand, the seam weld of a furnace butt-weld pipe was judged to have a seam weld factor of only 0.6. These values were established decades ago and were based on experience with the various methods of pipe fabrication at the time. The quality of U.S. pipe fabrication and mill inspections have greatly improved since these values were first established, and today the lower E values (which penalize the wall thickness) may well be too restrictive.

1.1.3 Yield and wall thickness

At ambient temperature, the yield stress used in the minimum wall design equation is the minimum value of the material's yield stress obtained from the pipe material specification.

Table 1-1 Examples of Longitudinal Weld Joint Factors E [ASME B31.8]

Material	Pipe Class	E
ASTM A 53,A106	Seamless	1.0
ASTM A 53	ERW	1.0
ASTM A 53	Furnace Butt Welded	0.6
ASTM A 134	Electric Fusion Arc Welded	0.8
ASTM A 135	Electric Resistance Welded (ERW)	1.0
API 5L	Seamless	1.0
API 5L	Submerged Arc Welded or ERW	1.0
API 5L	Furnace Butt Welded	0.6

Table 1-2 Examples of Yield and Ultimate Stress [ASME II Part D]

Temperature (°F)	A 106 Gr.B $S_Y\,[ksi]$	A 106 Gr.B $S_u\,[ksi]$	A 312 T.304 $S_Y\,[ksi]$	A 312 T.304 $S_u\,[ksi]$
100	35.0	60	30.0	75.0
200	31.9	60	25.0	71.0
300	31.0	60	22.5	66.0
400	30.0	60	20.7	64.4
500	28.3	60	19.4	63.5

Table 1-3 Location Design Factor F [ASME B31.8]

Location	F
Class 1 Div.1: Deserts, farm land, sparsely populated, etc	0.8
Class 1 Div.2: Class 1, with line tested to 110% design	0.72
Class 2: Industrial areas, town fringes, ranch, etc.	0.6
Class 3: Suburban housing, shipping centers, etc.	0.5
Class 4: Multistory buildings, heavy traffic, etc.	0.4

Note : Lowe location design factors apply at crossing, compressor station, etc. The pipeline designer must refer to codes and regulations for the applicable location design factor.

Table 1-4 Temperature Derating Factor [B31.8]

Temperature (°F)	T
250 or less	1.0
300	0.967
350	0.933
400	0.9
450	0.867

For older pipeline, where the material specification or the pipe wall thickness is unknown, the Code of Federal Regulation,[CFR 49] permits the use of an estimate of yield stress and wall thickness. The yield stress S_y can be based on measured values, in which case it is the smallest of 80% of the average measured yield stress, but not more than 52 ksi. The wall thickness t could be the average of four measurements at the pipe end. For example, if we measure 0.333", 0.333",0.330",0.330", the average is 0.3315", and the pipe is assigned a schedule 40 with a wall of 0.332", For pipe smaller than 20" nominal pipe size (NPS) the chosen wall (0.322") may not be larger than 1.14 times the smallest measured wall, in our case 1.14 x 0.330" = 0.376". For pipe 20" or more use 1.11 in place of 1.14. Since 0.322" < 0.376" the choice of a schedule 40 is valid. The material specification for line pipe [API 5L] permits wall under-thickness ranging from -8% to -12.5% depending on the pipe size and grade. For example, a 20" pipe grade X42, ordered to a nominal wall of 0.5" can have an actual wall 8% smaller or 0.46". The B31.4 and B31.8 pipeline allowable stress of 72% S_Y accounts for a 12.5% fabrication under-tolerance.

1.2 Pressure design of plant piping

1.2.1 Lame's formula

Without the thin wall approximation, the more general form of the three principal stresses in a closed cylinder subject to internal pressure P is given by Lame's formula.

$$\sigma_t = \mathrm{P}\frac{r_i^{\,2}}{r_o^{\,2} - r_i^{\,2}}\left(1 + \frac{r_o^{\,2}}{r^2}\right)$$

$$\sigma_r = \mathrm{P}\frac{r_i^{\,2}}{r_o^{\,2} - r_i^{\,2}}\left(1 - \frac{r_o^{\,2}}{r^2}\right)$$

$$\sigma_l = \mathrm{P}\frac{r_i^{\,2}}{r_o^{\,2} - r_i^{\,2}}$$

σ_t = tangential (hoop) stress, psi
σ_r = radial stress, psi
σ_l = longitudinal (axial) stress, psi
r_i = inner pipe radius, in
r_o = outer pipe radius, in
r = radial distance of a point in the pipe wall, in

1.2.2 *Early design equations*

Can Lame's formula be used to design piping systems? This question became of particular interest for the design of boilers in the early 1900's when the use of steam engines was quickly expanding and with it the use of higher steam pressures. At that time, several equations were used in the design of boilers, vessels and piping [Parsons]. The wall thickness rule applied by the U.S. Board of Supervising Inspectors of Steam Vessels in the early 1900's was

$$P = kSt / (3D)$$

P = working pressure, psi
k = 1 for single riveted, 1.2 for double riveted vessels
S = tensile strength, psi
t = wall thickness, in
D = mean diameter, in

Lloyd's rule was

$$P = C(T-2)B / D$$

C = safety factor
T = thickness in sixteenths of an inch
B = least percentage of strength of joint

The Board of Trade rule was

$$P = SB2t / (DC)$$

British Corporation's rule was

$$P = C(T-1)B / D$$

Let's use the U.S. Board of Supervising Inspectors of Steam Vessels as an example. With this rule, a double riveted 60" diameter vessel, 0.5" thick, with an ultimate strength of 50,000 psi would be permitted to operate at a pressure P = 1.2 x 50,000 x 0.5 / (3 x 60) = 167 psi. The hoop stress is PD / (2t) = 10,020 psi, five times less than the ultimate strength.

The factor of safety of 5 is consistent with the pre-World War II rules of the ASME pressure vessel code. Because of the shortage of steel during World War II, the safety factor was reduced to 4 in 1944, and then increased back to 5 until 1951. It was then permanently reduced to 4, until the late 1990's when it was further reduced to 3.5 in ASME B31.1 and ASME VIII Division 1.

1.2.3 *Piping design*

In 1951, in an attempt to make the design process more uniform, a Task group of the American Society of Mechanical Engineers investigated a number of pressure stress equations and failure criteria. In all, thirty different pressure design formulas and approximations were compiled. Seeking a balance between accuracy and simplicity, the task group proceeded as follows [Burrows]. It was first assumed that as the pipe deforms under pressure, the material maintains a constant volume. This corresponds to a material Poisson ratio ν of 0.5, in which case, the Saint Venant, Tresca and Von Mises equivalent stress can be written in a similar form:

Maximum strain energy (Saint Venant):

$$\sigma = \frac{3}{4}(\sigma_t - \sigma_r) = \frac{\sigma_t - \sigma_r}{1.33}$$

Maximum shear stress (Tresca):

$$\sigma = \sigma_t - \sigma_r = \frac{\sigma_t - \sigma_r}{1.0}$$

Maximum energy (Von Mises):

$$\sigma = \sqrt{\frac{3}{4}(\sigma_t - \sigma_r)} = \frac{\sigma_t - \sigma_r}{1.15}$$

All three stress expressions can be written in the form

$$\sigma = \frac{\sigma_t - \sigma_r}{K}$$

With K = 1.0,1.15 or 1.33. In addition, the average principal stresses through the wall are

$$\sigma_{t,avg} = P\left(\frac{D}{2t} - 1\right)$$

$$\sigma_{r,avg} = -\frac{P}{2}$$

$$\sigma_{r,avg} = P\left(\frac{D}{4t} - 0.75\right)$$

Substituting, we obtain

$$\sigma = \frac{\sigma_{t,avg} - \sigma_{r.avg}}{K} = \frac{P}{K}\left(\frac{D}{2t} - 0.5\right)$$

In 1943, a factor of 0.4 was recommended in place of 0.5 in the stress formula [Boardman]. The task group adopted the recommendation, and selected the maximum shear stress failure criterion, and therefore K = 1, which led to

$$\sigma = P\left(\frac{D}{2t} - 0.4\right)$$

At this point, questions were raised regarding the applicability of this equation to high temperature steam service. In light of burst test data available at the time, the task group judged that the choice of the 0.4 factor, while adequate for temperatures up to 900°F, would result in excessively thick pipe for service above 900°F. This would in turn lead to unnecessarily heavy and costly pipe, less flexibility to absorb thermal expansion and larger through-wall thermal gradients. To avoid these difficulties, and based on burst test results, it was decided that between 900°F and 1150°F the 0.4 factor would be gradually increased to 0.7. This was the origin of the y factor in today's ASME B31.1 and ASME B31.3 design equations. The values of y are listed in Table 1.5:

$$\sigma = P\left(\frac{D}{2t} - y\right)$$

Table 1-5 Coefficient y for t < D/6 for Temperatures T (°F) [ASME B31.3]

Material	$T \leq 900$	950	1000	1050	1100	≥ 1150
Ferritic Steel	0.4	0.5	0.7	0.7	0.7	0.7
Austenitic Steel	0.4	0.4	0.4	0.4	0.5	0.7
Ductile Metals	0.4	0.4	0.4	0.4	0.4	0.4
Cast Iron	0.0	-	-	-	-	-

Setting the stress σ equal to its maximum allowable value S multiplied by the weld quality factor E, and rearranging the terms, we obtain the ASME B31.1 and ASME B31.3 design equation

$$t = \frac{PD_o}{2(SE + Py)}$$

t = minimum required wall thickness, excluding manufacturing tolerance and allowances for corrosion (in)
P = internal design pressure, psi
D_o = outside diameter of pipe, in
E = joint efficiency factor
y = temperature coefficient (Table 1-5)
S = maximum allowable stress in material, psi

1.2.4 *Allowable stress*

The allowable stress for pipelines is 72%S_y and does not depend on the material's ultimate strength. The allowable stress for power and process plant piping systems is

$$S(T) = \min. \{S_y(T)/SF_y ; S_u(T)/SF_u\}$$

S(T) = allowable stress at design temperature T, psi
SF_y = safety factor applied to yield stress
SF_u =safety factor applied to ultimate strength
$S_y(T)$ =minimum specified yield stress at design temperature T,psi
$S_u(T)$= minimum specified ultimate strength at design temperature T, psi

For carbon steel pipe in ASME B31.1 applications

$$S(T) = \min.\{2 S_y(T)/3; S_u(T)/4\}$$

For carbon steel pipe in ASME B31.3 applications

$$S(T) = \min.\{2 S_y(T)/3; S_u(T)/3\}$$

For austenitic stainless steel in ASME B31.1 or B31.3 applications

$$S(T) = \min.\{90\% S_y(T); Su(T)/3\}$$

Where the values of yield stress S_y or ultimate strength S_u at design temperature are larger than at room temperature, the room temperature values are used. Some values of allowable stress are listed in Table 1-6.

Table 1-6 ASME B31.3 Allowable Stress

Material	100°F	200°F	300°F	400°F	500°F
A 106 Gr.B	20.0	20.0	20.0	20.0	18.9
API 5L X52	22.0	22.0	22.0	22.0	-
A 312 Type 304	20.0	20.0	20.0	18.7	17.5
B 241 6061 T6	12.7	12.7	10.6	5.6	-

In the late 1990's, the safety factor of 4 against ultimate strength, used in the design of B31.1 carbon steel pipe, and ASME B&PV Section VIII Division 1 pressure vessels, was reduced to 3.5 a larger safety factor SF_u applies to materials less ductile than steel. For example the allowable stress for cast iron is S_u / 10, and the allowable stress for malleable iron is S_u / 5.

1.2.5 *Wall thickness allowance*

Having established the minimum required wall thickness, the designer should add a corrosion allowance and, for piping systems but not for pipelines, a fabrication tolerance. The minimum wall thickness required by code plus allowance is then rounded up to obtain the commercial pipe size to be procured and used in construction.

It is up to the designer to select the corrosion allowance, based on experience with similar fluids, pipe material, temperatures and flow rates

The tolerance on wall thickness depends on the pipe material specifications. For example, an ASTM A106 carbon steel pipe may be furnished 12.5% below the specified nominal pipe wall thickness. Therefore, the minimum pipe wall thickness t_{min} calculated by the code design equation needs to be increased by the corrosion allowance C and the fabrication tolerance f (for example with a fabrication tolerance of 12.5% on the pipe wall thickness, f = 0.125). With these corrections, we now obtain the commercial pipe size to be procured:

$$t = (t_{min} + C)(1 + f)$$

t = pipe wall thickness, is
t_{min} = minimum wall required by Code, in
C = corrosion or threading allowance, in
f = pipe wall thickness fabrication tolerance

2. Layout Plan

2.1 Spacing Of Pipe Supports

The weight of piping and components must be supported to achieve five objectives: (1) minimize stresses in the piping, (2) maintain the intended layout and slope, (3) avoid excessive sag, (4) minimize reactions on equipment nozzles, and (5) optimize the type, size and location of pipe supports. To achieve these objectives, and given the pipe routing, the design process starts by placing weight supports at regular intervals, following a support spacing guide such as given in Table 2-1.

Table 2-1 Support Spacing for Steel Pipe [ASME B31]

Pipe Size (in)	Water (ft)	Gas (ft)
1	7	9
2	10	13
3	12	15
4	14	17
6	17	21
8	19	24
12	23	30
16	27	35
20	30	39
24	32	42

This is a classic spacing table for steel pipe. It is based on a maximum bending stress of 2300 psi and a maximum sag at mid-span of 0.10”. In practice, longer spans are usually feasible, with a deadweight bending stress in the order of 5000 psi to 10,000 psi; provided the sag between supports remains acceptable.

As a rule of thumb (by close examination of Table 2-1) the spacing of pipe supports for steel pipes in liquid service, expressed in feet, may be taken as the nominal pipe size, expressed in inches, plus ten. For example, the spacing of pipe supports on a 6” line will be approximately 6 + 10 = 16 feet.

This spacing changes at high temperatures and for materials other than steel. For example, for copper tubing support spacing varies from 8-ft for 1” tubing to 12-ft for 4” [Grinell]. For PVC pipe the spacing depends on the pipe schedule and operating temperature. The span is 4-ft for ¾ pipe, up to 6-ft for 4” schedule 40 pipe at ambient temperature; approximately 1-ft more for schedule 80, and half that spacing at 150°F [Grinell]. For fiber reinforced plastic pipe (FRP), support spacing in liquid service would vary from around 11-ft for 2” pipe to 22-ft for 8” pipe; and for gas service 17-ft for 2” to 40-ft for 8”. For high density polyethylene (HDPE) the spacing would vary from 7-ft for 4” pipe with SDR (diameter over thickness) of 11.0, to 16-ft for 24” pipe SDR 11.

In the simplest cases, the reactions, moments and deflections of pipe spans due to weight can be estimated using beam formulas. For example, in the cases illustrated in Figure 2-1, the reactions (R), moments (M_E at end, M_C at center and M_L under load) and sag (d) of pipe spans are:

Case (a):	R = wL/2	$M_C = wL^2/8$	$d = (5wL^4)/(384EI)$
Case (b):	R = wL/2	$M_E = wL^2/12$	$d = (wL^4)/(384EI)$
Case (c):	R = wL/2	$M_C = wL^2/9.3$	$d = (2.5wL^4)/(384EI)$
Case (d):	R = P	$M_E = Pb$	$d = Pb^2(3L-b)/(6EI)$
Case (e):	R = 5wL/8	$M_E = wL^2/8$	$d = wL^4/(185EI)$
Case (f):	R = Pb/L	$M_L = Pab/L$	$d = Pm_1m_2/(27EIL)$;with

$m_1 = ab(a+2b), m_2 = [3a(a+2b)]^{0.5}$

In these simplified representations, a straight pipe between supports acts as a beam, the supports and adjacent pipe spans act as end restraints, and the pipe span behaves as a beam with end conditions somewhere between simply supported and fixed. A reasonable approximation for the bending stress is

$$\sigma = M / Z = wL^2 / (10\ Z)$$

σ = bending stress, psi
M = approximate bending moment due to pipe weight, in-lb
w = weight of pipe, insulation and contents per unit length, lb/in
Z = section modulus, in^3 (Appendix A)

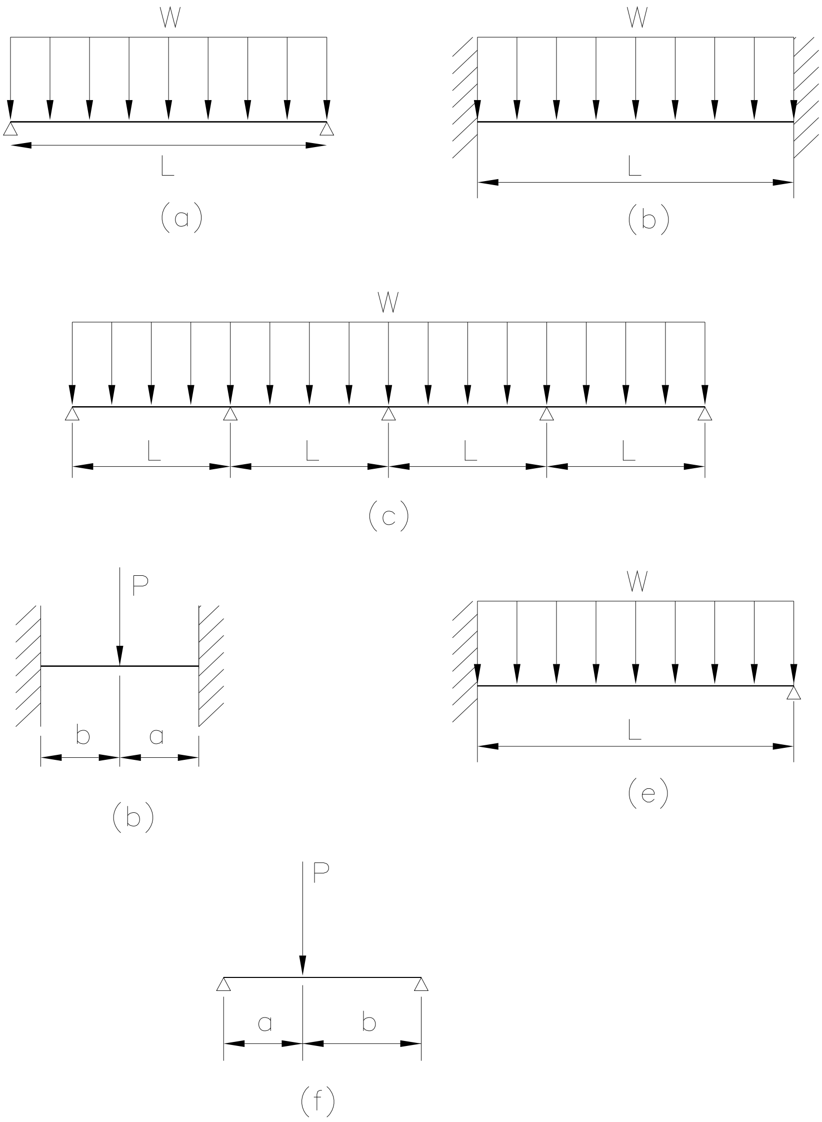


Figure 2-1 Simplified Pipe Spans

3. *Flexibility*

3.1 Layout For Flexibility

Changes in fluid or ambient temperature can have five effects on a piping system: (1) a global or flexibility effect in the form of movements and stresses as the pipe expands and contracts (2) a local effect in the form of local temperature gradients in the pipe wall as the temperature changes locally, for example when injecting cold water in a hot line, (3) at sufficiently high temperature, creep will take place accompanied by metallurgical changes, (4) changes in mechanical properties, with a loss of toughness at low temperature and a softening at high temperatures, and (5) changes in corrosion mechanisms or corrosion rate.

The global or flexibility effects that take place as a piping system or pipeline expands or contracts are: (1) movement of the line, (2) forces and moments along the pipe, (3) stresses in the pipe, (4) reactions on supports, and (5) reactions on equipment nozzles.

The first step in any good design and layout process is to understand the first of these five effects: movement of the line. The good designer immerses himself or herself into a three dimensional sketch (isometric) of the line, and tries to intuitively predict how the system will expand or contract as it is placed into service. This first step is often ignored by "analysts" whose understanding of flexibility analysis is to (1) make a model, (2) click on the "run" icon, and (3) check the last output line to read if "stress below code allowable"

Having a good feel for how the line should move, the designer then quantifies the magnitude of movement. Until the 1960's this was accomplished using thermal expansion charts and tables [Spielvogel]. As a tip of the hat to a bygone era, we illustrate how an expansion loop was designed. A vertical expansion loop has a width a and a height h in the middle of a span of total length L (L includes the width a), with an anchor at the both ends. The axial reaction force F_X at each anchor and the maximum bending stress σ_b in the loop are [Spielvogel, Grinnell]

$$F_X = k_X c \, I_p / L^2$$

$$\sigma_b = k_C c \, D / L$$

$$c = \Delta L E / 172{,}800$$

F_X = axial force, lb
c = tabulated expansion factor (for example c = 310 for mild steel at 300°F
I_p = moment of inertia of pipe cross section, in^4
L = total anchor to anchor length of loop, ft
D = pipe diameter, in
E = Young's modulus of material, psi
ΔL = expansion per 100-ft, in
k_X = tabulated force coefficient (for example k_X = 31.2 if L/a =3 and L/h=3)
k_b = tabulated stress coefficient (for example k_b = 37.4 if L/a = 3 and L/h=3)

Today, the designer uses piping analysis computer software to efficiently model the pipe, apply the temperatures and obtain movements and loads throughout the system. A first computerized flexibility analysis should be conducted, with the temperatures applied to the model and including only the restraining effect of weight supports. On the basis of this first analysis, the designer will gain a good understanding of the system flexibility. At this point, restraints, guides and anchors can be added. It is a fallacy to believe that anchors (support arrangement that fully restrain all six degrees of freedom of the pipe) must be avoided on hot lines. Rather, a few well-placed anchors help balance and direct pipe movements evenly in all directions In this process it may be necessary to add expansion loops, expansion joints or changes in direction to increase the system flexibility. This is an iterative process. Until an optimum configuration is achieved.

To verify that a design is sufficiently flexible, the ASME B31 power piping codes require that the longitudinal moment stress range due to thermal expansion from a cold to a hot condition be limited to a certain allowable valve Sa . This is written as

$$i\frac{M}{Z} < S_a$$

i = stress intensification factor
M = resultant moment range, in-lb
Z = pipe section modulus, in^3 (Appendix A)
S_A = allowable stress for thermal expansion, psi
S_A = f (1.25 $S_C + 0.25S_h)$
S_C = allowable stress at minimum (cold) metal temperature, psi
S_h = allowable stress at maximum (hot) metal temperature, psi
N = number of cold-hot temperature fluctuations.
f = stress range reduction factor
f = 6($N)^{-0.2} \le 1$,with the following values of “f” [ASME B31.3]

7,000 and fewer cycles, f = 1
7,000 to 14,000 cycles, f = 0.9
14,000 to 22,000 cycles, f = 0.8
22,000 to 45,000 cycles, f = 0.7
45,000 to 100,000 cycles, f = 0.6
100,000 to 200,000 cycles f = 0.5
200,000 to 700,000 cycles, f = 0.4
700,000 to 2,000,000 cycles f = 0.3

If the system undergoes a series of cycles N_i at different stress ranges S_i , then an equivalent number of cycles N must be calculated to select f, with

$$\mathrm{N} = \mathrm{N}_E + \sum_i r_i^5 N_i$$

N = equivalent number of cycles
N_E = number of cycles at the maximum stress range S_E
N_i = number of cycles at stress range S_i
r_i = S_i / S_E
S_E = maximum stress range, psi
S_i = stress range with N_i cycles, psi

The equation $iM / Z < S_a$ is also referred to as the flexibility stress equation or secondary stress equation. Let’s first explain what is the “moment range”. Consider a piping system or pipeline installed at an ambient temperature of 70°F. Assuming that the construction crew did not force the pipe into alignment during assembly, the bending moment at any point of the installed pipe is nearly zero. If the pipe is now put in service at say 150°F, the line expands and the bending moment at a given point, which we call point P, reaches say – 30 in-kips. The negative sign is a convention, which simply means that the line bends, for example, upward or left rather than downward or right. If the line is later put out-of service on a cold day, with an ambient temperature of say 40°F, the moment may reach for example + 11 in-kips at that same point. Note that in this case the contraction moment at 40°F (+11 in-kips) has an opposite sign to the expansion moment at 150°F (-30 in-kips). The moment range at point P is the largest absolute difference in moments from the cold to hot condition, or

$$M_p = \text{absolute value } (M_{p,150} - M_{p,40}) = 41 \text{ in-kips}$$

A change in moment magnitude occurs in each of three moment directions: two bending moments M_x, M_y and a torsional moment M_Z . The resultant moment range is the range of variation of the resultant moment M where

$$M = \sqrt{M_x^2 + M_Y^2 + M_Z^2}$$

The distribution of resultant moments M along a piping system, and therefore the stress are usually calculated by computer analysis of the piping system between anchor points. Today, charts and hand calculations are rarely used, and only in the simplest of system configuration.

3.2 Simplified Flexibility Analysis
ASME B31.3 gives an approximate formula for judging the adequacy

of the pipe flexibility between anchor points. It applies under the following conditions: (a) the pipe is of uniform diameter, and (b) the pipe is of very simple layout (a bend or two) with terminal anchors. This approximate formula is questionable and of little practical use in an age where piping analysis software is readily available. The formula is presented here for information

$$DY / [U^2 (R-1)^2 < 0.03$$

$$S_E = 33.3\ D\ Y\ S_A / [U^2 (R-1)^2]$$

D = pipe diameter, in
Y = thermal growth of end points, in
U = length of line from one anchor to the other, ft
R = ratio of developed pipe length / U
S_E = maximum stress range, psi
S_A = allowable expansion stress, psi

Another simplified equation can be applied to evaluate the effect of the thermal growth of a header on an attached branch pipe. We conservatively assume, as illustrated in Figure 3-1, that the branch is stiff, rigidly fixed at A (the second rigid restraint away from the header) and B (the header). If one more rigid restraint C is located between A and B, then if the header point B moves a distance d in the direction restrained by A and C, the reactions in the line are

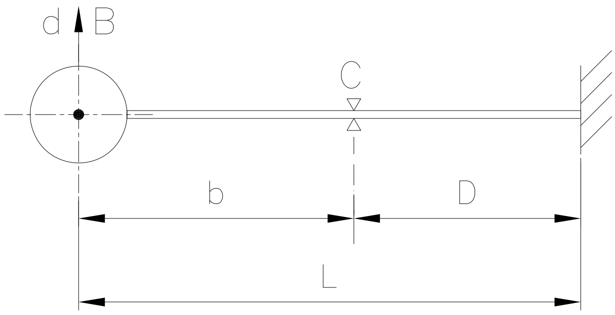


Figure 3-1 Evaluation of Header Movement

$$R_C = \frac{3EId}{L^3}\left(\frac{3L}{a} - 2\right)$$

$$M_B = \frac{3EId}{L^2}\left(\frac{3L-2a}{b^2} a + 2\right)$$

R_C = reaction load on support C, lb
E = Young's modulus of the pipe, psi
I =moment of inertia of the pipe cross section, in^4
d = movement of the header, in
M_B = bending moment at the header junction B, in-lb
L,a,b = lengths as indicated in Figure 3-1,in

If there is no support at C, and A and B are still built-in points

$$R_A = 12\ EId / L^3$$

$$M_A = 6\ EId / L^2$$

If there is no support at C, and A is built-in but B is simply supported

$$R_A = R_B = 3\ EId / L^3$$

$$M_A = 3\, EId / L^2$$

$$M_B = 0$$

Beam approximations are also useful to resolve field misalignment questions. When two ends of a piping system or pipeline are misaligned, the engineer is called upon to decide if one end can be pulled into alignment with the other, Figure 3-2. Given the pull distance d, we calculate the reactions R_A and R_C, and the stresses iM_A / Z, IM_B / Z, and iM_C / Z. The reactions are compared to the capacity of existing restraints at A and C, and the stresses are compared to a reasonably low limit for sustained loads, such as 5000 psi for steel at normal ambient temperature. For conservatism, it is also advisable to increase the calculated reactions and stresses by 1/3

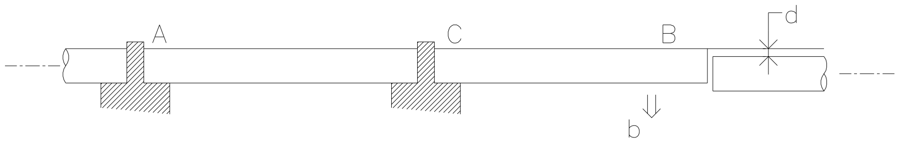


Figure 3-2 Field Misalignment of Joint

A few rules of good practice are in order when selecting a layout for a piping system operating at temperature: (a) Avoid stiff branched close to expanding headers. (b) A completely free pipe, although very flexible, is not necessarily best because the thermal growth of the pipe has to be guided and balanced, for example to avoid overloading nozzles. (c) Bends and elbows add flexibility to the piping system, reducing the bending moments.

4. Vibration

4.1 Root Cause

Trying to pinpoint the root cause of pipe vibration, let alone eliminate it, can be a vexing problem. A structured, systematic approach to the problem is required. We start by noting, as illustrated in Figure 4-1, that there are two possible causes for pipe vibration in service: mechanical or hydraulic. A mechanical induced vibration, Figure 4-1 block (2), is due to the mechanical vibration of a piece of equipment such as a pump or compressor which, in turn, causes the pipe to vibrate. The equipment vibration may be transmitted directly to the pipe through its nozzle attachment, Figure 4-1 block (3), or the equipment vibration may cause a structure or floor to vibrate, and the vibrating structure would then transmit the vibration to the pipe through the pipe supports, Figure 4-1 block (4). The pipe may just follow the equipment vibration, Figure 4-1 block (6) or, if the vibration frequency of the mechanical source f_{MS} is close to a mechanical natural frequency of the pipe f_{MP}, the pipe may amplify the vibration, Figure4-1 blocks (5) and (7). A hydraulic induced vibration, Figure 4-1 block (8),is due to continuous pressure pulses that cause the pipe to vibrate. The pressure pulses could be clearly periodical, Figure 4-1 block (9), or more random and turbulent, Figure 4-1 block (10). If the frequency of the pressure pulses, the hydraulic source frequency f_{HS} , is close to the acoustic frequency of the pipe cavity f_{AP}, Figure 4-1 block (11), the pipe will resonate and amplify the vibration, Figure 4-1 block (13). In this chapter, we will examine the logic behind Figure 4-1 in more detail to understand what causes pipe vibration in service, and how best to solve it.

One more basic point is in order: the vibratory motion of stiff pipe spans (spans with high natural frequencies, in the order of 50 Hz or more) is small, even when in resonance. In piping systems, large motion and therefore the real danger of fatigue failure due to vibration induced bending takes place most often at relatively low frequency, below about 50 Hz.

Finally, experience indicates that to minimize or eliminate piping vibration, it is best to rely on experience during the design stage, follow good construction practices during erection, and include pipe vibration monitoring as part of the preoperational or system startup test. Quantitative analysis at the design stage is complex and, with few exceptions (such as gas compressors [API 618], questionable

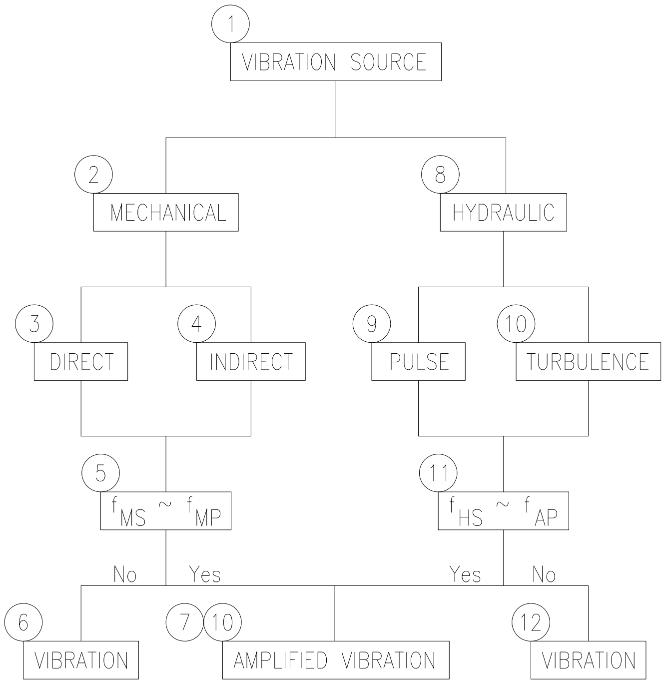


Figure 4-1 Systematic Approach to Pipe Vibration

Buried Pipe

The decision to bury a pipe or place it above ground depends on several factors: on one hand, a buried pipe (a) reduces plant congestion, (b) allows for the shortest route (fewer bends) from point to point, (c) avoids existing above ground obstructions, (d) is protected from ambient temperature changes, (e) is protected from wind loads, and (f) if buried deeply, is protected form surface traffic and activities. In certain cases, burying the pipe may be the only viable alternative.

On the other hand, a buried pipe (a) has unique corrosion challenges that may dictate the use of coating and cathodic protection, (b) requires more elaborate repairs, with the need to located the pipe, locate the leak, open the trench or resort to specialized trenchless repair techniques, (c) can be accidentally damaged by digging, (d) may leak for some time before the leak is detected, (e) requires careful trenching and backfill to avoid excessive soil settlement, and (f) has to be designed for soil and surface loads, which requires a good understanding of the soil condition and properties.

1. Soil loads

The simplest design rule for pipes installed in a trench with backfill, is to apply the prism formula, which states that the earth load on the pipe is equal to the weight of the soil prism right above the pipe, as shown in Figure 1-1.

$$P_v = \gamma H$$

P_v = earth load pressure on buried pipe, psi
γ = unit weight of backfill, lb/in^3
H = burial depth, in

If the pipe is below the water table, then the soil pressure is reduced by buoyancy and increased by the weight of water

$$P_v = \gamma H - 0.33 \frac{h}{H} \gamma H + \gamma_w h$$

h = height of water above pipe, in
γ_w = unit weight of water lb/in^3

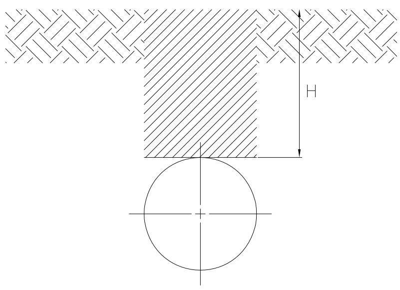


Figure 1-1 Soil Prism Above Pipe

If, instead of being placed in ditch with backfill, the pipe is tunneled through undisturbed soil, the earth pressure is lower by a factor 2c (H/D) where c is the soil cohesion [Moser]. Whether in a ditch or tunneled into place, the soil load on steel pressure on the pipe is 0.07 x 120 = 8.4 psi. Soil loads become important for large diameter thin pipes (large diameter / thickness ratio) as encountered in waterworks (water conduits made of corrugated sheet metal) and for materials with limited ductility, prone to fracture under external loads (concrete, cast iron).

2. *Surface loads*

Buried pipes crossing highways, runways, railroad tracks, construction sites, are exposed to loads due to the passage of heavy surface traffic. The pressure transmitted to the buried pipe by a surface load is [ALA,Moser, WRC]

$$P_P = 0.48 \frac{P_s}{H^2\left[1+(d/H)^2\right]^{2.5}}$$

P^2 = surface load, lb

d = offset distance from surface load to buried pipe, in

For example, an 180,000 lb surface load right above a pipe (d = 0) will cause a pressure of 6 psi if the pipe is buried 10 ft below ground. Under the effect of soil and external loads, the buried pipe will tend to ovalize, causing through wall bending stresses, with [ALA]

$$\sigma_b = 4E \frac{\Delta}{D}\frac{t}{D}$$

$$\frac{\Delta}{D} = \frac{0.15P}{\frac{EI}{R^3}+0.061E'}$$

σ_b = through-wall bending stress, psi

E = modulus of elasticity of pipe, psi

Δ = change in pipe diameter due to ovalization, in

D = pipe diameter, in

t = pipe wall thickness, in

EI = pipe wall stiffness, in-lb

E' = modulus of soil reaction, psi [AWWA C150]

For example, a downward pressure of 6 psi on a carbon steel pipe buried in poorly compacted soil will cause it to ovalize 0.8%, with a through-wall bending stress of 14.5 ksi. Where the surface load is both significant and repetitive, fatigue considerations may dictate a deeper soil cover. This subject has been extensively investigated for steel pipelines crossing highways and railroad tracks, and techniques have been developed to predict fatigue life and minimum depth of cover in this case [API 1102].

3. *Thermal expansion and contraction*

When the fluid temperature conveyed in a buried pipe differs from the soil temperature, the pipe will tend to contract or expand. In a straight pipe, fully restrained by the surrounding soil, unable to expand or contract, the temperature change will cause an axial stress [B31.4,B31.8,ALA]

$$\sigma_A = E\alpha(T_2 - T) - \upsilon\frac{PD}{2t}$$

σ_A = axial stress in a pipe fully constrained by the surrounding soil, psi

E = modulus of elasticity of pipe material, psi

α = coefficient of thermal expansion of pipe 1/^{0}F

T_2 = fluid temperature, ^{0}F

T_1 = burial installation temperature, ^{0}F

υ = Poisson ration of pipe material

P = internal pressure, psi

D = pipe diameter, in
t = pipe wall thickness, in

The situation is different if the buried pipe contains a bend, and is not assumed to be fully restrained by an infinitely stiff soil at the bend. In this case, the pipe will tend to flex around the bend, with the surrounding soil exerting a restraining force. The pipe acts as a beam on elastic foundation, as illustrated in Figure 7-1. The hand calculation of stresses is only possible in the simplest of configurations [ASME B31.1,ALA, WRC 425].

In most cases, it will be necessary to analyze the expansion or contraction around the bend using a pipe stress analysis program, with the soil modeled as spring elements around the pipe. The stiffness and spacing of soil springs depends on the soil and compaction properties [ALA, WRC 425, ASCE]. The bending stress in the pipe are calculated and compared to an allowable stress, such as defined in Appendix VII of ASME B31.1.

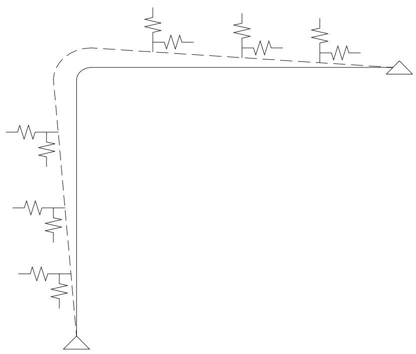


Figure 7-1 Partially Restrained Expansion at a Buried Pipe Bend

If the temperature rise in a pipeline becomes significant, the compressive stress in the line will increase and could cause the pipe to buckle up, what is referred to as upheaval buckling. The compressive force in the buried pipeline is

$$N = \sigma_A A_{pipe} = [E\alpha(T_2 - T_1) - \upsilon \frac{PD}{2t}]\pi Dt$$

The critical compressive buckling force in a perfectly straight pipeline is [Friedman, WRC 425]

$$N_{critical,perfect} = 2\sqrt{EIk_e}$$

$N_{critical,perfect}$ = compressive buckling force for a perfectly straight buried pipe, lb
E = pipe material Young modulus, psi
I = pipe cross section moment of inertia, in^4

The critical compressive buckling force in a real pipeline with an initial curvature is a fraction of

$$N_{critical,perfect} = \lambda N_{certical,perfect}$$

$N_{critical,perfect}$ = compressive buckling force for an initially deformed pipeline, lb
λ = fraction that depends on initial curvature of the pipeline

The uplift resistance of the soil can also be expressed in terms of force per linear foot of pipeline. In a cohesionless soil [Schaminee]

$$P = \gamma HD(1 + f_d \frac{H}{D})$$

γ = soil density, lb/in^3

H = buried depth, in

D = pipeline diameter, in

F_d = load factor 0.6 for gravel or rock dump, down to 0.15 for very loose soil

For cohesive soils

$$P = DC_u (1 + f_c \frac{H}{D}) \leq 5.14DC_u$$

C_u = shear strength of soil, psi

4. *Ground movement*

Ground movement (either a gradual settlement or spread, or a sudden failure due for example to a landslide, an earthquake or mining operations) could cause a buried pipe to fail by plastic tension or by compressive buckling. The assessment of ground movement consists of two parts: first, the prediction of the deformed pipe profile; second, the resulting stresses or strains in the deformed pipe. The first part, predicting the pipe profile, is not a simple proposition. The civil engineer must estimate the magnitude of movement and the distance over which it will take place. Given the soil deformation profile, the stresses or strains in the pipe can be estimated by computer analysis or hand calculation. A computer analysis will generally consist of an elastic-plastic model of the pipe, restrained by non-linear soil springs, with the ground movement imposed at the base of the springs [ALA,ASCE]. The stresses are obtained directly as output. An elastic analysis, in which the total stress in the pipe is kept below a fraction F_D of the material yield stress, can be accomplished by hand calculation. A pipe settlement X would be judged acceptable if it occurs over a distance at least equal to L, where [API 1117]

$$L = \sqrt{\frac{3.87x10^7 DX + 7.74x10^7 xX^2}{F_D S_Y - S_E}}$$

L = minimum required length (L=2L_1 in Figure 8-1), ft

D = outside pipe diameter, in

X = mid-span deflection, ft

F_D = design factor

S_Y = minimum yield stress of pipe material, psi

S_E = longitudinal stress in pipe prior to ground movement, psi

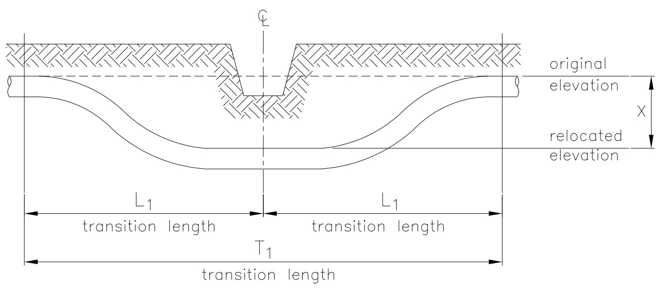


Figure 8-1 Mid-Span Deflection

The profile, along the pipe should be at least as gradual as that given by [API 1117]

$$X_a = \frac{16a^2 X(L-a)^2}{L^4}$$

X_a = vertical deflection, at a distance a, ft
a = distance along the trench from origin of deflection, ft

In addition to limiting the stress to $F_D S_Y$, the strain on the compressive side of the bent pipe should be less than the buckling strain [WRC 425]

$$\varepsilon_b = \frac{4\Delta D}{L^2} = \frac{D\kappa}{2} < 2.42\left(\frac{t}{D}\right)^{1.6}$$

ε_b = maximum compressive strain in bent pipe
Δ = maximum bow at mid-span, in
D = pipe outside diameter, in
L = length of pipe segment, in
κ = curvature of bent pipe (1/R where R is the radius of curvature), 1/in

5. Seismic

Earthquakes can fail buried pipes in one of two ways: (1) a large ground movement that fails the pipe by tension (particularly at corroded sections, poor weld joints and mechanical joints) or by compressive buckling, and (2) a large cyclic movement caused by the passage of the seismic wave. The effect of ground movement can be analyzed following the rules of Section 8 . It has been argued that wave passage alone could not fail modern (arc welded), well constructed (fabrication, NDE and hydrotest per ASME B31 code), and well maintained (little corrosion) steel pipe. Where wave passage must be analyzed, the upper bound of the strain in the pipe can be obtained by assuming that it is equal to the soil strain caused by wave passage [ALA, ASCE]

$$\varepsilon_a = \frac{V_g}{\alpha C_s}$$

ε_a = soil strain
V_g = peak ground velocity due to wave passage, ft/sec
α = factor 2 for shear waves, 1 for other seismic waves
C_s = apparent propagation velocity for seismic waves, 6560 ft/sec

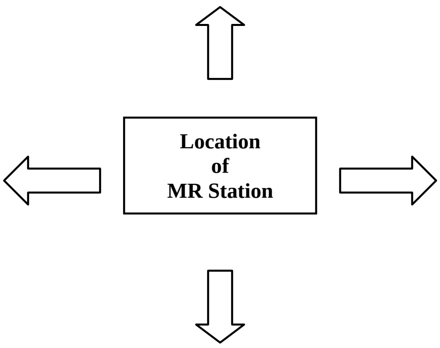
Where's the Gas Station ?

Hazardous Area

The gas stations have piping and the pressure control equipment. The standard gas stations equip with piping system, fitting and pressure control equipment. The safety restriction is launched by Department of Energy Business.(Thailand)

Specification of hazardous area uses the criterion of API 500 C (American Petroleum Institute Recommended Practice 500 C), Classification of Locations for Electrical Installations at Pipeline Transportation Facilities, classifies the M&R station as the hazardous area - Class 1 Division 2. Determination the area of M&R Station as following.

Level	*Distance in meter (feet)*	
1. Pressure 1900 kPa (275 PSIG) or less	3	(10)
2. Pressure above 1900 kPa (275 PSIG)	7.5	(25)



According to the specification of electric system, electric line must be equipped with metal pipe which have the E.I.T. Standard 2001-30 or the standard of NEC (National Electric Code, ANSI 70 or UL (America). The electric equipment must be covered by the box or closed with explosion proof and must be grounding.

Reference

- API 500 C (American Petroleum Institute Recommended Practice 500 C)
- NEC (National Electric Code, ANSI 70)
- มาตรฐาน ว.ส.ท. E.I.T. Standard 2001-30