Piping Design in Power Plants

Piping design in high temperature and high pressure service has become complicated in recent years. Simple approach to design aspects has been given in three successive parts.

Part Two

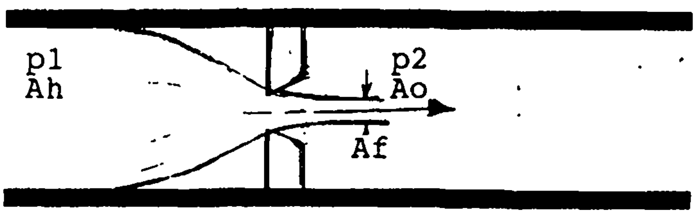
Sankar Chakrabarti†
Mannesmann Seiffert GmbH, Berlin
Germany

Part one of the series deals with the functional requirements and general design consideration for piping in power plants. The present part deals with the aspect of measurement of flow through pipes and design consideration for insulation and pipe strength as per different codes of practice.

2.6 Orifice Plate

The flow of fluid through a piping system can be measured by an orifice plate. A plate with a sharp edged hole in its centre is introduced and secured in a pipe so that the fluid passes through the hole only. Measurement of pressure drops across the plate provides a means of determination of flow rate. The orifice plate, again can be used to decrease the pressure of the flowing medium (Figure 2.6)

† S Chakrabarti—a graduate engineer from Jadavpur University, Calcutta— is associated with piping design and project management for more than twentyfive years. He is a professional engineer of the State of California, USA. Presently, he is head of Design and Calculation department of MSE, Berlin, Germany.



$$Ao = \frac{\pi}{4} * do^2 \quad mm^2$$

Fig. 2.6 : Flow through a orifice plate

2.6.1 Fluid

The required area of orifice is given as :

$$Ao = \frac{m}{miu * \sqrt{2*(p1-p2)/v1}} \quad m^2 \qquad [2.12]$$

where miu is the orifice factor, a correction factor for approach velocity which is defined as :

miu = Af/Ao where Af is the area of the fluid jet after the orifice.

Depending on the form of orifice opening, the fluid jet takes its shape. The correction factor for this reduction of diameter given by miu can be taken from table 2.6.1 and intermediate values can be interpolated.

Ao/Ah	Sharp	Broken	Round
≤ 0.1	0.62	0.70	0.90
0.2	0.63	0.71	0.90
0.4	0.66	0.73	0.91
0.6	0.71	0.77	0.92
0.8	0.81	0.85	0.96
1.0	1.00	1.00	1.00

Table : 2.6.1 Corrector Factor miu

If the orifice plate consists of multiple holes n of diameter do :

$$Ao = Ah*(do/di)^2 \quad mm^2 \qquad [2.13]$$

2.6.2 Steam or Gas

The orifice area can be calculated as :

$$Ao = \frac{m^* \sqrt{p1^*v1}}{miu^*p1^*\phi} \quad m\text{^}2 \quad [2.14]$$

The factor ϕ depends on the steam or gas condition and can be calculated as :

$$\phi = \sqrt{\left(\frac{2^*k}{k-1}\right) * \left(\frac{p^2}{pl}\right)^{2/k} - \left(\frac{p^2}{pl}\right)^{(k+1)/k}} \quad [2.15]$$

where k = Cp / Cv
= 1.4 for air
= 1.3 for superheated steam
= 1.135 for saturated steam
= 1.035 + 0.1*x for wet steam, for x refer below

Depending on the condition of the medium, the critical velocity is reached for a maximum ratio of p2/p1, and no further reduction of pressure or increase of flow is possible. These critical pressures are dependant on the steam or gas temperature and pressure.

$$pc = p1 * \{2/ (k+1)\}^{k/(k+1)} \quad N/m\text{^}2 \quad [2.16]$$

Some of the common critical values are given below :

Air pc	= 0.530 *p1
Superheated steam	= 0.546 *p1
Saturated steam	= 0.5775*p1
Wet steam	= (0.5982 – 0.0207*X)*p1 for 0.4 ≤ X ≤ 1.0
	= (0.5989 – 0.0225*X) *p1 for 0.0 ≤ X ≤ 0.4

The dryness fraction X = (h–h') / r

h	=	total heat of steam	kJ/kg
h'	=	sensible heat of steam	kJ/kg
r	=	latent heat of steam for the given temperature and pressure	kJ / kg

Substituting the critical pressure value in the equation ϕ

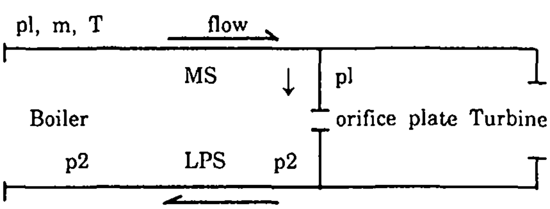
$$\phi max = \sqrt{\left(\frac{2^*k}{k-1}\right) * \left(\frac{2}{k+1}\right)^{2/(k-1)} - \left(\frac{2}{k+1}\right)^{(k+1)/(k-1)}} \quad [2.17]$$

ϕmax = 0.682 for air
= 0.669 for superheated steam
= 0.636 for saturated steam

The discharge coefficient is a function of jet size as well as viscous effect. The major disadvantage of flow measurement by orifice plate is the sizable pressure loss incurred because of the flow separation downstream of the plate. On the other hand, when required, the pressure can be substantially reduced by this arrangement.

2.6.3 Example of Orifice

In steam blowing operation, the pressure of MS is introduced back in the LPS line and blow out near the boiler, while the steam produced in LPS does not reach the required pressure to create disturbance required in steam purging. So the steam to the LPS requires pressure reduction which is simply done by orifice.



$$p1 = 18.8 \text{ bar}$$

$$T = 323°C$$

$$v1 = 0.1404 \text{ m}^3/\text{kg}$$

$$p2 = 13.0 \text{ bar (desired)}$$

$$m = 29.0 \text{ kg/s}$$

pipe dimensions : 219.1 da x 8.0 mm

$$di = 213.1 \text{ mm}$$

$$k = 1.29$$

The critical pressure for upstream condition :

pc = 18.8*(2/2.29) = 10.29 bar

p2 must be > pc p2 taken = 13.0 bar

ϕ = 0.632

Without any consideration of orifice correction factor (miu = 1) :

$$Aos = \frac{29 * \sqrt{1.88e{+}6 * 0.1404}}{1.88e{+}6 * 0.632}$$

= 0.01254 m^2 = 12540 mm^2

Ah = 0.7854*213.1^2 = 35666 mm^2

With Ao = Aos for Ao/Ah = 0.35 from table 2.6.1 for broken edge orifice :

miu = 0.725

Substituting and interpolating, miu = 0.75

Therefore,
Ao = 12540/0.75 = 17297 mm^2
do = 145.9 mm

The thickness of the orifice plate can be calculated considering it as a ring loaded uniformly.

2.7 Heat Transfer and Insulation

The pressure of the flowing medium decreases due to friction loss in the piping as described before. The medium also loses heat through the piping material and its insulation to the environment. Since the piping system in a power plant transfers superheated steam from one equipment to the other, such heat transfer through the piping and insulation material is loss of energy which should be minimized. In case of very high temperature piping vacuum insulation, double insulation or similar methods are adopted. Since the allowable stress of pipe material decreases rapidly at high temperature, inside insulation is recommended for very high temperature steam flow.

2.7.1 *Insulation*

The main objective of insulation is to serve one or combination of following aspects :

- withstand the medium temperature
- minimise undesirable temperature loss or maintain the desirable temperature at the end
- save fuel wastage
- provide comfortable surface temperature for user
- possess high insulation efficiency which is described as the ratio of difference of heat loss without insulation to the total heat loss with insulation.
- easily workable and available.
- economical
- moderate weight, normally 100-120 kg/m^3
- stable or resist deterioration over working life
- fire proof

There are various types of insulation available in the market flake, fibrous, granular, cellular, reflective etc. They may be bonded or unbonded. The insulation is selected in consultation with the customer and has not been described here.

2.7.2 Heat flow through plain wall

The heat flow through plain wall is given by :

$$q = \frac{Ti - Ta}{1/\alpha i + t1/\tau 1 + t2/\tau 2 + \ldots + 1/\alpha a} \quad \text{W/m} \quad [2.18]$$

The thermal conductivity τ of a material is dependant on the average temperature of the layer. For high temperature insulation, the normal practice is to use several layers of insulation and the heat transfer can be calculated in steps. The example at the end of this article illustrates the use of multi-layered insulation of a piping.

The convection part of coefficient of heat transfer can be calculated from the relation:

$$\alpha k = 1.4 * \sqrt[4]{(Ts - Ta)/h}$$
for h < 0.5 m W/(m^2*K) [2.19A]

$$= 1.5 * \sqrt[4]{(Ts - Ta)}$$
> 0.5 m W/ (m^2*K) [2.19B]

where h is the height of plane wall in meters.

If the object is subjected to wind of velocity w, the value of k can be calculated as :

$\alpha k = 6.2 + 4.2*w$
for $w < 5.0$ m/s W/(m^2*K) [2.19C]

$= 7.6*w^{0.8}$
for $w > 5.0$ m/s W/(m^2*K) [2.19D]

The radiation part of the coefficient of heat transfer can be obtained from the relation :

$\alpha c = a*C$ W/(m^2*K) [2.20]

where,

$$a = \frac{\{(Ts+273)/100\}^4 - \{(Ta+273)/100\}^4}{Ts - Ta}, K^3$$

C = coefficient of heat radiation for the outside surface of the piping or insulation W/(m^2*K^4)

αi or $\alpha a = \alpha k + \alpha c$ W/(m^2*K) [2.21]

Few standard values of C are given below:

Aluminium	polished	0.30
	unfinished	0.41
Lead	grey oxidized	1.62
Copper	polished	0.23
	black oxidized	4.49
Steel	oxidized	4.63
	bright oxidized	4.72
Glass	plain	5.41
Rubber	soft	4.95
Paper		5.39

Table 2.7.2 Values of coefficient of heat radiation

2.7.3 Heat flow through piping

Similar to the equation of heat flow through plain wall, the heat flow through an insulated pipe :

$$q = \frac{\pi*(Ti - Ta)}{\frac{1}{\alpha i*di} + \frac{1}{2*\tau 1}*\ln\frac{d1}{di} + \frac{1}{2*\tau 2}*\ln\frac{d2}{d1} + \ldots + \frac{1}{\alpha a*da}} \text{ W/m} \quad [2.22]$$

The distribution of temperature through a multilayered insulated pipe is shown in the diagram (Figure 2.7.3) in next page.

The different thermal constants can be calculated as per the approximate equations given below (for outside and inside surface):

$\alpha ka = 1.35*\sqrt[4]{(Ts - Ta)/da}$
W/(m^2* K) [2.23A]

For piping subjected to wind :

$\alpha ka = 4.13*w^{0.8}/da^{0.2}$
W/(m^2*K) [2.23B]

In case of steam at high velocity in the piping system, the inside heat transfer coefficient in convection can be obtained as:

$\alpha ki = 0.04*Pe^{0.75}*\tau m\ /di$
W/(m^2*K) [2.24A]

or,

$\alpha ki = (4.0 + 0.003*Ti)*wn^{0.75}/da^{0.25}$
W/(m^2*K) [2.24B]

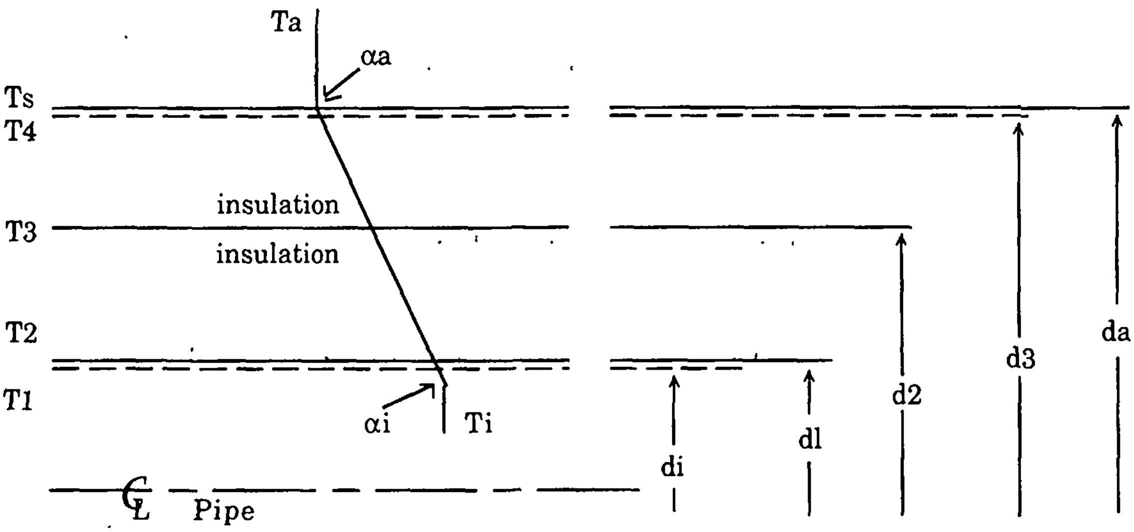


Figure 2.7.3 : Distribution of temperature through pipe and insulation

where Pe is Peclet's number depending on the condition of the fluid flowing through the pipe and is given by: Pe = w*di*Cp / (τm*v) —

where τm is the thermal conductivity of the medium

wn = w* (Ta+273)/(Ti+273) *pa/p m/s

The coefficient of heat transfer in radiation can be calculated as per the equation 2.20.

2.7.4 Axial Decrease of Temperature

The decrease of temperature in the axial direction for a flowing medium in a piping system due to radial loss of heat through the pipe wall and insulation can be calculated as :

$$dT = 3.6*q*1 / (m*Cp) \quad °C \qquad [2.25]$$

where q is the radial heat flow as given by equation 2.22.

2.7.5 Decrease of Temperature in a Vessel

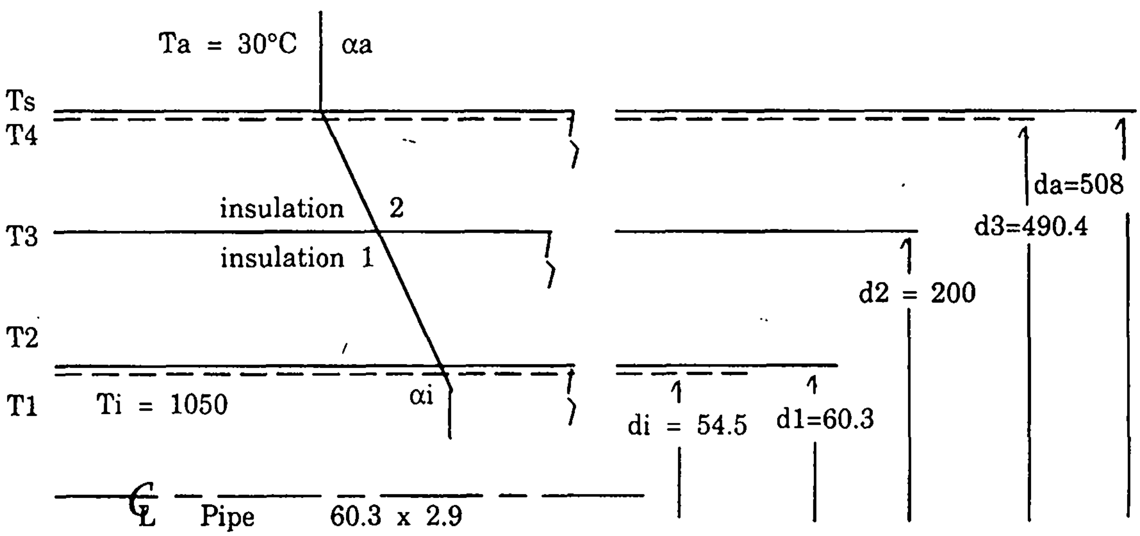
The decrease of temperature of the medium in a stagnant vessel for a given time can be calculated as:

$$dT = 3.6*q*A*t / (m*Cp) \quad °C \qquad [2.26]$$

where	A =	surface area of the insulated vessel	in	m^2
	t =	time	in	hours

2.7.6 Example of Heat Loss through multilayered Insulation

In a multilayered insulated piping system helium flows at a very high temperature. The radial heat flow, temperatures at every layer and axial decrease of temperature due to heat loss in the radial direction is to be calculated.



medium pipe	da 60.3 mm x 2.9 mm wall thickness		
Material	Incoloy 800H	τm = 25.1 W/(m*k)	for Tav = 1050°C
Insulation 1	Safil	τ1 = 0.5 W / (m*K)	for Tav = 850°C
Intermediate pipe	200.0 x 1.0 mm		
Insulation 2	kao-wool	τ2 = 0.2 W / (M*K)	for Tav = 390°C
Outside pipe	508.0 x 8.8 mm		
Material	15Mo3	τa = 52.0 W/ (M*K)	for Ta = 30°C

di = 54.5 mm	d2 = 200.0 mm	da = 508.0 mm
d1 = 60.3 mm	d3 = 490.4 mm	

Medium : Helium	m	=	90 0	kg/h	=	0.025 kg/s
	v	=	0.4602	m^3/kg		
	τm	=	0.4295	W/mK		for 1000°C
	Cp	=	5193.0	J/kgK		

Calculation :

Ah	=	0.7854*0.0545^2	=	0.002333	m^2
w	=	90.0 0.4602/(60*60*0.002333)	=	4 93	m/s
Pe	=	4.93*0 0545*5193/(0.4602*0.4295)	=	7059.0	–
αi	=	0 04 7059^0.75*0 4295/0 0545	=	316.7	W/(m^2*K)

Taking the outside surface temperature Tx = 80.0 °C

αk = 1.4 * $\sqrt{(80-30) / 0.508}$ = 4 25 W/(m^2*K)

$$\alpha c = 4.72 * [(\frac{80+273}{100})^4 - (\frac{30+273}{100})^4] / (80 - 30)$$

= 6.72 W/(m^2 K)

αa = αk + αc

αa = 4.25 + 6.72 = 10 97 W/(m^2*K)

1/(αi*di)	=	1/(316.7*0.0545)	=	0.0579
1/(αa*da)	=	1/(10.95*0 508)	=	0.1798
1/(2*τm) ln(d1/di)	=	1/(2*25.1)*ln(0.0603/0 0545)	=	0 0020
1/(2*τ1)*ln(d2/d1)	=	1/(2*0.5)*ln(0.2 / 0.0603)	=	1.1990
1/(2 τ2)*ln(d3/d2)	=	1/(2*0.2) ln(0 4904/0.2)	=	2.2423
1/(2 τa) ln(da/d3)	=	1/(2 52) ln(0.508/0 4904)	=	0.0003

The radial heat flow .

q = π*(1050 – 30) / (0 0579+0.002+1 199+2 2423+0.0003+0.1798) = 870.0 W/m

qa = π*αa da (Ts – Ta) = p 10 97*0 508 (79.5 – 30) = 866.0 W/m

q1 = π*(Ti – T2) / (0.0579+0 002+1 199) = π*(1050 – 700)/1.259 = 872.0 W/m

q2 = π (T2 – Ts) / (2.242 + 0.0003) = π*(700 – 79.5) / 2 243 = 869.0 W/m

So approximately, $q = q1 = q2 = qa$

Change of temperature in the axial direction :

The radial heat flow $q = qr = 870.0$ W/m

The loss of temperature in the axial direction for first meter :

$$dT = 3.6*q*1/(m*Cp)$$

$$= 3.6*870.0*1.0/(0.025*5193)$$

$$= 24.1 \text{ K/m}$$

Temperature distribution calculated :

T1			= 1050	°C
T2	=	T3	= 700	°C
T4	=	Ts	= 79.5	°C
Ta			= 30	°C

3. Pipe strength

3.1 Nomenclatures

C1/C1p	thickness tolerance in mm or %	mm, %
C2i/C2a	corrosion allowance inside or outside	mm
da/di	pipe or component outside or inside diameter	mm
E	Welding efficiency	–
P	design internal pressure	MPa
r	radius of pipe bend	mm
rm	mean radius of main pipe or header	mm
rw	radius of weld	mm

3.1 Nomenclatures (Continued from previous page)

Rm	specified min. tensile stress at room temp.	MPa
RmT	average tensile stress at temperature	MPa
Rp	specified min. yield stress at room temp.	MPa
RpT	average yield stress at temp. or 0.2% proof	MPa
Rp1T	average 1% yield stress at temperature	MPa
Rme5	mean stress at temp. rupture per 100000 hrs	MPa
Rmme5	minimum stress at temp. rupture per 100000 hrs when values are not available = 0.8 * Rme5	MPa
Rm2e5	average stress at temp. rupture per 200000 hrs	MPa
Rle5	average stress at temp. creep 1%, 100000 hrs	MPa
Rle3	average stress at temp. creep 0.01%, 1000 hrs	MPa
Rmz	mean stress at temp. rupture per time z	MPa
S	design stress intensity at design temperature	MPa
Sv	actual stress	MPa
tm	required calculated wall thickness without allowance	mm
t	required wall thickness with allowance	mm
T	design temperature	°C
z	number of hours or time under consideration	hr
b	included angle of reducer, diffuser, angle between header and breach	degree

suffix : b for branch, h for header, p for connecting pipe and r for reinforcement

The codes referred to are :

ASME American Society of Mechanical Engineers, Boiler and Pressure Vessel Codes

- III / NB Nuclear Class 1 components
- NC Nuclear Class 2 components
- VIII / 1 Pressure Vessels Division 1, design by rules
- 2 Pressure Vessels Division 2, design by analysis

B31.1	Power Piping	
B31.3	Process Piping	
BS	806	British Standard, Specification for design and construction of ferrous piping installations for and in connection with land boilers
TRD	300, 301	Technical Rules for steam boilers, Calc. of boiler strength, Cylindrical Shells etc.
IBR		Indian Boiler Regulations

3.2 Requirements

The strength design is prerequisite for piping design. This is performed to demonstrate :

- pressure integrity
- flexibility for thermal expansion
- pipe system weight
- withstanding capacity to dynamic loads
- stress concentration
- fatigue

3.3 Selection of Materials

Based on the temperature, pressure and requirement in material specification the piping material is selected and depending on the allowable maximum steam and water velocity, noise level, closing characteristics of valves, pump characteristics, flow requirements and pressure loss the diameters of the pipings are worked out.

Some of the criteria for material selections are :

- rules and standards specified
- design data and safety conditions
- cost considerations
- availability
- delivery period
- feasibility
- weight and handling facilities
- welding facility
- test procedure
- system considerations
- customer's desire

Materials commonly used for piping components in a thermal power plant are carbon steel, carbon molybdenum steel and chromemolybdenum steel.

3.4 Design Stress Intensity

To design the strength of a piping system the first requirement is the allowable strength of the material with which the pipe is manufactured. The most common rules and standards for design are ASME Codes which guide the designer throughout the period of design and construction of the piping system. The dimensioning or basic design may also be made using TRD, FDBR, IBR, BS, SNCT, GOST or any other standard or code as per the requirement of the country of power plant location. Since different codes have their own theory of design, wall thickness may vary.

For design of wall thickness a factor of safety is stipulated by the codes against yield of material to determine the basic design stress intensity. Although the normal trend is to have a 1.5 safety margin against yield strength at temperature, at higher temperature where no definite yield strength value of material is known, the creep or rupture stress values with appropriate safety factors are used. The ASME Codes use the tensile strength values for determination of basic design stress for strength calculation. Since every country has got their own experience with the materials normally available, it is better to use the safety factor and the material data of a country together with its testing methods. The factors of safety for calculation of basic design intensity values S in few Codes are given in table 3.4. The basic design stress intensity 'S' of any code is the minimum stress intensity calculated considering all the factors of safety against different stress values.

The pipe thickness calculated with these design stress intensities does not necessarily mean that the design is safe for the number of hours of operation taken into account in working out the design stress intensity value. It only stipulates the allowable stress required for the code. The code, however, takes into account of long time safe service.

The design stress intensity value as stipulated from different design codes shows that the stress failure theory of the codes are different. Again, whereas ASME NB/NC is applicable for relatively lower temperature range, all other codes referred above consider higher temperature and stress values at 100,000 to 200,000 hours rupture.

BS 806, Appendix B establishes the allowable stress for time dependant case as : S – Rmz/1.3.

TRD 300 also uses 200,000 hours values as: S = 0.8*Rm2e5

If the stress values for the required number of operating hours are not given in any code, the values corresponding to the required

Code	F/A	Rm	RmT	Rp	RpT	Rle3	Rme5	Rmme5	Rle5	Ref./Art.
ASME NB a)	F	3	2.73	1.5	1.5	-	-	-	-	App. III
	A	3	2.73	1.5	1.1b)	-	-	-	-	1132/2110
										and
ASME NC a)	F	4	3.64	1.5	1.5	1	1.5	1.25	-	III-3216
	A	4	3.64	1.5	1.1b)	1	1.5	1.25	-	Sec.II - App. 1 & 2
ASME B31.1	F	4	3.64	1.5	1.5	1	1.5	1.25	-	102.3
	A	4	3.64	1.5	1.1b)	1	1.5	1.25	-	VIII/1 Ap.P
ASME B.31.3	F	3	3	1.5	1.5	1	1.5	1.25	-	302.3.2(d)
	A	3	3	1.5	1.1b)	1	1.5	1.25	-	
BS 806	F	2.35	-	-	1.5d)	-	1.3j)	-	-	App. B
	A	2.5	-	-	1.5e, g)		1.3	-	-	
IBR	C	2.7	-	-	1.5	-	-	-	-	350 h)
	M	-	-	-	1.5g)	-	1.5	1.13	1	
TRD 300	F	2.4	-	-	1.5	-	-	1.0f)	-	9.2 & 9.4
	A	2.4			1.5g)			1.0f)		k)

a) not valid for creep temperatures

b) 1.1 only for austenitic steel, but for flange joints and other components where high deformation is not accepted, the safety factor is 1.5

d) 1.6 when elevated temperature values are not known

e) 1.5 upto 50°C, 1.35 above 150°C and 1.45 if elevated temperature values not available.

f) minimum stress for rupture at the end of 200,000 hours. In absence of 200,000 hours values, a factor of safety of 1.5 has to be taken against 100,000 hours average values.

g) for RpT / Rm < = 0.5, Rp1% for austenitic steel instead of Rp. 2%. Taken also for BS 806 and IBR.

h) only for seamless pipes, for stainless steel same factors as for carbon steel may be taken.

j) depending on the number of hours to be considered Rme5 or any other value as specified

k) for material with elongation > 15%

F Feritic steel — A Austenitic steel

C Carbon steel below 454°C — M Mo-steel above 454°C

Table 3.4 : Codes, corresponding stress values and factor of safety

number of hours may be obtained from the slope of materials used for this service or the time dependent stress mav be calculated from Larsen-Miller method.

Example :

Material : x20CrNiV121 DIN 17175 stress values are given in the standard upto for 650°C and 200,000 hours of service. Let us calculate the stress values for 225,000 hours.

1. Slope : The material shows a reduction of 13 - 22% in 100,000 hours to 200,000 hours values. Same reduction slope may be taken for further service hours (calc. : slope)

2. Larson - Miller : P = T * {C + 1g(z)}

   The Larson-Miller factor P is calculated from the time z and a material constant of C starting from 20 for high allow steel (calc. LM)

   For x20Cr Mo V121, C = 35

3.5 Straight Pipe

The minimum thickness of pipe wall required for design internal pressure and temperature can be calculated according to various Codes and Standards. Considering the nomenclatures given in para 3.1, some of the design rules for establishing the wall thickness for straight pipes are given as follows.

T (°C)	T (°K)	z(hours)	P (facor)	RmZ (MPa)	
540	813	10,000	31707	213.0	
		100,000	32520	147.0	
		200,000	32765	128.0	
	(calculated)	225,000	32806	124.8	LM
				123.3	slope
545	818	10,000	31902	202.5	
		100,000	32720	137.5	
		200,000	32966	119.5	
	(calculated)	225,000	33008	116.4	LM
				115.0	slope
550	823	10,000	32097	192.0	
		100,000	32920	128.0	
		200,000	33168	111.0	
	(calculated)	225,000	33210	108.1	LM
				106.8	slope

3.5.1 ASME III / NB-3641.1 and NC-3641.1 :

$$tm = \frac{p*da}{2*(S+P*y)} \qquad [3.01A]$$

y is the temperature coefficient and given in para 3.5.5

$$tm = \frac{p*di}{2*[S - P*(1-y)} \qquad [3.02A]$$

3.5.2 ASME B31.1 Art. 104.1.2 and B31.3 Art. 304.1.2 :

$$tm = \frac{p*da}{2*(S*E + P*y)} \qquad [3.01B)$$

$$tm = \frac{p*di}{2*[S*E - P*(1 - y) \}} \qquad [3.02B]$$

The ASME formulae are basically same in all ASME Codes mentioned above, only welding factor is considered in the allowable stress in B31.1 and B31.3. In nuclear power plant the welding factor is unity.

3.5.3 IBR 350, AD B0 Art.5 and BS 806 Art.4.2 :

$$tm = \frac{P*da}{2*S*E + P} \qquad [3.01C]$$

$$tm = \frac{P*di}{2*S*E - P} \qquad [3.02C]$$

3.5.4 TRD 301 Art. 5.1.1. :

$$tm = \frac{P*da}{(2*S - P) *E + 2*P} \qquad [3.01D]$$

$$tm = \frac{p*di}{(2*S - P)*E} \qquad [3.02D]$$

The ASME Code establishes that the membrane stress produced by internal pressure in the wall of a vessel should not be allowed to exceed one fourth or one third of the ultimate strength of the material at temperature. Actually the lack of any sound technical basis for developing the allowable membrane stress in the wall is reflected to the fact that every country has got its own way of establishing the allowable value. Figure 3.5.6 shows a striking comparison between the wall thicknesses calculated on the basis of different codes.

3.5.5 Temperature Coefficient : y

The temperature coefficient as used in the ASME Code B 31.1, Table 104.1.2(A) can be read off or interpolated between 28°C and the desired temperature from the values given in table 3.5.5.

3.5.6 Wall Thickness as per Different Codes

The wall thickness requirements of

Temperature °C	<482	510	538	566	593	621	649	>677
Ferrite	0.4	0.5	0.7	0.7	0.7	0.7	0.7	0.7
Austenite	0.4	0.4	0.4	0.4	0.5	0.7	0.7	0.7
Nickel Alloy	0.4	0.4	0.4	0.4	0.4	0.4	0.5	0.7

For Class 1 and Class 2 piping, y=0.4, since the design rules given are meant for ferrous material upto 373°C and for austenitic steel upto 427°C.

For pipe with tm>=da/6, the value of y for less than 482°C temperature can be taken as :
$y = di/(da + di)$

Table 3.5.5 : Temperature Coefficient for Material

straight pipes calculated as per different codes are shown in Figure 3.5.6. The welding efficiency has been taken as unity.

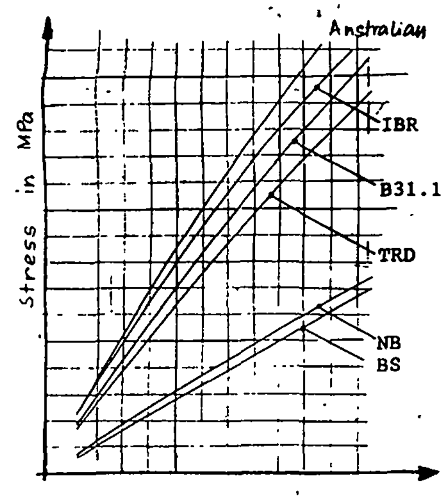


Fig 3.5.6 Trends of Wall Thickness of Straight pipes according to Different Codes

It can be seen that for the same design parameter, IBR requires the thickest pipes and BS the thinnest. The requirement of thin pipes as per NB can be explained as extensive testing requirement of Class 1 pipes and therefore the knowledge of mechanical and technological behaviour of the material used.

3.5.7 Allowances

The corrosion, erosion or manufacturing tolerances and allowances must be added to obtain the required wall thickness according to the code of design.

$$t = (tm + C2i + C2a) * (1 + C1P/100) + C1 \quad \text{mm} \qquad [3.03]$$

IBR recommands a minimum corrosion allowance C2i = 0.75 mm irrespective of material and mode of use i.e. t = tm + 0.75.

The corrosion allowance in other codes are not specified. As per temperature, pressure, velocity and composition of the flowing medium it is to be taken from the specification. For steam lines of power plants the internal corrosion is normally taken as zero.

Carbon and alloy steel pipes ordered with outside / inside diameter :

Diameter		DIN 17175 (std wall)	ASME SA530	IBR 345	BS 3602 / 3604 t/d(hot fin.)
da	<= 130mm	+15.0 − 10.0	±12.5	+no limitations	<=3% ± 15.0
	<=320 mm	± 12.5			<=10% ± 12.5 (DN 150)
	> 320 mm	+ 15.0 − 12.5		− 12.5 seamless or welded	>=10% ± 10.0
	*)				
di	<= 200 mm		+ 3.2	no	<=7.5% ± 15.0
	>= 700 mm	+ 15.0 − 12.5	forged & bored	value	<=15% ± 12.5 > 15% ± 10.0

*) pipes with inside diameter are ordered with minimum wall thickness.

Stainless steel pipes ordered with outside diameters :

Diameter	DIN 2464/1	ASME SA530	IBR 52	BS 3605
Cold formed (T3) da<= DN200	+10.0% + 0.2 mm	±12.5	cold drawn +10.0 − 5.0	t/da
hot formed (T1) da <= 610.0 mm	+ 15.0% + 0.6 mm		hot drawn + 15.0 − 5.0 (>DN50)	same as above

Table : 3.5.7 Tolerances in Percentage as per Different Codes

The manufacturing tolerance of wall thickness depends on the process of manufacture and given in piping standards. If the pipes are not machined after manufacture, the tolerance on wall thickness is given in percentage. The normally used tolerances are given in table 3.5.7 above.

3.6 Schedule Number

To standardize the pipe thickness for pressure and temperature, a schedule system has been created to facilitate the designer in selecting the pipe sizes. The preliminary design of wall thickness may be made with schedule number, which is defined as :

$$SN = 1000*P / (S*E) \qquad [3.04]$$

Knowing the internal design pressure and allowable stress for the pipe material at design temperature and welding efficiency, the pipe schedule may be calculated and next higher schedule available selected.

	da(mm)	tm(mm)	t(mm)	t, SN80(mm)	t,SN100(mm)
DN 400	406.4	17.6	20.1	21.44	26.19
DN 500	508.0	22.1	25.3	26.19	32.54
DN 600	610.0	26.5	30.3	30.96	38.89
As per Schedule system : SN = 1000*9.3/103.4 = 90 for all pipes					
Therefore, as shown above SN100 can be safely taken and as the calculation shows, SN80 is *just sufficient.*					

Examples :

P = 9.3 MPa, S*E = 103.4 MPa,

T = 400 °C, Material P22 y = 0.4

Manufacturing tolerance 12.5%

3.7 Pipe Class

Similar to the piping schedule, the flanges or fittings can be selected from the pressure rating or Class as :

$$CL = 8750*P / (S*E) \qquad [3.05]$$

The class of a fitting guarantees the temperature and pressure it can withstand for the material selected.

Pressure retaining wall thickness for forged fittings like elbows, reducers, tees, welding neck flanges etc. are not required to be calculated if these are taken from relevant recognised standards. The calculated thickness for the straight pipes or the schedule number should be specified for their connecting ends. For flanges, flanged fittings, valves and certain socket welding fittings, in place of wall thickness, the ability to withstand pressure / temperature is ascertained from the rating or class established by recognised standards to which their design conform. ANSI B16.5 (for flanges and flanged fittings) and ASME B16.34 (for flanged valves) have ratings class as follows : 150, 300, 400, 600, 900, 1500 and 2500 pounds per square inch.

But welding end valves either standard or special class may also be assigned intermediate rating, but such ratings normally find justifications only in very high pressure and temperature applications. ANSI B16.5 and ANSI B16.34 also have rating in metric unit. For flanged and flanged fitting, these are PN 20, 50, 68, 100, 150, 250, 420 and 760 where PN stands for nominal pressure rating number.

Pressure temperature rating tables are given in ANSI standards which indicate the maximum non-shock working pressure for which the fittings/flanges/valves as the case may be, are considered suitable corresponding to various working temperatures. Selection of the right class or rating is thus possible when service pressure and temperatures are known by consulting these tables. For components not covered by any recognised standards such as traps, strainers, rotameters etc. manufacturers rating or certification as to the maximum permissible working pressure for the service temperature are considered for their selection.

3.8 Pipe Schedule

This is a document containing information relevant to piping design, some of which are available also from the flow diagram and the P & I diagram and should not be mixed up with schedule number. This document has more information / data and the total information is presented in some systematic fashion. In fact, for preparation of this document the flow diagram and P & I diagram are necessary and must first be prepared. All piping connections appearing in the P & I diagram are listed here in order of line number sequence or if it can be justified, first servicewise and then line numberwise. The information / data provided for the listed lines could be as follows:

- identification of the line i.e. source and destination or from where it starts and where it ends, or KKS (*Kraftwerk Kennzeichen System*) number
- operating pressure, operating temperature
- design pressure, design temperature
- pipe wall thickness required as per specification and pipe wall thickness selected as per schedule number
- radius of bends, if used
- line insulated or not. If insulated, for what reason, ie, heat conservation, personal protection, avoiding condensation of atmospheric moisture on the pipe surface etc.
- Line heat traced or not. If steam traced, number and nominal diameter of tracers
- Hydraulic test pressure, if to be pressure tested
- Percentage of radiography or any other NDT to be carried out on the welds in line etc.

Sample design calculation for insulated pipes considering various parameters using different standard codes of practice will be interesting to the readers and be very useful specially to the practising engineers. The use of particular code of practice will, however, depend on the professional judgement of designer concerned[†]. ■

[†]Part 3 (the last part) will be published in October '99 issue