Piping Design in Power Plants

Piping design in high temperature and high pressure service has become complicated in recent years. Starting with functional requirement, design and pipe strength calculations, the piping engineers are able to perform the routing and elasticity analysis keeping in mind the supporting aspects. Simple approach to design aspects has been given in three parts of the article. Part I deals with the functional requirements and general design consideration for piping in power Plants.

Part One

Sankar Chakrabarti†
Mannesmann Seiffert GmbH, Berlin
Germany

1. Introduction

The progress of a country is normally assessed on the consumption of electricity which can be achieved by plants running on coal, oil, gas, water, wind, solar or nuclear power. As the coal reserve of India is very high, India has installed coal fired power plants of high temperature steam (550-570°C) for better efficiency in Singrauli, Korba, Ramagundam, Chandrapur, Farakka, etc – all of 500 MW capacity and going for waste heat generating power plants to utilize the heat of high temperature waste gas which would have otherwise burnt to atmosphere.

The main parts of a power plant, boiler and turbine, are connected with each other through piping. Though costing only 8 to 10% of the total cost of the plant, the piping must withstand the high temperature with less loss of energy and must be safe against any hazardous effect.

1.1 Considerations in piping design

The design of piping has become very complicated in recent years. The salient considerations of piping design and installation are summed up for better understanding.

---

† 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.

★ Functional requirements
Heat balance diagrams
Design parameters
Pipesizing
P & I diagrams
Pressure and temperature loss calculations
Insulations

★ Operational requirements
Fatigue stresses
Creep stresses
Life estimation

★ Pipe strength
Allowable stresses
Dimensioning of piping and components
Stress intensification factors
Elasticity and fluid flow analysis
Plant conditions and load cases
Load tables and check of connection loadings

★ Pipe layout 2-1-2 or 4-1-4 system
Piping layout composites for main and auxiliary lines
Cost and economy study
Interference checking
Steel structure drawings
Foundation drawings
Piping isometries of main and auxiliary lines

★ Pipe components
Detail component design and calculations
Detail engineering

★ Pipe supports
Stiffness of supports
Hanger, support and snubber schedules

★ Fabrication and erection
Fabrication and spool drawings
Cutting plans for piping
Welding lists (WL)
Welding Procedure Specifications (WPS)
Procedure Qualification Records (PQR)
Erection description and procedure
Erection calculations
Erection sequence plan
Installation

★ Testing and commissioning
Cleaning of piping, steam purging
Trail operation
Repairs
Commssioning

★ Inservice inspection
Creep gauge and expansion markers

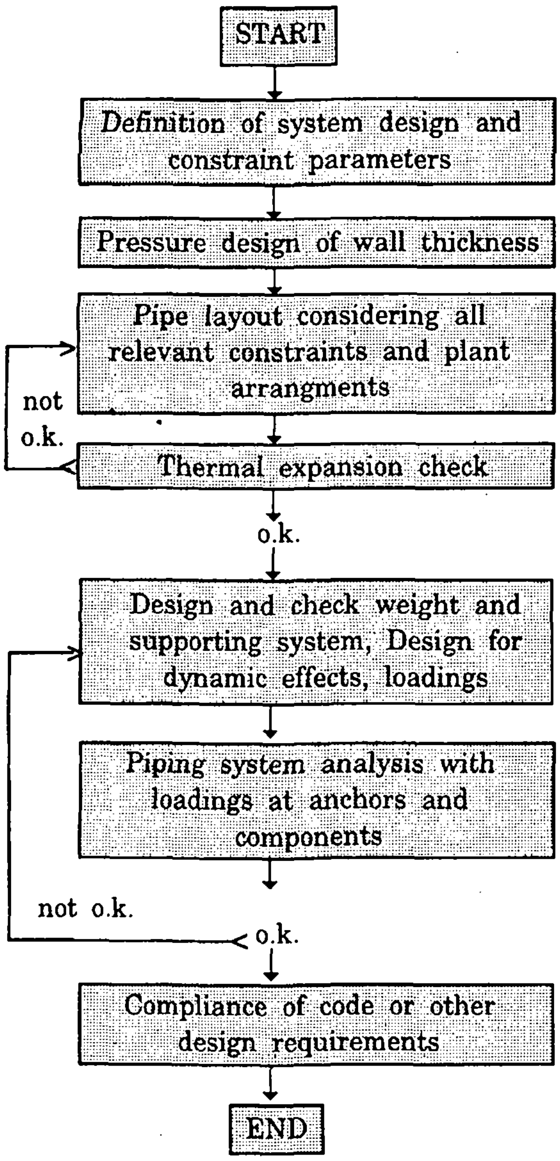
★ Quality Assurance (QA)
Technical Delivery Conditions (TDC) for pipes and components
Quality Plans (QP) for materials
Non-Destructive (NDC) and Destructive (DC) testing for materials
Field Qualtiy Assurance (FQA)

★ Material management
Bill of materials (BOM) for pipings, components, valves and supports
Enquiry for materials according to preliminary BOM
Ordering of materials
Receiving of materials and despatch to site
Erection follow-up

★ Project management

1.2 Standard design procedure

A standard pipe strength design procedure is given below :



2. Functional requirements

2.1 Nomenclatures

Ah	hydraulic flow area of pipe	mm^2, m^2
Ao	area of orifice	mm^2
d, d1, d2	pipe or component diameter	mm
da	pipe or component outside diameter	mm
di	pipe or component inside diameter	mm
dh	hydraulic diameter	mm
do	diameter of orifice hole	mm
dp	pressure loss, change of pressure, impulse	MPa, N/m^2
eta	dynamic viscosity	kg/m*s
f	friction factor of component or equipment	-
f1	friction factor of pipe	-
h, h1, h2	heights, geodetic heights at p1 and p2	m
k	ratio Cp/Cv	-
1, L, 1e	effective length of pipe	m
m	mass flow, fluid	kg/s
miu	orifice correction factor	-
nu	kinematic viscosity of medium = eta * v	m^2/s
P	design internal pressure	MPa
p, p1, p2	pressure of medium	Mpa, N/m^2
pc	critical pressure	N/m^2
q	distributed load = m*g	N/m

Re	Reyonlds Number	-
t, t1, t2	actual wall thickness	mm
T, T1, T2	design temperature	°C
Ta	ambient temperature	°C
Ti	inside wall temperature	°C
TK	absolute temperature = T + 273	K
Ts	surface temperature	°C
U	circumference of wetted area	mm
v, v1, v2	specific volume of the fluid	m^3/kg
w, w1, w2	velocity of fluid	m/s
αk	coeffieicient of heat transfer, convection	W/(m^2*K)
αc	coefficient of heat transfer, radiation	W/(m^2*K)
α	$\alpha k + \alpha c$	W (M^2*K)
αi	coefficient of heat transfer, inside	W/(M^2*K)
αa	coefficient of heat transfer, outside	W/(m^2*K)
ß	included angle of reducer, diffuser, angle between header and branch	degree
τ	thermal conductivity of material	W/(m*K)
Φ	orifice factor for medium	-
ε	smoothness factor	—

2.2 Properties of fluids

The piping should be designed to be able to deliver the rated flow at the required temperature and within the desired pressure loss from one equipment to the other. The flow rate is obtained from the applicable heat balance or flow diagram and the velocity of flow chosen for the transmission. For a particular flow rate, higher the velocity smaller is the internal diameter of the pipe and, therefore, lower could be the cost incurred on pipes, fittings and supports and even insulation where the pipe is insulated. But higher velocity results in higher pressure drop in piping and as a result costlier pump-motor set may have to be used. Also, the cost of energy in running the pump will be high and where a pump is not directly involved, high pressure drop would mean high energy loss and, therefore, more cost on energy replenishment. Further, high velocity could be detrimental from the point of view of erosion as well as corrosion specially in case of wet steam or liquids carrying abrasive solid in suspension. It may also cause vibration and noisy operation.

Knowing the flow parameters, the velocity of the fluid or the recommended velocity, the required hydraulic diameter of a pipe can be established.

$$w = 1{,}000{,}000 * m * v \;/\; Ah \quad \text{m/s} \qquad [2.01]$$

$$dh = 1{,}128 * \sqrt{(m * v / w)} \quad \text{mm} \qquad [2.02]$$

In calculating the pipe flow area or hydraulic diameter of equivalent piping for non-circular section, the flow diameter should be determined. For normal pipe, the flow diameter (d) and hydraulic diameter (dh) is same. But for non-circular section, the hydraulic diameter can be determined from the geometry as per the examples given below:

$$dh = 4 * Ah \;/\; U \quad \text{mm} \qquad [2.03]$$

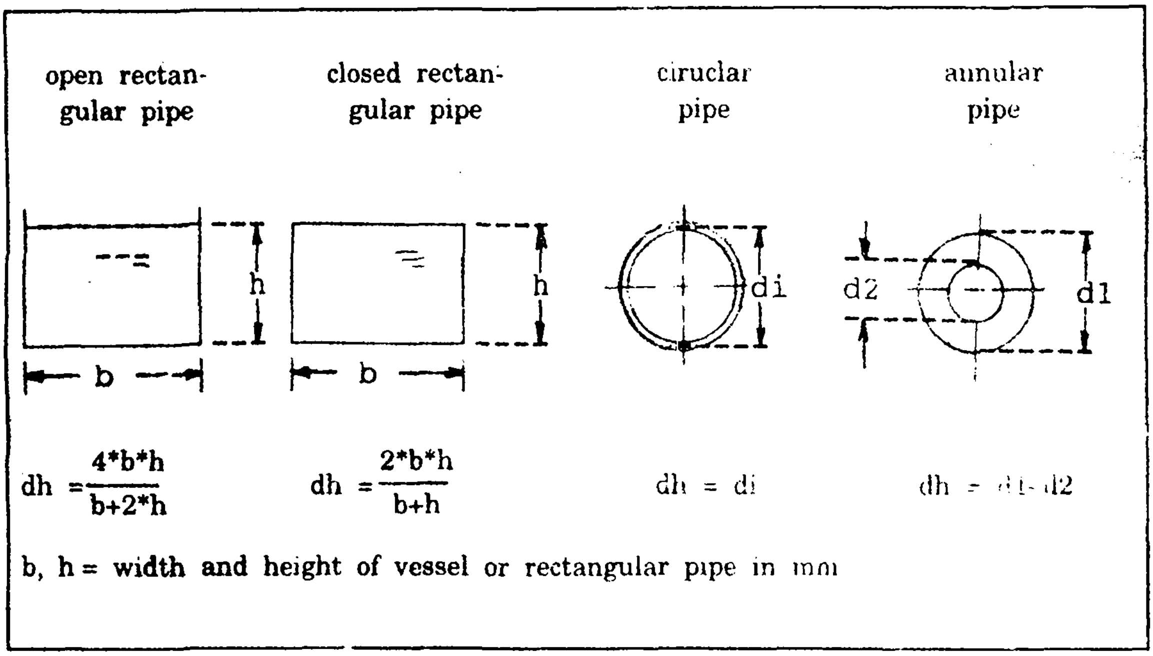


Fig. 2.2 : Example of hydraulic diameter

2.3 Recommended velocities

Unless limited by possibilities of accelerated corrosion/erosion or vibration, noise or pressure drop considerations, the choice of velocity can be optimized by techno-economic analysis of capital cost, operation and maintenance cost for the piping (inclusive of its supports and insulation) and associated pump-motor set and or other affected equipment or facilities. Many recommendations are available for the maximum velocity of flow of fluid through a pipeline. An actual optimization study is required for unusual cases such as long distance or cross-country pipelines.

Few recommended velocities in mass flow for specified services of a power plant are given in Table 2.3 in next page. These values are recommendations only and can be used as guideline for calculating the preliminary diameters.

Table 2.3 : Recommended velocities of a power plant piping

Medium	P (bar)	T (°C)	w (m/s)					
Superheated steam :			DN 150	200	300	400	500	800
Main steam	140	500/530	40	50	55	60	60	
	180	530/550	30	40	45	50	50	
	250	530/550	25	35	40	45	50	
Hot reheat	25	530/540			40	45	50	60
	40	530/540			35	40	45	55
	60	530/540			30	35	40	50
By pass		100% to 120% of steam line velocity						
Cold reheat	25	300			30	35	40	50
	40	340			25	30	35	45
	60	380			20	25	30	40
Feed water :								
Suction to pump					0.5 – 1.5			
Discharge					2.0 – 6.0			
Minimum flow, pumps					2.0 – 6.0			
Extraction	10 – 25		30	40	45	50	55	
	5 – 10		30	45	50	55	60	
	0 – 5				50	55	60	70
Saturated steam :		wetness X			(for all diameters)			
High pressure	80 – 100	1 – 2 %			35	–	40	
Medium pressure	8 – 20	8 – 2 %			20	–	25	
Low pressure	4 – 8	2 – 4 %			40	–	50	
	0 – 4	5 – 0 %			50	–	70	
Condensate :								
Suction to pump						0.5 – 1.0		
InFW tank						1.5 – 2.5		

Medium	P (bar)	DN 100	DN 200		DN 400
Condensate :					
Discharge	– MC 20 – 35	2.0	2.5		3.0
	– FW 100 – 150	3.0	3.5		4.0
	– FW 200 – 300	3.0	4.0		5.0
Heat exchanger drains :					
Upto CV	10 – 40		1.5	–	2.5
	5 – 10		1.0	–	1.5
	1 – 5		0.5	–	1.0
Loops			0.5	–	1.0
After CV (with flashing volume)			10.0	–	15.0
Limit valves – for start-up, shut-down or break down					
Superheat steam	40 – 250		60.0	–	70.0
Saturated steam	40 – 100		60.0	–	70.0
Blow out line	1 – 2		100.0	–	200.0

Cooling water : *Turbine condenser pump* :	DN 1000	DN 1800	DN 2500
Suction	1.0	1.3	1.5
Discharge	2.4	2.8	3.2

Intermediate cooler :	DN 400	DN 700	DN 100
Discharge	1.8	2.1	2.4

2.4 Pipe Sizing

Before material forecast, layout and engineering activities can proceed, fixing of pipe diameter and thickness of pipes must be completed and piping schedule or line list done.

Choosing first a velocity as per the recommendations of section 2.3 or as per the company standard, the preliminary diameter may be calculated for a velocity range or for a maximum velocity. For thin-walled pipes, the commercially available nearest internal diameter will be the size of the pipe. For high pressure or high temperature services, where wall thickness requirement for the pipe is quite high, it might be necessary to establish the wall thickness required for the outside diameter and then check back on the inside diameter available and the resulting velocity. If the velocity turns out to be unacceptable, the next higher commercial outside diamteter or a suitable non-standard outside diameter may be chosen if that be more economical. Normally, for high pressure piping, the pipes are manufactured according to required given inside diameter and minimum wall thickness. The pipe schedule, thus produced, should be checked for wall thickness to conform to required pipe strength.

2.5 Pressure Loss

The pressure loss of the fluid flowing through the piping is an irreversible process which occurs due to friction in the fluid or at the fluid circumference and produce heat energy. Before accepting the outside diameter and the nominal wall thickness established in accordance with the velocity and economy check, a pressure loss check is normally required for cirtical steam lines, such as, main steam, hot reheat and extraction steam and also for cooling water circuit.

For optimization of thermal cycle efficiency, commensurate with cost of piping materials inclusive of hangers, support and insulation, certain percentage pressure drops are normally allowed which are taken care of while preparing the heat balance diagrams and heat rate calculation. Sizing of these lines, therefore, are done without exceeding the allowable pressure loss values. The normal recommended allowable pressure drops in ciritical lines may be as given below:

★ ***Main Steam***
The pressure loss from superheater outlet of boiler to the high pressure turbine strainer should be within 90% of the pressure loss of the flow through the high pressure turbine valve wide open or within 5% of the main steam turbine inlet pressure. The main steam pressure losses are added to those of strainer and turbine to get the turbine outlet pressure or the cold reheat inlet pressure.

★ ***Cold Reheat***

The pressure loss of cold reheat with reheater line should be within 90% of the pressure difference between high pressure turbine exhaust and intermediate pressure turbine inlet.

In most of the cases the total pressure loss of cold and hot reheat is limited to a certain value. The increase of one may be compensated by the other.

Normally, all pipings and equipment shall be sized for flow, pressure and temperature at turbine valve wide open at rated pressure and auxiliary steam requirements.

If the pressure loss turns out to be in excess of that recommended, (may be ignored if marginal), reduction of the same may be attempted by route shortening, reducing number of fittings, changing radii of bends, type of valve, etc. Normally, these alternatives should not exist as the basic layout will be developed with all these considerations in mind for economy and a few such changes may not suffice. When these measures fail, only alternative left will be to go for higher ID or next higher commercially available OD pipes depending upon whether ID or OD based pipe was selected and repeat the design procedure.

Pressure loss calculations for liquid lines are required when it is necessary to establish the head for the pump delivering the liquid. Also, when it is necessary to size a line where the pump is existing or the pressures at the two ends are fixed or predetermined, pressure loss calculations must be made.

The resistance of flow through any pipe is represented by friction factor or drag coefficient (f 1) and is related to the Reynolds number which is a function of pipe inside diameter, velocity of flow and viscosity of the fluid.

As Reynolds number increases, flow pattern changes from the laminar to turbulent in the process of passing through a transient region where it is neither laminar nor turbulent and ultimately to hyper-turbulent state. In each of these regions, the friction factor is related to the Reynolds number in a different way. In laminar flow the friction factor is related only to the Reynolds number and for hyper-turbulent flow f 1 is related to the absolute roughness of the pipe wall.

Losses through fittings, denoted by the friction factor (f), can be converted to equivalent length for the fitting. Equivalent length for a fitting is defined as that length of a straight pipe of the same internal diameter as the fitting for which loss is the same as that through the fitting for identical flow conditions. Alternate way of calculation of losses through the fittings is to use the flow resistance coefficient f directly.

If the line or the circuit for which pressure loss calculation is performed has any special fitting like a strainer, steam separator, expansion joint, flow nozzle, orifice, rotameter, heat exchanger, control valve, etc, losses through them must also be added to arrive at the total friction loss. Loss through these items should be obtained from their manufacturers. In the beginning, assump-tions are made using, when available, previous experience or catalogue data and these are checked later on, when the actual data are available.

The basic equation of Euler for conservation of energy in a non-compressible medium as modified by Bernouli states that the summation of pressure energy, kinetic energy and potential energy are constant :

$$p1 + w1^2 /2*v1 + g*h1/v1 = p2 + w2^2 /2*v2 + g*h2 / v2 + dp$$

The pressure loss dp should be differentiated from other parts of the equation which are reversible from one form of energy to the other.

The pressure loss in a piping system irrespective of the flow pattern, eg, laminar or turbulent, with components as included in the above equation can be given as :

$$dp = (f + f1*1/dh)*w^2/(2*v) \quad N/M^2 \qquad [2.04]$$

The average value of specific volume and velocity of flow for the piping length should be employed in the equation, but if the length is too long and the flow condition changes very rapidly, the calculation is to be performed in small steps taking the average value for every step. The same procedure is usually adopted for non-isothermal flow also.

The temperature loss along the pipeline increases the density and the viscosity and hence the velocity decreases. This influence is small and can be neglected for normal purposes. In long pipeline the pressure loss is normally calculated with the specific volume calculated for the estimated average temperature of the line.

2.5.1 Friction factor f

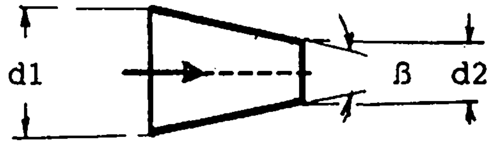
The friction factor f, for equipment and components dependant on their geometry and independant of the flow characteristics is mainly due to :

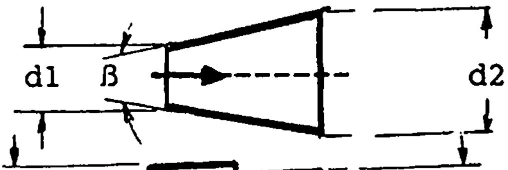
- ★ friction
- ★ change of flow section
- ★ change of flow direction
- ★ branches
- ★ components

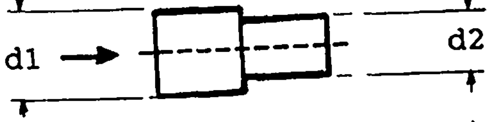
For quick reference the value of f for few common components are given herewith in the following table (Table 2.5.1 with figure) (refer VDI Heat Atlas / FDBR Handbuch):

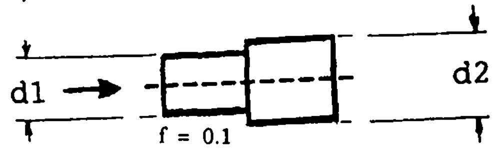
Table 2.5.1 with figure : Friction Factor f for Components and Fittings

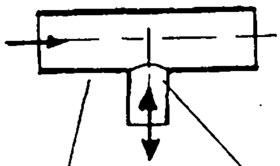
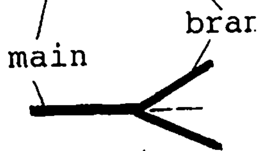
			r = 1.5*d	r = 3*d	r = 4*d
Bend	90°		f = 0.18	= 0.12	= 0.10
	60°		= 0.15	= 0.11	= 0.10
	45°		= 0.13	= 0.09	= 0.08
	30°		= 0.12	= 0.08	= 0.08
Reducer	ß	= 30°	f = 0.05		
		= 45°	= 0.06		
		= 60°	= 0.07		
Diffuser	ß	= 30°	f = 0.20		
d2/d1		= 45°	= 0.31		
1.5		= 60°	= 0.38		
d2/d1	ß	= 30°	f = 0.50		
3.0		= 45°	= 0.70		
		= 60°	= 0.81		
Abrupt change					
d2/d1	= 0.55		f = 0.37		
	= 0.70		= 0.30		
	= 0.85		= 0.18		
d2/d1	= 1.80		f = 0.48		
	= 1.40		= 0.25		
	= 1.20		= 0.10		
Branch flow	in main only			f = 0.1	
(T) • flow	from main to branch	100%		= 1.3	
		50%		= 0.9	
flow	from branch to main	100%		= 0.9	
		50%		= 0.3	
(Y) flow	main to branch branching			= 0.90	
flow	branch to main joining			= 0.50	
flow	main to branch in one leg			= 0.19	
flow	branch to main fr. one leg			= 0.24	
cross	pipe both branches to main			= 1.6	
	main to both branches			= 2.9	
(Y) flow	from/to vessel				
	sharp edge outlet			= 0.50	
	round edge outlet			= 0.10	
	inserted nozzle			upto 3.00	
	sharp edge inlet			= 1.00	
Valves (general)				f = 2.5	
Gate valve(depending on rated dia)				= 0.2 - 0.8	
Globe or throttle valve				= 3.0 - 15.0	
Non-return valves				= 3.5 - 6.0	
Angle valve				= 3.3 - 6.6	
Flow meter				= 2.5	
Strainer				= 2.5	
Water separator without baffle				= 4.0	

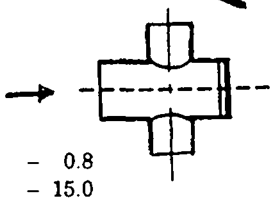










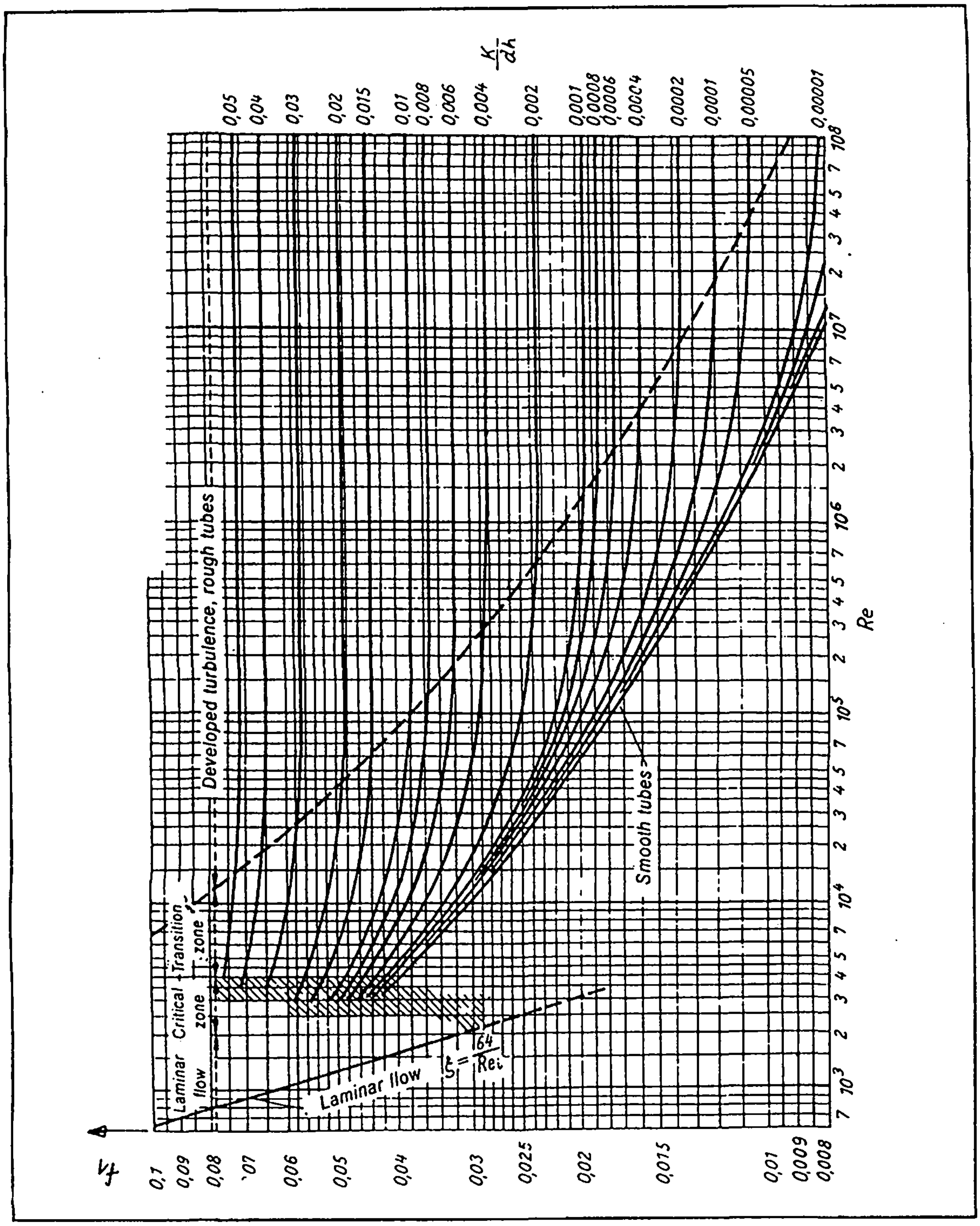


Figure 2.5.2 : Drag Coefficientf l

2.5.2 Drag coefficientf 1

The drag coefficient f1 for pipes may be taken from diagram, given in every standard work available, Figure 2.5.2, depending on the roughness of pipe and the nature of flow represented by Reynolds number, a non-dimensional number. The Reynolds number is calculated as:

$$Re = w*dh / nu \quad — \quad [2.05]$$

The flow is laminar below the critical Reynolds number of 2320, and above that in the transition zone of 2320 to 8000 the flow starts to be turbulent. At Re = 8000, the flow may still be laminar if it is very calm and quiet and the inside of the pipe is very smooth. With rough inside surface of pipe the flow turns to be turbulent with lower Re number but never below 2320.

According to Blasius, Hermann, Prandtl and von Karman, a rough estimation of f1 independent of roughness but for relatively smooth pipe can be made as :

fl =	0.3164/(Re)^0.25	for	3,000 > Re > 20,000
fl =	0.0054 + 0.3164 / (Re)^0.3	for	20,000 > Re > 2,000,000

A more accurate claculation of fl can be made for laminar or turbulent flow conditions determined by Re as given below.

★ For laminar flow, ie, for flow through smooth surfaces with moderate velocity, fl is independent of the roughness of pipe and is only related to Re, ie, for Re < 2320, Darcy-weisbach equation :

$$fl = 64 / Re \quad - \quad [2.06]$$

★ For flow changing from laminar to turbulent, the roughness of the pipe should be considered. if the surface of the pipe is rough, the fl value varies considerably. Regardless of the form of roughness a general term is relative roughness :

$$e = k / dh \quad - \quad [2.07]$$

where k denotes the absolute roughness or the average height of all the internal projections in the pipe in mm and can be taken from Table 2.5.2.

If the flow is completely governed by roughenss, fl can be calculated for flow of $Re*\sqrt{f1*\varepsilon} < 200$, from Colebrook-White equation :

$$\frac{1}{\sqrt{fl}} = 2.0*\log \left\{ \frac{\varepsilon}{3,71} + \frac{2.51}{Re*\sqrt{fl}} \right\} \quad [2.08]$$

★ For turbulent flow in hydraulic smooth pipe, the drag coefficient may be calculated for Re >> 2320 as :

$$1/\sqrt{fl} = 2.0*\log(Re*\sqrt{fl}) - 0.8 \quad [2.09]$$

★ If the roughness of pipe is considered, then the friction factor may be calculated for hydraulic raw pipe as per the formula given by Prandtl and von Karman :

$$1/\sqrt{fl} = 1.14 + 2*\log\varepsilon \quad [2.10]$$

For flow through pipes the absolute roughness factor of pipe k:

Steel pipes,	new	= 0.02	-	0.06	mm
	acid cleaned	= 0.03	-	0.04	mm
	old and rough	= 0.15	-	0.40	mm
	ERW pipes, new	= 0.04	-	0.10	mm
	very old and rusty	upto		3.00	mm
Concrete pipes	smooth	= 0.10	-	0.30	mm
	new or rough	= 0.30	-	3.00	mm
Cast iron pipes,	new	= 0.26	-	0.60	mm
	old and rough	= 1.50	-	4.00	mm
Copper, brass, bronze, glass pipes					
	smooth	= 0.001	-	0.002	mm
Aluminium pipe,	new	= 0.076			mm
Plastic tubes		= 0.0015			mm

Table 2.5.2 : Absolute roughness factors

2.5.3 Calculation of pressure loss

The salient points in calculating pressure loss can be summerized as :

★ The line is divided in sections with regard to change of state of flowing medium (diameter, valve, branch, etc.)

★ Temperature loss along the pipeline increases density and viscosity which is small in insulated lines and can be neglected.

★ The friction factors for the components should be based on the flow area and direction of flow.

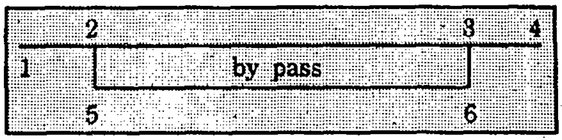
★ The pressure changes arising from velocity changes are to be included. In case of enlargement, the pressure increases due to velocity reduction and the 'f' values may be negative (ie, no pressure loss, rather pressure gain).

★ In case of inflow of fluid from a reservoir with w = 0 into pipeline, the pressure will drop due to acceleration. If the starting pressure of the reservoir is considered, f =1.0 may be taken.

★ During outflow from pipe in a reservoir, the velocity energy is normally lost, but the same pressure as at the end of pipe is measured in the reservoir. Therefore f = 0 is to be taken.

★ The pressure loss in series is simply added. But for parallel connection, the flow in the piping will adjust in such a manner that same pressure loss is generated in all parallel branches.

★ Pressure loss model for parallel connection.

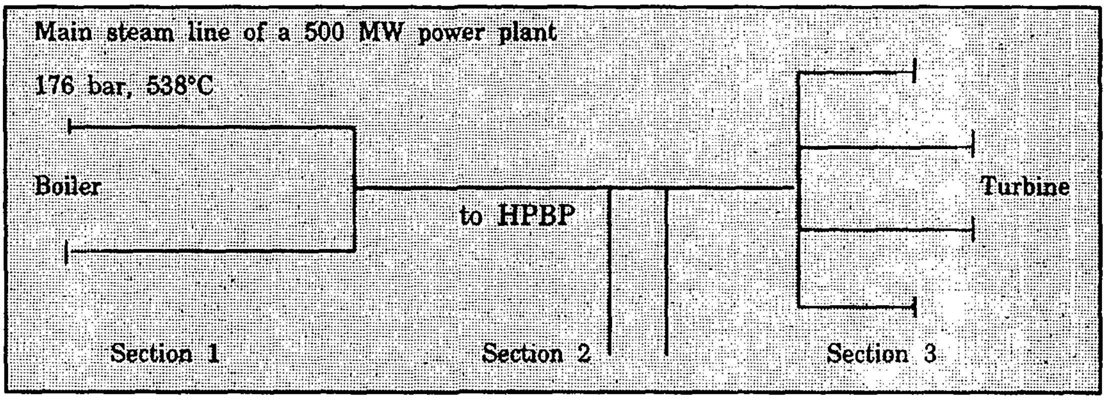


The flow adjusts in such a way that the pressure loss of the branches are the same. The error is insignificant for the practical cases if the pressure loss is calculated as follows:

$$dp(1-4) = dp(1-2) + dp(3-4) + dp(2-3) + dp(2-5-6-3)/2 \quad [2.11]$$

2.5.4 Example of pressure loss :

Example 1



Pressure Boiler outlet	P1	=	176 bar	=	17.6 MPa
Temperature	T1	=	538		°C
Specific volume	V	=	0.01871		m^3/kg, steam table
Dynamic viscosity	età	=	3.113e-5		kg/m*s, steam table
Kinetic viscosity	nu	=	eta*v		
		=	5.8243-7		m^2/s
Mass flow (total)	m	=	1670		t/h
		=	463.89		kg/s
Absolute roughness	k	=	0.04		mm (New pipe)

Pipe data :

Section 1, pro leg	Section 2	Section 3, pro leg
m = 231.95 kg/s	m = 463.89 kg/s	m = 115.97 kg/s
dh = 330 mm = 0.33 m	dh = 425 mm = 0.425 m	dh = 250 mm = 0.25 m

l = 68.5 m	l = 119.0 m	l = 31.0 m
h = 76.0 m	h = 40.0 m	h = 9.0 m
6 x 3*d, 90° bends	4 x 4*d, 90° bends	1 x 5*d, 90°C bends
2 x 3*d, 30° bends	2 x T, flow in main	1 x 1.5*d, 30° elbow
1 x Y, 90° br-main		1 x T, main to br
		1 x T, br to main
		1 x RD, ß = 30°

Turbine connection h4 = 22.5 m

Friction factors for components :

f = 6*0.12 + 2*0.08 + 1 x 0.5	f + 4x0.1 + 2 X0.1	f = 1x0.1 + 1x0.12 + 1*0.9 + 1*0 9 + 1 x 0.05
= 1.38	= 0.6	= 2.07
Ah = 0.08553 m^2	Ah = 0.14186 m^2	Ah = 0.04909 m^2
w = 50.74 m/s	w = 61.18 m/s	w = 44.2 m/s
Re = 2.875e7 –	Re = 4.465e7 –	Re = 1.897e7 –

Drag coefficient from eqn. 2.08

fl = 0.0125	fl = 0.0119	fl = 0.0132

Pressure loss due to loss of height : dph = g*dh/v

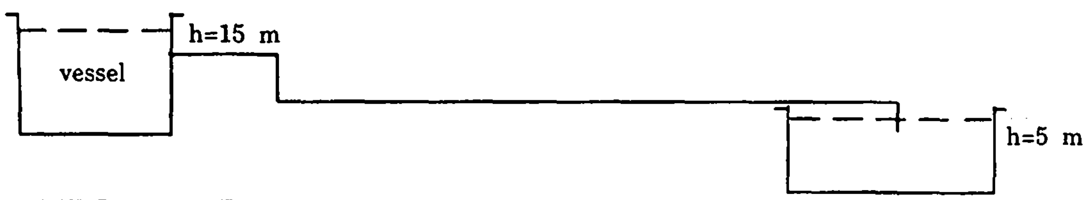
dh = h2 – h1	dh = h2 – h1	dh = h2 – h1
= –36 m	= – 31 m	= + 13.5 m
dph = –0.189e5 N/m^2	dph = –0.163e5 N/m^2	ph = 0.071e5 N/m^2
= –0.189 bar	= –0.163 bar	= 0.071 bar

Pressure loss due to friction : dps = (f+f*1/dh) *w^2/(2*v)

dps = 2.735e5 N/m^2	dps = 3.933e5 N/m^2	dps = 1.935e5 N/m^2
= 2.735 bar	= 3.933 bar	= 1.935 bar

Total pressure loss : dp = sum (dph + dps)
= 8.322 bar

According to the recommended pressure loss in main steam piping :
dp, all = 0.05*176 = 8.8 bar > dp



Example 2 :

In a piping system water to flow from 15 m to 5 m height with the following data:
m = 12 t/h = 3.333 kg/s

dh = 52.3 mm (pipe da 60.3 x 4.0 mm)		=	0.0523	m
T = 30 °C, v = 0.001 m^3/kg,		eta =	0.7972e-3	kg/m*s
1 = 25.3 m new pipe		k =	0.05	mm
2 x 1.5*d	90° bend, 3 x 1.5*d 45°bend,	1 x Gate valve		
3 x T	br- main, 1 x Y-90° br-main,	1 x RD B = 30°		

Calculations :

Ah = 0.7854*0.0523^2	=	0.002148	m^2
w = 3.333*0.001/0.002148	=	1.56	m/s
nu = 0.7972e-3*0.001	=	0.797e 6	m^2/s
Re = 1.56*0.0523/0.797 6	=	1.02e5	–

With k/dh ≅ 0.001 and neglecting the second part of the eqn 2.8 :
fl = 0.0196 and now with this value of fl in the same equation,
fl = 0.022

Converting the friction factors of the components and equipment in equivalent length of pipes : 1(e) = f*dh/fl

Pipe di	= 0.523 m			1	=	25.3	m
2 x 90	Bends	f = 2 x 0.18	= 0.36	lb1	=	0.9	m
3 x 45	Bends	f = 3 x 0.13	= 0.39	lb2	=	0.9	m
1 x	GV	f	= 0.8	lv	=	1.9	m
3 x	T	f = 3 x 0.9	= 2.7	lt	=	6.4	m
1 x	Y	f	= 0.19	ly	=	2.6	m
1 x	RD	f	= 0.05	lrd	=	0.1	m

Total equivalent length 1 = 38.1 m

Pressure loss due to friction :
dp = (fl*1/dh)*w^2/2^v
= (0.022*38.1/0.0523)*1.56^2/(2*0.001)
= 19501 N/m^2
= 0.195 bar

Pressure head available due to height :
dph= g*dh/v
= 9.81*(15-5)/0.001
= 98100 N/m^2
= 0.98 bar

Since dph > dp, the water flows from h2 = 15 m to h1 = 5 m.

Thus in this part of the article, some aspects of piping design in power plants—such as basic requirements, properties of fluid, pipe sizing, recommended velocity, pressure loss, calculations of pressure loss, etc, have been presented in a way which will be useful to the practising engineers. ■