Piping design in high temperature and pressure services for piping in power plants needs very careful consideration of different aspects which cause instantaneous enormous pressure rise due to high loading generating over-stressing of piping and pipe components and permanent deformation. In the present article, piping design in high temperature and high pressure is dealt with.

PIPING DESIGN IN POWER PLANTS

Part Four

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PIPING DESIGN IN POWER PLANTS
PART FOUR

With the invention of sophisticated materials and processes, the design requirements in powerplant piping became complicated. In 1999, simple approaches for functional requirements and pipe strength for common piping components have been presented. In the next few articles, pressure surge, elasticity analysis, supporting span and support and restraints will be presented.

4.0. Pressure Surge

4.1. Nomenclatures

a	wave propagation velocity	m/s
aw	adjusted wave propagation velocity	m/s
Ah	hydraulic flow area of pipe	mm^2, m^2
Ao, Ai, Al	area of pipe sections	mm^2
C	spring constant	N/mm, N/m
D	damping	kg/s
Dc	critical damping	kg/s
Df	damping factor DC	--
DLF	dynamic load factor	--
d,d1,d2 ...	pipe diameter (general)	mm, m
da	pipe outside diameter	mm, m
di	pipe inside diameter	mm, m
dh	hydraulic diameter	mm, m

dp,dp0,dp1	pressure loss, change of pressure, impulse	bar, N/m^2
dpm	surge pressure, maximum	N/m^2
dw	change of velocity	m/s
drh	change of specific weight	kg/m^3
F, Fm	force, surge force, maximum force	N, kN
f	friction fctor of component	--
f1	friction factor of pipe	--
fn	natural frequncy of system	1/s
fd	dampd frequency of system	1/s
h, h1, h2	geodetic heights, also at p1 and p2	m
I	impulse factor	--
k	isentropic exponent, ratio Cp/Cv	--
kt	transmission factor	--
l	length of pipe, general	m, mm
lo	wave length	m, mm
low	adjusted wave length	m, mm
m	mass flow	kg/s
nu	kinematic viscosity of medium = eta * v	m^2/s
nue	poisson's ratio (0.3)	–
p	design internal pressure	MPa, bar
p0, p1, p2 ...	pressure at different locations	N/m^2, bar
pc	critical pressure	N/m^2
Q	rate of fluid flow	kg/s
Re	Reynold's Number	–
rh, rh1	specific gravity of medium	kg/m^3
S	design stress intensity at desigh temperature	MPa
Sc	design stress intensity at room temperature	MPa

t	time, general	s
ts	valve opening or closing time, time for linear increase of pressure	s
t, t1, t2 ...	wall thickness	mm, m
tn	nominal wall thicknss of pipe	mm, m
T, T1, T2 ...	design temperature	°C
Ta	ambient temperature	°C
w, w0, w1 ...	velocity of medium	m/s
wn	natural circular frequency	rad/s
wd	circular frequency with damping	rad/s
v, v0, v1 ...	specific volume of the fluid	m^3/kg
x	displacement, path	mm, m
Y, Yt	elasticity mod. pf pipe ambient/design temperature	MPa, N/m^2
Ym, Ymt	elasticity mod. of medium ambient/design temperature	MPa, N/m^2
Γ	average density of a medium	kg/m^3
τ	period of free vibration	s

4.2. Introduction

When the transmission line circuit breakers trip under load, the turbine generator with almost zero output and some input, accelerates which may cause turbine-generator overspeeding. To prevent this, the turbine inlet valve must be closed quickly. This quick closing of turbine valve (0.05 second to 0.1 second) causes pressure wave in the piping which is a pressure difference with positive or negative sign propagating at a speed of sound in the piping and causing high transient unbalanced force with change of direction at every bifurcation and results high magnitude of vibration. These forces, however, die out after 2–3 seconds, but may cause overstressing of pipe and components and permanent deformation and hence need stronger restrains and structures and perhaps snubbers. The closing time of the valve must be carefully determined and adjusted to the system to compromise between the pressure rise in the piping and speed rise of the turbine.

This phenomena of high loading at restrains, vibration and overstressing of piping and pipe components cause by pressure surge is called steam hammer.

The pressure wave also arises in the pining system due to rapid local mass flow rate variation which are caused by condensation and evaporation processes, modification of pump heads or by lowering the pressure at one point of piping (pipe rupture). However, in this chapter, the quick closing of turbine valve generating pressure wave will be discussed.

The connecting tank or vessel to the piping may be disregarded as the pressure entering in them does not reflect back with the same amplitude, because the volume of the connecting vessel is much larger compared to that of the pipe.

As a result of newly acquired technology of material and system, the temperature and pressure of fossil power plants have been elevated nowadays. The use of sophisticated material and exact computerised analysis for piping on the other hand results in decreasing of the wall thickness. Again, the main steam velocity has gone up from 30 m/s to 60 m/s in the last 30 years. Therefore, in the capacity power plant, the safety valve must be operated more frequently due to exact requirement of change of loading and the thin designed pipes with less inertia against vibration are subjected to very high shock loads. Prediction of this high load requires surge wave analysis which depends solely on closing function, ie, characteristics of valves. So, the accuracy of the analysis depends entirely on the characteristics of these valves.

The analysis of water piping in nuclear and fossil power plants are analogous. But, for steam piping, there are some differences :

- The steam in fossil power plants has very high temperature (540°C–560°C) and is superheated, whereas in nuclear power plants the steam temperature in normally below 400°C.
- In the main steam line in fossil power plants, there is a HP — heater having substantial volume and hence reflection wave.
- The cold reheat line of the fossil power plant connected to the main steam line through HP Bypass has often modeled with the main steam with proper model of the turbine considering the reflection valve. The hot reheat is modeled separately.
- There are more leads from turbine and boiler connected together. A phase difference in the lead may cause unbalanced forces.

This shows that the requirement of fluid flow analysis is more in a modern fossil power plant than in a nuclear power plant. But due to probability of hazard in case of failure in nuclear power plant, nowadays it is primary requirement of the nuclear power plant safety laws that the pressure analysis is performed.

4.3. Pressure Wave Propagation

4.3.1. Opening of Valve

During opening and closing of valve in a pipe with flow medium, the mass flow rate and the pressure difference vary in accordance with the valve characteristics. The pressure wave propagates with a finite speed and requires a certain time to get the other end and reflects to the valve requiring almost the same time and gets again reflected. After a few reflections, depending on the influence exerted by friction, the perturbations are damped and the flow becomes steady state again.

A tank T1 with pressure p1 is connected to another Tank T2 with less pressure and isolated by a valve V. The pressure of the pipe up to valve V is p1, the full valve opening time is ts.

By opening the valve, an expansion wave propagates from the valve V in the steady flow of medium and expands the fluid in this process. During the expansion the specific volume increases resulting outflow from pipe through the valve in tank T2. The higher the velocity, the greater is the compressibility of medium.

After the expansion wave reaches the tank T1, a constant velocity prevails throughout the pipe downstream of wave, the pressure is equal to the pressure after valve (p2).

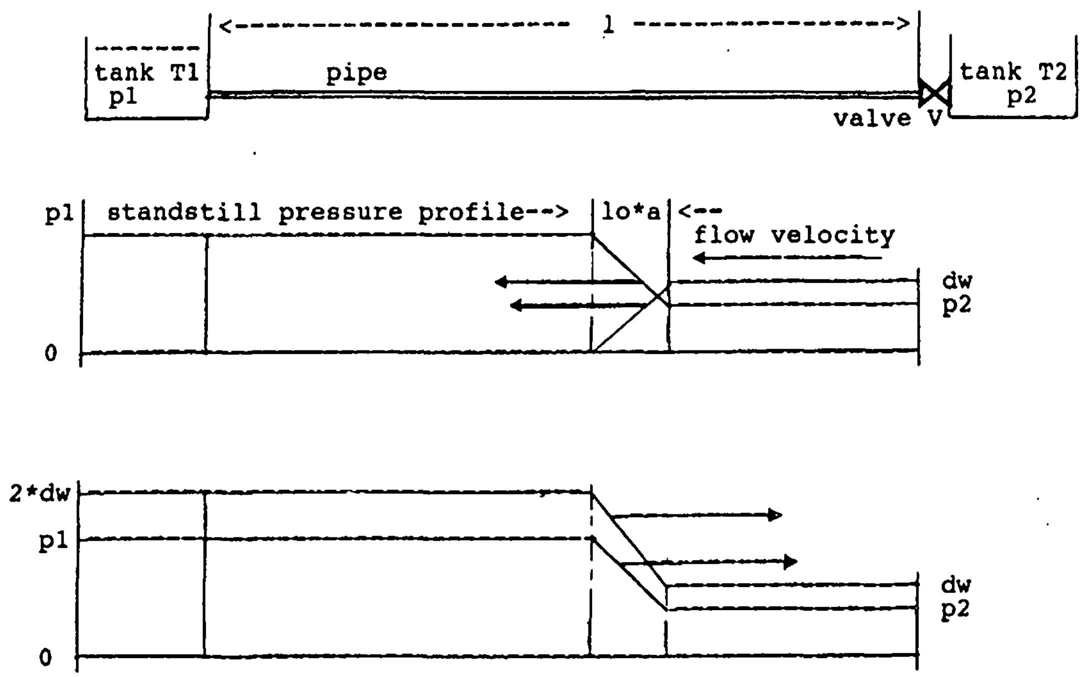


Fig. 4.1 : Pressure and Velocity Wave Propagation by Valve opening

But there is no equilibrium between p1 and the pressure in the pipe connection, therefore, a compression wave travels from T1 to T2. The medium will be compressed and accelerated. Downstream of this wave the flow velocity is nearly twice as high as before, and its pressure equals the pressure in the tank minus inlet pressure drop.

The compressive wave is reflected at T2 as expansion wave resulting a further increase of exit velocity. If the total pressure drop (inlet pressure drop + friction pressure drop in the piping) equals p1 – p2, the flow becomes steady.

If, however, the local speed of sound is attained, the flow is critical, ie, the flow expands up to critical pressure. This phenomenon occurs with great difference and small local speed of sound, eg, for gases and 2-phase mixture.

4.3.2. Closing of Valve

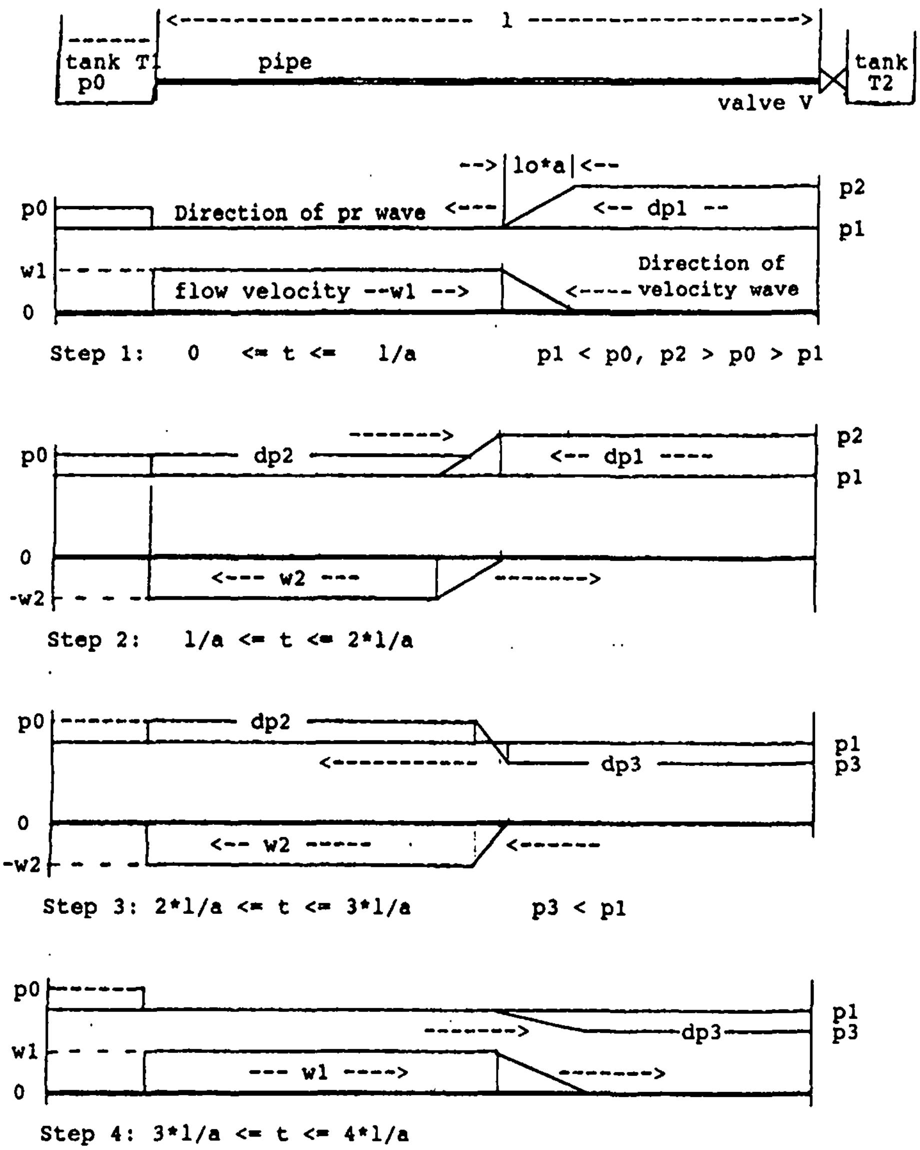
The valve completely closes the flow within time ts. Let us take the constant flow velocity w1 flowing from tank T1 to Tank T2. As the valve closes quickly, the flow comes down to a standstill and the steps of pressure surge are described below.

With index	0	condition of fluid in the vessel
	1	stationary condition of fluid with valve open
	2	maximum condition
	3	minimum condition, negative values,

the surge propagation, without consideration of friction loss, in the piping system by closing of the valve is explained in Fig. 4.2. in time steps.

- Step 1: Time t lies between 0 and 1/a : the compressive pressure generating the pressure wave dp1 = p2 – p1 move in direction of tank T1 with the velocity of sound a. In the region of pressure wave the velocity of the medium is zero, but between the end of pressure wave and the tank T1 the medium flows with uninterrupted velocity w1 towards the valve. This is the condition till the pressure wave dp1 reaches the vessel.
- Step 2: Time t lies between 1/a and 2*1/a : as the pressure wave reaches the tank T1 having a low pressure p0, adjustment of pressure takes place within the tank and the pressure wave dp2 = p0 –p1 now moves towards the valve with the velocity w2, slower than the velocity w1 in uninterrupted condition. The pressure wave dp1 exists between the new pressure wave and the valve corresponding velocity is zero.
- Step 3: Time t lies between 2*1/a and 3*1/a : The low pressure wave dp2 has now reached the valve and reflects back towards the tank t1 and underpressure dp3 = – (p1–p3) is created which move towards the tank T1. Between the pressure wave dp3 and tank T1 the medium has pressure dp2 and velocity w2.

- Step 4: Time t lies between 3*1/a and 4*1/a : The underpressure dp3 has now arrived the tank T1 and after a pressure adjustment the medium flows towards the valve with a velocity w1 and pressure is p1.
- Step 5: Time t > 4*1/a Step 1/a : (not shown in Fig. 4.2). The pressure wave starts again to move towards the tank T1 as in step 1. But the friction of the piping and other damping factors cause the flow to decrease and it stops altogether after a few cycles. It has been found that after 2 seconds there is practically no effect of surge.



← *Fig. 4.2 : Pressure and Velocity Wave propagation by valve closing*

If the valve closing time is higher than the wave propagation time, ie, the valve closes slowly, or ts > 2*1/a, where a is the wave propagation velocity, the influence of pressure wave may be neglected.

4.3.3. Pressure Wave due to Condensation

If water is injected in a pipe full of steam, simultaneous condensation of steam takes place and water surface thus produced strikes the undamped wall of another water surface. Such steam bubbles separating water column at elevated pipe locations of saturated water pipes cause pressure drop in the piping. At subsequent rise of pressure, the steam bubbles existing between the water columns collapse resulting high pressure wave. Similar effect is perceived when steam is injected in a condensate line.

4.3.4. Pressure Wave at Start-up of Pump

On start-up of a pump, a pressure difference depending on the mass flow rate variation characteristics is established between the suction and discharge nozzle and propagates in form of pressure wave. Similarly, pump shut-down produces a pressure wave. Constant pressure difference between pump nozzles suddenly disappears at decrease of mass flow and if the non-return valve does not close rapidly, the pressure wave traveling to the suction line, which is designed for lower temperature and pressure, may get damaged at the unpredicted loads from pressure wave.

4.3.5 Reflection of Wave

The change of pressure and the associated velocity due to surge for a straight pipe between a vessel of infinite volume (reactor, boiler, turbine, etc) and a valve (quick-closing value at turbine) 4.2.

There is a definite relation between the moving surge wave and the reflection part of it. Let normal surge part be NS and the reflected part RS. Then with flow sections as shown below :

$NS = RS + 1$
$RS = 2^{*} A1/A - 1$

In case of the reflection from the infinite vessel
$A = A1 + A2 + Ao$ = infinity and
$RS = -1$

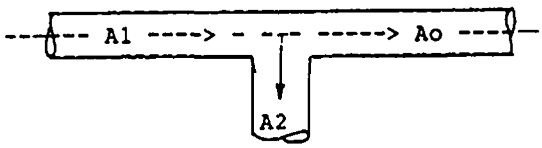
That shows that the wave is reflected back with full magnitude but with opposite sign from an infinite vessel.

4.3.6. Pressure Wave in Branch

If a pressure wave travels through a branch or a pipe with change of section, the amplitude is changed since the pressure wave travels back into the branch or to the first section of the pipe. The ratio of the amplitude of primary to the secondary wave is called the transmission factor kt which is given as

$$kt = \frac{2^{*}Ao}{Ao + A1 + A2} \quad \ldots \quad [4.01]$$

where Ao represents the outgoing and A1, A2 represent the incoming and branching hydraulic cross section of the pipes. This equation applies if the flow velocity is very small compared to the wave propagation velocity.



In branch connection with Ao = A1 = A2 :
RS = 2* Ao/3*Ao – 1 = – 1/3

The reflected wave is only one-third of the original volume and the main wave is also reduced to

NS = – 1/3 + 1 = 2/3 of the original value.

The *reflection wave magnitude at branching* changes to :

dp, r = kt*dp bar [4.01A]

4.4. Impulse Force

A special class of dynamic load is impulse force of relatively short duration which is normally absorbed in damping depending on the structural configuration of the system.

With

- Y as deflection
- Fy external force acting on the system
- Wf work done by force Fy
- Fd actual force on the system
- Wd deformation work by force Fd

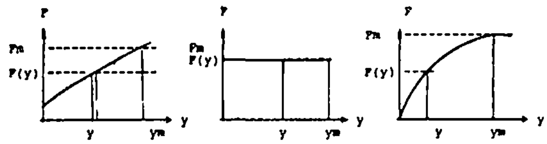


Fig. 4.3 : General, sudden and sine wave impulse

$$Wf = \int_{y=0}^{ym} F(y) * dy \qquad Fdm = C * ym$$

$$W = \int_{y=0}^{ym} F(y) * dy = C * ym^2/2 \qquad = Fdm * ym/2$$

since Wd = Wf,

$$Fdm = \frac{2}{ym} * Wf \quad = \frac{2}{ym} * \int_{y=0}^{ym} F(y) * dy$$

a) Sudden Load : FY = Fm

$$Wf = \int_{y=0}^{ym} F(y) * dy \quad = Fm * ym$$

$$Fdm = \frac{2}{ym} * Fm*ym = 2*Fm$$

Impulse factor I = Fdm/Ffm = 2.0 [4.02]

(b) Sine Wave Load : Fy = Fm * sin(wy)
w = $\pi/(2 * ym)$
Wf = Fm * sin(wy)*dy = Fm/w * $[\cos(wy)]_0^{ym}$ = Fm/w

$$Fdm = \frac{2}{ym} * Wf = 4 * Fm/\pi$$

Impulse factor I = Fdm/Ffm = 4*Fm/π*Fm)
= 1.27 [4.03]

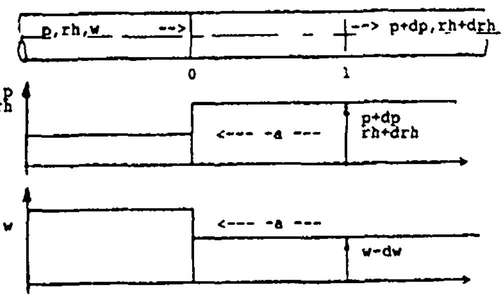
The impulse factor or dynamic load factor as normally named in surge analysis, represents the dynamic action of the flow induced loading. The maximum possible surge load as calculated later for every action can by multiplied by the impulse factor selected and assumed as acting at the change of direction of piping. The simplified elastic analysis may then be performed as against the dynamic analysis.

The impulse factor for triangular load $I = 1.7$

4.5. Basic Equation

The forces occurring at different bifurcations of piping system due to pressure surge arising from sudden closure of turbine valve may be calculated in a simplified way by using the formulae given by Joukowsky in 1898. This equation was developed for water hammer. But as the pressure surge occurs as a result of change of kinetic energy of the moving fluid medium irrespective of the cause of change of velocity, the formulae developed for water can also be adapted fro all other fluids. This equation is recognised as the basic equation in shock wave theory. But as a result of fluid friction and reflection of wave at branches or change of diameter, the pressure surge is normally lesser than that calculated by this basic equation. This calculation of surge with damping characteristics is complicated and can only be performed with computer.

The basic equation from impulse and mass equations is based on stationary conditions. The equation delivers no information about change of pressure with time and is valid for incompressible medium, such as, water, and with an established rh for the given condition , this equation can be used also for steam, gas, etc.



With pressure and velocity upstream and downstream of wave as p, w, and p+dp, w+dw and the control volume moving with wave velocity a in the opposite direction, the impulse theory yields :

before the surge = after the surge

$$p - a^2 * rh \quad = \quad (p+dp) - (a-dw)^2 * (rh+drh)$$

Considering the small value as $dw^2 \wedge 2 \rightarrow 0$, and the medium as incompressible, ie, $drh \rightarrow 0$; the above equation can be simplified as :

$$dp = -a^2 * rh + a^2 rh - 2 * a * dw * rh \qquad [4.04A]$$

Similarly from mass flow :

$$-a * rh = -(a-dw) * (r + drh)$$

and transfering, multiplicity by – a and neglecting the small value of drh as before :

$$0 = a^2 * rh - a^2 rh + a * dw * rh \qquad [4.04B]$$

Adding equation 4.04A and 4.04B,

$$dp = -a * dw * rh$$

Analogous for a wave propagating in the opposite direction, the general equation for pressure surge :

$$dp = p0 - p1 = a * h * (w1 - w0) \qquad N/m^2$$

where a >> w0

The piping system, after experiencing the effect of the first reflection wave, does not come to rest, because of pressure of different velocities at different sections. The exact analysis is complicated and time consuming and can be performed with computer programme as stated above. But for a conservative manual assessment of the maximum pressure and velocities can be done as given below :

With w1 = 0, the equation can be written as :

$$dpm = a * w0/v0 \qquad N/m^2 \qquad [4.04]$$

4.6. Calculation of Wave Propagation Velocity and Wave Length

The transient changes in pressure dp propagates in a compressive medium at a local speed of sound

$$a = 1/\sqrt{rh * (1/rh * drh/dp + 1/Ah * dAh/dp)} \qquad m/s$$

Considering that the pipe is rigid, ie, no change of area with change of pressure takes place, ie, dAh $\rightarrow$ 0 and single phase flow, the equation for sonic velocity can be written as :

$$a = \sqrt{dp/drh} = \sqrt{dp * dv} \qquad m/s$$

This equation implies that the sonic velocity of a fluid is dependant on the compressibility of the fluid.

As in techniques the compressibility is represented by the elastic module,

ie, Ym = rh * dp.drh,

the equation can be written as :

$$a = \sqrt{Ym/rh} \quad \text{or} = \sqrt{Ym * v} \qquad m/s \qquad [4.05A]$$

The value of Ym or fluid medium at ambient temperature for :

water = 20.3e8 N/m^2 and oil = 14.7e8 N/m^2

The sonic velocity of the medium can be calculated from the material data at different pressure and temperature. The influence of pressure for normal technical material is small, but the influence of temperature is high. For water, the sonic velocity can be calculated as :

$$\begin{aligned} a = {} & 1402.74 + 5.0336*T - 5.795e - 1*T^2 \\ & + 3.36e - 4*T^3 - 1.453e - 6T^4 \\ & + 3.045e - 9*T^5 \end{aligned}$$

This equation implies that the change takes place in an adiabatically reversible manner. The same equation for an ideal gas with isentropic exponent k is given by :

$$a = \sqrt{k*p*v} \qquad m/s \qquad [4.05B]$$

In good approximation, the steam may be an ideal gas with small pressure waves.

Velocity of the medium :

$$w0 = m*v0/Ah \qquad m/s \qquad [4.06]$$

Since the piping material is not infinitely rigid, the propagating pressure wave expands the piping and slows down the wave velocity. In normal steam pipe, this reduction of velocity is marginal but in case of water piping the velocity drop may be up to 2/3 of the sonic velocity.

If the pipe wall is not considered rigid and the wall of the pipe does not exceed the elastic limit of the material used, the local velocity of sound propagation decreases and can be calculated as :

$$aw = \frac{1}{\sqrt{\left(\frac{1}{a^2} + \frac{d}{Y^*t^*v}\right)}} \quad m/s \qquad [4.07]$$

The length of wave propagation (lo) in a pipe depends on the mechanism causing the wave, ie, the wave closing time (ts).

$$lo = (aw - w0) * ts \qquad m \qquad [4.08A]$$

and the maximum surge pressure is given in equation 4.4.

Since the velocity of flow varies due to pressure wave, the wave length lo varies accordingly, it decreases in case of a compressive wave and increases in case of an expansion wave.

In calculation of the forces exerted at the pipe ends due to surge, the wave propagation length lo as described above is of great importance. The decrease of wave length due to compressive wave can be calculated with a wave shortening factor kw :

$$kw = \left[1 - \frac{w0^*x}{aw * (aw - w0) * ts}\right] \quad \ldots \qquad [4.09]$$

where x denotes the travel length of the wave under consideration. The wave propagation length changes to

$$low = kw * lo \qquad m \qquad [4.08B]$$

4.7. Calculation of Surge Force

When a pressure wave propagates with the velocity of sound through a pipe, the pressure differences dp at front and back of the wave are different and also the resulting surge forces towards outside. These differences in magnitude of the forces cause the pipe to vibrate and exert extra forces to supports or stops. For calculation of surge force following simplifications can be made.

- at the valve end the starting values are dpl, aw and vl, corresponding to the uninterrupted flowing medium.
- The wave length pressure surge at front and back are linear as shown in Fig. 4.1.
- The influence of change of velocity and impulse force on bends is very small.
- The flow velocity w is small compared to the wave propagation velocity aw and hence may be neglected. Normally wmax << 0.2*aw.
- The change of wave velocity dw is small compared to the initial velocity. Normally dwmax << 0.1*aw.
- The pipe configuration is simple, ie, the flow propagation can be predicted.

The valve closing time is given by the manufacturer. To get a simplified ts a tangent to the curve is drawn to meet the axes and the intersection point is taken.

The force resulting from the primary pressure wave can be written as

$F = Ah * (p2 - p1)$

where $p2 - p1 = dp$ denotes the pressure difference between locations of change of direction of flow, ie, between bends. If the mass flow density linearity varies with the time at a location of pipe, the pressure gradient is

$dpg = dp/lo$

and the maximum surge force

$fm = kt * dpm * Ah$ N [4.11]

4.7.1. Long Pipe

In case of a long pipe, ie, l > lo or valve actuating time is very short so that the complete pressure wave can built up in the pipe section between bifurcations :

$dp = dpm$

$= m * aw/ah$ N/m^2 [4.12A]

and the corresponding axial surge force :

$Fm = kt * dpm * Ah$

$= kt * m * aw$ N [4.13A]

4.7.2 Short Pipe

If the valve actuating time is long or ts > t, the maximum pressure wave length corresponding to wave propagation velocity can not be built, the pipe is considered as short pipe or 1 < lo. The pressure surge developed in such a pipe between two bifurcations can be written as :

$dp = dpm * l/lo$

$= m * l/(Ah * ts)$ N/m^2 [4.12B]

and the corresponding axial surge force :

$F = Ah * dp$

$= m * l/ts$ N [4.13B]

If there is a 100% reflection, the wave propagation length may decrease to lo/2

The transmission factor kt can be introduced in case of branching of flow.

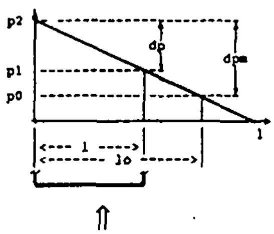


Fig. 4.4 : Case 1, Short Pipe

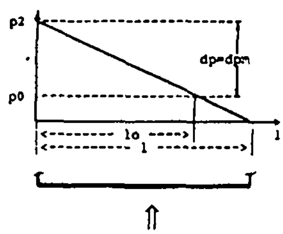


Fig. 4.5 : Case 2, Long Pipe

4.8. Dynamic Load Factor

The loads resulting from pressure waves on the piping system produce displacements and rotations of the elastic system. These excitations are oscillations of the pipes, the maximum amplitude of which depend on the frequency spectrum of the exciting force /time function.

A dynamic load factor is introduced for the conservative calculation of pipe deflections with maximum surge force taken as static force on the system.

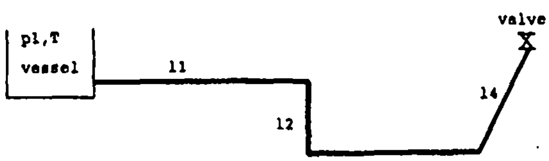
$$DLF = \frac{\text{Fm of a single mass oscillation}}{\text{Fm of fluid force}}$$

The value of DLF depends on the dynamic vibration behaviour of the structure and cannot be established as a standard as impulse factor discussed in section 4.4. But these factors, if not established, can be taken from experience irrespective of wave form as :

- for safety related section DLF = 2.0, as sudden force
- pipe mass with high mass coverage, vertical near turbine = 2.0
- other sections = 1.3 as sine wave force

4.9. Example of Pressure Surge

4.9.1. Example 1: Simplified Pressure surge of a Steam Line



Data:

Pipe inside dia.	d	= 0.5 m
Lengths	l1	= 42.0 m l2 = 135.0 m l3 = 68.0 m
	l4	= 38.0 m
Pressure	p1	= 157.0 bar = 157.0e5 N/m^2
Temperature	T	= 538.0 °C
Ratio Cp/Cv	k	= 1.291 –
Sp. volume	v	= 0.0213 m^3/kg (from steam table)
Mass flow	m	= 444.0 kg/s
Closing time of valve	ts	= 0.1 s

Calculation

Sectional area of pipe Ah		= 0.7845 *di^2	= 0.196	m^2
Velocity of sound	a	= √(k*p1*v)	= 649.5	m/s
Velocity of medium	w1	= m*v/Ah	= 48.3	m/s
Change of pressure	dp	= p2 – p1		
		= a*w1/v	= 14.7*10^5 N/m^2	
Wave length	lo	= a*ts	= 64.95	m

Forces

Section 1:	l1 = 42 m < l0	F1 = m*1/ts	= 1.86e5 N
Section 2:	l2 = 135 m > l0	F2 = m*a	= 2.88e5 N
Section 3	l3 = 68 m > l0	F3 = m*a	= 2.88e5 N
Section 4	l4 = 38 m > l0	F4 = ml/ts	= 1.69e5 N

Elasticity analysis may be performed with these forces acting axially in the flow direction at the bends.

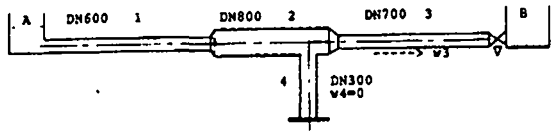
4.9.2. Example 2: Calculation of Water Pipe with Branch and with kw and kt

Data :

Pipe inside dia.	DN600 d1	0.582 m	t1 = 0.014 m
	DN800 d2	0.781 m	t2 = 0.0165 m
	DN700 d3	0.683 m	t3 = 0.014 m
	DN300 d4	0.306 m	t4 = 0.009 m
Pipe lengths	long pipe l = 2000 m > lo for all sections		
Pressure	p = 100.0 bar = 100.0e5 N/m^2		
Temperature	T = 20.0 ° C		
Sp. volume	v = 0.001 m^3/kg (water, from steam table)		
Density	rh = 990.0 kg/m^3		
Mod of elasticity	Ym = 20.3e8 N/m^2,	Water	
	Y = 213.0e9 N/m^2,	pipe material	

The medium flows from vessel A through 1 – 2 – 3 to B and through the valve V, Section 4 is dead end.

Velocity, normal $w3n = 10.1$ m/s taken
$w4n = 0$ no flow dead end
The valve is quickly closed, closing time of valve $ts = 1$ s



$a = \sqrt{(Ym \cdot v)} = (20.3e8 \cdot 0.001)$ = 1420 m/s [4.05A]

$$aw = \frac{1}{\sqrt{\left(\frac{1}{a^2} + \frac{d}{Y \cdot t \cdot v}\right)}} = 1160 \text{ m/s} \quad [4.07]$$

$lo = (aw - w0) \cdot ts$ = 1150 m [4.08A]

Since l >> lo, the wave front in section 3 has traveled less than half the length before the valve is completely closed.

Maximum surge pressure in section 3 :

$dpm3 = aw \cdot w3n/V3 = 115.0e5$ N/m2 = 115.0 bar [4.04]

The transmission factor kt3 at the branching 2-3-4 :

$$kt3 = \frac{2 \cdot A3}{A2+A3+A4} = \frac{2 \cdot 0.683^2}{0.781^2+0.683^2+0.306^2} = 0.797 \quad [4.01]$$

When the wave travels through the branch connection, the pressure amplitude changes and the reflection wave in section 3 which is same in section 2 and 4 :

$dp3R = dp2 = dp4 = kt3 \cdot dpm3$ = 91.7 bar [4.01A]

The velocity of the fluid in normal flow condition in section 2 :

$w2n = w3n \cdot 0.683^2/0.781^2$ = 7.65 m/s

The velocity is delayed by the pressure wave entering in sec. 2 by :

$dw2 = k3 \cdot w3n = 0.797 \cdot 10.0$ = 7.97 m/s to
$w2 = 7.65 - 7.97$ = − 0.32 m/s

The transmission factor in section 2 from section 1 :

$$kt2 = \frac{2 \cdot 0.781^2}{0.582^2+0.781^2} = 1.286$$

The wave passes from sec.1, DN600 with the kt2 :

$dp2R = dp1 = kt2 \cdot dp2 = 1.286 \cdot 91.7$ = 117.9 bar

The velocity of the fluid in normal flow condition in section 1 :

$w1n = w3n \cdot 0.683^2/0.582^2$ = 13.78 m/s

The velocity is delayed due to the pressure wave by :

$dw1 = kt2 \cdot w2n = 1.286 \cdot 7.65$ = 17.01 m/s to

$w1 = 13.78 - 17.01$ = − 3.24 m/s

Wave is reflected by the vessel PV and is subsequently superimposed on the secondary wave produced at the point of discontinuity.

The wave entering in the branch 4 with dp4 = 91.7 bar is reflected with full amplitude and the total pressure with the reflection wave pressure is

$p4R = 100 + 2 \cdot 91.7$ = 283.4 bar

Calculation of maximum pressure :

After the propagation of the first wave, fluid does not come to rest, because before closing of valve, different velocities existed in different sections. It is not possible to calculate manually the different velocities vs time and pressure vs time pattern in each section. For a conservative assessment the maximum pressure and velocities are calculated.

dpm = aw * w1n/v =1160*17.01/0.001=197.3 bar

pmax = p+dpm = 100+197.3 = 297.3 bar

In a very unfavourable condition, the pressure may rise twice as high in the branch pipe, here pmax, br = p4R = 283.4 bar which is less than pmax above.

Calculation of Maximum Force

Let us select the section 3 where the maximum force is required when the wave has moved 1500 m from the valve end. The maximum force occurs during propagation of first wave through section 3.

dpmax = dp3 = 115.0e5 N/m^2 = 115.0 bar

When the wave just passes half the pipe section 3, the wave length is shortened, with x = 1500 m :

$$kw = [\,1 - \frac{w3n * x}{aw^*(aw-w3n)^*ts}\,] = 0.988 \quad ... \quad [4.09]$$

low = lo * kw = 1136 m [4.08B]

dpg = dp3/low = 0.101 bar/m

For a length of pipe lg = 10 m at a bend 1500 m from the valve, maximum surge force can be calculated as

Fm = Ah*dpg*lg = 37.0e3 N = 37.0 kN

In this example, impulse or dynamic load factor has not been used.

4.10 Dynamic Surge

4.10.1. Introduction

The surge force as given in Article 4.7 is the maximum force occurring in the piping sections. These are considered as conservative, but do not take any amount of reflection as has been described before. If the pipes and components are designed for these forces, the anchor and support loads may be very high and snubber may be required. So the forces calculated with the basic equation can be used to check the maximum values of surge, but for critical systems dynamic analysis is recommended. To estimate the structural response of piping system, additional knowledge about instationary flow, conditions of the pipe is required and suitable computer programme developed for dynamic surge should be used.

4.10.2. Equations

Equations are derived for dynamic surge analysis with computer programme from conservation theory applied to mass. With x as axial coordinate, the basic equation can be written as :

mass : $d\Gamma/dt = -\Gamma^*dw/dx = w^*d\Gamma/dx$

impulse : $dw/dt = -w^*dw/dx - 1/\Gamma^*dp/dx - lam/2d^*w\,|w|$

Energy : isentropic behaviour of fuuid plan $p/\Gamma k$ = a constant, which can be used instead of the exact equation. For water the equation reduces to

Γ = f(p)

The equations can be transferred to differential equations and solved by finite difference techniques (solution for time dt and space dx). They can also be solved by using direct integration method of Runge-Kutta fourth order with fixed and *variable steps*. In finite difference method the variation of density with change of pressure is considered. The variation of density with change of pressure can be ignored for incompressible fluids as water, but to consider the change of pressure surge for changed density and the velocity of sound for the medium direct integration method is applied. The numerical integration to get the force time function is done with a small time and space stepping. Typical values are :

time step dt = 0.001 s

space step dx = 1.0 m

Criteria for stability dx/dt = K * a

K = a constant, value between 1 and 2

4.10.3. Effect of Valve Characteristics

The instationary effect of flow through the pipe is dependable on valve characteristics. It can be seen from the basic equation that the pressure wave varies directly to the change of velocity which is influenced by the opening of the valve. Valve wide open gives maximum velocity and completely closed gives zero velocity. The total valve setting time and the position with time are important factors for the analysis. Valve characteristics are often defined as the function of friction factor f — value over the stroke. For small opening area the curve asymptotically tends to infinity which cannot be clearly defined and so cannot be profitably computerised. Therefore, the more usual α– values based on maximum flow area are used for analysis. This curve starts from zero and goes to maximum with full open valve. Characteristics for few widely used valves are given in Fig. 4.6.

α may be defined as :

$\alpha = \sqrt{(1/(1+f1))}$

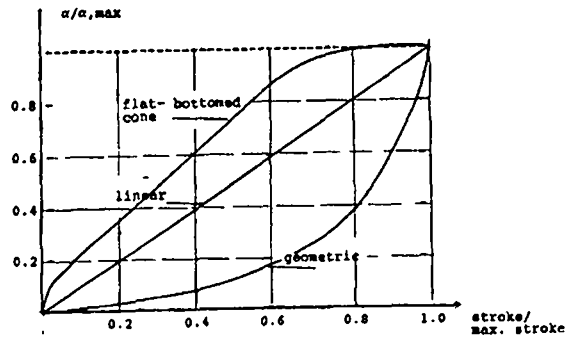


Fig. 4.6. ***Characteristics of Valve :***
Linear : high pressure reducer
Flat-bottomed : safety valves
Geometric : flow regulating valve,better for quick damping.

4.11. Flow Dynamic Model of Main Power Plant Piping

The condition of steam and the boundary conditions are important for modelling of high pressure piping system. The volume of the boundary vessels as superheater, reheater, etc, determines the amount of reflection of wave in the piping.

The main steam piping of a fossil power plant can be analysed independently because of critical mass flow in high pressure bypass valve. But hot reheat and cold reheat piping cannot be treated individually because part of the pressure surge generated either by non-return valve in cold reheat or by turbine control or stop valve in hot reheat travel through the reheater. So, to get the reflection effect from reheater the volume, length, diameter of reheater tubes, its pressure drop, etc, are generally modelled with hot reheat analysis. To model the cold reheat line with hot reheat, the full cold reheat up to non-return valve is modelled as equivalent pipe as the detailed force-time functions at bends of cold reheat are not required. The same procedure is taken with superheater, the waves are reflected from boiler drum or from within superheater has no significant influence on flow dynamics.

In modelling the main steam, constant flow of water to boiler drum can be assumed. The superheater volume must be included in the analysis as this influences the steam flow during turbine trip.

The piping system is subdivided into sections depending on change of direction. The cold reheat is modelled in the same way as hot reheat, only now equivalent pipe is taken for hot reheat line.

To suit the pressure drop of different sections, the friction coefficient is suited for geometry.

The volume of reheater can be divided in hot reheat and cold reheat headers together with the tubes between them. For simplicity, the same temperature as hot reheat can be taken for the reheater inlet pipes and header or cold reheat side. To get the effect of cold reheat temperature, the lengths of reheater tubes should be adjusted.

4.12. Input and Output Data of Flow Dynamic Analysis

The input and output data reassemble to those required for analysis with Joukowsky surge equations with only the difference that detailed data as given afterwards are required as input and the forces are obtained as a function of time. So, for every section of the system, the Force-Time history is obtained. Furthermore, the system pressure and mass flow function with time for superheater, reheater, valves and branches and other components are obtained.

The data required for the solution are given below ·

- Geometry data for pipe, bends and components : eg, lengths, bend angles to calculate the stretched lengths, diameters or, if the geometry is not known as in the case of turbine, the volume.
- Steady state data : Pressure, temperature, steam or water flow in steady state and friction factor. The pressure loss is calculated accurately or fitted to the different pressure and temperature given at every component.
- Valve characteristics including closing and opening with time lag of all valves with rate of mass flow.
- Boundary conditions : If the boundary conditions are not known exactly, then

approximate data are required so that the behaviour or the fluid in the system resembles the actual conditions.

4.13. Structural Analysis

The dynamic fluid flow in a piping system is not uniform. It varies in different legs depending on the configuration and mode of reflection. The dynamic surge can be analysed in the piping system by direct integration of different equations or by model time history method. Both the methods require high computer capacity for computation and are very sensitive to the input data and pipe geometry. For any change of geometry, for example, no rough estimation can be made and a new analysis is required. Therefore, a readily usable method similar to the spectrum-method for earthquake dynamic is normally used. This method needs less computation time, can be well predicted and remains within limits of uncertainties, whereas in time history method, the computation time being higher, sometimes part system is analysed to minimise the computation costs.

4.13.1 Mode Maximum Method

It has been seen that the force-time function and the dynamic pattern of the analysis has uncertainties. The exact time history needs fairly accurate data. The flow dynamic with the force-time function is imposed on the three dimensional piping system model and any assumption reflects directly to the time history analysis and deviations are very reflective in the high frequency area but not on the resonance. So, instead of time dependent resonance, spectrum maximum value for every eigenform can be used. For every eigenform, one spectrum for a small part of the system is created. The dynamic loads are imposed on the response spectrum of mass oscillator to which the load time functions, derived from flow dynamic calculation, is also imposed. The steps used to convert the time history to the spectrum method are :

- the forces-time function is imposed on a small mass oscillator with eigenfrequency function of the piping system
- the maximum amplitude of response is taken and plotted against fundamental eigenvalue f 1
- the other eigenfrequencies f2, f3 ..... fn are imposed similarly on the mass oscillator to get the complete responses : b = b(f), where b stands for acceleration
- to compensate for the uncertainties the response spectra, thus achieved, it is smoothened and the top widened (normal extension of peak is 10%)

4.13.2. Dynamic Load Factor in Programme

The dynamic load factor as used in semi-dynamic analysis has been described in article 4.8. But a more accurate calculated damping factor is used in dynamic analysis.

An oscillating mass m with force F(t) with a spring constant C in N/m and damping D in kg/s as in Fig. 4.7 is taken as free to vibrate in one direction x. The mass of spring is small compared to the oscillating mass. When the mass is put into motion, it starts to vibrate but the oscillation gradually diminished due to damping caused by air, friction or fluid resistance. When a body is vibrating in air or in a liquid and the velocity is small, the resisting force is nearly proportional to the velocity and nearly equal to c * x. The differential equation of motion can be written as :

$$m * \ddot{x} + d * \dot{x} + c * x = 0$$

where $\dot{x}$ and $\ddot{x}$ are the first and the second derivations of the displacement x with respect to time representing velocity and acceleration. The first part is mass matrix, the second one is damping matrix and the third is stiffness matrix.

With

$wn = \sqrt{C/m}$ $= 2*\pi*fn$ rad/s

$wd = \sqrt{wn*(1 - Df^2)}$ $= 2*\pi*fd$ rad/s

The damping factor

Df = D/Dc where the critical damping :

$Dc = 2 * \sqrt{C * m}$ ...

Then the dynamic load factor DLF :

DLF = x(t)max/x, static

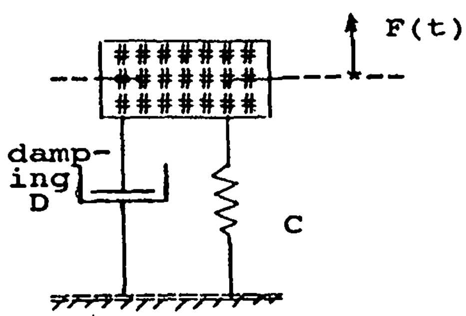


Fig. 4.7 : Mass oscillator

= x(t)max/Fm/c

= F(t)max/Fm ... [4.14]

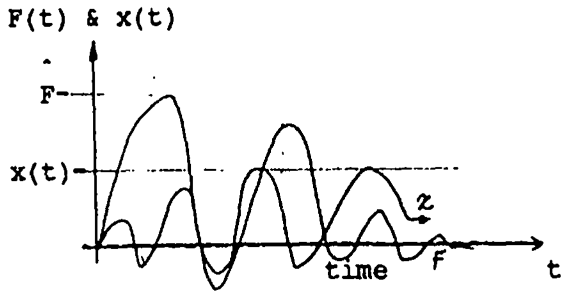


Fig. 4.8 : Dynamic force and acceleration

In general practical cases, the damping is small and the difference between wn and wd is a small quantity of second order. Assuming that the small damping does not affect the system, the period of free vibration :

$\tau = 1/fn = 2 * \pi * \sqrt{m/C}$

$\tau = 2 * \pi * \sqrt{x}$ s [4.15]

4.13.3. Analysis

The analysis can be divided into three parts :

- Determination of free-time functions separately and conversion to response spectra (DLF vs frequency) as described earlier.
- Piping system is divided into small lumped masses and eigenfrequencies and eigenvalues are determined by transmission method (matrix include all line masses) and stresses or loads are computed for each eigenvalues of the system.

The influence of change of fluid dynamic parameter can be judged better by splitting the calculation in two parts. The results of flow dynamic can be compared with the data of similar system to enable the designer to make some prediction for the end results.

The resonance derived from the dynamic analysis cannot be added arithmetically; they are added by root-mean square (RMS) method, which absorbs most of the uncertainties in geometry but the loading variations are compensated only by using maximum dynamic spectrum.

For higher frequencies the DLF becomes almost equal to 1 and the dynamic analysis is carried out for DLF = 1. The higher modes are considered in rest mode method.

Damping values are to be chosen according to Regulatory Guide 1.6.1 and the normal value is 3 to 4 per cent.

4.14. Example 1

4.14.1. System and Data

The dynamic flow model of a main steam of a 500 MW plant has been shown here. The salient calculation data have been given and results presented with a discussion.

Let us take a typical main steam 2 –1 – 4 piping system. The system sketch is given in Fig. 4.9.

The data for the example described here are as follows :

BD	Boiler Drum
SH	Superheater
HPBP	high pressure bypass
HPTV	high pressure turbine valve
HPT	high pressure turbine

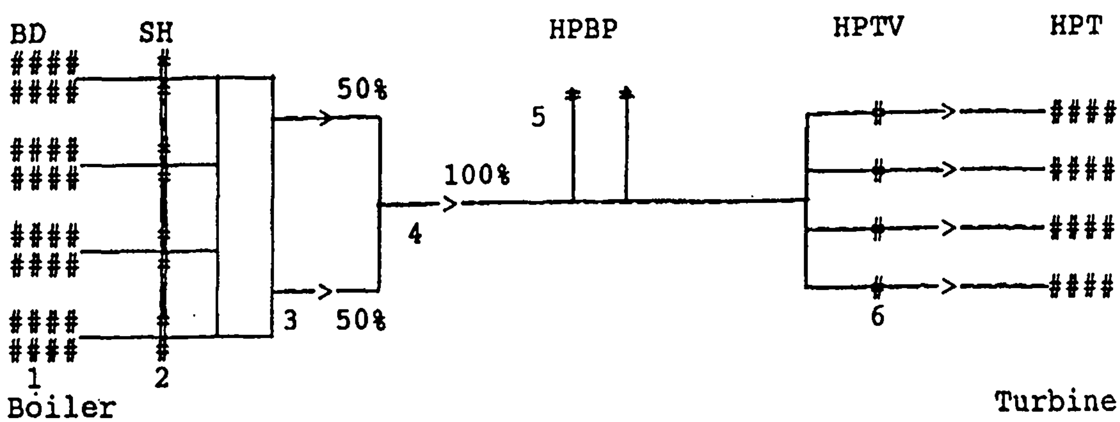


Fig. 4.9 : Main steam flow model of a 2 – 1 – 4 piping system

Pipe Data

Section	no.	d(m)	l(m)	m(kg/s)	p(bar)	T (C)	v(m^3/kg	w(m/s)
1 BD	-	29 M^3			190.0	361.4	saturated	—
2 SH	-	135 M^3		458.0	175.1	540.0	0.0189	—
3 BL	2	0.35		229.0	175.1	537.0	0.0188	44.7
4 SL	1	0.47		458.0	172.0	537.0	0.0192	50.7
5 HPBP60%	2	0.18		139.0	170.0	537.0	0.0194	106.2
6. TL	4	0.25		114.5	167.0	537.0	0.0198	46.2
7. HPT	1	1 M^3		458.0	166.7	537.0	0.0199	—

This system has been analysed with computer programme and the results obtained have been plotted and discussed.

4.14.2 Plots

The different curves showing the time history of pressure, mass flow forces obtained from the computer calculation are shown in Fig. 4.8 – 4.13 and explained in the following articles. The different observation points of the Fig. 4.9 are marked with numbers which correspond to the numbers in the plots.

Turbine trip : (Fig. 4.10) The opening of high pressure bypass valve and closing of high pressure turbine valve are taken as linear. The steam flow closing characteristic of HPT-valve is t = 0.05 seconds and the opening characteristics of HPBP-valve is t = 1.5 seconds with a starting delay of 0.12 seconds.

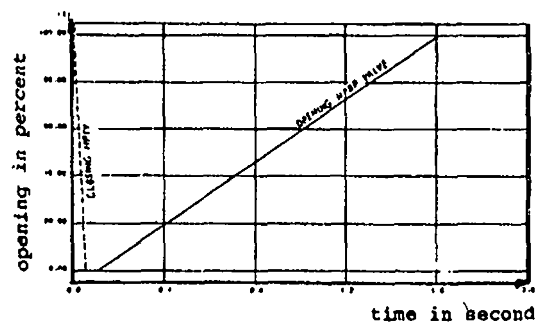


Fig. 4.10 : Turbine trip with HPBP valve opening

- Pressure plots (Fig. 4.11 – 4.12) At piping inlet or outlet, T-pieces or valves, the pressure changes rapidly due to change of state, so at these places the behaviour of traveling fluid can be studied with the pressure-time plot. The following are the main characteristics of pressure plots after closing of valve :
  - pressure increases
  - line packing effect is observed due to pressure compensation in piping with high friction losses.

- pressure amplitude decreases due to friction
- oscillation of pressure till it dies out due to friction

As a result of quick closing of HPTV (no. 6), the pressure before the valve rises very quickly, dp = 17.6 bar in t = 0.06 second. This surge pressure increases further to 192.0 bar due to steady state pressure difference between turbine valves and SH-outlet. The pressure in boiler drum increases slowly with a time lag of t = 0.8 second (no. 1). The pressure increases at SH (no. 2), header outlet (no. 3) after a time lag of 0.5 second and 0.3 second. Y-piece (no. 4) and HPBP-branch (no. 5) show the intermediate effects.

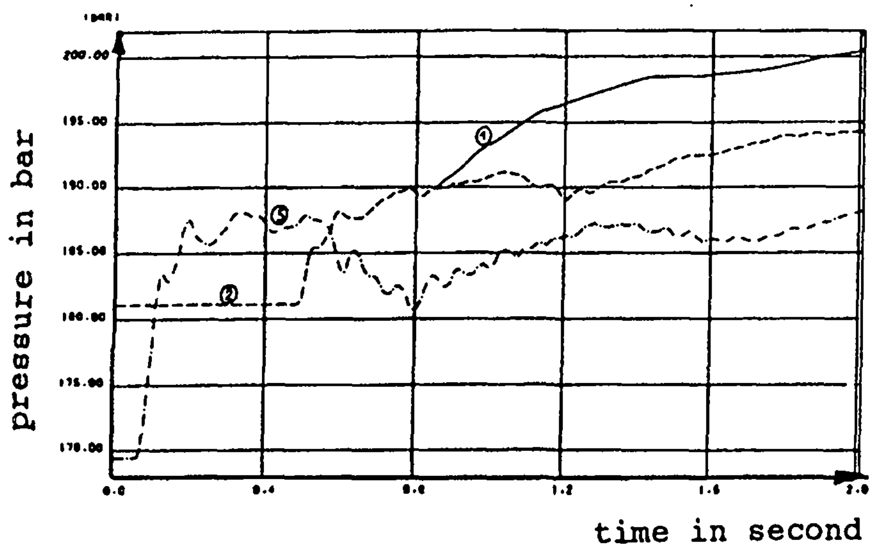


Fig. 4.11 : Pressure vs time at boilser drum, superheater and HPBP.

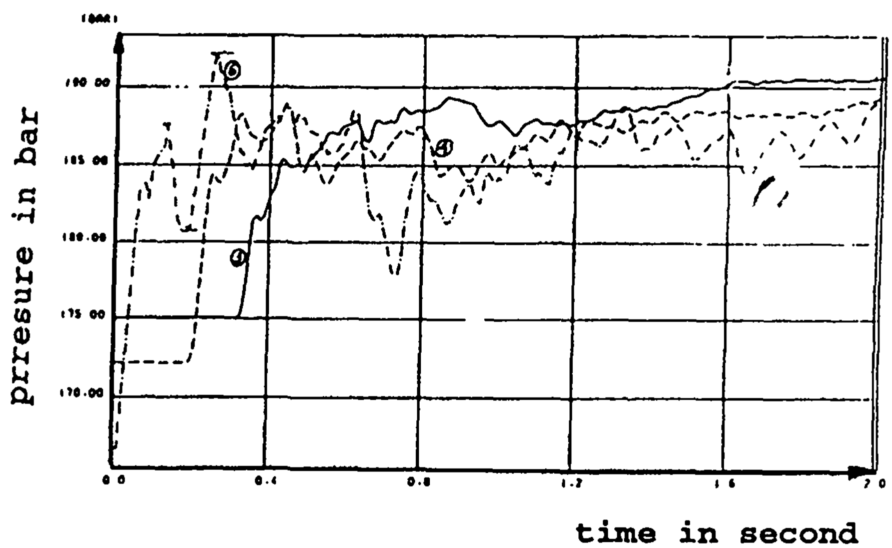


Fig. 4.12 : Pressure vs time at boiler legs, single line and HPTV

- Mass flow plot : (Fig. 4.13) The mass flow with time shows the effects of working of valves and similar components.

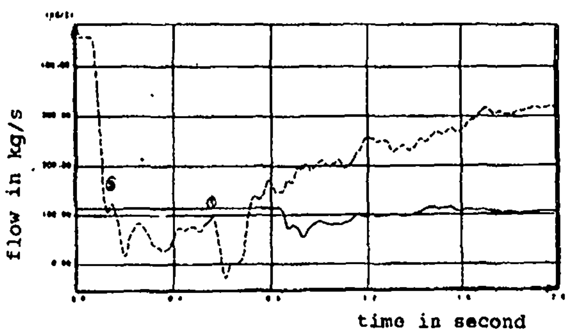


Fig. 4.13 : Mass flow vs time at boiler drum and HPBP

At boiler drum the flow is uninterrupted up to 0.9 second and after a small interruption comes back to steady state (no. 1). The flow before HPBP-branch (no. 5) has a small time lag and then it falls down rapidly and slowly regains almost steady state.

It is interesting to note that the surge pressure does not reach the spring operated safety valve blow pressure, 196.0 bar in this project and the pilot operated safety valve gets a signal at 1.5 seconds but after this time the turbine trip fluid surge effect has diminished and hence in occasional load case the safety valve blowing and surge force are not required to be considered simultaneously. This consideration is possible due to a dynamic fluid flow analysis.

- Force plots (Fig. 4.14 – 4.15) These plots with their maximum value can be compared with Joukowsky values. The forces are due to pressure difference, friction and momentum change. The forces, when acting in the flow direction, have positive sign.

The force time history for the marked points at superheater and at the entry of single line are given in these figures. Following are observations :

a) The force functions are in line with thumb rule;

b) The pick is always before 20 Hz;

c) The force amplitude dies down after 0.5 second.

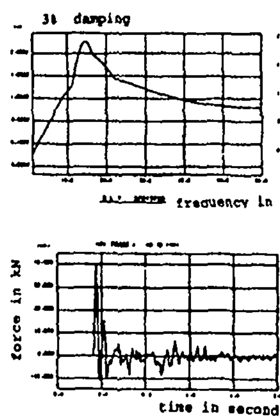


Fig. 4.14 : Force vs time at superheater

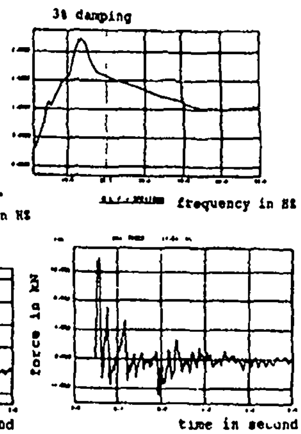


Fig. 4.15 : Force vs time at single line

4.15. Example 2

A 2 – 1 – 2 main stream system has been taken and a few system legs as shown in the isometric have been marked for manual calculation. This system has been analysed with computer programme. The results are compared in a table.

ts = 0.7 s

Sections calculated manually :

04 – 05
05 – 11
11 – 13

41 – 43
43 – 45

56 – 60
60 – 63

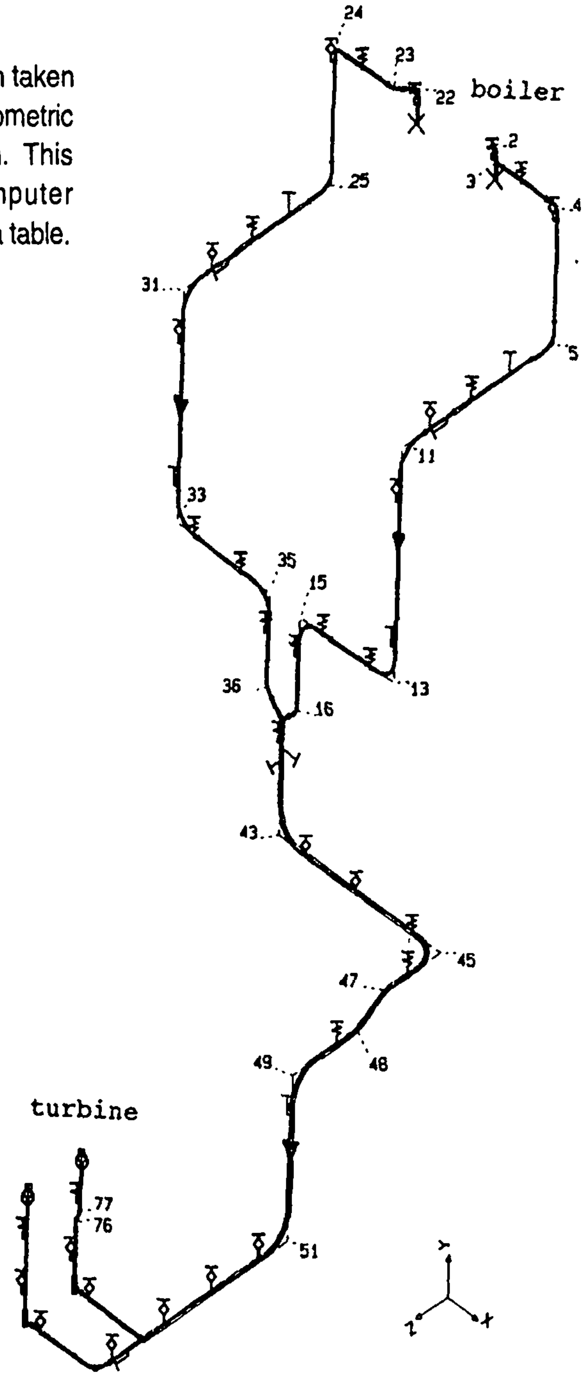


Fig. 4.16 : Main Steam System

4.15.1. Calculation table

No.	Ab	Description	steam data			pipe data			pressure surge data								Forces		
			P bar	T °C	v m^3/kg	No	di m	l m	f -	f1 -	m	w m/s	a m/s	aw m/s	lo m	dpm bar	F,manual kN	F, progr kN	t,max s
1	BD	Boiler drum	190,30	361	saturated	-	21 m^3	-	-	-	-	-	-	-	-	-	-	-	-
2	SH	Superheater	174,60	540	0,0190	2	0,700	240,0	520,00	-	232,70	-	-	-	-	-	-	-	-
2	BL	Boiler legs	174,60	540	0,0190	2	0,330	90,0	2,90	0,0125	232,70	51,69	655,19	654,00	-	-	-	-	-
		04 - 05	174,00	540	0,0190	2	0,330	13,0	0,40	0,0125	232,70	51,69	654,06	653,00	42,09	17,77	46,9	37,8	0,33s
		05 - 11	173,00	539	0,0191	2	0,330	19,0	0,60	0,0125	232,70	51,97	653,89	653,00	42,07	17,77	68,6	57,5	0,31s
		11 - 13	171,90	539	0,0193	2	0,330	22,0	0,20	0,0125	232,70	52,51	655,21	654,00	42,10	17,79	79,5	57,0	0,27s
4	SL	Single line	170,20	538	0,0194	1	0,450	85,0	2,60	0,0118	465,30	56,76	653,65	653,00	-	-	-	-	-
		41 - 43	169,70	538	0,0195	1	0,450	11,0	0,70	0,0118	465,30	57,05	654,37	654,00	41,79	19,13	80,1	61,9	0,22s
		43 - 45	168,60	537	0,0196	1	0,450	21,0	0,20	0,0118	465,30	57,34	653,92	653,00	41,70	19,10	153,0	103,8	0,19s
6	TL	Turbine legs	166,20	537	0,0199	2	0,330	26,0	4,70	0,0125	232,70	54,14	654,20	653,00	-	-	-	-	-
		56 - 60	165,00	537	0,0201	2	0,330	6,0	0,20	0,0125	232,70	54,69	655,10	654,00	41,95	17,79	21,8	22,0	0,11s
		60 - 63	165,00	536	0,0201	2	0,330	10,0	1,10	0,0125	232,70	54,69	655,10	654,00	41,95	17,79	36,3	28,2	0,10s
7	HPT	HP Turbine	164,90	536	0,0200	-	2,7 m^3	-	-	-	465,30	-	-	-	-	-	-	-	-

4.15.2. Plots

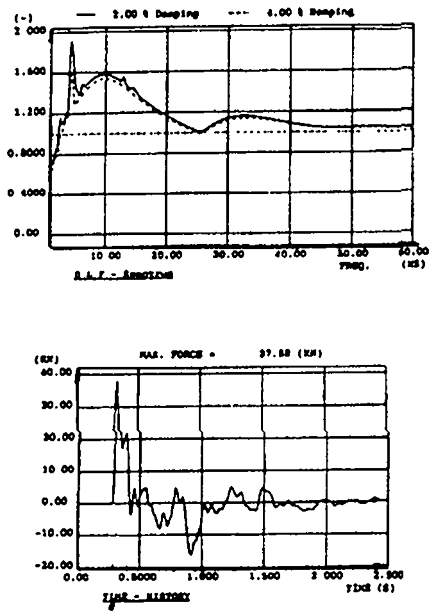


Fig. 4.17 : Forces for 04 – 05

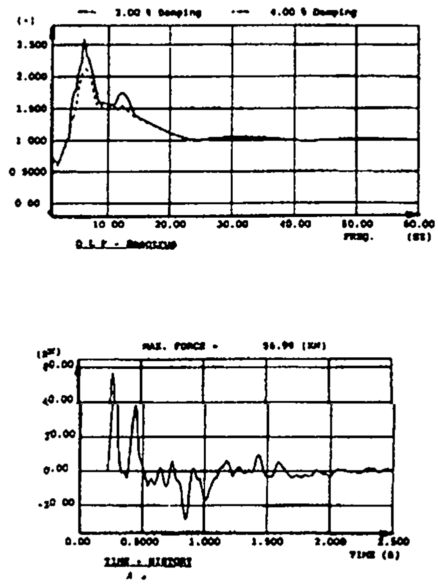


Fig. 4.18 : Forces for 11 – 13

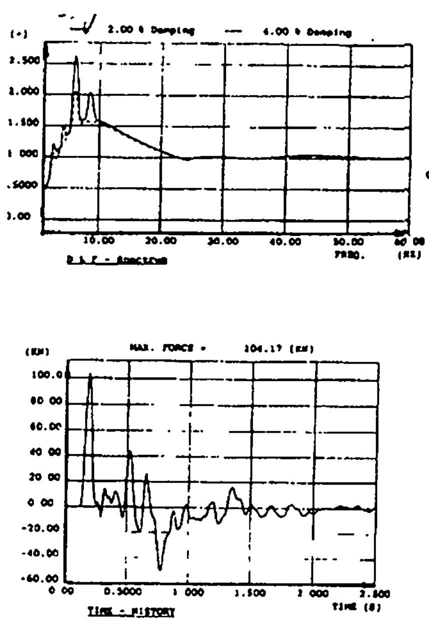


Fig. 4.19 : Forces for 43 – 45

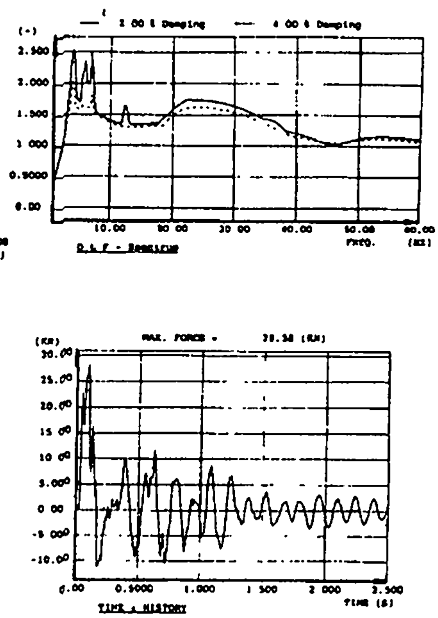


Fig. 4.20 : Forces for 60 – 63

4.15.3. Observations

- The steam pressure at different sections have been taken from the computer output and can be roughly checked from the friction factor, length and other data tabulated.

- The boiler drum, superheater and turbine have been modeled in the programme.

- The manually calculated forces are higher than those calculated by the dynamic programme.

- In some cases though the maximum forces are comparable, the acting phase difference results lower loading.

- Section 04 – 05 : The surge force characteristics are similar to those of example 1. The DLF is 1.6 with 4% damping.

- Section 11 – 13 : The section being relatively long, the simple calculation not considering different flow characteristics give very conservative values.

- Section 56 – 60 : The maximum values in both the calculations are almost the same, because the section length is small.

4.16. Snubber section

Snubbers protect complex piping system from excessive dynamic loads overstressing during seismic or other abnormal disturbances having an acceleration of 0.02 * g or greater. At the same time, the pipe is free to move as required for thermal expansion during abnormal operation, load change, start-up or any such cases.

Snubbers are mechanical or hydraulic operated. Mechanical snubbers are profitable for low loads, approximately near 10 kN. For high loading, hydraulic snubbers are taken.

Few usable information about snubber design are given below :

1. In the piping analysis the stiffness of snubber and support construction should be used.
2. The snubber must be so mounted that it can function under a small load.
3. Friction loss in snubber should not be more than 1.5% of rated load.
4. It should work by an acceleration of 02 mm/S^2 at 20°C.
5. The functional velocity is 4 – 6 mm/s.
6. Total movement at a frequency up to 35 Hz : at 20 °C < 4 mm and at 150 °C < 5 mm.

Before starting the exact dynamic analysis the impact forces due to valve closing are estimated by the equation for axial surge force given in article 4.7.

Pipe sections with F > 30kN is taken as a snubber requirement to start computation. When the exact location of snubber has been fixed (depending on the structure and the building), dynamic analysis is performed with snubber points as limit stop with a very high spring constant (usually 110^7 for tanslatory and 10^10 for rotatory stiffness.

The results of the dynamic analysis are checked and snubber position and requirement are checked once again. According to the requirement they are shifted, added or eliminated.

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