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Fluid transients in pipes Estimation of maximum pressures and forces in steam lines

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THE PREPARATION OF THIS DATA ITEM

This Item was developed in the Noise and Structural Dynamics Group on behalf of the Internal Flow Panel. The work of the permanent professional staff of the Engineering Sciences Data Unit on this particular Item was monitored and guided by the following Working Party:

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FLUID TRANSIENTS IN PIPES ESTIMATION OF MAXIMUM PRESSURES AND FORCES IN STEAM LINES

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FLUID TRANSIENTS IN PIPES

Estimation of Maximum Pressures and Forces in Steam Lines

1. NOTATION AND UNITS

Any consistent system of units may be used in the equations in this Item.

		<i>SI</i>	<i>British</i>
A	internal cross-sectional area of pipe	m^2	ft^2
A_V	partially-open valve flow area	m^2	ft^2
A_{Vo}	fully-open valve flow area	m^2	ft^2
a	speed of propagation of pressure waves (wave speed or speed of sound) in fluid	m/s	ft/s
C	steady-state non-dimensional valve area parameter (see Equation (4.4))	—	—
C_d	discharge coefficient of partially-open valve	—	—
C_{do}	discharge coefficient of fully-open valve	—	—
F	axial force along pipe section	N	lbf
f	Fanning friction factor	—	—
i	integer, referring to i^{th} section of pipeline	—	—
L_i	length of i^{th} section of pipeline	m	ft
L	pipe length	m	ft
M_0	Mach number of steady-state flow ($= V_0/a$)	—	—
p_0	initial pressure in a fluid (absolute)	N/m^2	$^*\text{lbf/ft}^2$
p_d	pressure drop across partially-open valve	N/m^2	$^*\text{lbf/ft}^2$
p_{do}	steady-state pressure drop across fully-open valve (see Equation (4.5))	N/m^2	$^*\text{lbf/ft}^2$
Δp	maximum pressure change value	N/m^2	$^*\text{lbf/ft}^2$
$\Delta p_1,$ $\Delta p_2,$ $\Delta p_3,$ <i>etc.</i>	instantaneous pressure change values (see Section 3.3)	N/m^2	$^*\text{lbf/ft}^2$

For footnotes see end of Notation section.

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t	time	s	s
t_c	elapsed time for complete closure of valve	s	s
R	gas constant	J/kg K	ft *lbf / [†] slug K
T	absolute temperature	K	K
V_0	steady-state flow velocity	m/s	ft/s
V_v	flow velocity at the valve throat	m/s	ft/s
ΔV	flow velocity change	m/s	ft/s
γ	ratio of specific heat capacity of fluid at constant pressure to that at constant volume (typically 1.3 for superheated steam)	—	—
τ_i	non-dimensional time scale for i^{th} pipe section (see Equation (4.6))	—	—
ρ_0	steady-state fluid density	kg/m ³	[†] slug/ft ³

* 1 lbf/ft² = 1/144 lbf/in²
† 1 slug = 32.174 lb (pounds mass)

2. BACKGROUND

2.1 Notes

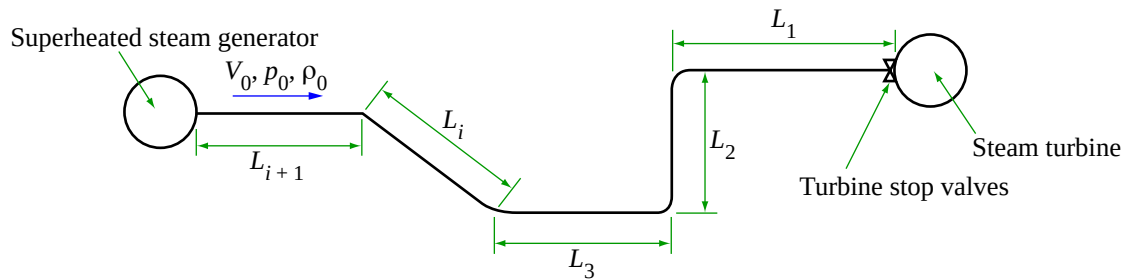
This Item is one in a series of ESDU Data Items concerning fluid transients in pipes. The magnitudes of the pressure changes in a compressible fluid following a change in the flow rate may be estimated using the analysis methods given in this Item. These methods are particularly applicable to steam systems. A graphical method is presented that provides a simple means of estimating the changing pressures and forces following valve closure in simple one-pipe lines such as steam turbine lines. A computer program is also provided that allows a full analysis of the pressure changes, and hence pipe force changes, in such systems following valve closure. The data in this Item are only applicable to compressible fluids. Data for the estimation of pressure changes in incompressible fluids (liquids) and guidance on various methods of liquid surge suppression are given in ESDU 83046, 84013, 84038, 85009 and 85044.

Fluid transients are the pressure changes that propagate through a fluid contained in a piping system following a change in the flow rate, and are commonly referred to as water hammer in liquid-filled systems. Other commonly used terms include pressure surge and hydraulic transients. In steam lines, these pressure changes are commonly known as steam hammer. The most common cause of significant steam hammer pressure changes in steam lines is the rapid closure of turbine stop valves. Such pressure changes may be sufficient to cause damage to the system by, for example, exceeding the maximum safe pressure of the system or causing fatigue failure of the piping or piping supports following repeated pressure cycles. As steam pipelines in turbine systems incorporate flexible connections to accommodate thermal expansion over wide temperature ranges, transient pressure changes may induce large scale vibrations as a result of large pressure differentials across sections of piping.

With the tendency towards the design of more efficient piping systems associated with the stringent requirements of the regulatory codes, it is necessary to determine pressure surge effects in order to gauge the integrity of the piping system and the piping supports. The methods presented in this Item allow simple, reliable estimates to be made of surge effects at the initial design stage to establish the need for more detailed investigations. Information concerning investigations into transient effects in steam-filled systems is given in References 9, 11, 17 and 18.

2.2 Scope and Applicability of the Item

This Item provides both graphical and computerised methods that may be used to estimate the magnitude of surge pressures in steam lines following a change in the flow rate. The graphical method for the estimation of pressure change and time-dependent piping forces following rapid closure of a turbine stop valve is applicable to one-line systems such as that shown in Sketch 2.1. The method used to generate the data is outlined in Appendix A. In deriving the graphical data, it was assumed that the fluid obeys the perfect gas law, which limits the applicability to superheated steam and other gases that follow this law. The data are therefore not applicable to saturated steam and wet vapours. In this Item, the term “steam” refers to superheated steam and other appropriate gases.



Sketch 2.1 One-line steam turbine system

In the one-pipe system shown in Sketch 2.1, superheated steam flows from the steam generator to the turbine, which is protected by a stop valve on the upstream side. It is assumed that the pipe system between the steam generator and the stop valve is of constant cross section, without branches, and that all reflections of pressure waves from pipe bends, *etc.* are negligible. It is further assumed that all pressure and density changes follow adiabatic processes, and that all friction effects can be neglected. In addition, all wave steepening effects (see Section 3.3) are neglected. The graphical data allow investigation of the effect on the pressure changes of varying the stop valve closure characteristics. Further guidance on these matters is given in Section 4.

The computer program introduced in Appendix B allows similar systems to be investigated without recourse to all of the simplifying assumptions that are necessary for the graphical presentation. The full scope of the program is described in Appendix B.

3. SURGE PRESSURES AND FORCES FOLLOWING VALVE CLOSURE

3.1 General

In a system such as that shown in Sketch 2.1, the magnitudes of the pressure changes in the fluid on the upstream side of the valve due to valve closure are dependent upon

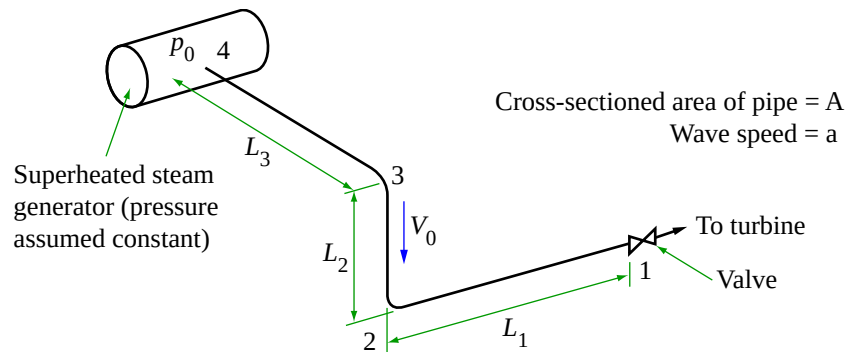
- (i) the ratio of the time taken for complete valve closure to the time taken for a pressure wave to propagate from the location of the valve to the constant pressure point (steam generator) and back to the valve,
- (ii) the way in which the particular valve shuts (*i.e.* the valve closure characteristic),
- (iii) the initial flow conditions of the fluid, and
- (iv) the initial pressure drop across the fully-open valve.

The time taken for a pressure wave to propagate from the turbine stop valve to the constant pressure point and back to the valve is known as the 'pipeline period', and is equal to $2L/a$, where L is the distance along the axis of the pipeline (including pipe bends, *etc.*) from the valve to the constant pressure point and a is the speed of propagation of the pressure waves, or wave speed. For steam, the wave speed is a function of pressure and density (or temperature) and is typically in the range 400 to 800 m/s (1300 to 2600 ft/s). Data for the estimation of wave speed in steam are given in ESDU 83046 and are reproduced for convenience in Section 4 and Figure 1.

As the valve closes, the flow velocity through the valve decreases as the pressure on the upstream side of the valve increases. The compression wave that is formed on the upstream side of the valve propagates along the line, at the speed of sound in the fluid, or wave speed, a .

3.2 Representation of Pressure Changes and Resulting Maximum Forces

To illustrate the changes in pressure following valve closure, consider the system shown in Sketch 3.1. The pipe indicated has three sections, and the total pipe length L is equal to the sum of the section lengths, L_i (where $i = 1, 2$ and 3 for this case). It is assumed that reflections of pressure waves from pipe bends, *etc.* are negligible. At time $t = 0$, the pressure at all points in the line is considered constant, as pressure changes due to elevation changes are minimal and therefore neglected. This pressure is the pressure at the steam generator, p_0 , (neglecting friction and potential heads). The flow velocity, V_0 , and fluid density, ρ_0 (and hence wave speed, a), are also considered constant along the line.

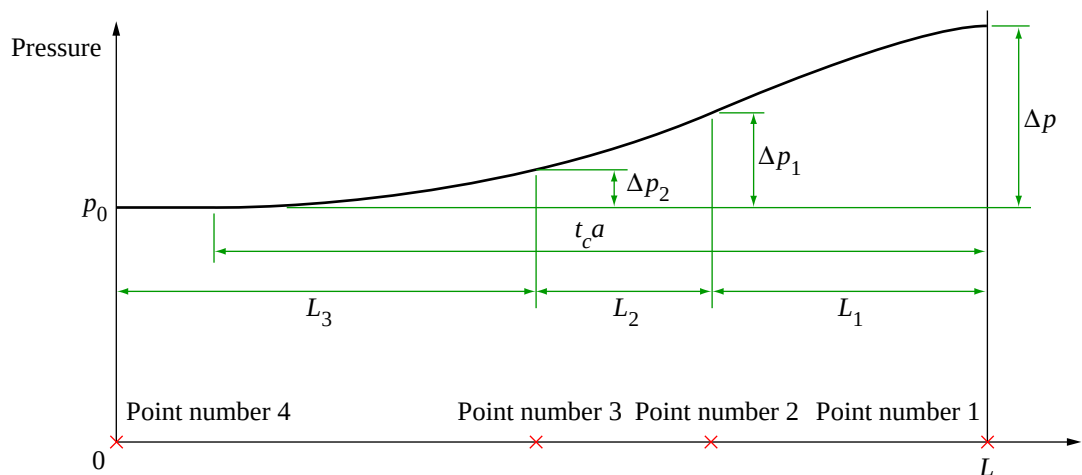


Sketch 3.1 Example system

It is assumed that the valve closes in a time, t_c , that is less than the time taken for pressure waves to propagate from the valve to the steam generator, L/a . The instantaneous magnitude of the pressure at the valve during valve closure is partly dependent upon the time taken for the valve to close and partly upon rate of change of effective valve flow area, which in turn is dependent upon the valve closure characteristic. The effective valve flow area is the open area of the valve modified by an appropriate discharge coefficient. As the valve closes, the pressure at the upstream side of the valve increases and the resulting compression wave propagates along the line at the wave speed, a . The compression wave has a characteristic shape, which is dependent upon the initial steam conditions and the valve closure characteristic.

In Sketch 3.2, the shape of a typical compression wave at particular time $t = t_c$ is superimposed on a longitudinal diagram of a piping system. At this instant, the flow velocity has fallen to zero at the valve and the pressure on the upstream side of the valve is at the maximum value of $p_0 + \Delta p$, where Δp is the maximum value of the pressure change. The instantaneous pressure at any location along the line, such as the four points shown in Sketch 3.1, may be determined if the magnitude and shape of the pressure wave, giving the pressure distribution, are known. The pressures at time $t = t_c$ at each of the four points, calculated using the pressure distribution shown in Sketch 3.2 are given in Table 3.1. Due to the pressure difference along a section of the pipeline, out of balance forces act along the axis of the pipe section that vary with time when the pressure difference changes as the wave propagates down the line. These forces are given by the product of the mean value of cross-sectional flow area along the specified pipe section and the pressure difference across it. The values of the forces in each section of the pipeline shown in Sketch 3.1 are also tabulated.

The maximum value of the force across any particular pipe section may occur at a time other than t_c . This maximum is obtained when the pressure difference along the section of the pressure wave under consideration is at a maximum value. This may be determined using the data presented in Section 4. Whilst the forces are generally short-lived, their duration increases over a section of pipe with increasing section length. It may be necessary in some systems to consider the frequency of the forces in order to predict the structural response of the pipework. The graphical method presented in Section 4 does not allow the frequency of pressure and force fluctuations to be predicted directly as it is dependent upon the locations of points of pressure wave reflection relative to the points of interest. This information may be obtained using the computer program. Guidance on the response of pipe systems to transient pressure changes is given in Derivations 1, 2 and 4 and References 7, 13, 14, 19, 20 and 28.



Sketch 3.2 Pressure wave shape along pipe length at time $t = t_c$

TABLE 3.1 Maximum Pressure and Force Values for Example System

Point No.	Pressure
1	$p_0 + \Delta p$
2	$p_0 + \Delta p_1$
3	$p_0 + \Delta p_2$
4	p_0

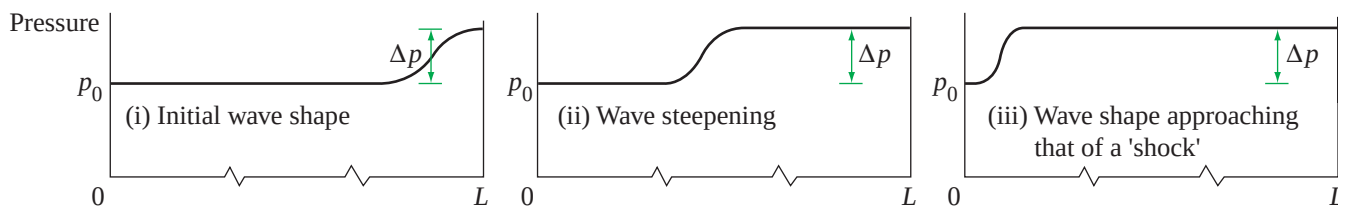
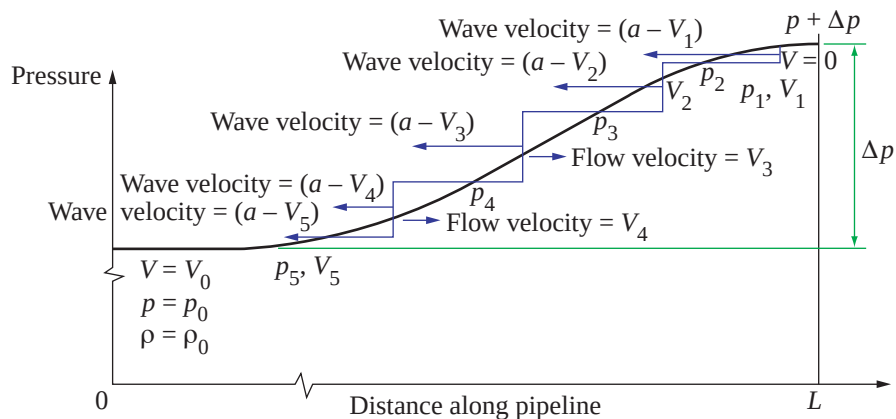
Pipe Section No.	Pressure difference	Force
1	$\Delta p - \Delta p_1$	$A(\Delta p - \Delta p_1)$
2	$\Delta p - \Delta p_2$	$A(\Delta p - \Delta p_2)$
3	Δp_2	$A\Delta p_2$

(i) Maximum pressure values

(ii) Maximum force values

3.3 Wave Steepening Effects

In pipelines where the length of a compression wave ($= t_c a$) is less than about a quarter of the straight pipe length, wave steepening effects are likely to be significant. Wave steepening is the tendency for a smooth wave to steepen until it becomes a ‘shock’. This phenomenon may be explained as follows. Consider a pressure wave broken into a number of segments, as shown in Sketch 3.3. For a certain valve closure characteristic, the smooth wave shown in the sketch is produced. At time $t = 0$, the flow velocity is V_0 and the pressure p_0 .



Sketch 3.3 Wave-steepening representation

As the compression wave propagates along the line from the closing valve to the steam generator, the pressure at each point increases as the flow velocity decreases. Hence, pressure and flow velocity gradients exist along the pipeline. In the system shown in Sketch 3.2, for example, the flow velocities at each of the four points differ such that, at time $t = t_c$, $V_1 (= 0) < V_2$, $V_2 < V_3$ and $V_3 < V_4$. However, at each point, the relative speed of propagation of pressure waves increases as the flow velocity decreases, with the result that succeeding increments of the wave front 'catch up' with preceding increments. Consequently, the wave front steepens, increasing the pressure difference across a given section of pipe. However, this effect is reduced by dissipation of flow energy by friction, and is usually unimportant after the first few pressure wave cycles. The effects of wave steepening can be investigated using the computer program introduced in Appendix B.

4. ESTIMATION OF PRESSURE CHANGES AND STEAM HAMMER FORCES FOLLOWING VALVE CLOSURE

The data presented in this Section enable estimates to be made of the maximum pressure changes in a compressible fluid represented as an ideal gas, contained in a single pipe system such as that shown in Sketch 4.1, following closure of the downstream valve. The maximum pressure difference across any section of pipeline may be calculated, and hence the maximum axial force across that section of pipe may be estimated. The magnitudes of pressure changes are dependent upon the initial conditions of the flow, the pressure drop across the valve at steady-state, and the closure characteristic of the valve. The data are applicable to systems where the valve closure is 'rapid', i.e. the time for complete valve closure, t_c , is less than the time taken for the pressure waves to propagate from the valve to the upstream constant pressure point. This time is equal to L/a , where L is the pipe length and a is the wave speed in the fluid. The initial conditions are represented by the parameter M_0 , which is the Mach number of the flow, and is given by

$$M_0 = \frac{V_0}{a}, \quad (4.1)$$

where V_0 is the initial flow velocity.

For the estimation of wave speed it is assumed that the perfect gas laws apply, and the wave speed is given by the equation*

$$a = \sqrt{\frac{\gamma p_0}{\rho_0}}, \quad (4.2)$$

where γ is the ratio of the specific heat capacity of the fluid at constant pressure to that at constant volume, and p_0 and ρ_0 are the absolute pressure and density of the fluid at steady-state conditions. The density, ρ_0 , may be obtained for a given temperature and pressure using steam tables such as those presented in ESDU 68010. Since, for an ideal gas, $p/\rho = RT$, the wave speed may also be given by the expression

$$a = \sqrt{\gamma RT}, \quad (4.3)$$

where R is the appropriate gas constant and T is the absolute temperature of the fluid (in kelvin). Some data from which an estimate may be made of the wave speed in compressed gases are given in ESDU 83046 (Reference 23). The wave speed in steam at temperatures up to 800°C (1472°F) and pressures up to 200 atmospheres may be estimated using Figure 1. These data were calculated using Equation (4.2) and the standard Tables of the thermodynamic properties of steam (e.g. ESDU 68010), and show very good agreement with available experimental values. The ratio of specific heat capacities, γ , may be estimated using†ESDU 00.01.08.

* The general equation for the wave speed in a liquid or gas contained within an elastic conduit is usually expressed in the form

$$a = \left[\rho \left(\frac{1}{\gamma K} + \frac{1}{A} \frac{\partial A}{\partial p} \right) \right]^{-1/2},$$

where K is the bulk modulus of the fluid, and the term $(1/A)(\partial A/\partial p)$ is the variation of the pipe area strain with the internal pressure. For compressible fluids, this pipe wall elasticity term is neglected and the bulk modulus K is equal to the pressure p . Hence, this equation reduces to Equation (4.2).

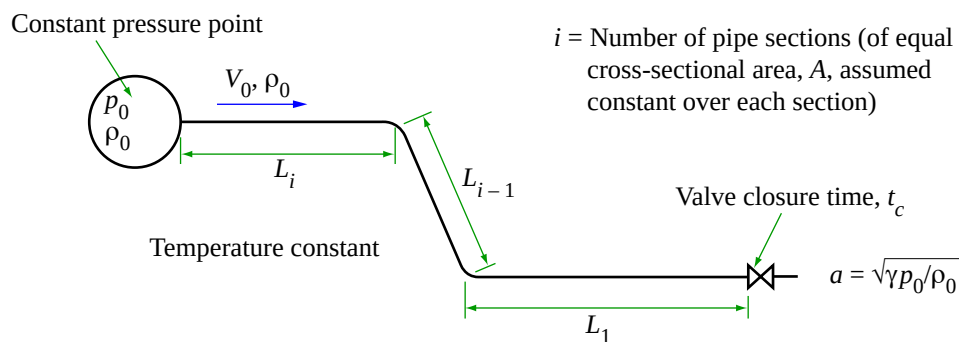
† 'Specific heat capacities and their ratio as functions of temperature for several common gases' in the Aerodynamics Series.

The steady-state non-dimensional valve area is represented by the parameter

$$C = \frac{C_{do} A_{Vo}}{A}, \quad (4.4)$$

where C_{do} and A_{Vo} are the discharge coefficient and the cross-sectional flow area of the fully open valve respectively, and A is the pipe flow area. Assuming incompressible flow through the valve (which is valid for values of M_0 less than about 0.3), the parameter, C , is related to the steady-state pressure drop across the fully-open valve, p_{do} , by the equation

$$C = V_0 \sqrt{\frac{\rho_0}{2p_{do}}} \quad (4.5)$$



Sketch 4.1 System model

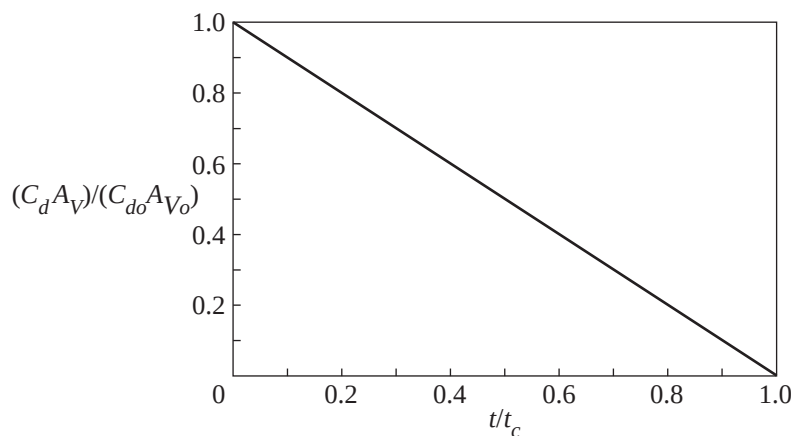
The following assumptions are necessary for the graphical presentation.

- (1) The flow is frictionless and the pipe has constant cross-sectional area, so that the pressure at all points in the line is the same at steady-state and the magnitudes of the pressure changes are not affected by pipe wall friction.
- (2) All reflections of pressure waves from sources of minor pressure losses are neglected. The shape of the pressure wave is not altered by reflections from changes of diameter.
- (3) The pressure in the steam turbine at the downstream end remains constant over the period of valve closure (due to the inertia of the turbine).
- (4) Pressure and density changes follow an adiabatic process (*i.e.* p/ρ^γ is constant), as all flow changes are too rapid for external heat transfer to be significant.
- (5) Wave steepening effects are negligible (see Section 3.3).
- (6) The ratio of the steady-state flow velocity, V_0 , to the wave speed, a , is small (typically $M_0 (= V_0/a)$ is less than 0.3).
- (7) The ratio of the specific heat capacity of the fluid at constant pressure to that at constant volume, γ , is equal to 1.3, typical of superheated steam.

Data are presented in Section 4 that allow pressure changes and piping forces to be estimated following closure of valves with closure characteristics that vary linearly with closure time (see Section 4.1) and also for valves with other forms of valve closure characteristics (see Section 4.2). The valve closure characteristic defines the ratio of the effective flow area of the valve, $C_d A_V$, at non-dimensional time t/t_c to the effective flow area of the valve, $C_{do} A_{Vo}$ at steady-state conditions, where t_c is the time taken for the valve to close completely. A linear variation of the valve closure characteristic with time is unlikely to occur in practice, and it must be noted that pressure change estimates made using the data presented in Section 4.1 are likely to be lower than those made using a more realistic valve closure characteristic. Therefore, if the estimates made using the linear valve closure characteristics of Section 4.1 are close to acceptable limits, it is recommended that the effect on these estimates of varying the valve closure characteristic is investigated using the data presented in Section 4.2. These data were derived using information presented in Derivations 1, 4 and 5.

4.1 Procedure for Estimation of Pressures and Forces Due to Closure of Valves with Linear Variation of Valve Effective Area with Time

Data are given in this Section that allow estimates to be made of the pressure changes at any location along a pipeline such as that shown in Sketch 4.1 following closure of a valve with valve characteristic that varies linearly with time, as shown in Sketch 4.2.



Sketch 4.2 Valve closure characteristic

For closure of valves with valve closure characteristic as shown in Sketch 4.2, pressure change data are presented in Figures 2a to 2e for a range of values of C and M_0 . The procedure for using the data given in these figures to estimate the maximum pressure difference and piping forces across any section of the pipeline is as follows.

- (1) Calculate the wave speed in the fluid using Equation (4.2),

$$a = \sqrt{\frac{\gamma p_0}{\rho_0}},$$

or, for steam, using Figure 1.

- (2) Calculate the Mach number of the flow, M_0 , at steady-state using Equation (4.1),

$$M_0 = \frac{V_0}{a}.$$

- (3) Calculate the steady-state non-dimensional valve effective area parameter, C , using Equation (4.4),

$$C = \frac{C_{do} A_{Vo}}{A},$$

or, if applicable, Equation (4.5),

$$C = V_0 \sqrt{\frac{\rho_0}{2p_{do}}}.$$

- (4) Obtain the closure time of the valve under consideration, t_c .
- (5) Compare the closure characteristic of the valve under consideration with that shown in Sketch 4.2. If their forms differ significantly, use the data in Section 4.2 for the pressure change calculation.
- (6) For the section of the pipeline under consideration, denoted by the integer i (see Sketch 4.1), calculate the non-dimensional time scale, τ_i , given by

$$\tau_i = \frac{L_i}{at_c}, \quad (4.6)$$

where L_i is the length of the i^{th} pipe section.

- (7) Data are presented in Figures 2a to 2e that correspond to values of the steady-state non-dimensional valve area parameter, C , of 0.2, 0.4, 0.6, 0.8 and 1.0, respectively. From these figures, for the value of C that is closest to the calculated value from step (3), and for the value of M_0 given by step (2), obtain the appropriate pressure wave profile for the flow conditions. The changing pressure difference over the length of the pipe section under consideration is determined by considering non-dimensional time scale segments, τ_i , of the pressure wave profile. Plot the difference in values of $\Delta p/p_0$ over the segment against t/t_c to determine the time varying pressure difference over the length of the considered pipe section. The procedure for this calculation is shown in Section 5.1.
- (8) Estimate the maximum steam hammer force across the section of pipeline under consideration, neglecting effects due to pipe movement, using

$$F = A\Delta p, \quad (4.7)$$

where A is the cross-sectional flow area of the pipeline and Δp is the maximum pressure difference calculated from $\Delta p/p_0$ in step (7).

The calculation procedure is illustrated in the worked example of Section 5.1.

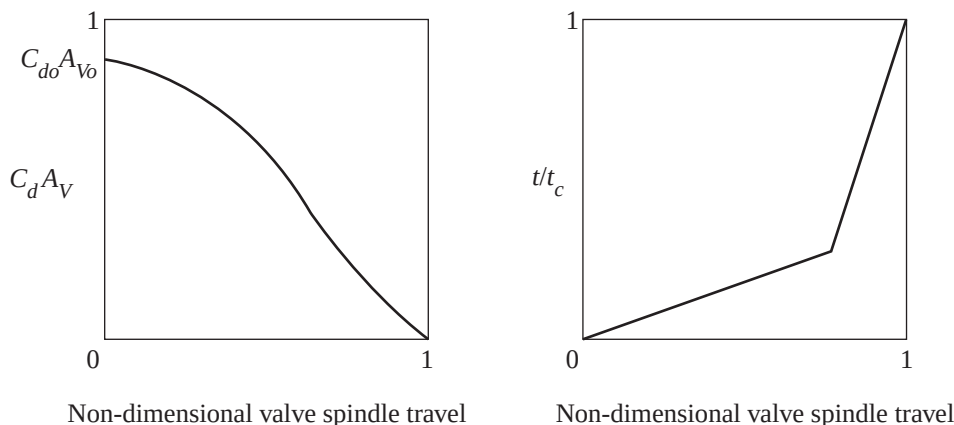
4.2 General Procedure for Estimation of Pressures and Forces Due to Valve Closure

Data are given in this Section that allow estimates to be made of the pressure and forces over any section of a pipeline such as that shown in Sketch 4.1 following closure of valves with characteristics other than that shown in Sketch 4.2. This procedure involves the construction of a pressure change versus non-dimensional time graph. Again, it is assumed that the valve closes in a time t_c , and the ratio of the effective area of the valve, $C_d A_V$, at non-dimensional time, t/t_c , to the effective area of the valve, $C_{do} A_{Vo}$, at steady-state conditions can be determined over the whole closure cycle.

The procedure for performing these calculations is as follows.

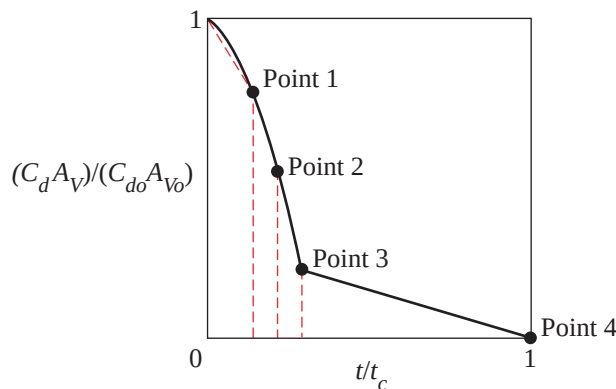
Steps (1) to (4) are the same as those in Section 4.1.

- (5) Plot the curve of $(C_d A_V)/(C_{do} A_{Vo})$ versus t/t_c . It may be necessary in some cases to construct this characteristic curve, particularly when non-linear valve closure, such as two-stage closure, is specified. In general, the characteristic curve may be constructed by combining the curve of effective valve area versus spindle movement with that for spindle movement versus non-dimensional closure time. Example curves of this type are shown in Sketch 4.3. This procedure is as follows.



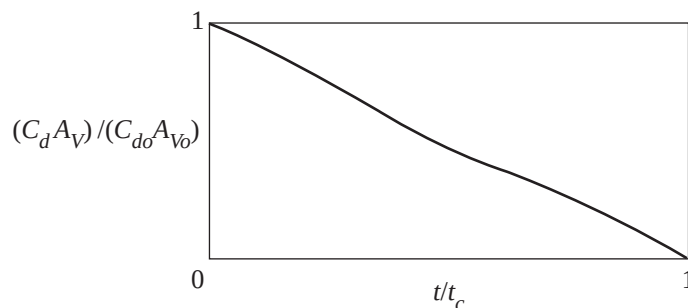
Sketch 4.3 Typical valve closure curves

- (i) From a plot of the effective flow area versus spindle movement, determine the fully-open effective valve area, $C_{do} A_{Vo}$, at steady-state conditions (corresponding to time $t = 0$).
- (ii) Determine the values of the non-dimensional spindle movement at a suitable number of points on the time scale from a plot of the non-dimensional spindle movement versus non-dimensional time. Note that, in Sketch 4.3, values of the non-dimensional spindle movement of 0 and 1 refer to the fully-open and fully-closed positions respectively.
- (iii) For each value of spindle movement determined from (ii), estimate the corresponding value of the effective area, $C_d A_V$, from the plot of effective area versus spindle movement sketch.
- (iv) Divide the values of $C_d A_V$, determined from (iii), by $C_{do} A_{Vo}$ and plot these values of $(C_d A_V)/(C_{do} A_{Vo})$ versus the appropriate value of the non-dimensional time, t/t_c . An example of this figure is derived using Sketch 4.3. and shown in Sketch 4.4.



Sketch 4.4 Valve closure characteristics

It is then necessary to divide the abscissa scale (t/t_c) of this figure into a number of sections such that the shape of the valve closure curve is suitably approximated by a series of straight lines as shown in Sketch 4.4. These sections do not necessarily have to be of equal length. Data are presented in Figures 2a to 2e that correspond to values of C of 0.2, 0.4, 0.6, 0.8 and 1.0, respectively. From these Figures, for the value of C that is closest to that calculated in step (3), and for the value of M_0 calculated in step (2), estimate the values of $\Delta p/p_0$ for each value of $(C_d A_V)/(C_{do} A_{Vo})$ at each time segment as chosen in step (5). This allows the construction of a non-dimensional pressure change versus non-dimensional time figure such as that shown in Sketch 4.5.



Sketch 4.5 Pressure change figure

The procedure for estimating the maximum pressure difference and steam hammer forces across a chosen section of the pipeline is the same as that in steps (7) and (8) of Section 4.1.

The calculation procedure is illustrated in the worked example of Section 5.2.