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Computer program for the prediction of fluid transients in liquid-filled systems

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(Continued on inside back cover)

COMPUTER PROGRAM FOR THE PREDICTION OF FLUID TRANSIENTS IN LIQUID-FILLED SYSTEMS

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COMPUTER PROGRAM FOR THE PREDICTION OF FLUID TRANSIENTS IN LIQUID-FILLED SYSTEMS

1. NOTATION AND GLOSSARY

1.1 Notation and Units

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

		<i>SI</i>	<i>British</i>
A	cross-sectional area of pipe	m^2	ft^2
A_{AV}	cross-sectional area of air vessel*	m^2	ft^2
A_{in} , A_{out}	cross-sectional areas of air valve inlet and outlet orifices, respectively	m^2	ft^2
$A_{T_{in}}$, $A_{T_{out}}$	cross-sectional areas of air vessel throat for inflow and outflow, respectively*	m^2	ft^2
A_v	instantaneous cross-sectional area of valve	m^2	ft^2
a	speed of propagation of pressure wave (wave speed)	m/s	ft/s
C_d	instantaneous non-dimensional discharge coefficient of valve	—	—
$(C_d A_v)_0$	non-dimensional product of discharge coefficient and cross-sectional area of valve (<i>i.e.</i> effective flow area) at steady-state conditions	m^2	ft^2
$(C_d A_v)_p$	non-dimensional product of discharge coefficient and cross-sectional area of valve (<i>i.e.</i> effective flow area) at point p (see Section 3.3.5)	m^2	ft^2
C_{in} , C_{out}	non-dimensional discharge coefficients of air valve for flow through inlet and outlet orifices, respectively	—	—
$C_{T_{in}}$, $C_{T_{out}}$	non-dimensional discharge coefficients for flow into and out of air vessel, respectively†	—	—
dt	time step ($= L/na$, see Section 4.2)	s	s
D	internal diameter of circular pipe	m	ft
D_E	equivalent internal diameter of pipe of non-circular cross section, defined as $D_E = 4A/P_E$	m	ft

For footnotes see end of Notation section.

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E	Young's modulus of elasticity of pipe wall material	N/m ²	‡lbf/ft ²
f	Fanning friction factor (see Equation (3.4))	–	–
g	acceleration due to gravity, $g = 9.81 \text{ m/s}^2$ or 32.2 ft/s^2	m/s ²	ft/s ²
H	head above datum	m	ft
$\bar{H}$	**atmospheric head, $\bar{H} = 10.3 \text{ m}$ or 33.8 ft of water at sea level and 15°C	m	ft
H_0	head above datum at constant pressure point (see Sections 3.3.2 and 3.3.3), equal to free surface elevation above datum for reservoir or unpressurised tank	m	ft
H_d	head drop across valve (see Section 3.3.5)	m	ft
H_i	pump inlet head	m	ft
H_p	pump head	m	ft
H_R	pump rated head	m	ft
H_v	vapour head or gas release head (see Section 3.2.2)	m	ft
h	non-dimensional pump head ($= H_p / H_R$)	–	–
$H_{q=0}$	non-dimensional no-flow pump head at rated speed	–	–
$H_{q=0.5}$	non-dimensional pump head at half rated flow and at rated speed	–	–
$H_{q=q_s}$	non-dimensional pump head at non-dimensional flow, $q = q_s$, and at rated speed	–	–
ΔH	change in pressure head due to flow disturbance	m	ft
∂H	head loss due to distributed minor losses (see Section 3.1.5)	m	ft
I	inertia of rotating components of pump	kg m ²	††slug ft ²
K	air vessel throttle loss characteristic or bulk modulus of liquid (see Section 3.1.4)	N/m ² –	‡lbf/ft ² –
k_l	loss factor for minor loss in pipe	–	–
L	pipe length	m	ft
N_p	instantaneous pump rotational speed	rev/min	rev/min
N_R	pump rated rotational speed	rev/min	rev/min

For footnotes see end of Notation section.

N_{SS}	pump rotational speed at steady-state	rev/min	rev/min
$NJUNCT$	number of junctions in network (see Section 4)	—	—
$NPIPES$	number of pipes in network (see Section 4)	—	—
$NPOINTS$	number of selected points on valve closure characteristic (see Section 3.3.5)	—	—
n_i	number of segments in pipe i (see Section 4)	—	—
P_E	internal perimeter of pipe	m	ft
P_p	instantaneous pump shaft power	W	††ft lbf/s
P_R	rated pump shaft power	W	††ft lbf/s
p	pressure or non-dimensional pump shaft power (= P_p/P_R)	N/m ²	†lbf/ft ²
$p_{q=0}$	non-dimensional no-flow pump shaft power at rated speed	—	—
$p_{q=0.5}$	non-dimensional pump shaft power at half-rated flow and at rated speed	—	—
$p_{q=q_s}$	non-dimensional pump shaft power at non-dimensional flow, q_s , and at rated speed	—	—
Q	liquid flow rate (general) or flow into or out of network at a junction (see Section 3.3.1)	m ³ /s m ³ /s	ft ³ /s ft ³ /s
Q_p	instantaneous flow rate of pump	m ³ /s	ft ³ /s
Q_R	rated volume flow rate of pump	m ³ /s	ft ³ /s
q	non-dimensional flow rate delivered by pump (= Q_p/Q_R)	—	—
q_s	non-dimensional flow (see Section 3.3.4)	—	—
R	gas constant, typical value for air = 287 J/kg K or 3090 ft lbf/slug K	J/kg K	***ft lbf/slug K
Re	Reynolds number (= $\rho V_0 D_E / \mu$, see Section 3.1.5)	—	—
s	integer number of time steps over period of valve closure (see Section 4.2)	—	—
T_a	absolute temperature of air in air cavity	K	K
T_l	absolute temperature of liquid	K	K

For footnotes see end of Notation section.

t	time or pipe wall thickness	s m	s ft
t_c	time for change in flow conditions to be completed (e.g. for valve to close completely)	s	s
t_{rev}	elapsed time between pump trip and flow reversal	s	s
V	flow velocity	m/s	ft/s
V_{fo}	flow velocity through one-way valve at instant of opening (see Section 3.3.4)	m/s	ft/s
V_0	steady-state flow velocity through pipe	m/s	ft/s
V_R	maximum permitted reverse flow velocity through one-way valve (see Section 3.3.4)	m/s	ft/s
v_0	initial volume of air in air vessel (see Section 3.3.6)	m ³	ft ³
x	distance along pipe axis from upstream end	m	ft
z	elevation above datum	m	ft
z_b	height of air vessel base above centre line of adjoining section of pipe (see Section 3.3.6)	m	ft
z_t	liquid level in air vessel prior to pump trip	m	ft
ε	effective surface roughness of pipe wall (see Section 3.1.5)	m	ft
ϕ	pipe restraint factor in Equation (3.3) (= 1 in this Item)	—	—
η	polytropic index of gas (usually 1.2 for air)	—	—
η_R	pump rated efficiency (see Section 3.3.4)	—	—
μ	liquid dynamic viscosity (see Section 3.1.5)	kg/m s	^{††} slug/ft s
ν	Poisson's ratio (see Table 6.2)		
ρ	density of liquid	kg/m ³	^{‡‡} slug/ft ³

* Alternatively, these terms may be the sum of cross-sectional areas of a bank of air vessels.

[†] For more than one air vessel, the average value of the discharge coefficients may be used.

[‡] 1 lbf/ft² = 1/144 lbf/in².

^{**} $\bar{H}$ is taken as the value of head of water in the International Standard Atmosphere at sea level and 15°C.

^{††} 1 slug = 32.174 lb (pounds mass).

^{‡‡} 1 ft lbf/s = 0.001818 hp (horsepower).

^{***} 3090 ft lbf/slug K = 96.03 ft lbf/lb K.

Subscripts

The following additional subscripts are not defined in the above notation

down refers to conditions at downstream end of pipe

equiv refers to equivalent value

i refers to pipe number *i*

0 refers to steady-state or non-varying conditions

up refers to condition at upstream end of pipe

ss refers to steady-state conditions

1.2 Glossary of Terms Used in This Item

AIR INLET/OUTLET VALVE

Valves that allow inflow and outflow (usually throttled) of air into liquid-filled pipelines at locations at which pressure would otherwise fall below atmospheric by an unacceptable amount following pump run-down or valve closure (see Section 3.3.7). Air valves are also used to allow controlled release of contained gases during system filling, and to release air or gas coming out of solution in low pressure regions of pipelines during normal operation.

ACCUMULATOR

A device for suppressing pressure transients comprising a vessel containing a gas separated from the main liquid (*e.g.* by a flexible medium such as a rubber membrane in water applications). See AIR VESSEL.

AIR VESSEL

A device for suppressing pressure transients often used in water applications comprising a vessel containing pressurised air or gas and a quantity of the main liquid (see Section 3.3.6). See ACCUMULATOR.

BOUNDARY CONDITION

Component or junction of three or more pipes or change in pipe properties at the ends of a section of pipe having constant flow area, wave speed and friction factor.

BULK MODULUS

A measure of the compressibility of a liquid. For a fluid transients analysis the isentropic tangent bulk modulus is normally used when estimating wave speeds (see Section 3.1.4).

CHECK VALVE

One-way valve. Commonly used term in the water industry. Also known as reflux valve and non-return valve.

FLUID TRANSIENTS

Pressure changes that propagate through an initially steady flow following a disturbance to the flow.

HEAD

Representation of local pressure referred to a chosen datum, expressed as $H = p/(\rho g) + z$, where p and z are the local values of gauge static pressure and pipe elevation respectively (see Section 3.1.6).

HYDRAULIC GRADE LINE

Representation of local pressure along a pipe as the sum of the pressure head and elevation above a chosen datum. This line would pass through the surface levels in piezometer tubes located along the pipe. See HEAD.

JUNCTION

Considered to be the ends of a pipe for the purposes of this Item. A junction has a boundary condition type associated with it. See Section 4.1.

METHOD OF CHARACTERISTICS

Technique allowing the unsteady values of head and flow velocity at a future time in a pipe system to be calculated using values of these parameters at a previous time.

NODE (OR NODAL POINT)

For the computer model of the pipe network, each individual pipe must be divided for the purposes of calculation of flow parameters into two or more segments of equal length. The ends of these segments are termed the nodes or nodal points (see Section 4.2).

NON-RETURN VALVE

One-way valve. Commonly used term in fluids engineering.

PIPE LENGTH

Length of line of fluid between the ends of a pipe (including bends, *etc.*). Equal to the axial distance between boundary conditions.

PIPELINE PERIOD

Time taken for a pressure wave to travel the full PIPE LENGTH from one end and back again ($= 2L/a$).

PRESSURE SURGE

Alternative nomenclature for fluid transient.

PRESSURE TRANSIENT

Alternative nomenclature for fluid transient.

PRESSURE WAVE

Arising from the change in kinetic energy in a fluid following a disturbance to the steady flow. Analogous to a shock wave in air.

REFLUX VALVE

One-way valve. Commonly used term in fluids engineering.

WATER HAMMER

Alternative (dated) term for fluid transient, with special application to water flow.

WAVE SPEED

Speed of propagation of pressure waves through a fluid contained in a pipe or tunnel.

2. BACKGROUND

2.1 Notes

This Item is one in a series of Data Items concerning fluid transients in pipes. A computer program is provided that may be used to predict the changes in the pressure and flow of a liquid contained in a piping system following changes in the flow rate or disturbances in the flow. The data are applicable to a wide range of liquid flow systems, from water supply systems to oil pipelines. The program is also applicable to tunnels, thick-walled pipes and non-circular closed conduits running full.

Fluid transients are the pressure changes that propagate through a fluid following a disturbance to the flow rate. Other terms that are used to describe these flow changes include pressure surges, water hammer and hydraulic transients. The pressure changes occur at the location of the disturbance and propagate rapidly through the fluid as pressure waves. The most common causes of fluid transients in liquid-filled pipe systems are valve closure and pump trip, although other events such as rapid changes in flow demand at some point in a system could cause significant flow changes. These flow changes may arise due to both normal operational requirements or fault conditions, such as power failure to pumps or malfunctioning of system components. Maximum or minimum transient values of pressure at some or all points in the line may impair the performance. For example, changes in the internal steady pressure in the fluid may lead to significant changes in the stress levels in the pipe walls, which can cause local or widespread damage to pipes or fittings. Alternatively, regions of sub-atmospheric pressure may occur, which can lead to seepage of contaminants through pipe joints or the possibility of the formation of vapour cavities (which can cause very high pressures when the vapour cavity collapses). If the magnitude of the pressure changes associated with the fluid transients lead to the design limits for the pipeline or pipe fittings being exceeded, it may be necessary either to strengthen the system to allow for the pressure changes or to consider methods of reducing these pressure changes. Both increases and decreases of pressure need to be considered. Reduction of pressure will normally result in vapour cavity formation if the pressure drops below vapour pressure.

The computer program in this Item provides a means of predicting pressure and flow changes in a liquid, contained in a piping system, arising from fluid transients. The steady-state conditions of head and flow at all points in the network are also calculated by the program prior to starting the transient analysis. These conditions are calculated from the steady-state operating conditions of all the components in the network (such as pumps and valves) and the characteristics of the pipes. The program may be used to investigate the effects of variations in system parameters on these pressure and flow changes, either at steady-state or during transient conditions. In addition, the program allows the engineer to investigate the effectiveness of certain types of surge suppression devices in situations where the magnitudes of pressure or flow changes in the unprotected system are excessive (*e.g.* the onset of vapour cavity formation is predicted, although the behaviour of the resulting vapour cavity is not modelled by the program).

The changing flow conditions following a flow disturbance in a liquid-filled piping system are estimated from the fundamental equations of flow continuity and momentum in uniform lengths of pipes, together with suitable equations that describe the flow conditions at components such as pumps, valves and pipe junctions. The most convenient technique for the solution of the governing equations is the method of characteristics. This technique involves the transformation of the two partial differential equations, describing the pressure and flow rate in terms of the dependent variables of time and pipe position, to a set of two ordinary differential equations with time and pipe position as the independent variables. The version of the method of characteristics used in the computer program in this Item allows the pressure and flow velocity to be estimated at a number of time increments following the initiation of the flow disturbance. The method of characteristics is discussed in detail in Derivations 2, 3 and 4 and References 12, 13, 14, 15 and 19.

2.2 Scope and Applicability of This Item

The computer program provides a means of estimating the flow in piping systems consisting of a pipe or number of pipes running full of liquid, either at steady-state conditions or following disturbances to the flow. Flow changes may result from the closure of a valve or valves, run-down of pumps or a change in external flow demand at some point in the line. Other pumps, valves and components over which pressure losses occur at steady-state may be included in the analysis. The effect on the flow changes of varying pump run-down characteristics or valve closure times, and the inclusion in the system of air vessels, air inlet/outlet valves and pump bypass non-return circuits (used for surge suppression) may be investigated. The full scope of the program is detailed in Section 4.

2.3 Graphical Methods of Analysis Using Other Related Items

The computer program in this Item should be used for the detailed analysis of systems for which simplified graphical methods are not readily applicable. However, for some systems, the assessment of the scale of fluid transients problems may be made using graphical methods. Graphical methods are also useful to engineers considering the nature of transient problems in complex systems, and familiarising themselves with the various methods available for suppression of excessive pressure changes. The data are, in many cases, sufficient to enable engineers to decide whether there is a potential fluid transient problem in a given system. Their use is also invaluable as a guide to the interpretation of data resulting from both graphical and computer methods.

Previous Items in this series provide graphical prediction methods for the magnitudes of pressure changes following pump trip and valve closure in simplified one-pipe systems, and means of assessing the effect on these pressure changes of suppression devices are included. A general guide to fluid transients in piping systems is given in ESDU 84013, which also includes graphical data for the prediction of pressure changes following valve closure and pump trip. Further graphical data are presented in ESDU 84038, 85009 and 85044 for the prediction of pressure surge in pipelines with air vessels, air inlet/outlet valves and pump bypass circuits, respectively, used as suppression devices. ESDU 83046 provides data necessary for the estimation of the speed of propagation of pressure waves. ESDU 86015 presents graphical data and a computer program for the estimation of pressures and forces in steam lines following valve closure.

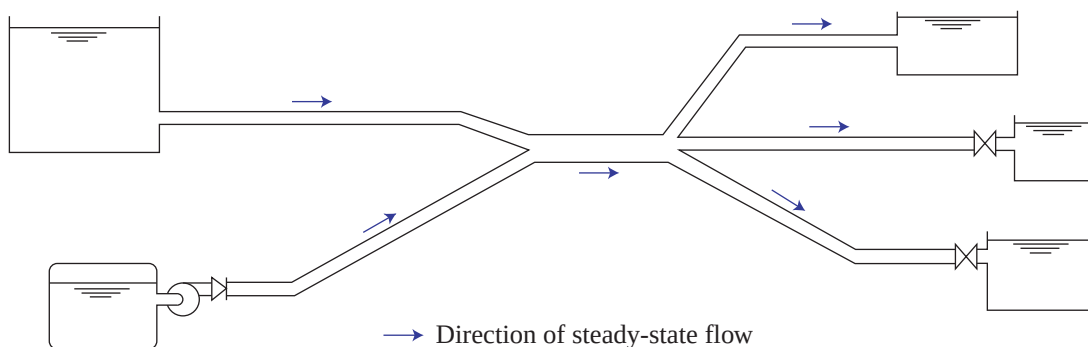
Graphical methods of fluid transients analysis are also given in References 7, 9, 10, 13 and 14.

3. GUIDANCE ON DATA REQUIRED FOR FLUID TRANSIENTS ANALYSIS

The accuracy and validity of a fluid transients analysis is dependent upon the reliability of the basic data available for the system and components. Mathematical models that closely predict the behaviour of the pipes and components must be used for a realistic solution. In a computer analysis, it is necessary to model the pipes and components carefully, and to give particular attention to the prediction of flow changes at components such as valves and pumps and at pipe junctions. The data given below provide guidance on the information required for a fluid transients analysis using the computer program provided in this Item. Guidance is given on the prediction of the numerical values of each parameter where appropriate. The program allows the input data to be expressed in either the SI or the British system of units (though not mixed). For some parameters, non-consistent units are used in the data input scheme and converted to consistent units within the program (*e.g.* pipe internal diameter is expressed in inches for the British system of units). The units required throughout the data input scheme are illustrated in Section A.3.

3.1 Piping Data

A diagram showing the layout of the piping system and the location of all components that are to be considered in the analysis should be drawn. An example is shown in Sketch 3.1. In a typical piping system, there may be a number of components connected by pipes that may terminate at either another component or a junction with another pipe. The piping system is considered to be running full of liquid, and is modelled as a network of pipes. Each pipe within the network is treated as being of constant cross section with constant friction factor and wave speed. Non-circular pipes can be modelled using their equivalent diameters (see Section 3.1.2).



Sketch 3.1 Example diagram of a typical system

The following data are required for each pipe in the system.

3.1.1 Pipe length, L

The pipe length, L , is taken as the true axial distance (including distance around pipe bends, *etc.*) between the ends of the pipe. Note that here, unlike some pressure drop calculations, equivalent lengths of straight pipe are not used to represent the pressure loss due to fittings.

3.1.2 Pipe diameter, D (or D_E)

The appropriate value is the internal diameter, D , of a circular pipe or the equivalent diameter, D_E , of a

non-circular pipe. The equivalent diameter (also known as the hydraulic diameter) is given by the expression

$$D_E = \frac{4A}{P_E}, \quad (3.1)$$

where A is the cross-sectional flow area of the pipe and P_E is the internal perimeter. The program provides the option of specifying circular pipes throughout the network to avoid inputting the pipe cross-sectional flow area.

3.1.3 Pipe flow area, A

This parameter is only required for non-circular pipes.

3.1.4 Wave speed, a

The speed of propagation of pressure waves or wave speed, a , is dependent upon both the properties of the fluid and the elasticity of the pipe that contains it. The general equation for the calculation of wave speed in a liquid contained in a flexible conduit is

$$a = \left[\rho \left(\frac{1}{K} + \frac{1}{A} \frac{\partial A}{\partial p} \right) \right]^{-1/2}, \quad (3.2)$$

where the term $(1/A)(\partial A/\partial p)$ is the variation of the area strain with the internal pressure in the pipe or conduit. The parameter K is the bulk modulus of the liquid and values for some commonly-used liquids are given in Table 6.1. Expressions for the area strain term of some common types of conduit are given in Table 6.2. The wave speed in water contained in circular pipes with frequent expansion joints may be estimated from Figure 1. These data are given for a range of values of t/D , the ratio of the pipe wall thickness to internal diameter, and a range of values of E/E_{ref} , the ratio of Young's modulus of elasticity of the pipe wall material to a reference value ($E_{ref} = 10 \text{ GN/m}^2$ or $2.09 \times 10^8 \text{ lbf/ft}^2$). Some typical values of Young's modulus of elasticity for common pipe materials are given in Table 6.3. The wave speed in liquids contained in tunnels, thin-walled pipes or non-circular pipes and in homogeneous liquid/gas and liquid/solid mixtures may be estimated using ESDU 83046 (Reference 18).

The value of wave speed is required for each pipe in the network. However, if all pipes are circular and thin-walled (*i.e.* the pipe wall thickness is less than approximately 1/10th of the internal diameter), the program provides the option of calculating the wave speed in each pipe, using the equation

$$a = \left[\rho \left(\frac{1}{K} + \frac{D\phi}{tE} \right) \right]^{-1/2}. \quad (3.3)$$

In this equation, ρ and K are the density and bulk modulus of the liquid, respectively, t is the pipe wall thickness, E is the modulus of elasticity of the pipe wall material and ϕ is a pipeline restraint factor. For pipelines with a high elastic modulus (such as steel or concrete) or for pipelines with frequent longitudinal expansion joints, a value of ϕ of unity may be taken. In the program, ϕ is assumed to be unity. For the wave speed calculation, values of t and E are required for each pipe.

The program allows single values of the above parameters to be specified for all pipes in the network.

3.1.5 Frictional losses

If frictional losses in the pipes due to pipe wall friction and minor losses (at components, bends, *etc.*) are included in the analysis, their magnitudes are calculated by specifying either

- (i) friction factors, or
- (ii) effective pipe wall surface roughness values and the summation of all loss factors for each pipe in the network.

These options are discussed below. The program allows single values or pairs of values to be specified as appropriate for all pipes in the network.

The numerical values of friction factors may be estimated using data presented in ESDU 66027 and 79014, based upon the head losses due to friction over uniform straight pipe lengths. The friction factor, f , over a uniform straight length of pipe is given by the expression

$$f = \frac{(H_{up} - H_{down})}{4} \frac{D_E}{L} \frac{2g}{V_0^2}, \quad (3.4)$$

where H_{up} and H_{down} are the steady flow values of head at the upstream and downstream ends of the pipe respectively, D_E is the equivalent diameter, L the length of the pipe and V_0 the steady-state flow velocity through the pipe. In many practical situations, the sum of the head losses at components in a pipe such as pipe bends may be a significant proportion of the overall head loss in the pipe, and the calculation of the friction factor using Equation (3.4) includes these individual losses. It should be noted that individual components over which there is a large head loss at steady-state may have a significant effect on the transient flow conditions. Such components may be modelled individually in the analysis, using the procedure described in Section 4.3.

If friction factors for pipes in a network are not known, the program can calculate an effective friction factor for each pipe in the network. It is assumed that friction losses arise from both the roughness of the pipe wall surface and the pressure losses over individual components in the pipe. The losses due to pipe wall roughness are represented by the friction factor, f , which is calculated for most flow conditions within the program using the equation (due to Colebrook and White)

$$\frac{1}{\sqrt{f}} = -4 \log_{10} \left[\frac{1.25}{Re \sqrt{f}} + \frac{\epsilon}{3.7D} \right], \quad (3.5)$$

where ϵ is the effective roughness of the pipe wall surface, Re is the Reynolds number of the flow and D_E is the effective diameter of the pipe. The Reynolds number, Re , is given by the expression

$$Re = \frac{\rho V_0 D_E}{\mu}, \quad (3.6)$$

where ρ is the density of the liquid, V_0 is the steady-state flow velocity and μ is the dynamic viscosity of the liquid. Typical values of ϵ for a number of common pipe wall surfaces are given in Figure 2. Typical values of μ for a number of common liquids are given in ESDU* 66024 and 78040. Further guidance on the calculation of f using Equation (3.5) and for flow conditions where this equation does not apply is

* ESDU 66024, 'Approximate data on the viscosity of some common liquids', and ESDU 78040, 'Viscosity of water and steam' are in the Physical Data, Mechanical Engineering Series.

given in ESDU 66027 (Derivation 1).

The head loss in a pipe due to the sum of the minor losses at components is expressed as

$$\delta H = \frac{\Sigma k_l V_0^2}{2g}, \quad (3.7)$$

where Σk_l is the sum of head loss factors for all the components in the pipe. If this total head loss, δH , is negligible compared with the gross head loss over the pipe, $H_{up} - H_{down}$, the minor losses may be neglected and the friction losses are assumed to be solely due to pipe wall roughness. Head loss factors for pipe bends, branches and junctions may be estimated using ESDU 73022, 73023, 77009, 77029 and 83037. Head loss factors for duct fittings and equipment may be estimated using ESDU 66030, 72009, 81021 and 81039.

The sum of the head losses in a pipe due to surface roughness effects and minor losses is represented by the static head loss between the upstream and downstream ends of the pipe using the equation

$$H_{up} - H_{down} = \frac{2fLV_0^2}{gD_E} + \frac{\Sigma k_l V_0^2}{2g}. \quad (3.8)$$

For the convenience of the friction factor calculation, it is assumed that the minor losses are distributed equally along the line, and an equivalent friction factor, f_{equiv} , is used that includes the effects of the minor losses. Hence, Equation (3.8) can be re-expressed as

$$H_{up} - H_{down} = \frac{2f_{equiv}LV_0^2}{gD_E}, \quad (3.9)$$

where

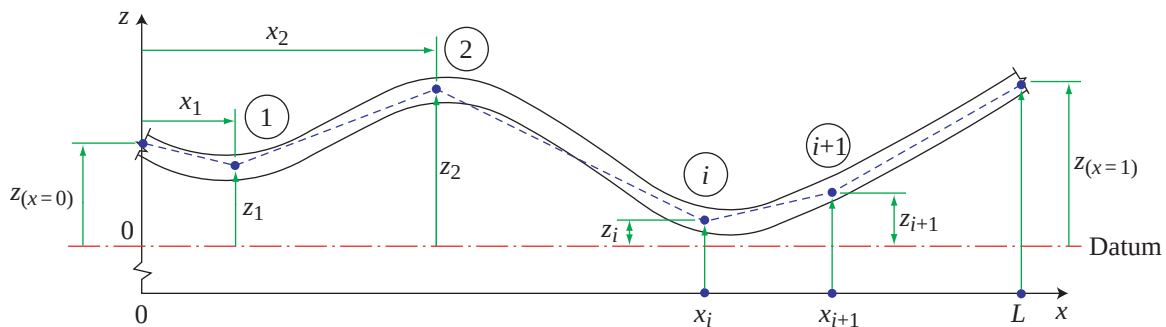
$$f_{equiv} = f + \frac{\Sigma k_l D_E}{4L}. \quad (3.10)$$

3.1.6 Pipe elevations, z

All values of head are expressed relative to a datum, the position of which is arbitrary. In water systems, for example, it is usually convenient to use the surface level of the lowest reservoir or tank as the datum. The relationship between pressure and head is illustrated on Sketches 3.2a and 3.2b.



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Sketch 3.3 Example pipe elevation

Once a suitable datum is chosen, it is used as a reference for all head and pipe elevation values. The program calculates the head above datum, H , throughout the network. Both head above datum, H , and pressure head, $(H - z)$, are given in the output. Where the local head, H , coincides with the local elevation, z , the liquid is at atmospheric pressure, and the pressure head is zero. Head may also be expressed relative to the absolute value, given by $(H - z + \bar{H})$, where $\bar{H}$ is the absolute head, typically equal to 10.3 m (or 33.8 ft) of water at sea level. In a transients analysis, it is necessary continuously to monitor the minimum pressure throughout the network in order to predict whether the pressure at any point falls to a value such that gases are released from the liquid, which can cause vapour cavities to form. This condition occurs when the pressure head, $(H - z)$, falls below atmospheric by an amount large enough to allow gases to be released from the liquid. Numerically this implies that the pressure head has fallen below zero to H_v , where H_v is the vapour head or gas release head. If this condition is reached at any point throughout the network the remaining results from the program may not be realistic because vapour cavities are not modelled by the program. A message is printed by the program if this condition occurs and the analysis continues for ten further time steps (see Section 4.2) to allow the user to gauge the extent of the problem.

The local elevation, z , along each pipe in the network is required by the program. This is achieved by specifying the elevation at a number of points at distances x along the longitudinal profile of each pipe, such that straight line segments between the points describe the shape of the elevation profile with sufficient accuracy. An example profile of one pipe in a network is shown in Sketch 3.3. The program requires the user to select a number of internal points along each pipe, and input for each point a longitudinal distance, x_i , from the upstream end of the pipe. By considering each end of the pipe to be a junction (see Section 4.1), the elevations at the ends of each pipe are input as the junction elevations, z_{up} and z_{down} . The program allows straight pipe lengths to be considered in which case the junction elevations only are required. A further option is provided to specify the same elevation for each point in the network.

3.2 Liquid Properties

3.2.1 Liquid density, ρ

Typical values of the density of some commonly-used liquids are given in Table 6.1. These data are reproduced from ESDU 83046. The liquid density is assumed to be uniform throughout the network, so only one value is requested by the program.

3.2.2 Vapour head or gas release head, H_v , at the operating conditions

This is the smallest value of head below atmospheric at which any gases dissolved in the liquid begin to be released, leading to vapour cavities or bubble formation in the regions of such low pressure. The value is dependent upon the operating conditions and is usually temperature-dependent. Vapour cavities are formed as the liquid columns separate and intense local pressures may result from collapse of vapour cavities. They are most likely to occur at high points in pipelines and at the downstream sides of tripped pumps or closing valves.

In many types of piping system, such as water supply lines, one of the objectives of a fluid transient analysis is to check that the pressure at all points in the system does not fall to atmospheric pressure, as seepage of contaminants through joints or even pipeline collapse may occur. The formation and subsequent behaviour of vapour cavities cannot usually be predicted accurately, and no attempt to do so is made in the computer program presented with this Item. However, their occurrence at sub-atmospheric and sub-vapour head conditions is predicted. The liquid vapour head is assumed to be uniform throughout the network, so only one value is required by the program.

3.2.3 Liquid dynamic viscosity, μ

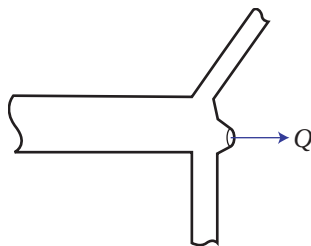
This parameter is required if friction factors are to be calculated by the program. Typical values of μ for several common liquids are given in ESDU 66024 and 78040 (see footnote to Section 3.1.5).

3.3 Boundary Conditions and Component Characteristics Data

The characteristics of all components in the system (pumps, valves, *etc.*) need to be specified. In particular, the prediction of pressure and flow changes following pump trip and valve closure requires the careful and accurate simulation of the behaviour of the devices. The following information may be of assistance when modelling boundary conditions.

3.3.1 Multi-pipe junctions with outflow or inflow

The program allows the effects of outflow or inflow, Q , from the network at junctions between two or more pipes to be considered (see Sketch 3.4).



Sketch 3.4 Multi-pipe junction with outflow

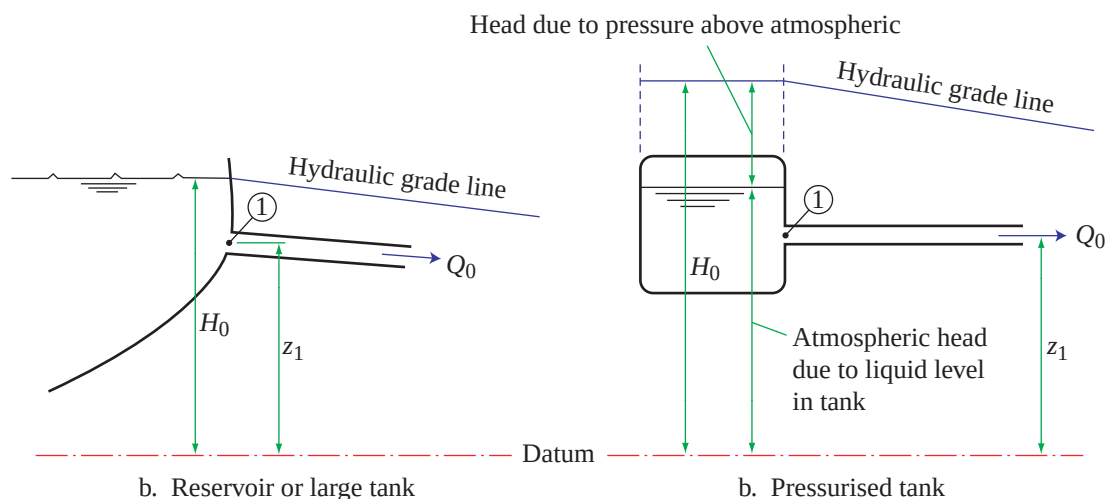
It is assumed that the pressure is the same at the junction end of each pipe connected at the junction. The program allows the following conditions to be considered.

- (1) No outflow (or inflow), Q , from the network.
- (2) Constant outflow (or inflow), Q , from the network.

- (3) Instantly changing outflow (or inflow), Q , from the network. The program requires two values of Q , calculating steady-state conditions using the first value and using the second value for fluid transient calculations.
- (4) Instantaneous shut-off of outflow (or inflow), Q , at the start of the transient calculations due to instantaneous closure of valves at the ends of each pipe at the junction, isolating the junction. The pressure at the ends of each pipe will vary independently following closure of these valves.

3.3.2 Upstream constant pressure points

It is assumed that the elevation of the hydraulic grade line above the chosen datum at the inlet to the pipeline does not change over the considered flow period. In a reservoir, or large tank, this is the elevation of the liquid surface above the chosen datum. In a pressurised tank, this is the elevation of the liquid surface plus an extra increment added to account for the pressure above atmosphere (see Sketch 3.5). Pressure losses at the inlet to the pipe are usually small and are neglected in the analysis.



Sketch 3.5 Reservoir and pressurised tank

3.3.3 Downstream constant pressure points

The elevation of the hydraulic grade line at the boundary is again assumed to remain constant over the considered flow period. Pressure losses at the inlet to the reservoir or tank are neglected (see Section 3.3.2).

3.3.4 Pumps

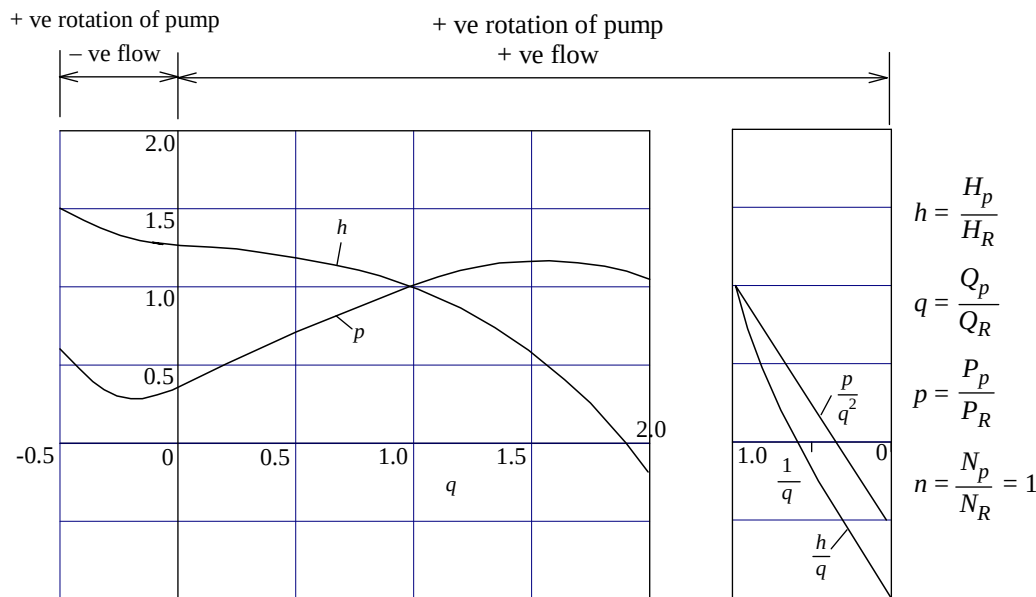
The pump head and flow at a given speed are calculated using the pump characteristic curves, usually available from the manufacturer. The head, flow and power of a pump at any speed can be estimated from the curves of head versus flow and power versus flow at a reference speed (or rated speed). This rated speed is usually the predicted operating speed of the pump, and the rated values of head, flow and power at that point can be obtained from the characteristic curves.

Typical non-dimensionalised centrifugal pump characteristic curves are shown in Sketch 3.6, from References 6 and 11. The data required by the program to model these curves are as follows.

- (i) Non-dimensional values of pump head, $h_{q=0}$, and power, $p_{q=0}$, at zero flow, $q = 0$.

- (ii) Non-dimensional values of pump head, $h_{q=0.5}$, and power, $p_{q=0.5}$, at half-rated flow, $q = 0.5$.
- (iii) Non-dimensional values of pump head, $h_{q=q_s}$, power, $p_{q=q_s}$, and flow, q_s , at a suitable point on the pump characteristic such that $1 \leq q_s \leq 2$. A value of q_s of 1.5 is recommended for convenience.

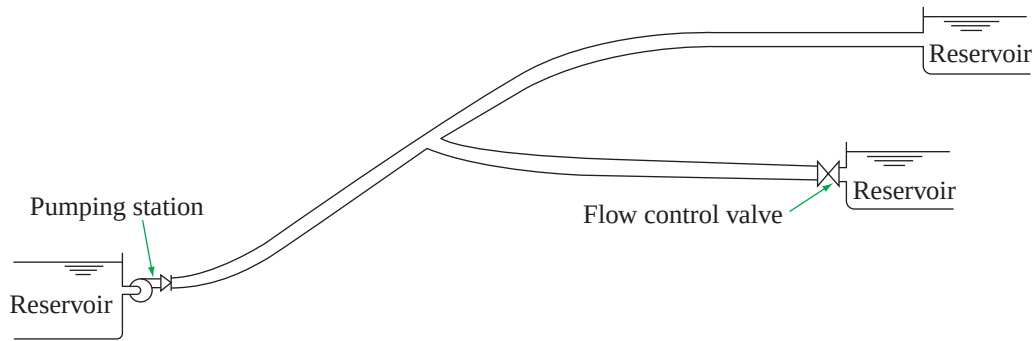
The shapes of the curves in the regions $-0.5 \leq q \leq 0$ and $q > 2$ are rarely available from the manufacturer, and are modelled by the computer program using typical values for centrifugal pumps. Centrifugal pumps only are modelled by the program.



Sketch 3.6 Typical centrifugal pump characteristic curves

The rated conditions of the pumps are required in the analysis. These are the pump rated head, H_R , flow rate, Q_R and rotational speed, N_R . The steady-state operating speed of the pump, N_{SS} , is also required. The pump inertia, I , is required for run-down calculations, and is usually available from the manufacturer. However, an estimate of the inertia may be made from data presented in Figure 3. In this Figure, values of inertia are presented for a range of values of the ratio of pump shaft input power to pump rotational speed at rated conditions. For this estimation the pump rated efficiency, η_R is required, and is in the range 0 to 1 (usually between 0.65 and 0.9). Figure 3 shows a mean line and a band containing the measured data. A smaller value of I usually leads to higher values of transient pressure changes. Figure 3 was derived using data for British and U.S. pumps, presented in References 6 and 8, respectively.

A typical pumping system is shown in Sketch 3.7. The pump is protected by a one-way valve (commonly referred to as a reflux valve or check valve) intended to prevent all reverse flow through the pump by closing completely when the discharge falls to zero. However, in many practical situations some reverse flow through the pump occurs which may lead to the problem of check valve slam, resulting in pressure fluctuations and possible pipeline vibrations.



Sketch 3.7 Typical pumping system

The program allows the effects of backflow through the pump to be investigated by specifying a percentage of the steady-state flow velocity (or flow rate) through the pump that is allowed to flow back through the pump. An upper limit of 30 per cent of the steady-state forward flow velocity is recommended to ensure the estimates of flow changes due to backflow are realistic.

A reliable estimate of the magnitude of the maximum backflow velocity, $|V_R|$, is required in order to select the type of one-way valve that must be used to protect the pump and pipework. The type(s) of valve that is suitable for a particular pump is determined using the steady flow characteristics of the pump, the deceleration of the liquid following pump trip and the dynamic performance characteristic of the valve. The dynamic performance characteristics of one way valves should be available from the valve manufacturers. Typical examples are shown in Sketch 3.8a. The way in which these data may be used to select a suitable one-way valve for a particular application is as follows (see Derivation 5).

- (1) For the range of valves that are considered, construct a non-dimensional valve selection diagram by converting the dynamic performance characteristic curves provided by the manufacturers (see Sketch 3.8a) into the non-dimensional form of Sketch 3.8b. The diagram is constructed by non-dimensionalising $|V_R|$ by V_0 and $|dV/dt|$ by D/V_0^2 and is therefore solely applicable to the particular valve under consideration.
- (2) Estimate the maximum and minimum values of head at the inlet and outlet of the pump assuming the one-way valve closes instantly on flow reversal. This may be done using graphical methods for simple systems, or using the computer program for more complex systems. Estimate an additional maximum head change value, ΔH , for which these estimated maximum and minimum head values plus and minus ΔH respectively remain within acceptable limits. It is further essential that a shock load due to an instantaneous pressure change of magnitude ΔH would not cause unacceptably high stress values in the pipework and pump casing.
- (3) Estimate the maximum permitted reverse velocity, V_R , using the equation

$$V_R = \frac{g}{a} \Delta H, \quad (3.11)$$

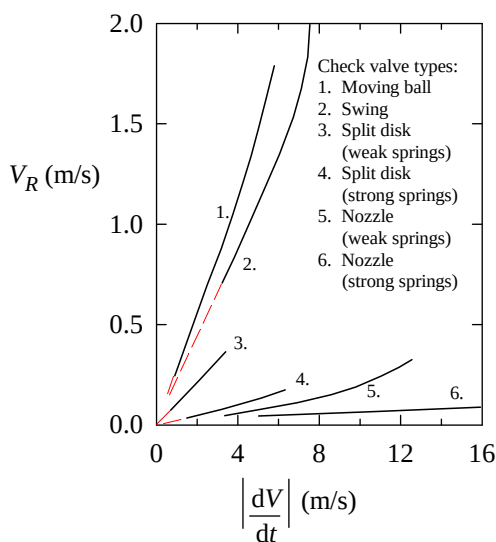
where a is the wave speed in the outlet pipe. This value should not exceed 30 per cent of the steady-state flow velocity.

- (4) Normalise V_R by dividing by the flow velocity through the valve at the instant the one-way valve opens fully during start-up, V_{fo} . This value is typically 75 per cent of the steady-state flow velocity,

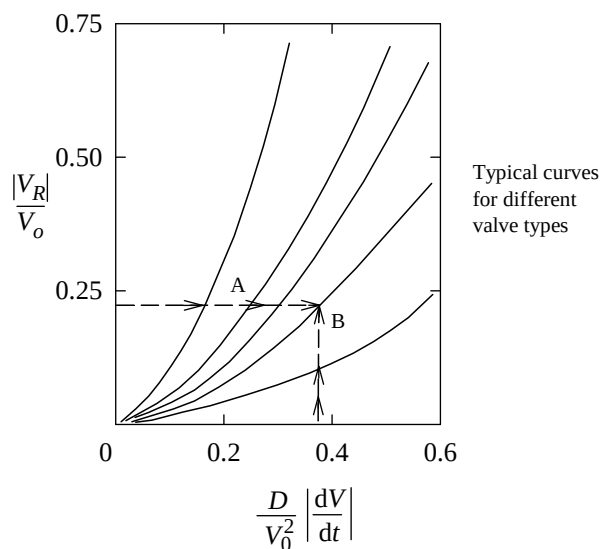
but confirmation from the manufacturer should be sought.

- (5) Repeat the velocity analysis procedure of (1) using the computer program, except that a backflow of magnitude V_R (expressed by V_R/V_{fo}) is allowed. Estimate the modulus of the rate of change of flow velocity $|dV/dt|$, up to the point at which the flow reaches the reverse velocity, V_R (or the maximum value of reverse flow velocity if this value is not reached), from the computed values of flow velocity versus time at the pump discharge side. Multiply the parameter dV/dt by D/V_{fo}^2 to enable a suitable one-way valve to be selected using the valve selection diagram, constructed in (1). On this diagram, construct the two bounds A and B, where A lies at the ordinate value $|V_R|/V_0 = |V_R|/V_{fo}$ as calculated in (4), and B lies at the abscissa value $(D/V_0^2)(|dV/dt|)$. Suitable valves are those that cross the bound B below A, as shown on Sketch 3.8b.

More guidance on the use of one-way valves for surge protection is given in Reference 17. It must be noted that the check valve will not close in the transients analysis if an unrealistically large value of V_R is chosen, and the values of the flow parameters calculated after flow reversal are likely to be unrealistic.



a. Manufacturer's data

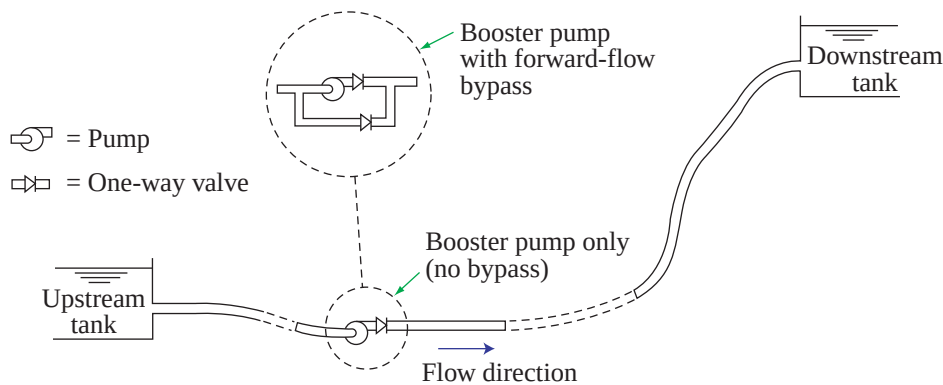


b. Derived performance curves

Sketch 3.8 Typical one-way valve selection curves

The system may be further protected by a bypass circuit that allows the full flow of liquid to bypass the pump when the pressure on the discharge side falls below the pressure on the suction side following pump trip (see Sketch 3.9). This bypass includes a one-way valve that closes completely when, at a later stage in the transient cycle, the flow reverse velocity through the bypass falls to V_R . The bypass is normally isolated from the flow during normal steady-state flow conditions.

The effect of bypass circuits on the suppression of pressure surges following pump trip is explained in ESDU 85044.



Sketch 3.9 Simple discharge line with booster pump

The computer program will also allow pumping from pressurised tanks to be considered, provided tank head values include the extra amounts to allow for the increase in pressure above atmospheric.

The way in which the pump characteristic data are used in the fluid transients analysis is illustrated in Sections [A.3.7.4](#) and [A.3.7.5](#).

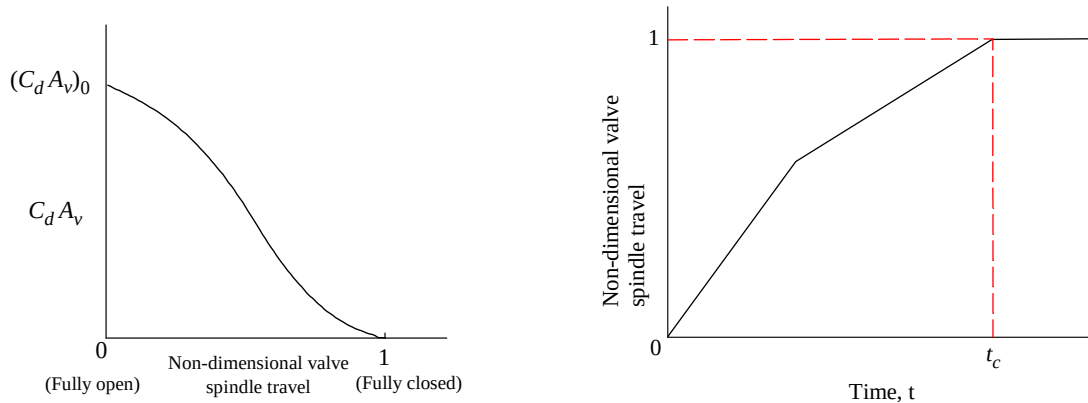
3.3.5 Valves

The magnitudes of pressure changes resulting from the closure of valves are dependent upon the type of valve and the time of closure in relation to the pipeline period, $2L/a$. For example, the way in which a valve shuts off may lead to a gentle decrease in the flow velocity and small increase in pressure as the valve starts to close, followed by a rapid decrease in velocity and increase in pressure as the valve nears total closure. In such a situation, the maximum pressure change is likely to be greater than that obtained for a linear rate of decrease of flow velocity (assuming total closure in the same period). The valve closure may be described by the variation of values of a dimensional parameter, $C_d A_v$, with time, which are instantaneous values of the effective flow area of the valve.

The flow velocity on the upstream side of the valve, V , is related to the head drop across the valve, H_d , by the equation

$$V = \frac{C_d A_v}{A} \sqrt{2gH_d}. \quad (3.12)$$

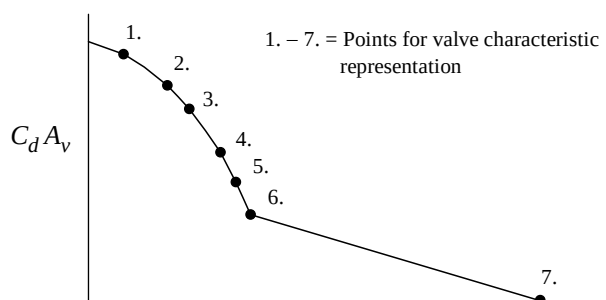
It may be necessary in some cases to construct the valve characteristic curve, particularly when non-linear valve operation, such as two-stage closure, is specified. The characteristic curve may be constructed by combining the curves of effective valve area versus valve spindle travel and of spindle travel versus non-dimensional time. Example curves of these types are shown in Sketch [3.10](#). The procedure for estimating the valve characteristic curve is as follows.



Sketch 3.10 Typical valve closure curve

- (i) From a plot of the effective flow area versus spindle movement, determine the fully-open effective valve area, $(C_d A_v)_0$, 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.
- (iii) For each value of spindle movement determined from (ii), estimate the corresponding value of the effective flow area, $C_d A_v$, from the sketch of effective area versus spindle movement.
- (iv) Plot these values of $C_d A_v$ against time.

A valve characteristic curve for valve closure is shown in Sketch 3.11. This curve has been obtained from information given in Sketch 3.10 using the above procedure.



Sketch 3.11 Valve closure characteristics

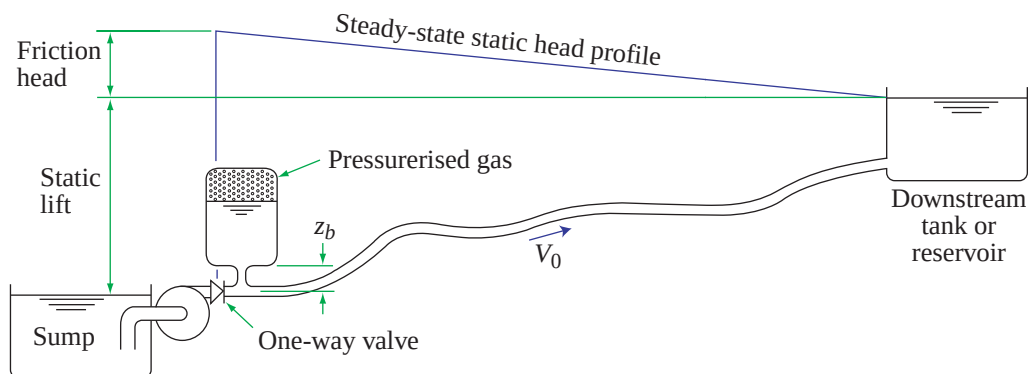
If the shape of the valve characteristic is to be modelled accurately in the computer analysis, values of $C_d A_v$ are input as a series of points at different values of time after time zero, the time at which the transient analysis begins. The value of $C_d A_v$ at any time t on the valve characteristic is calculated, within the program, by linear interpolation between the given values of $C_d A_v$. It may be seen that the closure characteristic on Sketch 3.11 may be described with reasonable accuracy by seven values of $C_d A_v$ (denoted by $(C_d A_v)_p$ at time t_p).

The way in which the valve characteristic data are input for the computer program is described in Sections A.3.7.6 to A.3.7.8.

The program will also model in-line valves, used to prevent backflow either when filling the lines or during surge. In practice, some reverse flow through the valve will occur, which can lead to higher values of surge pressure than those predicted assuming no backflow. The program allows an amount of backflow, expressed as a non-dimensional allowable reverse flow velocity, V_R/V_0 , where V_0 is the steady-state forward flow velocity through the valve. The procedure for one-way valve sizing for pump protection given in Section 3.3.4 may be used to size in-line one-way valves.

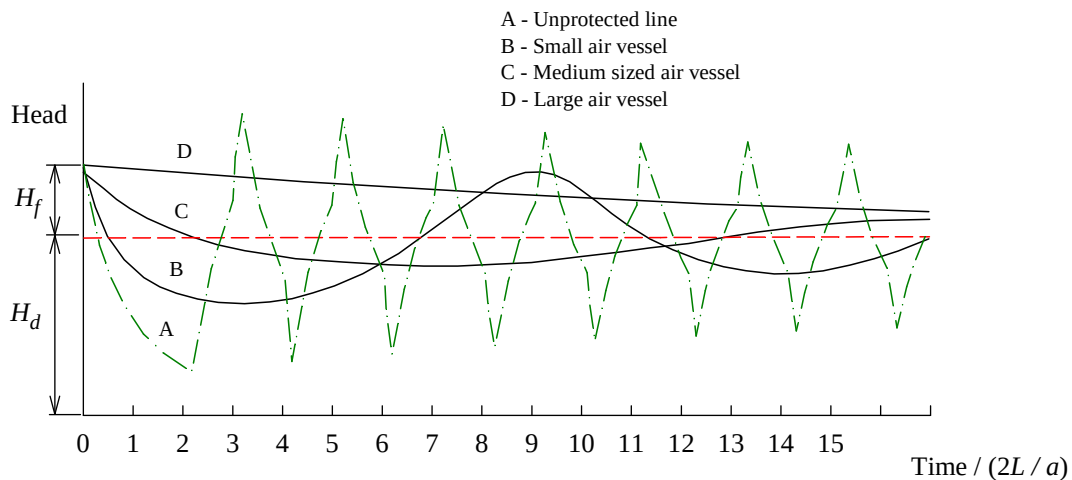
3.3.6 Air vessel

In certain systems, particularly rising water mains, an air vessel (or a group of air vessels), usually situated just downstream of the pump outlets and their one-way valves, may be an efficient and effective means of reducing the magnitudes of the surge pressures following pump trip. They may also be used to protect the upstream side of a line from surge pressures following valve closure; in this case they are located just next to the valve on the upstream side. The principles of air vessel operation, used for the protection of a pump discharge line (such as that shown in Sketch 3.12) from the effects of pump trip for example, are as follows. At the instant of pump trip, the pressure at the pump outlet falls as the liquid decelerates. Liquid is forced out of the air vessel into the pipeline, the pressure in the air vessel is reduced and the pressure in the liquid is maintained to some degree by this extra liquid reducing the deceleration of the flow. Eventually the velocity of the flow in the line falls to zero and the flow reverses. The pressure at the upstream end of the line starts to rise, and will reach a value exceeding the pressure in the air vessel when reverse flow into the vessel occurs until the liquid eventually decelerates and reverses direction again. The cycle of events then continues with decreasing pressure amplitudes as frictional effects in the pipe and energy losses in the chamber damp out the pressure changes.



Sketch 3.12 Typical pump discharge line protected by an air vessel

The rate of change of flow of the liquid, and hence the rate of change of pressure, is primarily dependent upon the initial volume of air in the vessel, v_0 . The lower the initial air volume, the more rapidly the pressure falls after pump trip. The total volume of liquid in the vessel must be large enough to prevent air being drawn into the pipe during downsurge. The effect of air vessel size on the head changes in a typical system is shown in Sketch 3.13. The rate of change of pressure is also governed to a lesser degree by the details of the losses associated with flow into and out of the vessel. Air vessels are generally throttled to increase the frictional damping effects. Usually, the resistance to flow into the vessel is made greater than that for flow out. This arrangement is usually beneficial in the reduction of the maximum upsurge and downsurge pressure changes, and may be achieved using a pipe with a one-way valve to allow flow out of the vessel and a small bypass to allow flow in. Alternatively, a multi-door one-way valve (or check valve) with one or more doors missing may be used to achieve this effect.

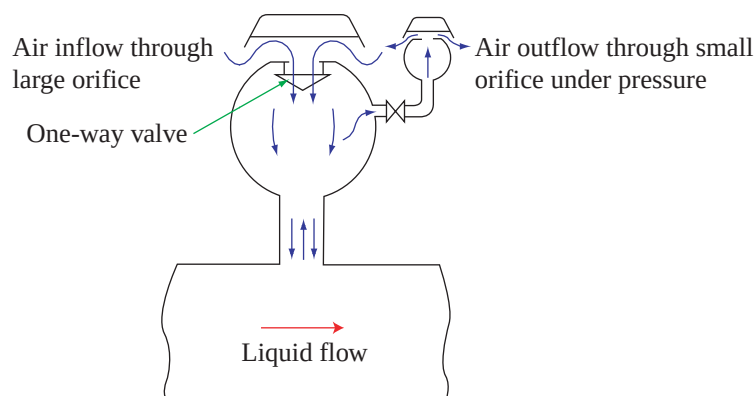


Sketch 3.13 Effect of air volume on surge suppression

Further guidance on the use of air vessels for surge suppression is given in ESDU 84038 (Reference 20). Design charts for air vessel sizing and other information concerning air vessel design and maintenance is also given in that Item and in Reference 10.

3.3.7 Air valves

Air inlet/outlet valves may be installed in certain types of pipelines to admit air into the line when the pressure at the air valve location falls to atmospheric following a surge event such as pump trip. In order to maintain the pressure near atmospheric, the air valve should open automatically to allow air freely into the line. The liquid column breaks at the air valve location and the resulting cavity is filled with air, reducing the pressure differential between the pressure in the pipeline and atmospheric pressure at the inlet vent of the valve. The air is retained in the pipeline temporarily and must be released slowly as the pressure increases. Throttling the outflow of air acts to prevent the violent collapse of the cavity between the two ends of the broken liquid column and helps prevent the large surge pressures that are associated with cavity collapse. Air inlet/outlet valves must allow sufficient air to flow freely into the line and ensure that the air is released slowly. This may be achieved using a differential throttle arrangement, with outflow through a small orifice for slow air release. It may be necessary to use a one-way valve arrangement to prevent outflow through the large inlet orifice. An idealised air valve arrangement suitable for surge control is shown in Sketch 3.14.



Sketch 3.14 Air inlet/outlet valves used for surge suppression

Air inlet/outlet valves may have a beneficial effect when used in pipelines that are protected against pressure surge by specially designed surge suppression devices. For example, air valves are often used as surge suppression devices on pump discharge lines protected by air vessels. Generally, smaller air vessels are required when high points in the line are protected by air valves.

Further guidance on the design and maintenance of air valves is given in ESDU 85009 (Derivation 4). Design charts that may be used to size air valves are presented in that Item. Further guidance on the use of air valves as surge suppression devices is given in Reference 16.

3.4 Statement of Objectives of Analysis

A clear statement should be given of the possible normal operating and fault conditions that may cause pressure changes in the system, and that (if not all) are to be investigated. This statement should include notes on such features as valve positions (*i.e.* those normally left open/closed), *etc.*

Having specified the conditions to be studied, there should be a proper understanding of what results are expected. These are, for the computer program in this Item,

- (i) computed maximum and minimum pressures over the whole system, allowing maximum and minimum head envelopes from computed data to be compared with those from upper and lower design limits, and
- (ii) head versus time data for chosen locations in the system.

These data allow the engineer to decide whether the magnitudes of pressure changes exceed the design limits of the system. If these limits are exceeded, the engineer may then decide on one of the following courses.

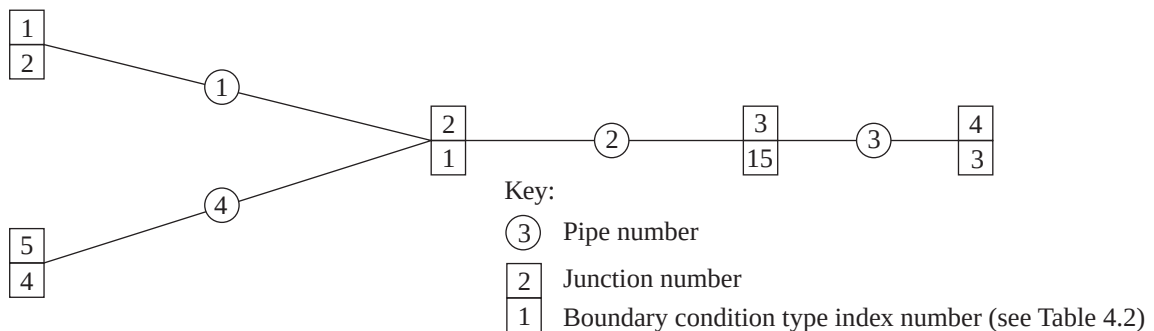
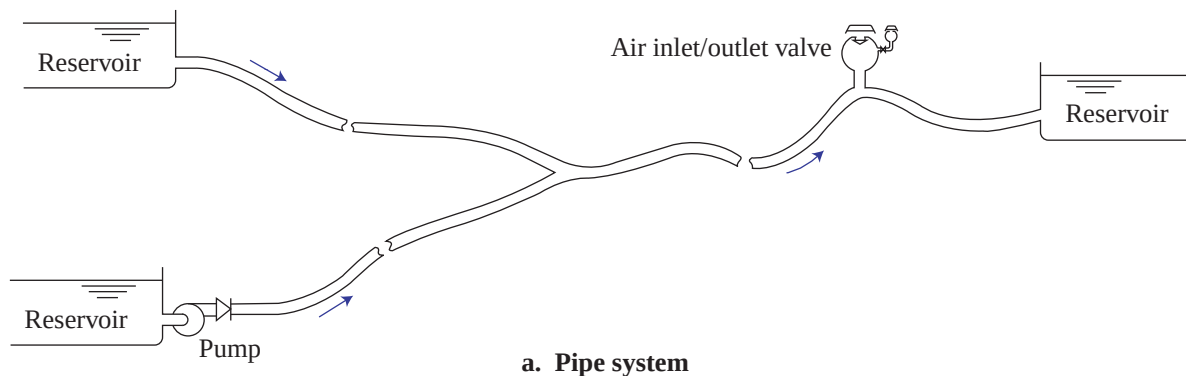
- (i) Alter the characteristics of the device causing the transients (*e.g.* increase pump inertia or valve closure time), or change appropriate pipeline properties (*e.g.* re-route a section of pipeline to avoid an elevated section), or
- (ii) use a surge suppression device (or a number of devices) to reduce the magnitude of the pressure changes.

It is advisable to consult simple design guides, such as those in the other Items in this series, to investigate the likely effect on the pressure changes of these options. Graphical data may allow the effect of changing certain parameters to be assessed at a glance, and may help in the initial sizing of surge suppression devices such as air vessels or air inlet/outlet valves. Further guidance on graphical methods of transient analysis is given in Section 2.3.

4. COMPUTER MODEL OF PIPING SYSTEM

4.1 System Representation

The piping system is represented by an arrangement of pipes that connect either to another pipe or a component (such as a pump, valve or constant pressure point) at both the upstream and downstream ends. Junctions are numbered individually from 1 to N_{JUNCT} , the number of junctions in the network, and pipes are numbered individually from 1 to N_{PIPES} , the number of pipes in the network. The way in which the junction and pipe numbers are assigned is not important, although it is advisable to adopt a simple sequential scheme for ease of interpretation. A typical numbering scheme is illustrated in Sketch 4.1.



Sketch 4.1 Example pipe system

Each numbered junction, which may be either an extremity of the network or an internal connection of more than one pipe or a component, requires additional data to specify the appropriate boundary condition. The types of boundary condition that may be modelled using the computer program in this Item are shown in Table 4.2.

4.2 Steady-state and Transient Solution Using Method of Characteristics

The program calculates the steady-state values of head and flow velocity throughout the network prior to starting the transient analysis. This steady-state solution is obtained as a special case of a fluid transients analysis in which head and flow velocity values converge to the stable, steady-state solution. This solution is calculated using the steady-state operating conditions of the components at the pipe junctions.

The version of the method of characteristics used in this Item requires each pipe in the model to be divided into a number of segments. Although the segments in a given pipe must be of equal length, the segment