



# ESDU 84013

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## **Fluid transients in pipes. Reduction and control of pressure surges in liquids**

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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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(Continued on inside back cover)

## FLUID TRANSIENTS IN PIPES. REDUCTION AND CONTROL OF PRESSURE SURGES IN LIQUIDS

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## FLUID TRANSIENTS IN PIPES

### Reduction and control of pressure surges in liquids

#### 1. NOTATION AND UNITS

|              |                                                                                                              | <i>SI</i>       | <i>British</i>     |
|--------------|--------------------------------------------------------------------------------------------------------------|-----------------|--------------------|
| $A$          | cross-sectional area of pipe                                                                                 | $\text{m}^2$    | $\text{ft}^2$      |
| $A_v$        | instantaneous cross-sectional area of valve                                                                  | $\text{m}^2$    | $\text{ft}^2$      |
| $a$          | speed of propagation of pressure waves                                                                       | $\text{m/s}$    | $\text{ft/s}$      |
| $D$          | internal diameter of circular pipe                                                                           | $\text{m}$      | $\text{ft}$        |
| $D_E$        | equivalent diameter (also known as hydraulic diameter) of pipe of non-circular cross section, $D_E = 4A/P_E$ | $\text{m}$      | $\text{ft}$        |
| $f$          | *Fanning friction factor,<br>$f = \frac{p_f D_E}{2L\rho V_0^2} \text{ or } \frac{H_f D_E g}{2L V_0^2}$       | —               | —                  |
| $g$          | acceleration due to gravity, $g = 9.81 \text{ m/s}^2$ or $32.2 \text{ ft/s}^2$                               | $\text{m/s}^2$  | $\text{ft/s}^2$    |
| $H$          | pressure head of liquid                                                                                      | $\text{m}$      | $\text{ft}$        |
| $H_a$        | throttle head loss                                                                                           | $\text{m}$      | $\text{ft}$        |
| $H_d$        | difference in head at location of disturbance and at point of reflection                                     | $\text{m}$      | $\text{ft}$        |
| $H_f$        | friction head loss                                                                                           | $\text{m}$      | $\text{ft}$        |
| $H_0$        | local elevation of pressure profile at steady-state conditions                                               | $\text{m}$      | $\text{ft}$        |
| $H_R$        | pump rated head                                                                                              | $\text{m}$      | $\text{ft}$        |
| $H_v$        | vapour head of a liquid                                                                                      | $\text{m}$      | $\text{ft}$        |
| $\Delta H$   | change in pressure head due to flow disturbance                                                              | $\text{m}$      | $\text{ft}$        |
| $\Delta H_D$ | maximum downsurge head (see Section 5.2.1)                                                                   | $\text{m}$      | $\text{ft}$        |
| $\Delta H_U$ | maximum upsurge head (see Section 5.2.1)                                                                     | $\text{m}$      | $\text{ft}$        |
| $I$          | inertia of rotating components of pump                                                                       | $\text{kg m}^2$ | $\text{slug ft}^2$ |

For footnotes see end of Notation section.

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|              |                                                                                                                             |                             |                             |
|--------------|-----------------------------------------------------------------------------------------------------------------------------|-----------------------------|-----------------------------|
| $K$          | pipe and air vessel throttle loss characteristic (see Equation (5.3))<br>or<br>bulk modulus of liquid (see Equation (B1.1)) | $\text{N/m}^2$              | $^{\dagger}\text{lbf/ft}^2$ |
| $K_3$        | pipe loss characteristic (see Equation (5.4))                                                                               | –                           | –                           |
| $K_4$        | air vessel throttle loss characteristic (see Equation (5.5))                                                                | –                           | –                           |
| $L$          | pipe length                                                                                                                 | m                           | ft                          |
| $M_R$        | rated pump torque                                                                                                           | N m                         | ft lbf                      |
| $N_R$        | rated pump rotational speed                                                                                                 | $^{\ddagger}\text{rev/min}$ | rev/min                     |
| $P_E$        | internal perimeter of pipe                                                                                                  | m                           | ft                          |
| $P_R$        | rated pump shaft power                                                                                                      | W                           | $^{**}\text{ft lbf/s}$      |
| $p$          | pressure                                                                                                                    | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $p_1, p_2$   | static pressures at locations of disturbance or point of reflection (see Sketch 4.1)                                        | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $p_d$        | difference between pressures at location of disturbance and point of reflection                                             | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $p_f$        | pressure loss due to pipe friction                                                                                          | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $p_0$        | local pressure at steady-state conditions                                                                                   | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $p_v$        | vapour pressure of a liquid                                                                                                 | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $\Delta p$   | change in pressure due to flow disturbance                                                                                  | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $\Delta p_D$ | maximum downsurge pressure (see Section 5.2.1)                                                                              | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $\Delta p_U$ | maximum upsurge pressure (see Section 5.2.1)                                                                                | $\text{N/m}^2$              | $\text{lbf/ft}^2$           |
| $Q_R$        | rated volume flow rate of pump                                                                                              | $\text{m}^3/\text{s}$       | $\text{ft}^3/\text{s}$      |
| $t$          | time                                                                                                                        | s                           | s                           |
| $t_c$        | time for change in flow conditions to be complete (e.g. for valve to close completely)                                      | s                           | s                           |
| $V$          | flow velocity                                                                                                               | m/s                         | ft/s                        |
| $V_0$        | initial flow velocity at steady-state conditions                                                                            | m/s                         | ft/s                        |
| $\Delta V$   | change in flow velocity                                                                                                     | m/s                         | ft/s                        |

For footnotes see end of Notation section.

|           |                                                                                                                     |          |                            |
|-----------|---------------------------------------------------------------------------------------------------------------------|----------|----------------------------|
| $v$       | total volume of air in air vessel (see Section 5.2.1)                                                               | $m^3$    | $ft^3$                     |
| $v_0$     | initial volume of air in air vessel (see Section 5.2.1)                                                             | $m^3$    | $ft^3$                     |
| $z$       | elevation above datum                                                                                               | $m$      | $ft$                       |
| $\alpha$  | non-dimensional parameter, $\alpha = \frac{p_d}{\rho a V_0}$ or $\frac{g H_d}{a V_0}$<br>(see Sections 4.2 and 4.3) | —        | —                          |
| $\delta$  | pressure or head change factor (see Equations (4.6) and (4.11))                                                     | —        | —                          |
| $\eta_R$  | pump efficiency at rated conditions (0 to 1)                                                                        | —        | —                          |
| $\rho$    | density of liquid                                                                                                   | $kg/m^3$ | $\dagger\dagger slug/ft^3$ |
| $\rho'$   | non-dimensional pipeline characteristic (see Equation (5.1))                                                        | —        | —                          |
| $\sigma'$ | non-dimensional air vessel characteristic (see Equation (5.2))                                                      | —        | —                          |
| $\tau$    | non-dimensional parameter, defined by Equation (4.5) for<br>valve closure or Equation (4.9) for pump run-down       | —        | —                          |
| $\phi$    | non-dimensional valve closure parameter (see Section 4.2.2)                                                         | —        | —                          |

\* Another form of the friction factor that is four times the Fanning friction factor is in common use. As the same symbol,  $f$ , is often used for the other (larger) form, great care should be taken to ensure that the friction factor used is consistent with the definition used in this Item for the friction head loss. More guidance and charts for the estimation of the friction factor used in this Item are given in Reference 12.

$\dagger$   $1 \text{ lbf}/ft^2 = 1/144 \text{ lbf}/in^2$ .

$\ddagger$  The pump speed is given in rev/min for the pump data given in this Item.

\*\*  $1 \text{ ft lbf}/s = 0.001818 \text{ hp}$  (horsepower).

$\dagger\dagger$   $1 \text{ slug} = 32.174 \text{ lb}$  (pounds mass).

## Subscripts

$max, min$  refers to maximum or minimum value respectively

$rev$  refers to flow reversal

## 2. INTRODUCTION

### 2.1 Notes

This Item is one in a series of ESDU Data Items concerning fluid transients in pipes. In this Item, data are presented that may be used to calculate approximately the magnitude of transients in liquid-filled pipework systems. The data are applicable to a wide range of liquid flow systems, from rising water mains to oil pipelines, or domestic water systems. This Item is also applicable to tunnels, thick walled pipes and non-circular closed conduits running full.

Fluid transients are the pressure changes propagated through a contained fluid following a dynamic event, or disturbance, in the fluid such as a change in the flow rate. Other commonly used terms for these transient pressure changes include pressure surges, hydraulic transients and, in water applications, water hammer. The pressure changes arise from the change in the kinetic energy in the fluid following a dynamic event. This energy change leads to a change in the pressure in the fluid at the location of the disturbance, which then propagates through the fluid as a pressure wave. In addition to the changes in the internal steady pressure in the fluid, pressure transients may lead to significant changes in the stress levels in the walls of the pipe or tunnel. The magnitude of these stresses may be sufficient to cause local damage to the pipeline or fittings. Repeated pressure cycles may lead to fatigue failure of the pipe. 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.

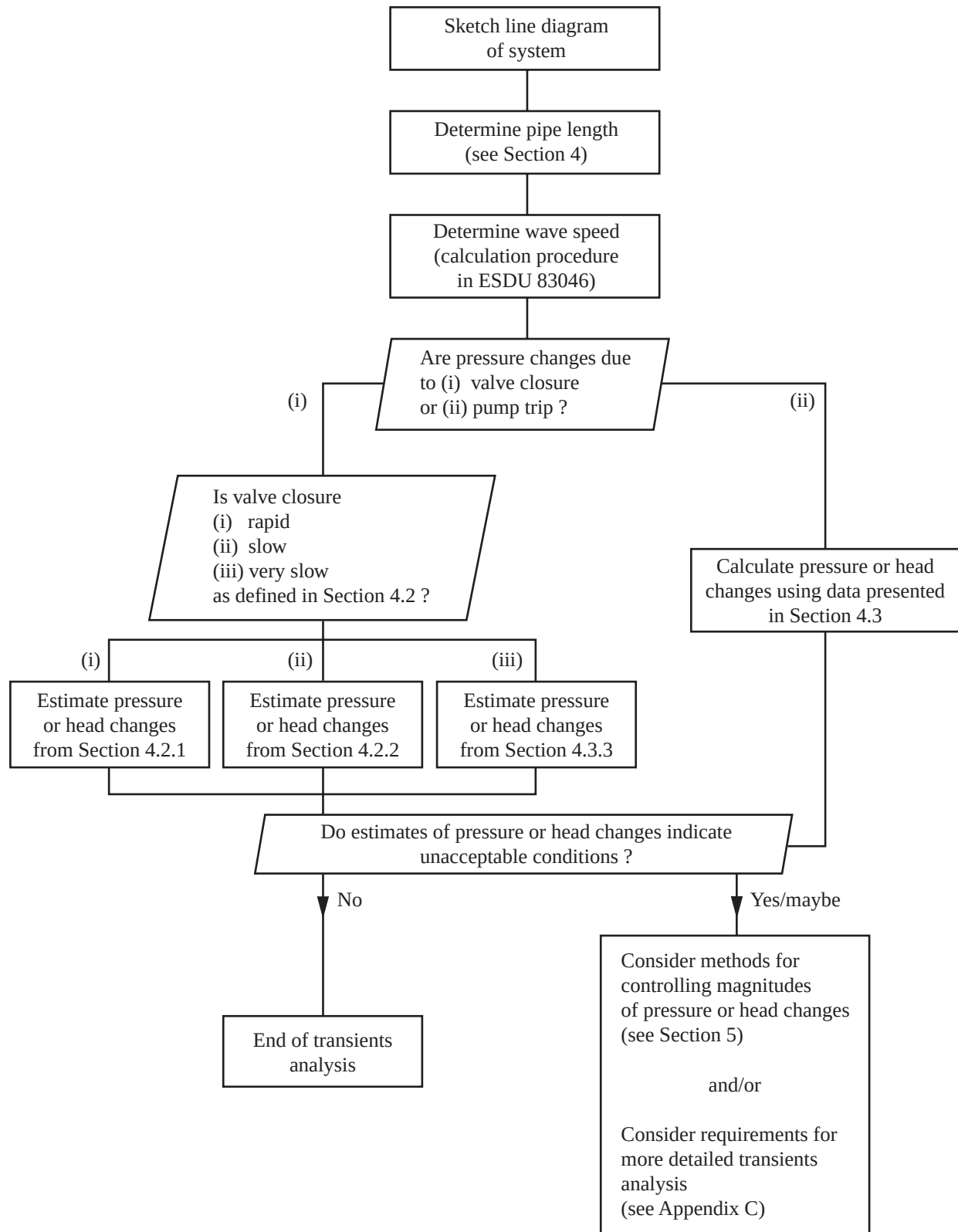
This Item is concerned with methods of reduction or control of fluid transients in liquids, and may involve either the alteration of the characteristics of the device causing the pressure change (such as increasing the inertia of a pump to increase the pump run-down time) or the attachment of auxiliary control devices to the system.

### 2.2 Scope and Applicability of the Item

This Item contains data necessary for assessing the scale of fluid transients problems, but by virtue of the large amount of computation required, the data presented are not sufficient for the detailed analysis of fluid transients in complex liquid flow applications. However, the data should enable engineers considering a complex liquid flow system to define the nature of the problem and familiarise themselves with various methods for the suppression of excessive pressure changes. The data are, in most cases, sufficient to enable engineers to decide whether there is a potential fluid transients problem in such a system. Some guidance is given on the basic data required for a fluid transients analysis. Guidance is also given to help engineers in the interpretation of the data resulting from this analysis.

The procedure for estimating the pressure changes in a fluid following a flow disturbance is shown schematically in Sketch 2.1. In Section 3, some typical causes of transient pressure changes are discussed, and simple graphical or numerical techniques for estimating the associated maximum pressure changes in such simple systems are given in Section 4. In Section 5, various methods for reducing these maximum pressure changes are considered. The effects of pipe friction and vapour cavities (and column separation) on the analysis of the pressure changes, and the problems of resonance in pipelines or components are discussed in Section 6. Three worked examples illustrating the use of the approximate methods of analysis are given in Section 6. A glossary of various terms used in the Item is given in Appendix A. The propagation of transient pressure waves in a simple system is illustrated in Appendix B, and the basic data required for a full transient analysis are detailed in Appendix C.

\* 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.



**Sketch 2.1 Procedure for system assessment**

## 3. TYPICAL CAUSES OF TRANSIENT PRESSURE CHANGES IN A CONTAINED LIQUID

Transient pressure changes will occur in a liquid whenever the flow or pressure is varied, during normal operation or due to a malfunction (or incorrect operation) of the system. The most common causes of disturbances in practical systems are outlined here.

- (i) **Changes in valve settings.** The maximum pressure changes in a system on valve closure may be estimated using the numerical and graphical techniques presented in Section 4.2. These data allow the user to investigate the effect of the variation of valve closure characteristics upon the maximum pressure changes in order to ensure that the pressures are within the maximum design limits for the pipe and pipe fittings under consideration.
- (ii) **Pump run-down.** The maximum pressure changes on pump run-down may be estimated using the data presented in Section 4.3. The effect of varying the pump run-down characteristics may be investigated using these data.

Some other less common causes of disturbances are listed below.

- (iii) **Pump start-up.** Large pressure changes may occur, depending upon the rate of change of pump speed, the length of the pipeline and the nature of the particular system. Serious disturbances associated with pump start-up could result from the rapid filling of empty pipelines, hoses and closed vessels and the rapid venting of gas release valves.
- (iv) **Filling of empty pipelines and hoses.** The maximum pressure changes are likely to be much higher than predictions based upon normal flow conditions, due to higher flow velocities. In order to keep the pressure changes low, the rate of filling of the lines must be kept as low as possible.
- (v) **Rapid chemical reactions and thermal changes (e.g. vapour generation).** Large pressure changes may be generated under these conditions. Some examples of these conditions are listed below.
  - (a) The introduction of water in hot oil; the vaporisation of water will lead to the instantaneous formation of a vapour cavity and an associated pressure rise.
  - (b) The collapse of steam pockets (rapid condensation) as either the temperature falls below vapour temperature or the pressure rises above vapour pressure.
  - (c) The reaction between liquids and molten metal coolants following coolant system fracture. For example, the very fast volume changes caused by the reactions between two streams that are normally separated, such as liquid sodium and water in nuclear reactor cooling circuits, may lead to the generation of very high pressures.
- (vi) **Resonance.** Some typical causes of resonance in liquid flow systems include the effects of reciprocating pumps, mechanical vibration of system components and instabilities in control equipment (see Section 6.4).

## 4. ESTIMATION OF THE MAXIMUM TRANSIENT PRESSURE CHANGES USING APPROXIMATE METHODS OF ANALYSIS

The data presented in this Section will enable estimates to be made of the maximum pressure changes in a liquid, contained in the single pipe systems modelled as shown in Sketches 4.1a and 4.1b, when the rate of flow of the liquid is changed.

These systems consist of

- (i) a constant pressure source (*e.g.* a reservoir or large tank) feeding liquid through a pipe to a closing valve, and
- (ii) a simple pump discharge line connected to a constant pressure source by a single pipe.

These data may also serve to give an estimate of the maximum pressure changes in a liquid contained in a branch of a complex pipe network in which a disturbance to the flow occurs. Usually, assuming a single disturbance in the system, this pressure change will be the maximum pressure change in the system. One exception to this rule occurs when a dead-end is located close to the source of the disturbance. The appropriate value of the pipe length,  $L$ , to be used in these approximations is discussed in Section 4.1.

Data are presented in Section 4.2 that enable estimates to be made of the maximum pressure changes in liquids resulting from valve closure. In Section 4.3, similar pressure change data are presented for pump run-down. For all these data, the effects of pipe or tunnel wall friction on the maximum pressure changes are neglected. In general, this friction has a beneficial effect in that it tends to damp down the oscillations in pressure. Even so, in some circumstances, such as cross-country oil pipelines with high friction head loss, the pressure may continue to rise above that which would be predicted if there was no friction in the pipeline (see Section 6.3).

The maximum pressure change data are expressed in terms of changes in the pressure,  $\Delta p$ , or head,  $\Delta H$ . The maximum pressure or  $H_{max}$ , (neglecting any build-up due to friction) at the location of the disturbance in a pipeline following the disturbance to the flow is given by

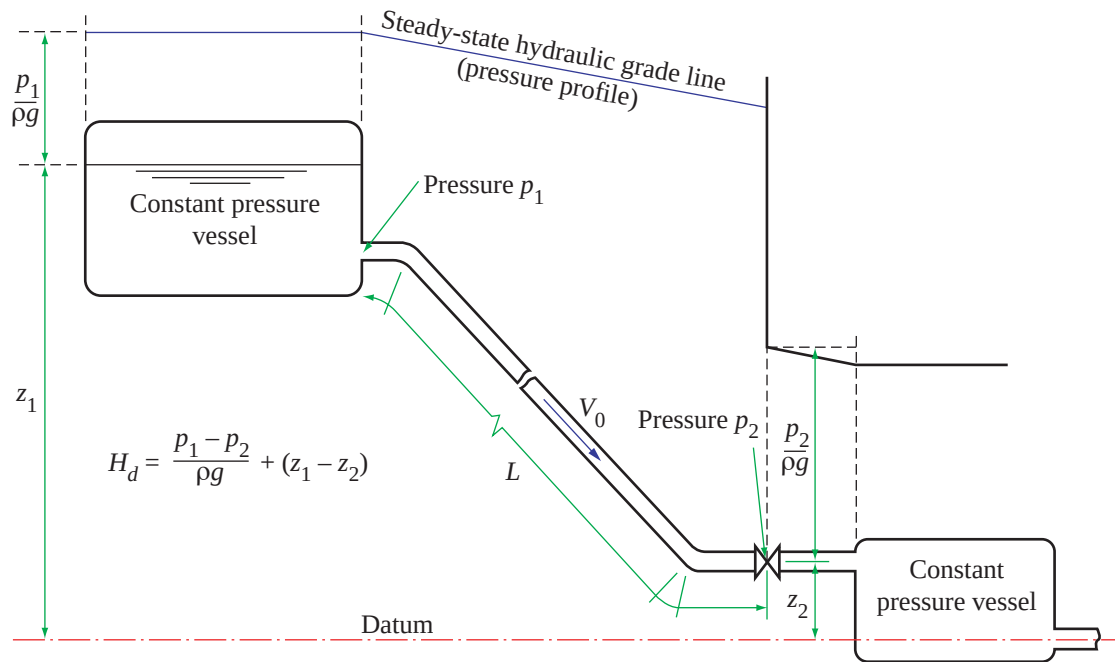
$$p_{max} = p_0 + \Delta p \text{ or } H_{max} = H_0 + \Delta H, \quad (4.1)$$

where  $p_0$  and  $H_0$  are the steady-state pressure and head at that location before the disturbance. The maximum pressure or head will be expressed in absolute or gauge pressure terms depending on the way in which  $p_0$  or  $H_0$  is expressed.

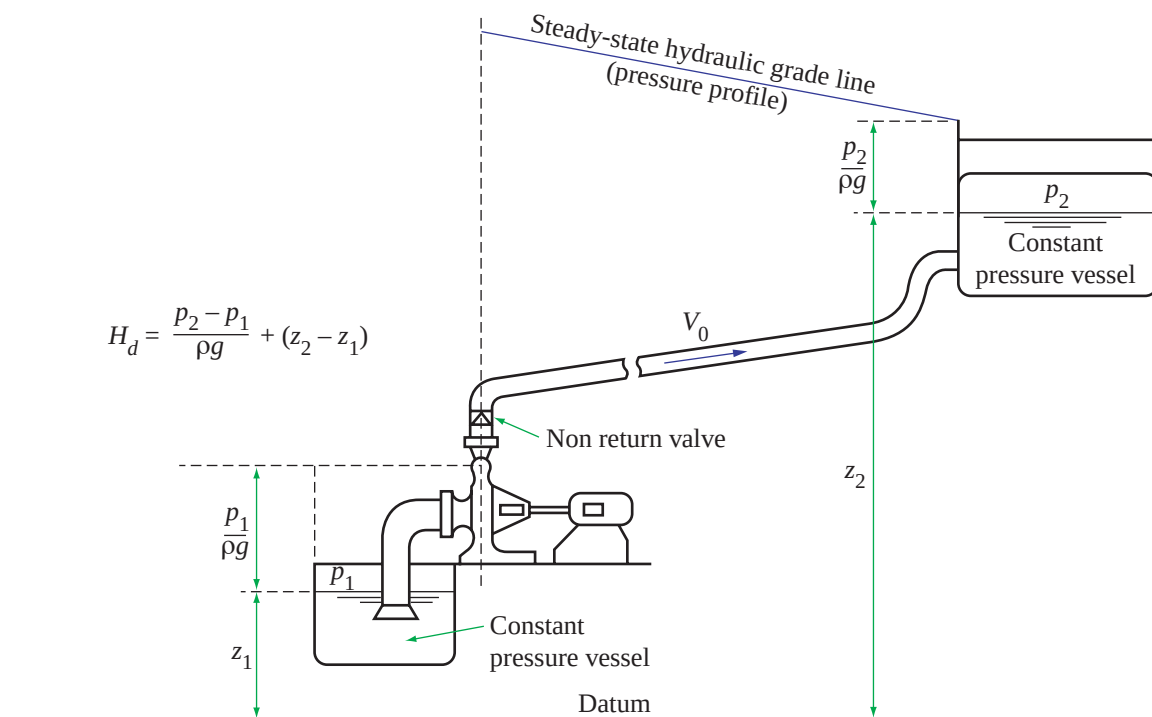
Similarly, the minimum pressure or head,  $p_{min}$  or  $H_{min}$ , at that location in the pipeline following the disturbance to the flow, may be expressed as

$$p_{min} = p_0 - \Delta p \text{ or } H_{min} = H_0 - \Delta H, \quad (4.2)$$

assuming that the minimum pressure or head,  $p_{min}$  or  $H_{min}$  is not less than vapour pressure or head,  $p_v$  or  $H_v$ . If it is, the minimum pressure or head achieved will generally be  $p_v$  or  $H_v$ . However, for very clean, pure liquids (unusual in practice) it is possible to achieve pressures below vapour pressure.



(a) Constant pressure source feeding to a valve

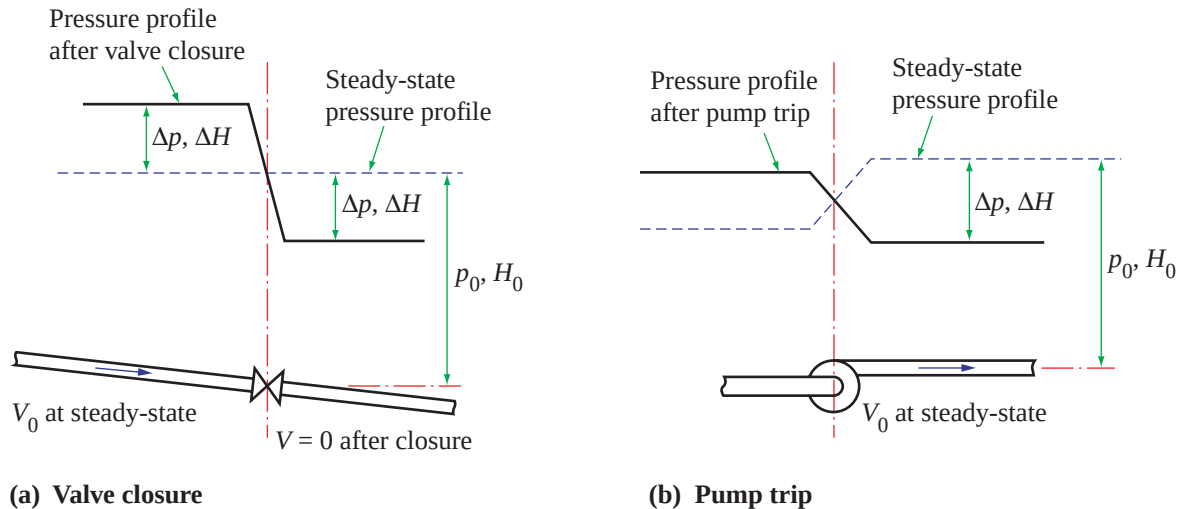


(b) Simple pump discharge line

## Sketch 4.1 Pressure and head terms for two simple one-pipe systems

Following an instantaneous event in a liquid, the maximum pressure change will normally occur at the location of the disturbance. The magnitude of the pressure change at a location in the pipe away from the location of the disturbance is dependent on the nature of the point of reflection. For the simple systems shown in Sketch 4.1 for example, the pressure change along the length of the pipe will vary from  $\Delta p$  (or

$\Delta H$ ) at the location of the disturbance to zero at the (constant pressure) point of reflection. The pressure profile at a valve following instantaneous valve closure and at a pump following instantaneous pump trip are shown in Sketch 4.2 to illustrate the senses of these pressure changes. However, these conditions may not be true if vapour cavities are present or in systems with a large head loss due to friction (see Section 6.3). An explanation of the propagation of transient pressure waves and their variation with time after a flow disturbance is given in Appendix B.



**Sketch 4.2 Locations of maximum and minimum pressure change after flow disturbances**

## 4.1 Pipe Length, $L$

The magnitude of the pressure changes in a liquid following a flow disturbance is dependent upon the time taken for a pressure wave to propagate through the liquid from the location of the disturbance to a point of reflection and back again. Hence, in order to estimate the pressure changes associated with fluid transients in a contained fluid, it is necessary to determine the effective length of the pipe along which these transient pressure waves propagate. The first step in the estimation of the pressure changes is to idealise the system and draw a simple line diagram. The pipe length,  $L$ , is then taken as the distance along the axis of the pipe line (including pipe bends *etc.*) from the location of the disturbance to the location of the first point of reflection of the pressure waves (unlike some pressure drop calculations, equivalent lengths of fittings are not added to the straight pipe lengths). Such points of reflection occur at the ends of uniform sections of pipe; typical examples are pumps, partially or fully-closed valves, dead-ends, constant pressure points, junctions of two or more pipes, or even changes in pipe wall material or thickness. In general, significant pressure waves are not reflected at pipe bends, *etc.* Typical examples of these points of reflection are discussed below.

- (i) **Constant pressure points (such as reservoirs, pressure vessels or large tanks).** Pressure waves are reflected with opposite sign and undiminished amplitude.
- (ii) **Tanks.** Pressure waves are reflected with opposite sign but their amplitude may be diminished depending upon the pressure loss to the tank (or geometry of the tank).
- (iii) **Constrictions (such as partially-closed valves, orifice plates, filters, *etc.*).** Pressure waves are partially reflected upon reaching a constriction. The proportion of the original pressure wave reflected is dependent upon the loss coefficient of the constriction.
- (iv) **Dead-ends.** Pressure waves are entirely reflected without change of sign. The net pressure change is twice that of the incident pressure wave, unless modified by the presence or formation of vapour cavities.

- (v) **Junctions of three or more pipes.** A proportion of the energy associated with the pressure wave returns to the point of disturbance as a reflected wave. The transmission factors are dependent upon the relative cross-sectional areas and wave speeds associated with each pipeline at the junction. The remainder of the energy associated with the initial pressure wave is transmitted down the other pipes at the junction as pressure waves. The magnitudes of these pressure waves in the other pipes are equal irrespective of their relative cross-sectional areas.
- (vi) **Changes in pipe cross-sectional area.** Pressure waves are partially reflected upon reaching a change in the pipe cross-sectional area. The proportion of the pressure wave reflected is dependent upon the relative cross-sectional areas and wave speeds in the contained liquid in the two sections of the pipe.
- (vii) **Changes in wave propagation speed for constant internal diameter.** Pressure waves are partially reflected upon reaching a change in the wave speed. This change may occur for constant internal diameter pipes where the wall thickness or wall material changes. The proportion of the pressure wave reflected at the location of the change in wave speed is dependent upon the relative cross-sectional areas and wave speeds of the two sections of the pipe.

Additionally, pressure waves may be reflected at the locations of equipment in the pipeline if significant changes in flow conditions are experienced at these locations. More guidance upon pressure wave reflection is given in Appendix B. Additional guidance is given in Derivations 3 and 4 and References 5, 11, 13 and 14.

## 4.2 Pressure Changes After Valve Closure

The pressure change data in this Section are applicable to a wide range of valves, for either complete or partial closure. The data are given for three distinct regimes dependent upon the relationship between the time taken for the change in flow velocity to be complete,  $t_c$  (i.e. the time taken for complete closure of a valve) and the pipeline period,  $2L/a$ . These pressure regimes are defined as follows.

- (i)  $t_c \leq 2L/a$  : the change in the flow conditions is said to be 'rapid' (see Section 4.2.1).
- (ii)  $2L/a < t_c < 500 \times 2L/a$  : the change in the flow conditions is said to be 'slow' (see Section 4.2.2). Note that for  $t_c$  equal to 2 or 3 times the pipeline period the pressure changes are of the same order as for a 'rapid' change in flow conditions.
- (iii)  $t_c \geq 500 \times 2L/a$  : the change in the flow conditions is said to be 'very slow' (see Section 4.2.3).

The pipe length,  $L$ , is defined in Section 4.1, and the wave speed,  $a$ , may be estimated using the data presented in ESDU 83046 (Reference 16).

The maximum pressure change predicted for 'rapid' changes in flow conditions is an upper limit on the pressure change, neglecting friction and cavity collapse. These effects are discussed in Section 6.

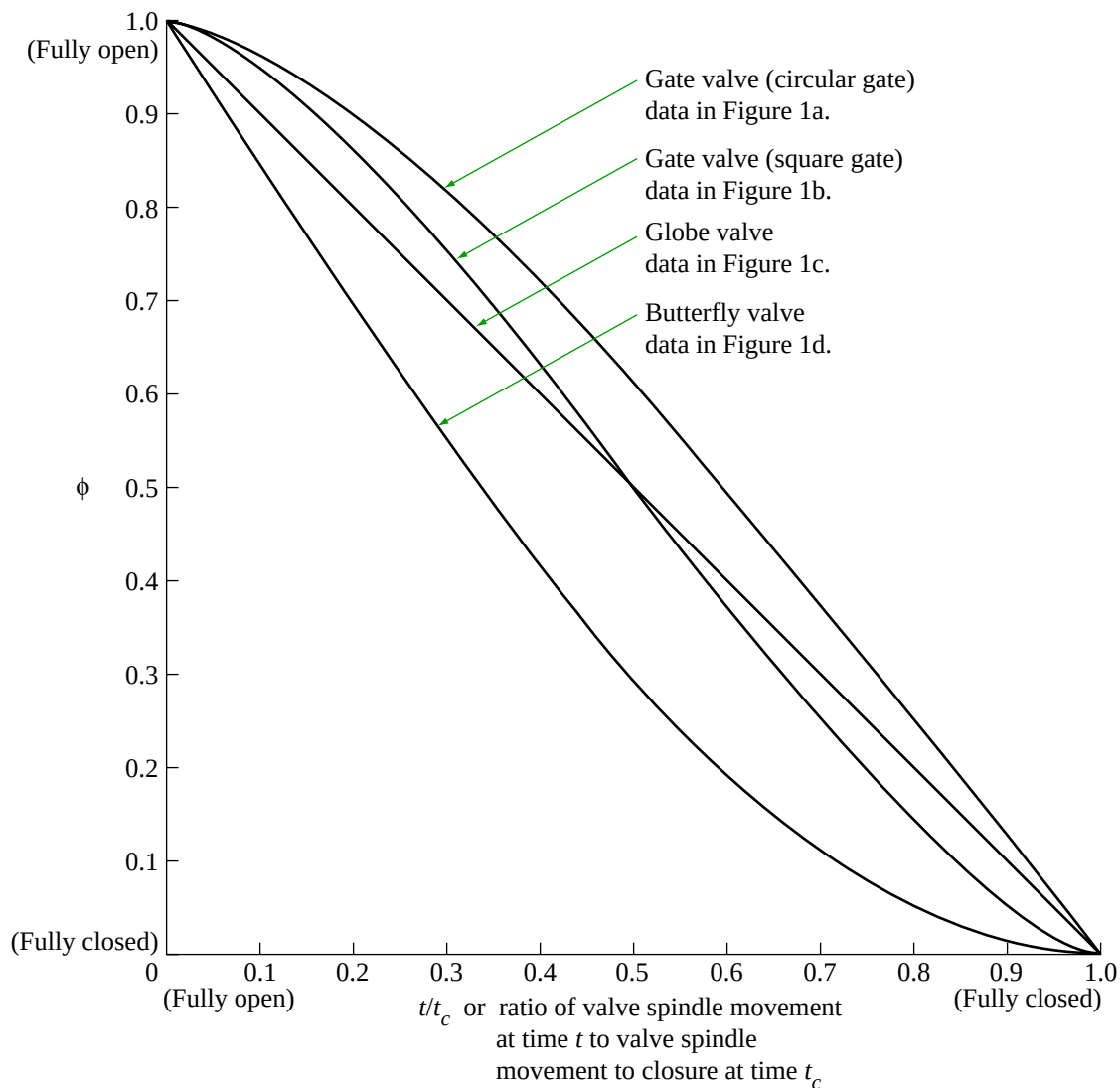
### 4.2.1 'Rapid' change in flow conditions

If the duration of a flow disturbance,  $t_c$ , is less than or equal to the pipeline period,  $2L/a$ , the magnitude of the resulting maximum pressure change (or head) in the pipeline is given by

$$\Delta p = \rho a \Delta V \quad \text{or} \quad \Delta H = \frac{a \Delta V}{g} \quad (4.3)$$

Such a maximum pressure change (known as the Joukowsky head) can occur at the upstream side of a closing valve and will propagate through the pipeline from the location of the disturbance to a point of reflection. The maximum and minimum pressures in the pipeline (neglecting friction effects) are given by Equations (4.1) and (4.2) (except if the pressure falls to vapour pressure where the minimum pressure will be vapour pressure).

## 4.2.2 'Slow' change in flow conditions



**Sketch 4.3 Valve closure characteristics**

If the time taken for the valve to close is greater than the pipeline period,  $2L/a$ , the maximum pressure change is less\* than the pressure change resulting from 'rapid' flow shut-off. This is due to the beneficial effects of reflected pressure waves. In such cases, graphical or numerical techniques must be used to estimate the maximum pressure changes. Some data are presented in Figures 1a to 1d that enable estimates of the maximum pressure changes at the valve in a simple, one-pipe system to be made. These data (obtained using Derivation 1) are applicable to systems with flow from a constant pressure source (such as a reservoir or a large tank) along a frictionless pipeline to discharge through a valve, but may be used to give estimates of the pressure changes in other similar simple one-pipe systems. The magnitude of the pressure change is

\* The reduction in the pressure change increases as  $t_c$  increases. However, for  $t_c$  up to 2 or 3 times the pipeline period, this reduction is usually insignificant.

dependent upon the method of valve closure, and hence the way in which the flow changes. For example, the way in which a valve shuts off may lead to a gentle decrease in the flow velocity as the valve starts to close followed by a rapid decrease in the velocity as the valve nears total closure. In such a situation, the pressure change would be greater than that obtained for a linear rate of decrease of flow velocity (assuming total valve closure in the same period). The way in which the flow changes may be expressed using the non-dimensional parameter,  $\phi$ . For a valve closure,  $\phi$  is expressed as the ratio of the instantaneous area of the valve (partly-closed) to the fully-open area of the valve, and will vary with the ratio of the valve spindle movement at any instant to the total movement to closure. Alternatively,  $\phi$  may be plotted against valve closure time. Some data are presented in this Section for the estimation of the pressure changes following the closure of four types of commonly-used valves. These valves are:

- (i) gate valve (circular gate),
- (ii) gate valve (square gate),
- (iii) globe valve, and
- (iv) butterfly valve.

The closure characteristics for these four valve types are shown in Sketch 4.3.

To use the data, two dimensionless parameters defining the initial conditions in the system and the way in which the valve closes must be defined. The parameter,  $\alpha$ , defining the initial conditions in the system, may be expressed using either pressure or head terms, and is given by

$$\alpha = \frac{p_d}{\rho a V_0} \quad \text{or} \quad \alpha = \frac{g H_d}{a V_0}. \quad (4.4)$$

In Equation (4.4),  $p_d$  is the difference between the pressure at the supply and at the exit in the pipeline, and  $H_d$  is the difference between the heads at the same points. In Sketch 4.1a, these terms are shown for the simple system of a constant pressure feed to a discharge valve. The dimensionless parameter defining the rate of flow change is given by

$$\tau = t_c \times \frac{a}{2L}, \quad (4.5)$$

where  $t_c$  is the time taken for the change in flow conditions to be complete. For valve closure this is equal to the time taken for the valve to close completely.

The pressure change,  $\Delta p$ , or head change,  $\Delta H$ , may be estimated using the data presented in Figures 1a to 1d. Data are presented in Figures 1a, 1b, 1c and 1d for linear closure of gate valves (circular gate), gate valves (square gate), globe valves and butterfly valves, respectively. These data are presented for a wide range of values of  $\alpha$  and  $\tau$ . The maximum pressure or head change may be expressed as a proportion of the full pressure or head change predicted for rapid changes in flow conditions, and is given by

$$\Delta p = \delta \times \rho a \Delta V \quad \text{or} \quad \Delta H = \delta \times \frac{a}{g} \Delta V, \quad (4.6)$$

where  $\delta$  denotes the proportion of the 'rapid-change' pressure or head.

The effect of changing the valve closure time or the type of valve may be readily investigated using the data presented in this Section. An estimate of the pressure changes following the closure of other types of valves may be made using the data for valves with closure characteristics most similar to those of the valve under consideration. Alternatively, the pressure changes may be estimated for valves with any chosen form of closure characteristic using the computer program ESDUpac A8727 introduced in ESDU 87027.

## 4.2.3 'Very slow' change in flow conditions

The maximum change in pressure (or head) following a valve closure in a time much greater than the pipeline period,  $2L/a$  (of the order of 500 times the pipeline period or longer) may be estimated using the approximate expression

$$\Delta p = \rho L \frac{dV}{dt} \quad \text{or} \quad \Delta H = \frac{L}{g} \frac{dV}{dt}. \quad (4.7)$$

The term  $dV/dt$  is the rate of change of the fluid velocity, or deceleration, and in the sign convention adopted in this Data Item is always taken as positive in Equation (4.7). In practice,  $dV/dt$  is not normally constant, although in most cases it is sufficient to use the approximation  $dV/dt = V_0/t_c$  which will not introduce appreciable error.

## 4.3 Pressure Changes After Pump Trip

Data are presented in this Section that enable estimates to be made of the maximum pressure changes following pump trip in a liquid contained in a simple one-line pump discharge system (shown in Sketch 4.1b). The magnitude of the pressure changes is dependent upon the initial conditions in the system and the way in which the pump runs down. The initial conditions may be expressed by the parameter  $\alpha$ , which is given by

$$\alpha = \frac{p_d}{\rho a V_0} \quad \text{or} \quad \alpha = \frac{g H_d}{a V_0}, \quad (4.8)$$

where  $p_d$  is the difference between the pressures at the supply and the exit in the pipeline, and  $H_d$  is the difference between the heads at the same points (see Sketch 4.1b). The pump run-down characteristics may be expressed by the non-dimensional parameter,  $\tau$ , given by

$$\tau = \frac{\pi^2 I N_R^2 \eta_R}{450 \rho g H_R Q_R} \times \frac{a}{2L}, \quad (4.9)$$

where  $H_R$ ,  $Q_R$  and  $N_R$  (in rev/min) are the rated values of the pump discharge head, flow and speed, respectively,  $I$  is the rotational inertia of the pump (including the gearbox, motor and fittings) and  $\eta_R$  is the pumping efficiency at the rated conditions (in the range 0 to 1). Equation (4.9) may also be expressed in terms of the shaft input power at the rated conditions,  $P_R$ , as

$$\tau = \frac{\pi^2 I N_R^2}{450 P_R} \times \frac{a}{2L}. \quad (4.10)$$

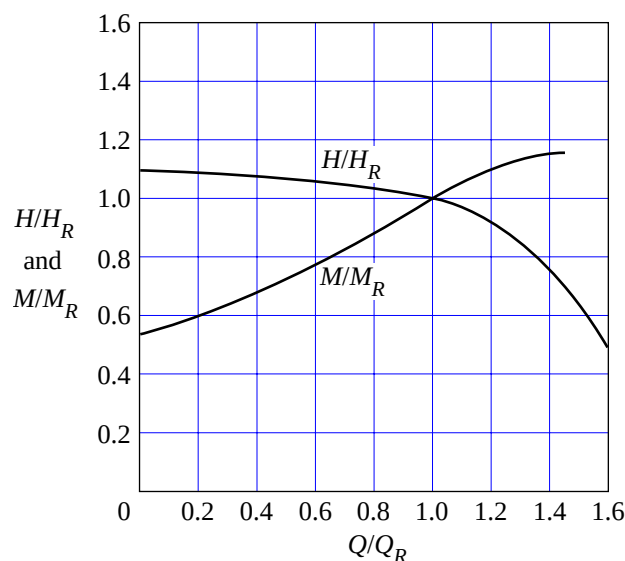
Data are presented in Figure 2a from which the maximum pressure changes at the pump outlet after pump trip may be estimated. These data, given for values of  $\alpha$  from 0.001 to 50 and values of  $\tau$  from 0.1 to 100, are given as a proportion,  $\delta$ , of the full 'rapid' pressure change (see Section 4.2.1). The pressure changes

may therefore be estimated using the equation

$$\Delta p = \delta \times \rho a \Delta V \quad \text{or} \quad \Delta H = \delta \times \frac{a}{g} \Delta V, \quad (4.11)$$

where  $\delta$  may be estimated from Figure 2a. The maximum and minimum pressure may then be estimated using Equations (4.1) and (4.2). These data were derived specifically for pumps with the characteristics shown in Sketch 4.4, but may be used to provide estimates of the pressure changes following run-down of other types of pumps with similar characteristic curves. In the pump curves shown in Sketch 4.4, the head, torque and flow are referred to their rated conditions by the non-dimensional terms  $H/H_R$ ,  $M/M_R$  and  $Q/Q_R$ , respectively.

The pressure change data were derived assuming that no reverse flow through the pump will occur. Consequently, it is assumed that the pump is protected by a one-way valve that shuts immediately upon flow reversal, thus preventing reverse flow. For centrifugal pumps that are not protected by such valves, the reverse flows may lead to very high reverse speeds of rotation of the pump. Some data are presented in Figure 2b that will enable the time taken for the flow to reverse to be estimated. These data are given for the same ranges of  $\alpha$  and  $\tau$  as the pressure change data. The time taken for the flow to reverse,  $t_{rev}$ , may be estimated for a chosen set of values of  $\alpha$  and  $\tau$  from this figure. The time for flow reversal is expressed by the non-dimensional parameter  $t_{rev} a / 2L$ , the number of pipeline periods.



Sketch 4.4 Pump head, torque and flow characteristics

## 5. METHODS FOR CONTROLLING EXCESSIVE TRANSIENT PRESSURE CHANGES

If the maximum pressure changes in a contained liquid are too large for the pipe or pipe fittings or system performance, it will be necessary either to consider techniques for controlling the magnitude of the pressure changes (in order to keep them within acceptable limits) or to strengthen the pipeline or pipe fittings (to accept the pressure changes). Techniques for controlling the pressure changes involve either the alteration of the characteristics of the device causing the flow disturbance, or the attachment of a protective device to limit the rate of change of the flow, and hence reduce the magnitude of the pressure changes.

The effects of altering the characteristics of the device causing the flow disturbance are discussed in Section 5.1, and various protective devices are considered in Section 5.2.

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## 5.1 Effects of Altering the Characteristics of the Device Causing the Flow Disturbance

In some cases, it may be possible to restrict the magnitude of the maximum pressure changes to within acceptable levels by altering the way in which the flow is affected by the device causing the transient pressure changes. Usually, this will involve slowing down the device to reduce the rate of change of flow velocity. For a valve, it may be possible to slow down the rate of valve closure to achieve this end. However, this may not be acceptable in some cases for operational reasons. Alternatively, it may be possible to use a two-part opening or closure, which may involve a slower rate of movement over the last 10 to 20 per cent of valve spindle travel. This technique would lead to a lower rate of change of flow velocity, and hence reduce the magnitude of the maximum pressure changes. The effect of altering the closure time of a valve or the effect of using a different type of valve may be investigated using the data presented in Section 4.2.2. Alternatively, the computer program ESDUpac A8727 introduced in ESDU 87027 may be used to give estimates of the pressure changes for any chosen form of valve closure characteristic.

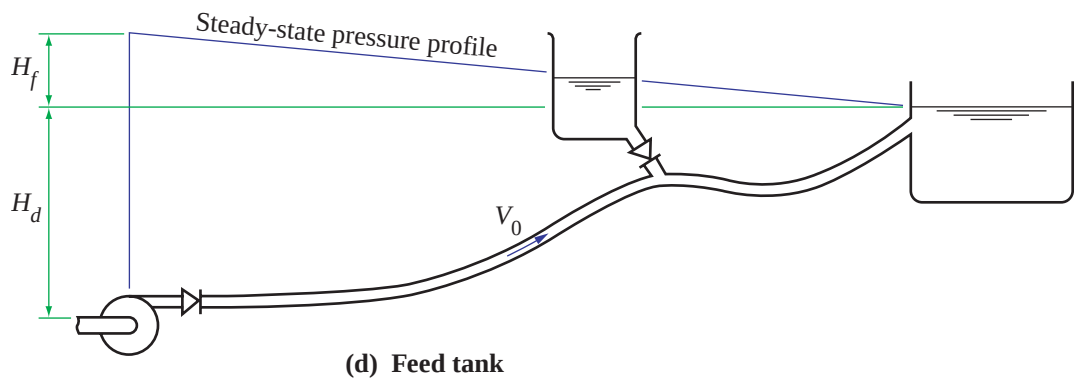
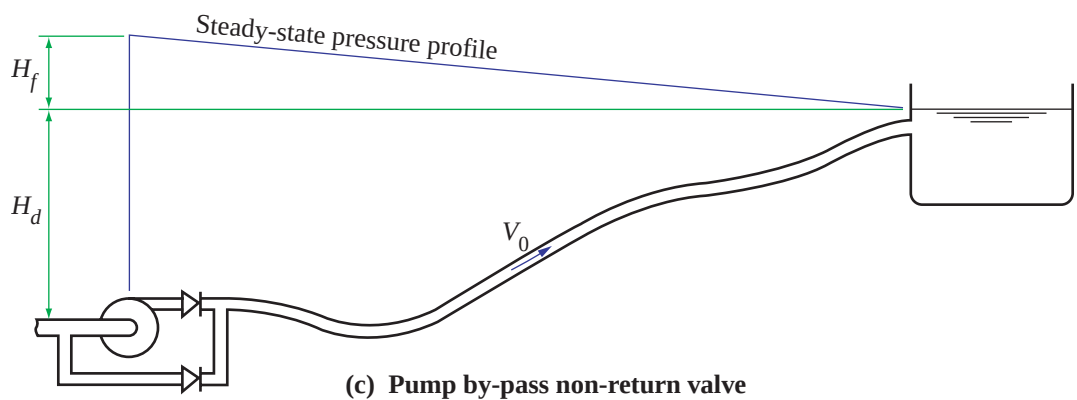
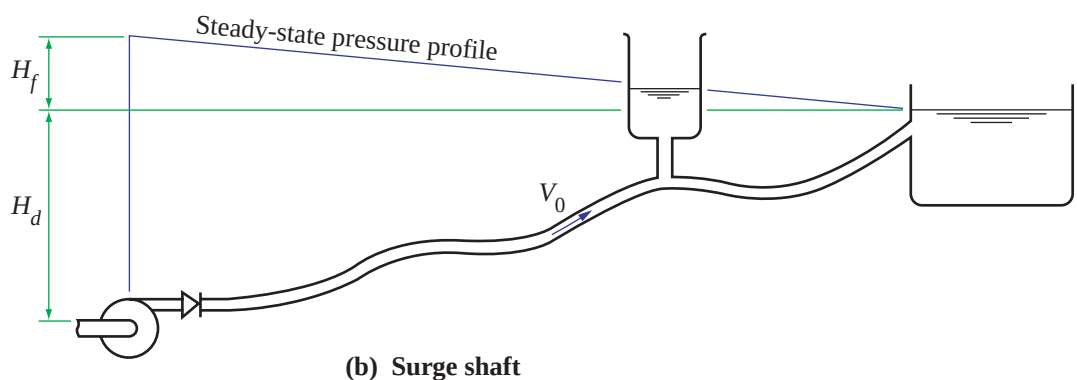
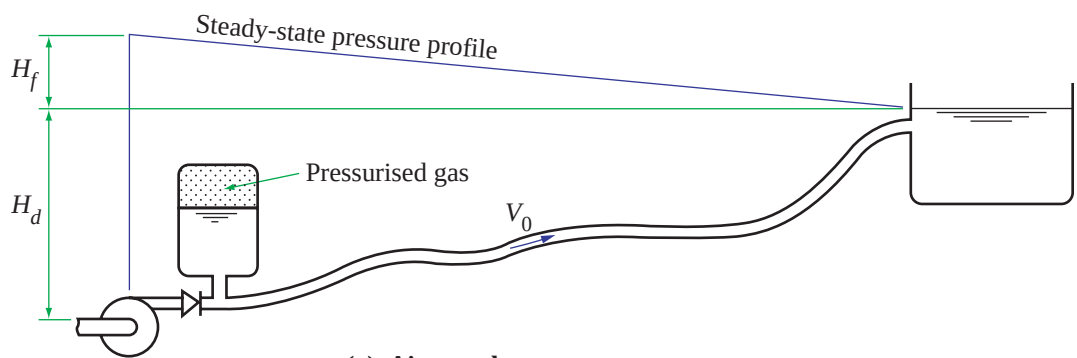
For a pump, it may be necessary to increase the start-up or run-down time by increasing the inertia of the rotating parts. This may be achieved by fitting a flywheel to the rotating parts of the pump. The effect of this measure is to decrease the rate of change of the flow velocity and hence the maximum pressure change (provided the flow is stopped or started in a time greater than  $2L/a$ ). The effect of altering the inertia of the rotating parts of a pump may be investigated using the data presented in Section 4.3. The computer program ESDUpac A8727 introduced in ESDU 87027 may be used to give a more detailed estimation of these effects. It may be noted that the addition of a flywheel to a pump in order to increase the rotational inertia may involve certain penalties, such as increased power loss in windage and the requirement for a more powerful starting system. This limits the application of this remedy to relatively short systems.

## 5.2 Other Methods of Controlling Excessive Pressure Changes

### 5.2.1 Air vessels and accumulators\*

Air vessels and accumulators are relatively small pressurised tanks containing a gas and the system fluid. A common application is to maintain the flow of fluid for a short period at a location in a system where the pressure falls following a flow disturbance. In this case, the flow of fluid may be brought to rest more slowly, reducing the magnitude of the pressure drop. An example of this application is in the control of fluid transients in a pipeline after pump failure, where air vessels are normally situated just downstream of the pump. Another example of their use in the limitation of large pressure rises is the protection of a pipeline following valve closure; in such cases, they are commonly situated at the upstream side of the closing valve. An air vessel (see Sketch 5.1a) will require a compressed air supply to ensure the air pressure remains within prescribed limits at all times that the pump is operational. An accumulator is effectively an air vessel in which the gas (usually nitrogen) is separated from the liquid by an elastic membrane, and is relatively maintenance-free (although generally more expensive to install). These devices are frequently used in oil and oil-hydraulic systems. Air vessels and accumulators are normally fitted with differential throttles for flow into and out of the vessel to allow the designer to optimise frictional damping.

\* Additional information on the use of these suppression devices will be found in ESDU 85009, and ESDU 87027 introduces a computer program that treats their design and use.



Sketch 5.1 Some control devices