



ESDU 85044

Issued December 1985
Reprinted at Amendment A
March 2007

Fluid transients in pipes Pressure surge following booster pump trip. Suppression using pump bypass

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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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This Item was also approved by the Vibration and Structural Dynamics Steering Group, which had the following constitution:

(Continued on inside back cover)

FLUID TRANSIENTS IN PIPES

Pressure surge following booster pump trip. Suppression using pump bypass

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FLUID TRANSIENTS IN PIPES

Pressure surge following booster pump trip. Suppression using pump bypass

1. NOTATION AND UNITS

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

		SI	British
A	internal cross-sectional area of pipe	m^2	ft^2
a	speed of propagation of pressure waves (speed of sound) in a liquid	m/s	ft/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
E	Young's modulus of elasticity of pipe wall material	N/m^2	$^*\text{lb/ft}^2$
E_{ref}	reference value of Young's modulus of elasticity of pipe wall material, $E_{ref} = 10 \text{ GN/m}^2$ or $2.09 \times 10^8 \text{ lb/ft}^2$	N/m^2	$^*\text{lb/ft}^2$
f	† Fanning friction factor, $f = gH_f D_E / (2LV_0^2)$ (Note: $D_E = D$ for circular pipe)	—	—
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_D	steady-state head on discharge side of pump	m	ft
H_d	steady-state head value of downstream tank	m	ft
H_D^{max}	maximum head on discharge side of pump following pump trip	m	ft
H_D^{min}	minimum head on discharge side of pump following pump trip	m	ft
ΔH_D	head change due to pump trip on discharge side of pump	m	ft
H_f	friction head loss	m	ft
H_p	static head across pump at steady-state conditions	m	ft
H_R	pump head at rated conditions	m	ft
H_S	steady-state head on suction side of pump	m	ft
H_S^{max}	maximum head on suction side of pump following pump trip	m	ft
H_S^{min}	minimum head on suction side of pump following pump trip	m	ft
ΔH_S	head change due to pump trip on suction side of pump	m	ft
H_u	steady-state head value of upstream tank	m	ft

For footnotes see end of Notation section.

I	inertia of rotating components of pump	kg m ²	‡slug ft ²
L	total pipe length	m	ft
L_D	length of pipe on discharge side of pump	m	ft
L_S	length of pipe on suction side of pump	m	ft
N_R	pump rotational speed at rated conditions	rev/mi n	rev/min
P_R	pump shaft input power at rated conditions	W	**ft lbf/s
Q_b	flow rate at instant bypass circuit opens	m ³ /s	ft ³ /s
Q_0	steady-state liquid flow rate	m ³ /s	ft ³ /s
Q_R	pump volume flow rate at rated conditions	m ³ /s	ft ³ /s
t	pipe wall thickness	m	ft
t_b	lapsed time between pump trip and bypass opening	s	s
t_{rev}	lapsed time between pump trip and flow reversal	s	s
V_0	steady-state liquid flow velocity	m/s	ft/s
x	non-dimensional pump position parameter, $x = L_s / L$ (see Section 4)	—	—
α	non-dimensional parameter, $\alpha = gH_p / (aV_0)$ (see Equation (4.1))	—	—
δ_x	head change factor for pump located at non-dimensional position $x = L_s / L$	—	—
η_R	pump efficiency at rated conditions (in the range 0 to 1)	—	—
ρ	density of liquid (typically 1000 kg/m ³ or 1.940 slug/ft ³ for water at 1 atmosphere and 15°C)	kg/m ³	‡slug/ft ³
τ	non-dimensional pump run-down parameter (see Equations (4.2a) and (4.2b))	—	—
ϕ_b	non-dimensional bypass flow rate parameter (see Equation (4.6))	—	—

* 144 lbf/ft² = 1 lbf/in²

† Another form of the friction factor that is four times as large as 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 of the friction head loss used in this Item. More guidance and charts for the estimation of the friction factor used in this Item are given in Reference 6.

‡ 1 slug = 32.2 lb (pounds mass).

** 1 ft lbf/s = 1/550 hp (horsepower).

2. BACKGROUND

2.1 Notes

This Item is one in a series of ESDU Data Items concerning fluid transients in pipes. In this Item, graphical data are presented that may be used to estimate the magnitude of the surge pressures (expressed in head of liquid) following power failure to a booster pump station located along a liquid-filled line. These data are applicable to centrifugal type pumps, with or without forward-flow bypass circuits fitted for surge suppression, in lines running full of liquid. General data for the estimation of transient pressure changes in liquids contained in pipelines and guidance on other methods of surge suppression are presented in ESDU 84013, 84038 and 85009 (References 11, 12 and 13). A computer program for the prediction of fluid transients in liquid-filled networks is provided in ESDU 87027.

Fluid transients are the pressure changes that propagate through a fluid contained in a pipe or tunnel following a change in the flow rate. Other commonly used terms for these pressure changes include pressure surges, water hammer and hydraulic transients. These pressure changes typically occur when pumps run down, and may cause local or widespread damage in the line and may lead to fatigue failure following repeated pressure cycles. The pressure drop associated with the run-down phase of pumps may lead to sub-atmospheric conditions at either side of the pump or at high points in the pipeline profile (or even along the whole length of the pipeline), which can lead to unacceptable conditions such as cavitation at the pump, seepage of contaminants through pipe joints or the formation of vapour cavities at high points in the line.

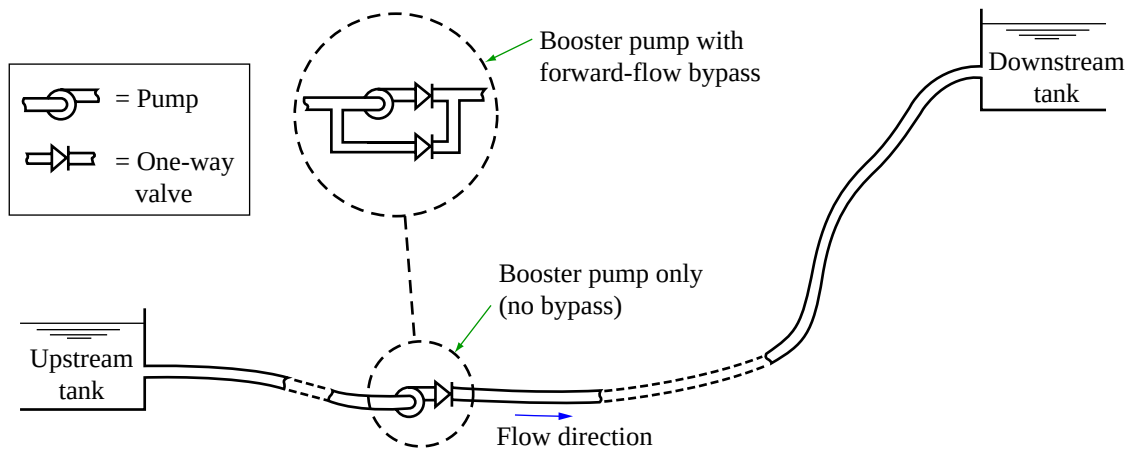
Without some means of surge suppression, the pressure rise that occurs after the discharge from the pump has fallen to zero may be high enough to cause local damage to the pipeline or fittings. The use of pump bypass circuits provides a means that may reduce the pressure changes at both the suction side and discharge side of a pump or set of pumps, so preventing damage to the system.

2.2 Scope and Applicability of the Item

This Item is intended to provide simple approximate methods that may be used to estimate the magnitude of surge pressures following complete power failure to a booster pump or booster pump station. The effect on these surge pressures of a pump bypass, fitted with a one-way valve to allow forward flow only, may also be investigated. A simple idealised model of a typical system with a booster pump station, shown in Sketch 2.1, is considered in this Item. It is assumed that the pipe diameter (and friction factor, where applicable) in both the suction and discharge lengths of pipe (relative to the pump) are constant.

Component losses are assumed to be small compared with the pipe friction and are neglected. The pump is protected by a one-way valve (commonly referred to as a reflux valve or check valve) that prevents reverse flow through the pump by closing completely when the discharge falls to zero. The system may also 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 to the pressure of the suction side following pump trip. This bypass includes a one-way valve that closes completely when the flow through the bypass falls to zero at a later stage in the transient cycle, thus preventing all reverse flow round the pump. The bypass is normally isolated from the flow during normal steady-state flow conditions.

Following bypass operation, some liquid will, in most cases, continue to be drawn through the pump but this is usually small relative to the flow through the bypass and is neglected. The model will allow systems with pressurised tanks to be considered, provided suitable adjustments are made to the tank head values.



Sketch 2.1 Simple discharge line with booster pump

The effect of bypass circuits on the suppression of pressure surges following pump trip is explained in Section 3.1 using a series of sketches illustrating a typical sequence of events. The use of bypass circuits is explained in more detail in Section 3.2. Data are presented in Section 4 that allow estimates to be made of the head and flow conditions at the suction and discharge sides of a booster pump with or without a bypass circuit. The effect of pipe friction on the pressure changes is also considered in that section. A worked example that illustrates the use of these data is given in Section 5. The computer program, ESDUpac A8727, introduced in ESDU 87027 allows the pressure changes to be estimated after pump trip without the simplifying assumptions required here for a graphical presentation.

3. SURGE PRESSURES FOLLOWING PUMP TRIP

3.1 Sequence of Events

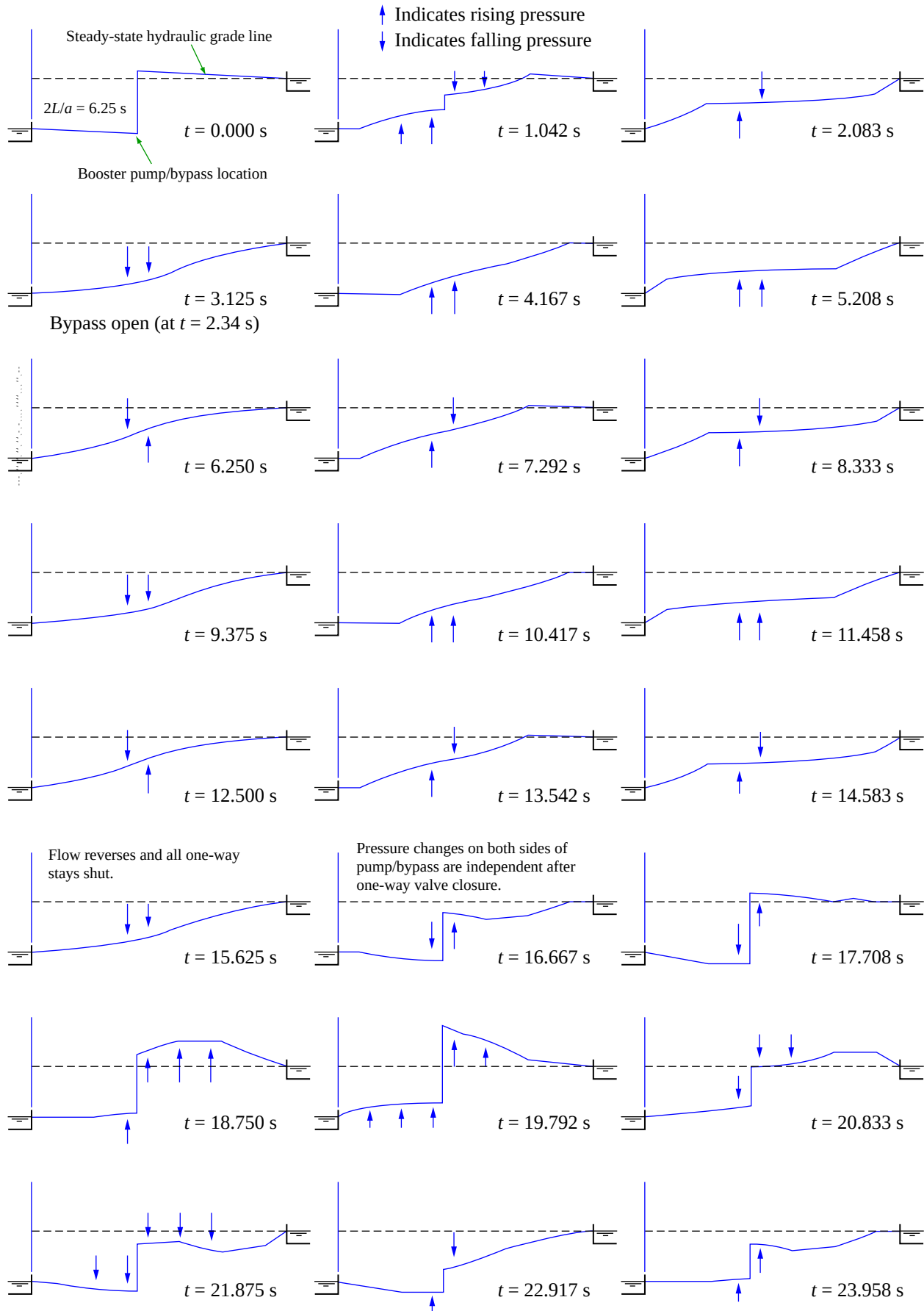
Following power failure to a pump pumping liquid along a pipeline, the discharge from the pump falls and the pressure difference across the pump decreases as the pressure on the discharge side drops and the pressure on the suction side rises. On the discharge side, expansion waves propagate downstream from the pump outlet, and are reflected at a later time at the downstream end of the line as compression waves. Conversely, compression waves propagate from the pump inlet and travel upstream until they are reflected as expansion waves from the upstream end of the line. The magnitude of the pressure waves is dependent primarily upon the time taken for the pump to run-down and the time taken for the pressure waves to propagate from the pump to the upstream and downstream ends of the line. For the discharge and suction lengths of pipe, these periods are equal to $2L_D/a$ and $2L_S/a$, respectively, where L_D and L_S are the discharge and suction pipe lengths respectively, and a is the speed of propagation of the pressure waves (see ESDU 83046).

As the pressure difference across the pump decreases with time, one of two situations may occur. Firstly, the discharge from the pump may fall to zero whilst the pressure on the discharge side still exceeds the pressure on the suction side. In this case, which is common in high-head systems with relatively long pump run-down times for instance, the one-way valve closes to prevent reverse flow through the pump. Secondly, the pressure difference across the pump may fall to zero whilst forward flow through the pump continues. In this case, which is common in relatively low head systems with low inertia pumps, the forward flow due to the inertia of the liquid results in the liquid doing work on the pump with a head loss in the direction of the flow. This condition is known as 'windmilling', and may be prevented by fitting a pump bypass that opens to allow the full flow of liquid to bypass the pump and hence maintain the pressure. This helps prevent a further drop in pressure on the discharge side and a further rise in pressure on the suction side. The use

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of pump bypass circuits may also reduce the magnitude of the resulting pressure rise and pressure drop at the discharge and suction sides respectively. It may be noted that, for pumps located at the upstream end of the line, the pressure at the suction side of the pump is assumed constant, and the pressure changes at the discharge side will in general be significantly larger. The typical sequence of events of Sketch 3.1 illustrates the conditions following power failure to a pump protected by a one-way bypass circuit, located at an interior point along a pipeline (*i.e.* remote from the pipe end). The system represented in this sketch is also considered in the graphical example of Section 5 and the computed example of Section 5.5. The instantaneous head and flow values at each of the time steps shown on Sketch 3.1 are given in the printed computer output of Section 5.6, and may be used for comparison purposes.

In general terms, the maximum and minimum head conditions that occur at a location in a pipeline as a consequence of pump trip must be within upper and lower design limits specified for that location in the pipeline. Usually, this requirement means that the maximum pressure head should not exceed the maximum design pressure head, and the minimum pressure head should not fall to vapour head or gas release head. The maximum design head is usually specified to prevent damage to the pipe or fittings, although head values may be allowed to exceed this value for the occasional surge occurrence. As surge effects are generally greater for higher flow rates, it is recommended that the maximum design flow is used in a surge analysis.



Sketch 3.1 Sequence of events following booster pump trip

3.2 Use of Pump Bypass in Surge Suppression

In addition to the use of pump bypass circuits to protect a system following the total loss of pumping power, forward-flow bypass circuits may be used to maintain flow round a pump in systems with more than one pump in series following the tripping of only one of the pumps.

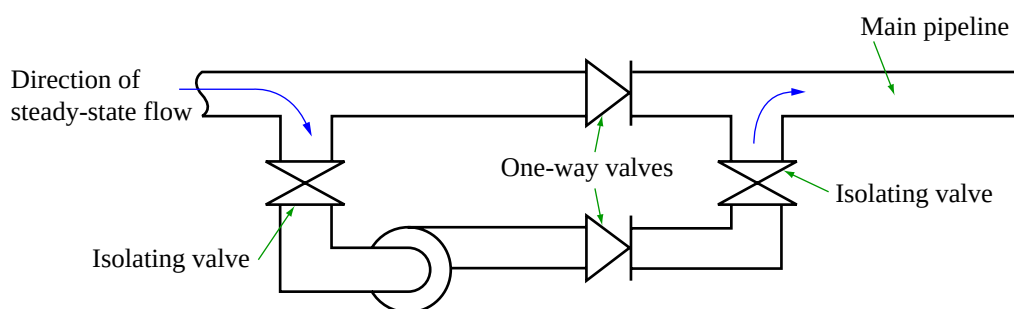
It may be necessary in some cases to consider head losses in bypass circuits, as the assumption that the bypass opens at the instant the pressure on the pump discharge side falls to the pressure on the pump suction side is no longer true for bypass circuits with high flow resistance. In such a case, a head loss in the direction of the flow will occur (proportional to the square of the flow velocity) and, in the initial stages at least, a significant proportion of the flow may continue through the pump. The computer program introduced in ESDU 87027 allows this condition to be considered in a surge analysis.

Although the assumption that one-way valves shut immediately and stay shut at the instant of flow reversal is not true in practice, the effects of one-way valve closure on the pressure surges may be negligible provided that the valves are selected correctly to suit the system. In general, one-way valves must be capable of responding rapidly to changes in flow conditions. This requires the following conditions to be met:

- (i) all moving parts must be of low inertia, and
- (ii) the distance of travel of the moving parts from steady-state to closure must be as small as possible.

In some cases, spring assistance may be required to achieve rapid closure. Some data concerning the choice of one-way valves are given in Reference 9.

As an alternative to the typical bypass circuit configuration shown in Sketch 2.1, it may in some systems be desirable to locate the pump in the 'bypass' circuit as shown in Sketch 3.2. This configuration reduces the flow resistance through the bypass, and also allows easier access to the pump for maintenance purposes.



Sketch 3.2 Alternative pump/bypass arrangement

4. ESTIMATION OF HEAD AND FLOW PARAMETERS FOLLOWING PUMP TRIP

Data are presented in Section 4.1 that allow estimates to be made of the maximum head changes at the discharge and suction sides of a pump located at some point along a line such as that shown in Sketch 4.1. The pump has a one-way valve on the discharge side that closes immediately and stays shut when the flow reverses, and data are provided in Section 4.2 that allow the lapsed time between the instant of pump trip and flow reversal to be estimated. The data in Sections 4.1 and 4.2 are applicable to pumps with or without a forward flow bypass fitted. For a pump that is not protected by a bypass, the pressure on the discharge side may fall below the pressure on the suction side under certain conditions, and forward flow may persist through the "windmilling" pump due to the momentum of the liquid. If the minimum or maximum head values resulting from this condition exceed the design limits, a pump bypass may reduce the head changes to within acceptable values. However, it must be noted that under some conditions, the use of a bypass may increase the head change values. For the case where a bypass is fitted, the maximum flow rate through the bypass may be estimated using data presented in Section 4.3, and the lapsed time between the instant of pump trip and the opening of the bypass may be estimated using data presented in Section 4.4. The time of bypass opening is the same as the time at which the discharge side pressure falls below suction side pressure for a pump with no bypass. For all these data, it is assumed that the line is not otherwise protected against surge effects (*i.e.* there are no air vessels, flow control valves, active air inlet/outlet valves, *etc.*), and no vapour cavities are formed or gas is released from the liquid.

It is assumed that the steady-state flow velocity and the speed of propagation of pressure waves are the same in both the suction and discharge lengths of pipe. This restriction implies that the pipe diameter is constant.

For the calculation of the data given in the graphical presentation, the following assumptions are made.

- (i) All losses (pipe friction losses, component losses, *etc.*) are neglected,
- (ii) reverse flow through the pump is prevented by the instantaneous closure of a one-way valve (situated at the pump outlet) when the pump discharge falls to zero, and
- (iii) the pressure at all points in the line remains above vapour pressure or gas release pressure.

The following assumptions are also made for the case where a bypass is used to protect the line.

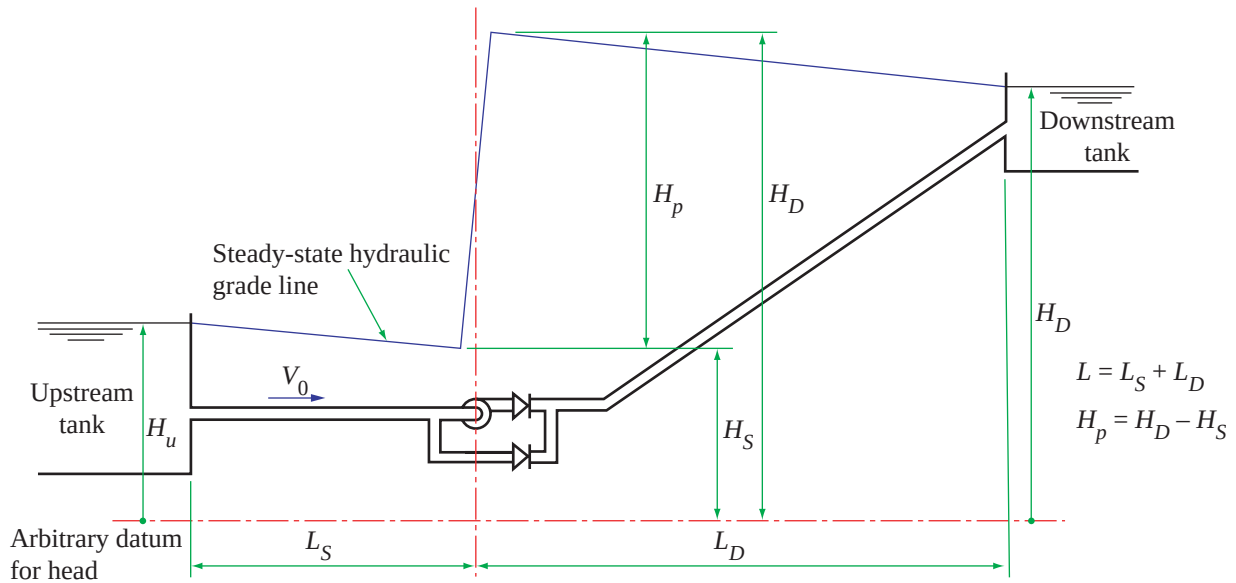
- (i) The bypass opens to allow the full flow of liquid to bypass the pump completely at the instant the pump head falls to zero,
- (ii) reverse flow through the bypass is prevented by the instantaneous closure of a one-way valve in the bypass circuit when the discharge through the bypass falls to zero, and
- (iii) the friction head loss across the fully-open bypass is small compared to the pipe friction losses and is also neglected.

The data are given assuming only one pump, or one set of identical parallel pumps, are located in the line. The effect of pipe wall friction on the head changes is discussed in Section 4.3.

The model of the system for which data are presented is shown in Sketch 4.1. The steady-state head/flow conditions are described by non-dimensional initial head/flow conditions parameter α , given as

$$\alpha = \frac{gH_p}{aV_0}, \quad (4.1)$$

where g is the acceleration due to gravity ($g = 9.81 \text{ m/s}^2$ or 32.2 ft/s^2), V_0 is the maximum design value of the steady-state flow velocity and H_p is the corresponding steady-state static head difference across the pump (which is equal to the difference in the head values in the downstream and upstream tanks, $H_d - H_u$, plus the head loss due to friction, H_f). The effect of friction on the estimation of the head and flow parameters is discussed in Section 4.5.



Sketch 4.1 Simple booster pump/bypass model

The speed of propagation of the pressure waves (or wave speed) may be estimated using the data presented in ESDU 83046. For water contained in circular pipes that are unrestrained against longitudinal pipe expansion and contraction (usually requiring closely-spaced expansion joints), wave speed data are reproduced in 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 (10 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 7.1. The wave speed data are given assuming that no free gas is present in the water. If a small quantity of free gas is present in the liquid, the wave speed will be reduced, sometimes drastically. Similarly, the presence of solid matter in the liquid will alter the wave speed. Further guidance on the estimation of the wave speed in liquid/gas or liquid/solid mixtures is given in ESDU 83046 (Reference 10). Data are given in ESDU 83046 for the estimation of the wave speed in liquids contained in circular or non-circular, lined or unlined, buried or unburied pipes.

The pump run-down characteristics are expressed by the parameter τ , given by

$$\tau = 2 \left(\frac{2\pi}{60} N_R \right)^2 \frac{I \eta_R}{\rho g H_R Q_R} \times \frac{a}{2L}, \quad (4.2a)$$

where I is the rotational inertia of the rotating components of the pumps (including gearbox, motor and fittings and the liquid contained in the pump impeller), ρ is the density of the liquid, L is the pipe length and N_R , η_R , H_R and Q_R are the rated (or reference) values of the pump rotational speed (in rev/min), efficiency, head and flow, respectively. The rated efficiency, η_R , is in the range 0 to 1. The pipe length, L , is taken as the distance along the axis of the pipeline (including pipe bends, etc.) from the upstream tank outlet to the inlet of the large tank or reservoir at the downstream end of the line (noting that, unlike standard pressure drop calculations, equivalent lengths of fittings are not added to the straight pipe length). Alternatively, this parameter may be expressed as

$$\tau = 2 \left(\frac{2\pi}{60} N_R \right)^2 \frac{I}{P_R} \times \frac{a}{2L}, \quad (4.2b)$$

where P_R is the pump shaft input power (using the unit of watts in the SI system) at the rated conditions. The inertia term, I , is usually available from the pump manufacturer. However, an estimate of the inertia term may be made from data presented in Figure 2. 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. Figure 2 shows a mean line and a band containing the measured data. Mean value lines calculated from the data presented in Derivation 5 and given in Derivation 4 show very good agreement, using British and US pumps, respectively.

4.1 Estimation of Maximum and Minimum Heads

Data are presented in Figures 3a to 3i that enable estimates to be made of the head changes, expressed as a proportion of the maximum theoretical head change (which is known as the “Joukowsky head” and is given by aV_0/g) at the discharge and suction sides of a pump with and without a bypass. These head change data allow the minimum head at the discharge side and the maximum head at the suction side to be estimated over a range of values of α and τ . These minimum and maximum head values are given by the following expressions:

$$H_D^{min} = H_D - \Delta H_D \quad (4.3a)$$

$$\text{and} \quad H_S^{max} = H_S + \Delta H_S, \quad (4.3b)$$

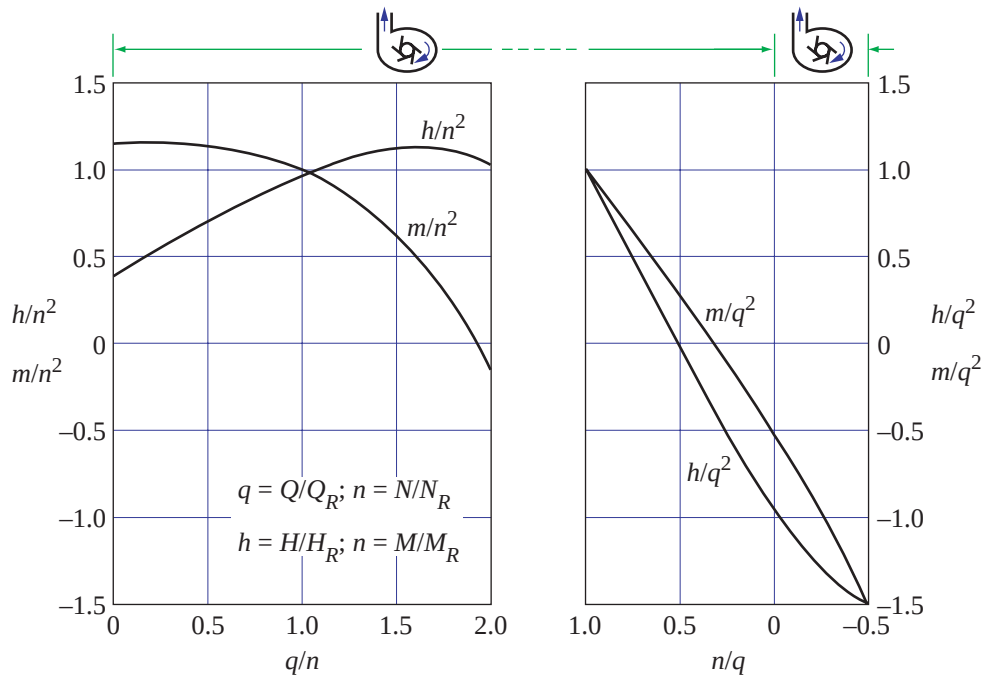
where H_D^{min} and H_S^{max} are the minimum and maximum head values due to surge effects at the discharge and suction sides of the tripped pump, respectively, H_D and H_S are the steady-state head values at the discharge and suction side of the pump, respectively, and the corresponding head change values, ΔH_D and ΔH_S , are estimated using the graphical presentation discussed in Sections 4.1.1 and 4.1.2, respectively. The use of these data is illustrated in the worked example of Section 5.

The maximum head at the discharge side and the minimum head at the suction side may not be determined accurately from a graphical presentation, except for the case where the pump is at the upstream end of the line (i.e. the suction line length is short and may be neglected, see Section 4.1.1). However, the discharge side head rise is less than the discharge side head drop and, as τ increases, the difference in the two values decreases. Similarly, the suction side head drop is less than the suction side head rise. Therefore, conservative estimates of the discharge side maximum head and suction side minimum head may be obtained from the expressions

$$H_D^{max} = H_D + \Delta H_D, \quad (4.3c)$$

$$\text{and} \quad H_S^{min} = H_S - \Delta H_S. \quad (4.3d)$$

The head change data were calculated using the pump characteristic curves given in Derivations 1 and 4, and shown in Sketch 4.2. These are typical centrifugal pump type characteristics, and allow the pumping head, flow rate, torque and rotational speed to be expressed non-dimensionally for a range of pumps with similar operating characteristics. Although the data are applicable to one particular type of pump, the data may be used for other pumps with similar characteristic curves. Other typical pump characteristic data are given in Derivation 4.



Sketch 4.2 Typical centrifugal pump characteristic curves

4.1.1 Estimation of ΔH_D

The value of the discharge side head change, ΔH_D , is dependent upon the non-dimensional parameters α and τ (see Equations (4.1), (4.2a) and (4.2b)), and also upon the location of the pump in the line. For the estimation of ΔH_D , the pump position may be accounted for by the non-dimensional pump position parameter, x , given by $x = L_S / L$. Hence, the minimum and maximum head values at the discharge side of the pump are given by Equations (4.3a) and (4.3c) respectively, where ΔH_D may be estimated from the equation

$$\Delta H_D = \delta_x \times \frac{a V_0}{g}. \quad (4.4)$$

In this equation, δ_x is a non-dimensional head change parameter, and may be estimated for a range of values of α and τ from Figures 3a to 3h. The data in each of these figures are presented for a given pump position, defined by the pump position parameter x . The appropriate figures to use in the calculation of ΔH_D may be determined from the table below.

Figure No.	x	α	τ
3a	0	0.05, 0.1, 0.2, 0.3, 0.4, 0.5, 0.7, 1, 2	In the range 0.1 to 100
3b	0.2		
3c	0.4		
3d	0.45		
3e	0.5		
3f	0.7		
3g	0.9		
3h	0.95		

For values of x intermediate to those used on the figures, the discharge side non-dimensional head change parameter, δ_x , may be estimated by graphical interpolation. It is recommended that values of δ_x derived from at least three of the presented figures, for the appropriate values of α and τ , be used for interpolation purposes.

Note that, for the case where $x = 0$ (i.e. the pump is located at the upstream end of the line) the appropriate value of δ_x may be taken directly from Figure 3a. It may be seen from Figures 3a to 3i that regions exist where the pump head falls below zero and, if a bypass is present, the bypass opens to prevent a further head drop. Within this region, the value of δ_x due to bypass action is indicated by the dotted lines. The full lines indicate the appropriate value of δ_x for systems in which forward flow through the pump is allowed for by a head drop in the direction of the flow. These data allow the effect on the head changes of a bypass to be determined. The minimum and maximum heads on the discharge side of the pump may now be determined from Equations (4.3a) and (4.3c) respectively. Note that, for $x = 1$, the value of δ_x , and hence the value of ΔH_D , is equal to zero.

For the case where the pump is located at the upstream end of the line ($x = 0$), the appropriate value of ΔH_D , for the calculation of the maximum head at the discharge side using Equation (4.3c), may be estimated from Equation (4.4) with a value of δ_x taken directly from Figure 3i. These data are given for the same range of values of the parameters α and τ as Figures 3a to 3h.

4.1.2 Estimation of ΔH_S

The value of the suction side head change, ΔH_S , may be estimated using the same procedure as the estimation of ΔH_D , detailed in Section 4.1.1. However, for this estimation of δ_x , the appropriate value of x is given by $x = 1 - L_S / L$. The value of ΔH_S is given by the equation

$$\Delta H_S = \delta_x \times \frac{aV_0}{g}, \quad (4.5)$$

where δ_x is estimated for the appropriate value of α , τ and x . The maximum and minimum head values on the suction side of the pump may now be calculated from Equations (4.3b) and (4.3d) respectively. Note that for the case where $x = 1$, δ_x and hence ΔH_S are equal to zero.

4.2 Estimation of Lapsed Time Between Pump Trip and One-way Valve Closure Time, t_{rev}

Following pump trip, the flow velocity in the line decreases until the flow eventually reverses. The time after the initiation of pump trip that the flow reverses (which is the time at which the discharge from the pump falls to zero or the flow through the bypass reverses) and hence all one-way valves close, may be calculated using Figures 4a to 4c for a range of values of α , τ and x (where $x = L_S / L$). This reversal time, t_{rev} , is expressed as the non-dimensional flow reversal time $t_{rev} / (2L/a)$, where $2L/a$ is the pipeline period. The appropriate figures to use in the estimation of t_{rev} may be determined from the table below.

Figure No.	x	α	τ
4a	0 and 1	0.05, 0.1, 0.2	In the range 0.1 to 100
4b	0.25 and 0.75	0.3, 0.4, 0.5, 0.7	
4c	0.5	1, 2	

For values of x intermediate to those used on the figures, the value of t_{rev} may be estimated by graphical interpolation between the values of α and τ on all three of the figures presented for individual values of x . For values of τ below approximately 1, it may be necessary to take four or more points for individual values of x on each figure in order to estimate t_{rev} accurately. The effect of a bypass in the line on t_{rev} may be investigated using these data.

4.3 Estimation of Maximum Flow Rate Through Pump Bypass, Q_b

For the case where a bypass is used to protect the line, it is assumed that the bypass opens immediately when the pump head falls to zero and all flow is then taken through the bypass. The maximum flow rate, which is a proportion of the steady-state flow rate, Q_0 , occurs at the instant the bypass opens, and is given by the equation

$$Q_b = \phi_b Q_0 \left(1 - \frac{\alpha}{2}\right). \quad (4.6)$$

In this equation, ϕ_b is a non-dimensional bypass flow rate parameter, and may be estimated for a range of values of α and τ from Figures 5a to 5g. The data in each of these figures are presented for a given value of α , and for a range of values of τ and x (where $x = L_S/L$). The appropriate figures to use in the estimation of ϕ_b may be determined from the table below.

Figure No.	α	τ	x
5a	0.05	In range between 0.1 and 100	0 to 0.5 (1 to 0.5)
5b	0.1		
5c	0.2		
5d	0.3		
5e	0.4		
5f	0.5		
5g	0.7		

For values of α intermediate to those used on the figures, the non-dimensional bypass flow rate parameter, ϕ_b , may be estimated by graphical interpolation. It is recommended that values of ϕ_b derived from at least three of the presented figures, for the appropriate values of α and τ , be used for interpolation purposes. Note that the curves on the figures are symmetrical about $x = 0.5$ (so that values at $x = 0.7$ equal those at $x = 0.3$, for example).

4.4 Estimation of Lapsed Time Between Pump Trip and Bypass Opening, t_b

The lapsed time between the initiation of pump trip and the pump head falling to zero (which is the time at which the bypass opens) may be estimated using the data presented in Figures 6a to 6h for a range of values of α , τ and x (where $x = L_S/L$). This lapsed time, t_b , is expressed by the non-dimensional parameter $t_b/(2L/a)$, where $2L/a$ is the pipeline period. The procedure for estimating t_b is the same as that for estimating ϕ_b , as detailed in Section 4.3. Note that the curves on the figures are symmetrical about $x = 0.5$ (so that values at $x = 0.8$ equal those at $x = 0.2$, for example).

4.5 Effect of Friction on Head Change and Flow Rate Values

It is not possible to predict accurately the effect of friction on the head changes on the discharge and suction sides of a tripped pump. However, for low values of τ , the head change values may be significantly increased for systems where the head loss due to friction, H_f , is a significant proportion of the total pumping head, H_p . In general, friction effects lead to an increase in the lapsed time between pump trip and flow reversal, t_{rev} .

For systems where the head loss due to friction, H_f , is greater than (say) 20 per cent of the total pumping head, H_p , it is recommended that the effects of friction on the head changes are investigated using a suitable computer program.