Water Hammer

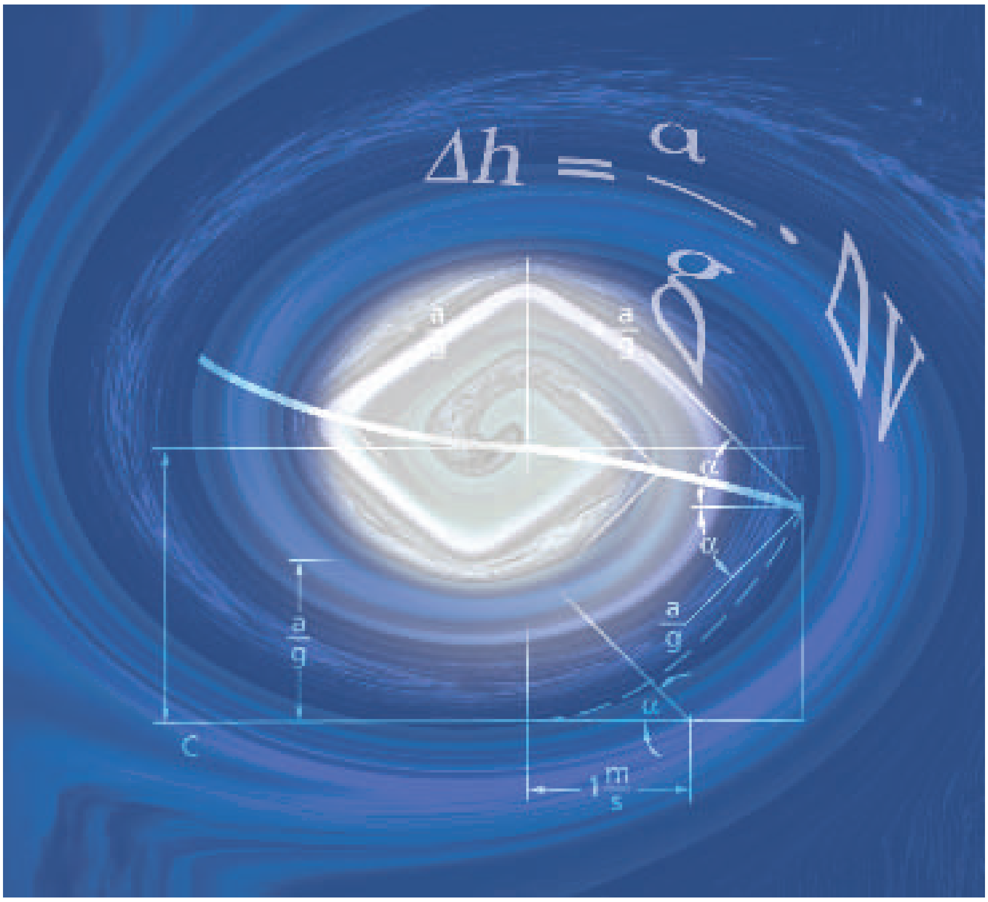

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Table of Contents

1 Introduction

Most engineers involved in the planning of pumping systems are familiar with the terms "hydraulic transient", "surge pressure" or, in water applications, "water hammer". The question as to whether a transient flow or surge analysis is necessary during the planning phase or not is less readily answered. Under unfavourable circumstances, damage due to water hammer may occur in pipelines measuring more than one hundred metres and conveying only several tenths of a litre per second. But even very short, unsupported pipelines in pumping stations can be damaged by resonant vibrations if they are not properly anchored. By contrast, the phenomenon is not very common in building services systems, e.g. in heating and drinking water supply pipelines, which typically are short in length and have a small cross-section.

The owners or operators of systems affected by water hammer are usually reluctant to pass on information about any surge damage suffered. But studying the photos taken of some "accidents" (**Figs. 1-a, 1-b, 1-c**) one thing is clear: the damage caused by water hammer by far exceeds the cost of preventive analysis and surge control measures.

The ability to provide reliably designed surge control equipment, such as an air vessel or accumulator[1], flywheel and air valve, has long been state of the art. The technical instruction leaflet W 303 "Dynamic Pressure Changes in Water Supply Systems" published by the German Association of the Gas and Water Sector clearly states that pressure transients have to be considered when designing and operating water supply systems, because they can cause extensive damage. This means that a surge analysis to industry standards has to be performed for every hydraulic piping system at risk from water hammer. Dedicated software is available for this purpose – an important tool for the specialist surge analyst to use. Consultants and system designers are faced with the following questions, which we hope to answer in this brochure:

- How can we know whether there is a risk of water hammer or not?
- How significant are approximation formulas for calculating water hammer?
- Can the surge analysis of one piping system be used as a basis for drawing conclusions for similar systems?
- Which parameters are required for a surge analysis?
- What does a surge analysis cost?
- How reliable is the surge control equipment available and how much does it cost to operate it?
- How reliable is a computerised analysis?

System designer and surge analyst have to work together closely to save time and money. Water hammer is a complex phenomenon; the purpose of this brochure is to impart a basic knowledge of its many aspects without oversimplifying them.

Fig. 1-a: Completely destroyed DN 600 discharge pipe (wall thickness 12 mm)

Fig. 1-b: Destroyed support (double T profile 200 mm, permanently deformed)

Fig. 1-c: DN 800 check valve following a pressure surge in the discharge pipe

[1] *Air vessels, sometimes also called "accumulators", store potential energy by accumulating a quantity of pressurised hydraulic fluid in a suitable enclosed vessel.*

2 General – The problem of water hammer

2.1 Steady and unsteady flow in a pipeline

When discussing the pressure of a fluid, a distinction has to be made between pressure above atmospheric [p bar], absolute pressure [p bar(a)] and pressure head h [m]. Pressure head h denotes the height of a homogeneous liquid column which generates a certain pressure p. Values for "h" are always referred to a datum, (e.g. mean sea level, axial centreline of pipe and pipe crown etc.).

As a rule, system designers start by determining the steady-state operating pressures and volume rates of flow. In this context, the term **steady**[2] means that volume rates of flow, pressures and pump speeds do not change with time. **Fig. 2.1-a** shows a typical steady flow profile:

With a constant pipe diameter and a constant surface roughness of the pipe's inner walls, the pressure head curve will be a straight line. In simple cases, a pump's steady-state operating point can be determined graphically. This is done by determining the point where the pump curve intersects the piping characteristic.

A pumping system can never be operated in steady-state condition all the time, since starting up and stopping the pump alone will change the duty conditions. Generally speaking, every change in operating conditions and every disturbance cause pressure and flow variations or, put differently, cause the flow conditions to change with time. Flow conditions of this kind are commonly referred to as unsteady or transient. Referring specifically to pressures, they are sometimes called dynamic pressure changes or pressure transients. The main causes of transient flow conditions are:

- Pump trip as a result of switching off the power supply or a power failure.
- Starting or stopping up one or more pumps whilst other pumps are in operation.
- Closing or opening of shut-off valves in the piping system.
- Excitation of resonant vibrations by pumps with an unstable H/Q curve.
- Variations of the inlet water level.

Fig. 2.1-b may serve as a representative example showing the pressure envelope[3] with and without an air vessel following pump trip.

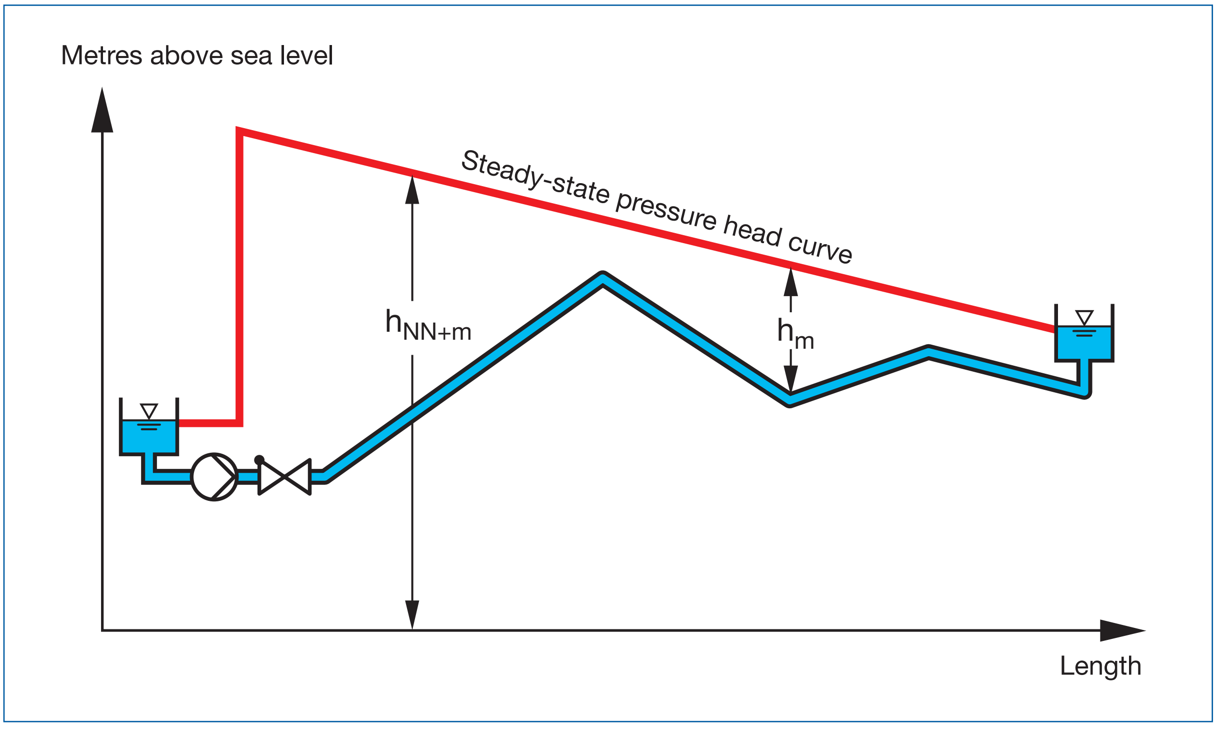


Fig. 2.1-a: Steady-state pressure head curve of a pumping system

[2] *Not to be confused with the term "static".*

[3] *The term "pressure envelope" refers to the area defined by the minimum and maximum head curves along the fixed datum line resulting from all dynamic pressures occurring within the time period under review.*

h_{steady} in **Fig. 2.1-b** is the steady-state pressure head curve. Pressure head envelopes h_{minWK} and h_{maxWK} were obtained from an installation with, h_{min} and h_{max} from an installation without air vessel. Whereas h_{minWK} and h_{maxWK} are within the permissible pressure range, h_{min} gives evidence of vapour pressure (macro-cavitation) over a pipe distance from 0 m to approximately 800 m. Almost across the entire length of the pipe, the value of h_{max} exceeds the maximum permissible nominal pressure of the pipe PN 16 (curve marked “PN pipe“) and is, therefore, inadmissibly high. As a rule, vapour pressure is a most undesirable phenomenon. It can have the following harmful effects:

- Dents in or buckling of thin-walled steel pipes and plastic tubes.
- Disintegration of the pipe’s cement lining.
- Dirty water being drawn into drinking water pipelines through leaking connecting sockets.

We will come back to the subject of macro-cavitation, i.e. liquid column separation, in section 3.1.

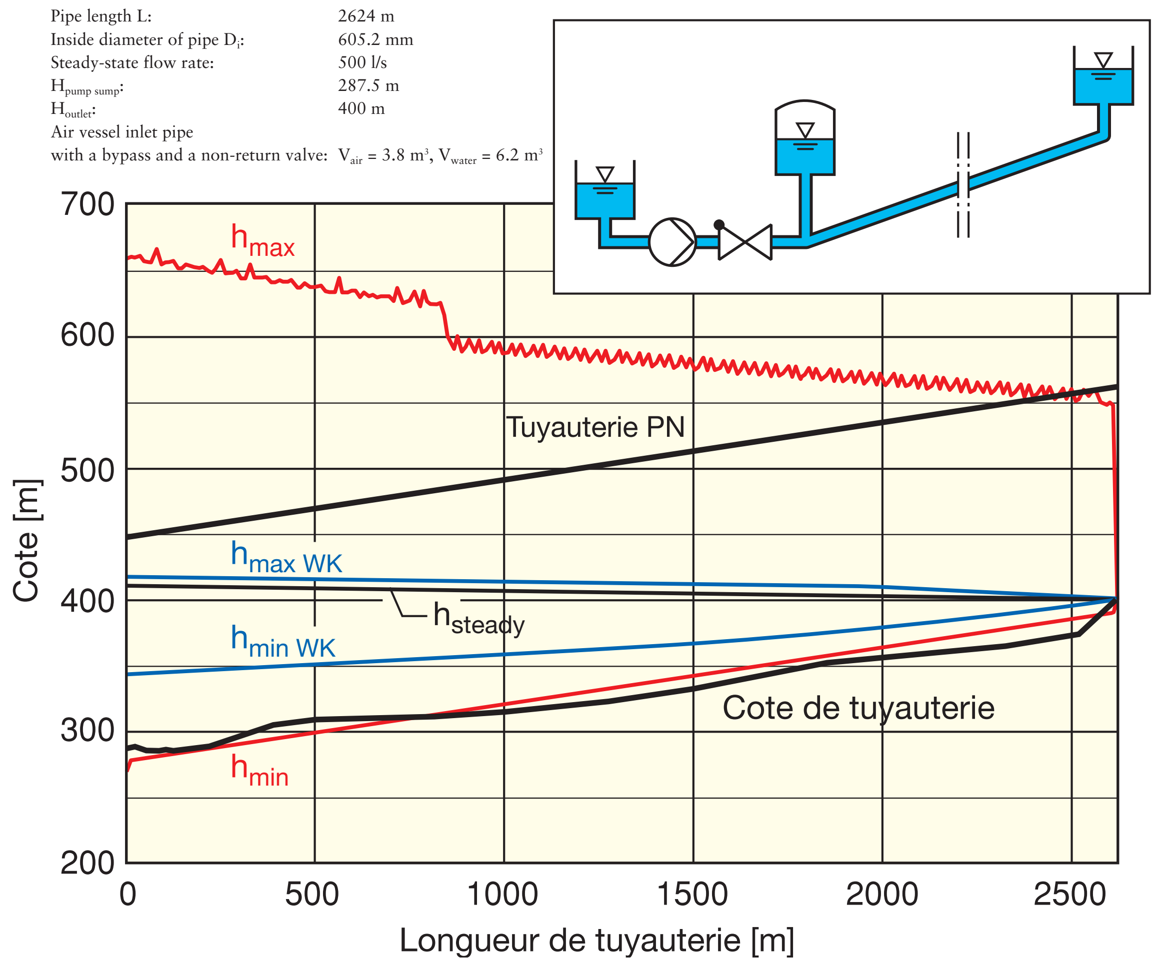


Fig. 2.1-b: Pressure head envelope of pressure transients following pump trip

3 Water hammer

Pressure transients are also referred to as surge pressure or, if - referring to water systems, water hammer. The latter term suitably reflects the harmful effects that the hammer-like blows accompanying the pressure surges can have on pipes and system components. Water hammer causes piping, valves, pipe fixtures, supports, system components, etc. to suffer the added strain of dynamic loads. The term "water hammer" is used to describe the phenomenon occurring in a closed conduit when there is either an acceleration or retardation of the flow. In contrast to a force, pressure is non-directional; i.e. it does not have a vector. Not until a hydrostatic pressure starts acting on a limiting area, is a force exerted in the direction of the area normal.

As it is not possible to altogether avoid pressure transients when operating a piping system, the art lies in keeping the pressure transients within controllable limits. What makes matters even more complex, is the fact that the damage caused by impermissibly high surge pressures is not always visible. Often the consequences do not become apparent until long after the event, for example a pipe rupture, loose or disconnected flanges. The root cause of damage then tends to remain in the dark. Some representative incidents caused by water hammer are listed below:

Pressure rise:

- Pipe rupture
- Damaged pipe fixtures
- Damage to pumps, foundations, pipe internals and valves

Pressure fall:

- Buckling of plastic and thin-walled steel pipes
- Disintegration of the cement lining of pipes
- Dirty water or air being drawn into pipelines through flanged or socket connections, gland packing or leaks
- Water column separation followed by high increases in pressure when the separate liquid columns recombine (macro-cavitation)

3.1 Inertia

The sudden closure of a valve in a pipeline causes the mass inertia of the liquid column to exert a force on the valve's shut-off element. This causes the pressure on the upstream side of the valve to increase; on the downstream side of the valve the pressure decreases. Let us consider an example: for a DN 200 pipe, L = 900 m, v = 3 m/s, the volume of water in the pipeline is calculated by

$$m_{water} = \frac{0.2^2 \pi}{4} \cdot 900 \cdot 1000 = 28274 \text{ kg}$$

This is more or less the same as the weight of a truck; v = 3 m/s corresponds to 11 km/h. In other words, if the flow is suddenly stopped, our truck – to put it in less abstract terms – runs into a wall (closed valve) at 11 km/h (water mass inside the pipe). In terms of our pipeline, this means that the sequence of events taking place inside the pipe will result in high pressures and in high forces acting on the shut-off valve.

As a further example of inertia, **Fig. 3.1-a** shows a pump discharge pipe. At a very small moment of inertia of pump and motor, the failing pump comes to a sudden standstill, which has the same effect as a suddenly closing gate valve, only this time on the downstream side of the gate valve. If mass inertia causes the fluid flow on the downstream side of the pump to collapse into separate columns, a cavity containing a mixture of water vapour and air coming out of solution will be formed. As the separate liquid columns subsequently move backward and recombine with a hammer-like impact, high pressures develop. The phenomenon is referred to as liquid column separation or *macro-cavitation*[4].

[4] *Macro-cavitation in pipelines is not to be confused with microscopic cavitation causing pitting corrosion on pump and turbine blades. The latter always strikes in the same place and is characterised by local high pressures of up to 1000 bar or more that develop when the microscopically small vapour bubbles collapse. With macro-cavitation, repetitive strain of this kind, or the bombarding of a sharply contoured area of the material surface, does not occur since the pressure rises are considerably lower.*

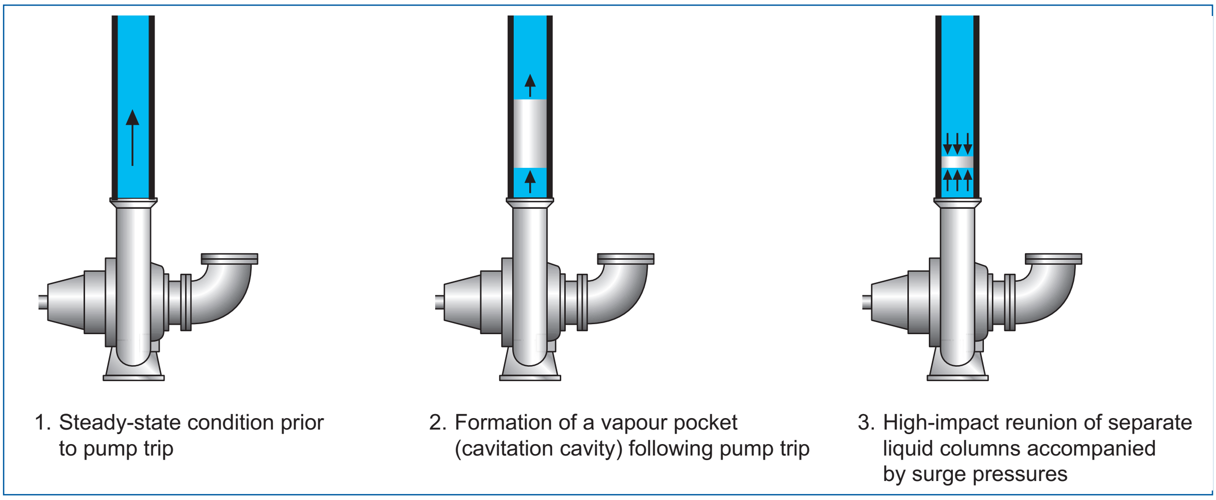


Fig. 3.1-a: Macro-cavitation following pump trip

3.2 Elasticity of fluid and pipe wall

The attempt at visualising water hammer resulting from the inertia of a body of water made in section 3.1 is only partly correct, because no allowance was made for the elasticity of fluid and pipe wall. As long as safety belts are worn and the barrier impact speed is not too high, even a head-on collision will not put drivers in too much danger today, because the vehicle's momentum is converted to harmless deformation heat[5]. Contrary to the body of a car, however, water and pipe walls are elastic, even though they are so hard that this property is not noticeable in every day use.

What actually goes on inside the pipe will, therefore, be described using the following example of a heavy steel spring sliding through a pipe. This spring suffers elastic deformation when it is suddenly stopped (**Fig. 3.2-a**):

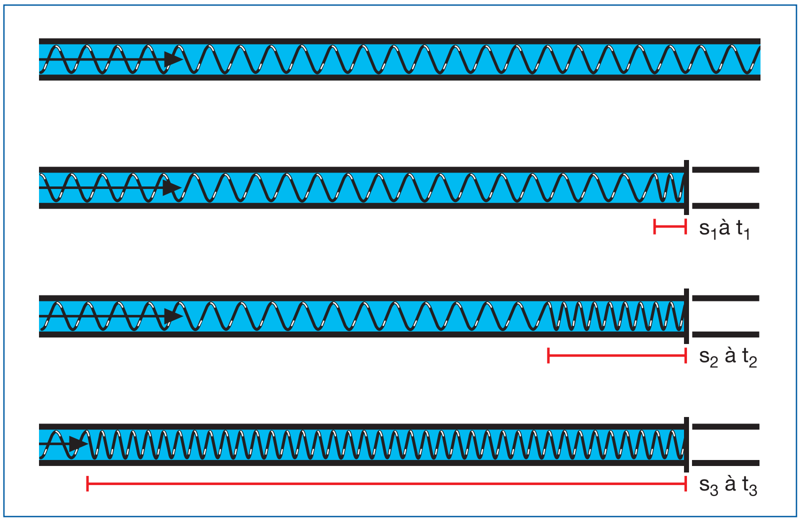
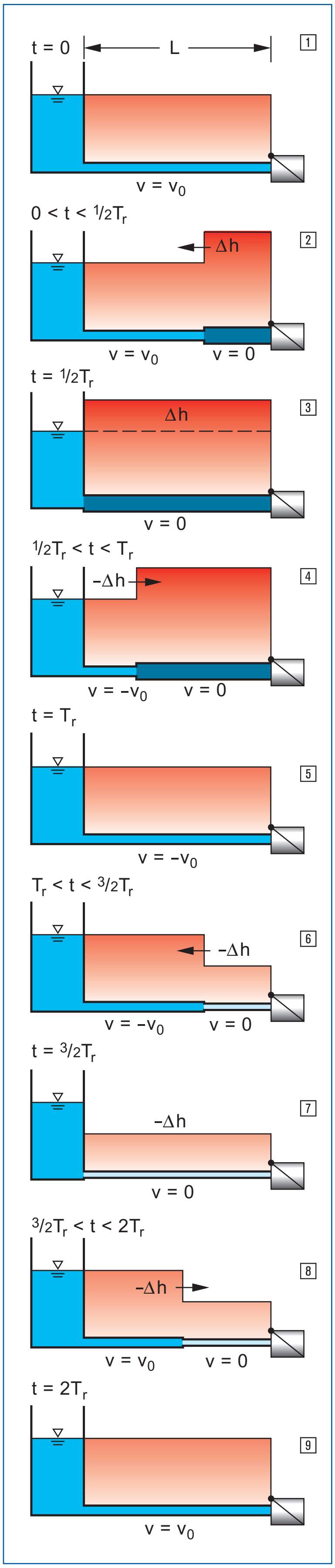


Fig. 3.2-a: Sudden closure of gate valve, visualised by a heavy steel spring

The front end deformation travels in the opposite direction to the original direction of movement at the speed typical for the steel spring, i.e. wave propagation velocity a in m/s. In the compression zone, the velocity of the steel spring is v = 0 everywhere.

Following these, admittedly poor but hopefully helpful, examples chosen to illustrate the subject, we will now go back to the real situation inside the pipe, which is shown in **Fig. 3.2-b**, with friction being neglected. The shut-off valve installed at the downstream end of a horizontal pipeline with a constant inside diameter, which is fed from a reservoir at constant pressure, is suddenly closed:

[5] *To withstand the regular pushing and shoving over rare parking spaces, cars have to be elastic. To minimise the damage of a collision at high speed, however, carmakers spend vast amounts of time and money to make their products as inelastic as possible!*



1 For t = 0, the pressure profile is steady, which is shown by the pressure head curve running horizontally because of the assumed lack of friction. Under steady-state conditions, the flow velocity is v_0.

2 The sudden closure of the gate valve at the downstream end of the pipeline causes a pulse of high pressure Δh; and the pipe wall is stretched. The pressure wave generated runs in the opposite direction to the steady-state direction of the flow at the speed of sound and is accompanied by a reduction of the flow velocity to v = 0 in the high pressure zone. The process takes place in a period of time $0 < t < \frac{1}{2} T_r$, where T_r is the amount of time needed by the pressure wave to travel up and down the entire length of the pipeline. The important parameter T_r is the reflection time of the pipe. It has a value of 2L/a.

Fig. 3.2-b: Pressure and velocity waves in a single-conduit, frictionless pipeline following its sudden closure. The areas of steady-state pressure head are shaded medium dark, those of increased pressure dark, those of reduced pressure light. The expansion and contraction of the pipeline as a result of rising and falling pressure levels, respectively, are shown. To give an idea of the relationship involved: With a 100 bar pressure rise, the volume of water will decrease by about 0.5%.

3 At $t = \frac{1}{2}T_r$ the pressure wave has arrived at the reservoir. As the reservoir pressure p = constant, there is an unbalanced condition at this point. With a change of sign, the pressure wave is reflected in the opposite direction. The flow velocity changes sign and is now headed in the direction of the reservoir.

4 A relief wave with a head of $-\Delta h$ travels downstream towards the gate valve and reaches it at a time $t = T_r$. It is accompanied by a change of velocity to the value $-v_0$.

5 Upon arrival at the closed gate valve, the velocity changes from $-v_0$ to v = 0. This causes a sudden negative change in pressure of $-\Delta h$.

6 The low pressure wave $-\Delta h$ travels upstream to the reservoir in a time $T_r < t < \frac{3}{2}T_r$, and at the same time, v adopts the value v = 0.

7 The reservoir is reached in a time $t = \frac{3}{2}T_r$, and the pressure resumes the reservoir's pressure head.

8 In a period of time $\frac{3}{2}T_r < t < 2T_r$, the wave of increased pressure originating from the reservoir runs back to the gate valve and v once again adopts the value v_0.

9 At $t = 2T_r$, conditions are exactly the same as at the instant of closure t = 0, and the whole process starts over again.

So, one might ask, what happened to the original steady-state kinetic energy of the fluid following the sudden closure of the gate valve? A closer look at **Fig. 3.2-b** will reveal the answer. According to the law of the conservation of energy, it cannot simply disappear. First it is converted into elastic energy of the fluid and the pipe wall, then changes into kinetic energy again as a result of reflection, then becomes elastic energy again, and so forth. Let's look at **Fig. 3.2-b** up to the point where $t = \frac{1}{2}T_r$. The conversion into elastic energy takes place within this period of time. Immediately preceding the reflection of the wave at the reservoir, the velocity of the liquid column is $v = 0$ everywhere, and it is totally devoid of kinetic energy. Instead, the kinetic energy has been changed into elastic energy, comparable to the situation of a compressed steel spring. The energy conversion in reverse also becomes apparent from **Fig. 3.2-b** – specifically from the condition prevailing at $t = 2T_r$. If the gate valve were to be suddenly opened at this point, we would have the old steady-state condition at $t = 0$ again without change, and there would be no elastic energy left.

Without friction, the pressure fluctuations would not diminish. In actual fact, no system is ever entirely without friction, but the reduction in pressure fluctuation is relatively small in reality, because the energy conversion into frictional heat as a result of the fluid rubbing against the pipe walls, the inherent fluid friction and, finally, the deformation of pipe walls and fixtures is relatively small.

To show the process in a less abstract manner, **Fig. 3.2-c** provides the results of a computerised simulation of the example given in **Fig. 3.2-b** for a real pipeline with the following parameters:

L = 100 m, DN 100, k = 0.1 mm, h_{inlet} = 200 m, linear throttling of Q = 10 l/s at the outlet of the pipe to Q = 0, starting at t = 0.1 s in a period of time Δt = 0.01 s.

Based on **Fig. 3.2-b**, the reflection of pressure waves at the upstream and downstream ends of the pipeline can be explained in a general manner as follows:

- If a pressure wave Δp reaches the closed end of a pipe, Δp becomes twice the amount with the same sign, i.e. $p = p \pm 2{\cdot}\Delta p$. The velocity at the pipe ends is always $v = 0$.
- At the open end of a pipe with a constant total head (e.g. reservoir with a constant water level), the pressure change always equals zero.
- At valves, throttling sections, pumps and turbines, pressure and velocity are always found on the resistance or machine characteristic curve.

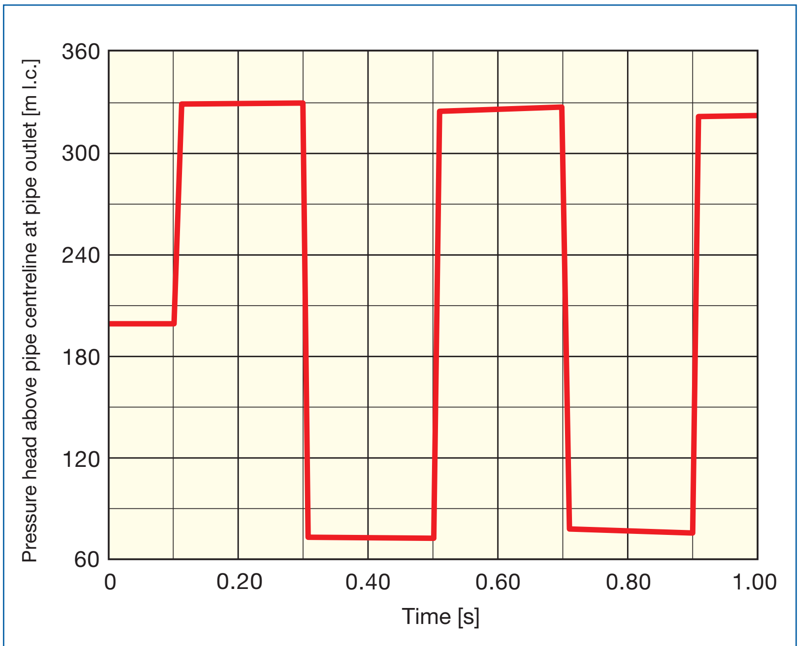


*Fig. 3.2-c: Pressure head upstream of gate valve. Compared with the situation shown in **Fig. 3.2-b**, small differences are apparent. For example, the pressure flanks are not perfectly perpendicular, because of the finite closing time of Δt = 0.01 s. As a result of friction, the pressure planes are not perfectly horizontal – this phenomenon will be discussed in greater detail in section 4.1.*

Water hammer occurs when the kinetic energy of a fluid is converted into elastic energy. But only rapid[6] changes of the flow velocity will produce this effect, for example the sudden closure of a gate valve or the sudden failure or tripping of a pump. Due to the inertia of the fluid, the flow velocity of the liquid column as a whole is no longer capable of adjusting to the new situation. The fluid is deformed, with pressure transients accompanying the deformation process. The reason why surge pressure is so dangerous is that it travels at the almost undiminished speed of sound (roughly 1000 m/s for a large number of pipe materials) and causes destruction in every part of the piping system it reaches.

Surge pressures travel at a very high wave propagation velocity, for example a = 1000 m/s in ductile or steel piping (see 4.1). They dampen out only gradually and, therefore, remain dangerous for a long time. The time needed to subside depends on the length of the pipeline. In an urban water supply installation, they only last several seconds. In long pipelines, it can take a few minutes until a pressure surge has dampened out.

Knowing these facts, the basic working principles of all surge control equipment, such as air vessel or accumulator, flywheel, standpipe and air valves can be deduced. They prevent the dangerous conversion of steady-state kinetic energy into elastic deformation energy. Air vessels are ideally suited to explain the underlying principle. The pressurised air cushion in the air vessel stores potential energy. If there were no air vessel, the dreaded conversion of kinetic energy into elastic deformation energy following a pump trip would take place at the pump outlet, which could cause the liquid column to separate (**Fig. 3.1-a**). However, this does not happen, because the energy stored in the air cushion in the vessel takes over the work of the pump. Immediately following pump trip, the air cushion starts expanding and takes over the pump's job of discharging the water into the pipeline. Provided that the vessel is properly designed, it will prevent rapid changes in the flow velocity in the pipeline. Instead, the water level in the vessel and the undreformed liquid column in the pipeline will continue to rise and fall over a longer period of time. The process is kept in motion by the energy discharged by the air cushion each time fluid flows out of the vessel and by the energy absorbed again by the air cushion on the fluid's return. The energy stored in the air cushion is only gradually dissipated. That is why it takes many minutes for air vessel oscillations to die away, in longer pipelines in particular.

3.3 Resonance

Resonant vibrations are an exception. These occur when exciter frequencies of whatever origin, generated, for example, by the pump drive or by flow separation phenomena in valves and pipe bends, happen to coincide with a natural frequency of the pipeline. Improperly anchored, unsupported pipeline sections in pumping installations are particularly prone to resonant vibrations transmitted by the fluid pumped and by the piping structure. By contrast, resonance is all but negligible for buried piping. In order to design adequately dimensioned anchoring, all pipe anchors in pumping installations should be examined using structural dynamics analysis, with the pump speed serving as the exciter frequency.

[6] *The adjective "rapid" is to be seen in relation to the system's operating conditions. For example, the pressure transients caused by the closure of a valve in a 2 km long pipeline may well stay within the permissible range, whereas the same closing process could generate unacceptably high pressures in a 20 km long pipeline.*

4 The Joukowsky equation

The pressure change Δp_{Jou} in a fluid caused by an instantaneous change in flow velocity Δv is calculated by:

$$\Delta p_{Jou} = \rho \cdot a \cdot \Delta v \qquad (4.1)$$

Δv: Flow velocity change in m/s

ρ: Density of the fluid in kg/m^3

a: Wave propagation velocity through the fluid in the pipeline in m/s

Δp_{Jou}: pressure change in N/m^2

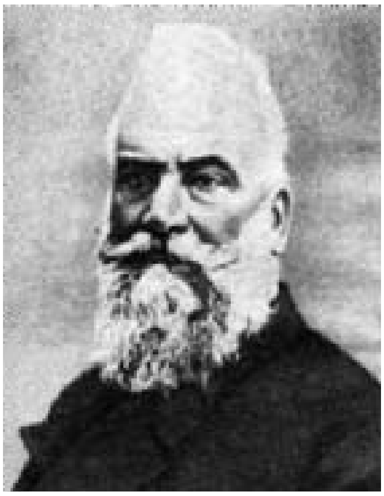
The Δp_{Jou} formula is referred to as the Joukowsky equation. As well as Δv, equation (4.1) contains the density ρ and wave propagation velocity a. *The relationship only applies to the period of time in which the velocity change Δv is taking place.* If Δv runs in opposite direction to the flow, the pressure will rise, otherwise it will fall. If the liquid pumped is water[7], i.e. $\rho = 1000\ kg/m^3$, equation (4.1) will look like this:

$$\Delta h_{Jou} = \frac{a}{g} \cdot \Delta v \approx 100 \cdot \Delta v \qquad (4.2)$$

g: Acceleration due to gravity $9.81\ m/s^2$

Δh_{Jou}: Pressure head change in m

In 1897, Joukowsky conducted a series of experiments on Moscow drinking water supply pipes of the following lengths / diameters: 7620 m / 50 mm, 305 m / 101.5 mm and 305 m / 152.5 mm. He published the



Nikolai Egorovich Joukowsky

results of his various experiments and theoretical studies in 1898.

It may seem inconsistent that Δp_{Jou} in the Joukowsky equation (4.1) seems to have nothing to do with the mass of the flow inside the pipeline. For example, if the water hammer described in the first example in section 3.1 had been based on a pipe diameter twice that of the diameter used, $A = D^2\pi/4$ would have caused the fluid mass and its kinetic energy to turn out four times as large. What seems to be a paradox is instantly resolved if one considers the force exerted on the shut-off valve, i.e. force $F = \Delta p \cdot A$, the defining parameter for the surge load. Because of A, it is now in actual fact four times as large as before.

This shows that one must also consider the fluid mass to judge the risk of water hammer, although that does not seem necessary after a superficial glance at Joukowsky's equation. At the same time, this explains why the pressure surges occurring in domestic piping systems with their small diameters and lengths are usually negligible. In these systems, the kinetic energy levels and fluid masses are very small. In addition, it is practically impossible to close a valve within the very short reflection time of a domestic water system.

The Joukowsky equation can be used to calculate simple estimates. Let's consider three examples:

Example 1:

In a DN 500 pipeline, L = 8000 m, a = 1000 m/s and v = 2 m/s, a gate valve is closed in 5 seconds. Calculate the pressure surge. Calculate the force exerted on the gate.

Answer:

$5\ s < T_r = 16\ s$, i.e. Joukowsky's equation may be applied. If the flow velocity is reduced from 2 m/s to zero as the valve is closed, $\Delta v = 2$ m/s. This gives us a pressure increase $\Delta h = 100 \cdot 2 = 200$ m or approximately $\Delta p = 20 \cdot 10^5\ N/m^2$, which is 20 bar. The valve cross-section measures $A = D^2 \cdot 0.25 \cdot \pi \approx 0.2\ m^2$. The force acting on the gate is $p{\cdot}A = 0.2 \cdot 20 \cdot 10^5 = 4{\cdot}10^5$ N= 400 kN.

[7] *Despite the high flow velocities common in gas pipes, these do not experience surge problems, because p · a is several thousand times smaller than for water.*

Example 2:

A pump delivers water at Q = 300 l/s and a head Δh = 40 m through a DN 400 discharge pipe measuring L = 5000 m into an overhead tank; a = 1000 m/s. The inertia moments of pump and motor are negligible. Is there a risk of liquid column separation, i.e. macro-cavitation, following pump trip? If so, what is the anticipated pressure increase?

Answer:

Q = 300 l/s in a DN 400 pipeline roughly corresponds to a flow velocity v = 2.4 m/s. As a result of pump trip and the loss of mass inertia moment, the pump comes to a sudden standstill, i.e. Δv = 2.4 m/s. According to the Joukowsky equation, this causes a head drop of Δh = -100 · 2.4 m = -240 m. Since the steady-state head is just 40 m, vacuum is reached, the liquid column collapses and macro-cavitation sets in. Following the liquid column separation near the pump outlet, the two liquid columns will recombine with great impact after some time. For reasons of energy conservation, the highest velocity of the backward flow cannot exceed the original velocity of the steady-state flow of 2.4 m/s. Under the most unfavourable conditions, the cavitation-induced pressure rise will, therefore, be Δh = 100 · 2.4 = 240 m, which is the equivalent to 24 bar.

Example 3:

A pump delivers water at Q = 300 l/s and a head Δh = 40 m into a 2000 m long pipeline DN 400; a = 1000 m/s. The mass moment of inertia[8] of all rotating components (pump, motor, etc.) is J = 20 kgm^2, the speed of rotation n_0 = 24 s^{-1} and the total efficiency = 0.9, i.e. 90%. Is there a risk of liquid column separation, i.e. macro-cavitation, following pump trip?

Answer:

For the instant of pump failure, the change in speed $\dot{n}$ may be derived from the inertia equation as follows:

$$M_p = 2 \cdot \pi \cdot J \cdot \dot{n}$$

Assuming as an (extremely rough) approximation a linear speed reduction $\dot{n} = \dfrac{n_o}{\Delta t}$, then, if $M_p = \dfrac{\Delta p \cdot Q}{2\pi \cdot n_0 \cdot \eta}$,

we obtain a time Δt in which the speed has dropped to zero, and, if Δp = 1000 · 9.81 · Δh,

$$\Delta t = \frac{(2\pi \cdot n_0)^2 \cdot J \cdot \eta}{\Delta p \cdot 0.001 \cdot Q} \approx 4 \cdot \frac{n_0^2 \cdot J \cdot \eta}{\Delta h \cdot Q} = 3.4 \text{ s}$$

The reflection time of the pipeline is T_r = 4 s (for a = 1000 m/s), which means that the reflected pressure relief wave will not reach the pump until after the speed has dropped to zero and it is too late for the relieving effect to take place. It is, therefore, probably safe to say that macro-cavitation will develop.

4.1 Scope of the Joukowsky equation

The Joukowsky equation only applies to:

- Periods of time which are equal to or shorter than the reflection time of the piping T_r
- The period of time which falls within the velocity change Δv
- Pipes characterised by friction losses within the limits typical of water transport systems

Reflection time T_r:

In **Fig. 3.2-b** the wave of reduced pressure reflected by the tank has arrived at the gate valve after T_r has lapsed, and evens out some of the pressure increase Δp. If the change in flow takes place in a period of time Δt longer than T_r, the rise in pressure Δp_{Jou} will only occur at the wave's source, whereas it will have diminished to the value given by the boundary condition by the time it reaches the opposite end of the pipeline.

Fig. 4.1-a shows the pressure envelope, which applies to a case of this kind:

[8] *Mass moment of inertia J: J expressed in kgm^2 is the correct physical quantity. Flywheel moment GD^2, which was used in the past, should no longer be used, because it can easily be confused with J!*

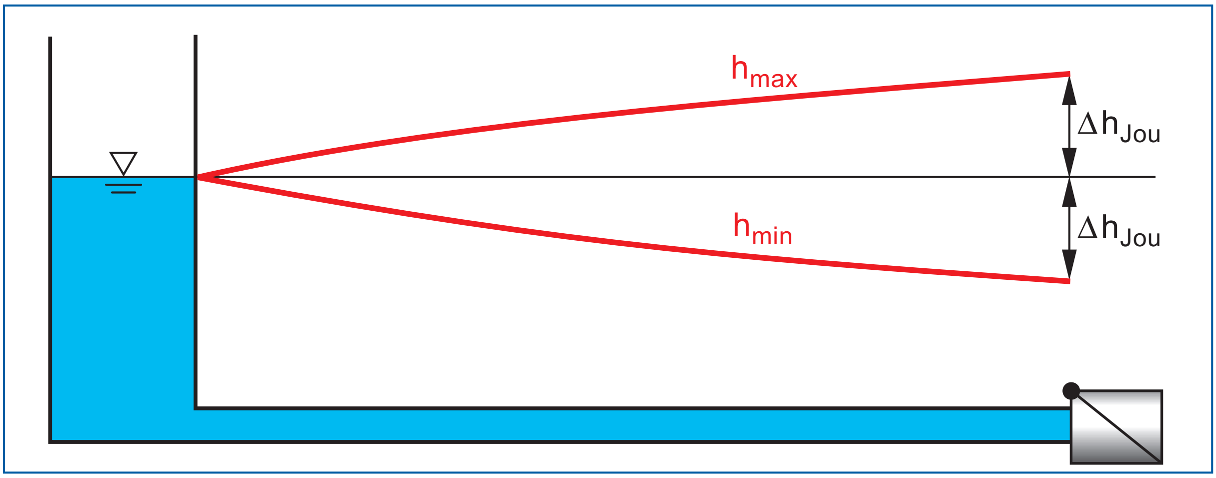


Fig. 4.1-a: Pressure head envelope for closing times > reflection time T_r .

Friction

If the liquid pumped is highly viscous or if the pipeline is extremely long (say, 10 km and more), the work done by the pump only serves to overcome the friction produced by the pipeline. Changes of geodetic head due to the pipe profile, by comparison, are of little or no importance. The Joukowsky equation no longer applies, not even within the reflection time of the pipeline. In a case like this, the actual pressure rise following the sudden closure of a gate valve can be several times that of Δp_{Jou} as calculated by the Joukowsky equation! The phenomenon caused by the pipe friction is commonly called line packing. The following flow simulation calculation gives an example of this:

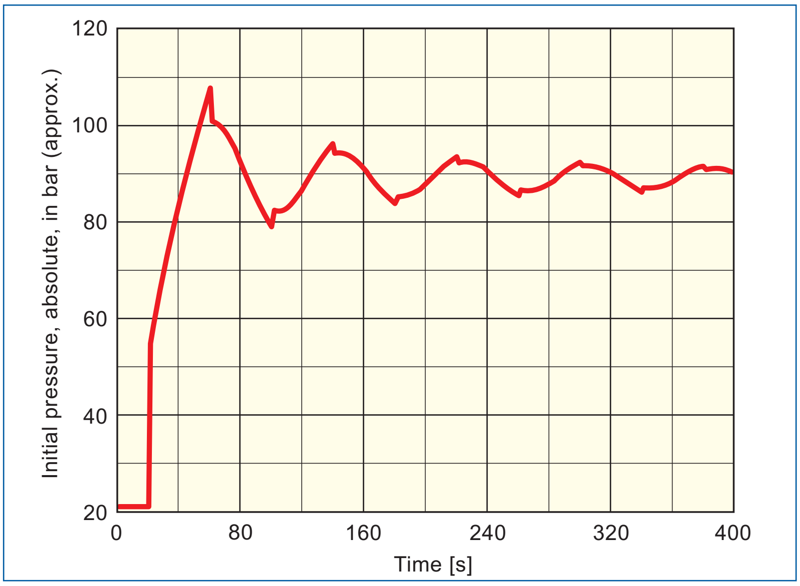


Fig. 4.1-b: Pressure curve at the outlet of a 20 km long crude oil pipeline following a sudden gate valve closure. Calculation parameters: DN 300, k = 0.02 mm, inlet pressure 88 bar constant, Q = 250 l/s, fluid pumped: crude oil, ρ = 900 kg/m³

The gate valve in the example shown in **Fig. 4.1-b** closes 20 s after the start of the calculation. The first steep increase by approx. 20 bar to approx. 55 bar is Δp_{Jou} according to the Joukowsky equation; the continued increase to almost 110 bar is caused by line packing. Line packing is only of significance for long pipelines or highly viscous media. It is unlikely to occur in urban water supply and waste water disposal plants.

Wave propagation velocity

The wave propagation velocity is one of the elements of the Joukowsky equation and, therefore, a vital parameter for defining the intensity of a surge. It is calculated by solving equation (4.1).

$$a = \frac{1}{\sqrt{\frac{\rho}{E_F} + \frac{\rho \cdot d_i \cdot (1-\mu^2)}{E_R \cdot s}}} \text{ m/s} \qquad (4.1)$$

ρ: Density of the fluid in kg/m³
E_F: Modulus of elasticity of the fluid in N/m²
E_R: Modulus of elasticity of the pipe wall in N/m²
d_i: Inside pipe diameter in mm
s: Pipe wall thickness in mm
μ: Transverse contraction number

Equation (4.1) produces a range of values from approximately 1400 m/s for steel pipes to around 300 m/s for ductile plastic pipes. Wave propagation velocity "a" in an unconfined body of water is approximately 1440 m/s. To all intents and purposes, the validity of equation (4.1) should not be overestimated; surge analyses are often performed without it, in which case the value of "a" is estimated. The volume of air contained by the fluid, which equation (4.1) does not take into account, can have a strong impact on "a", as is shown by some examples in Table 4-1: In drinking water supply pipelines the gas content is negligible; in waste water installations it normally is not. Further elements of uncertainty with regard to "a" mainly concern pipes made of synthetic material. An unknown and varying modulus of elasticity, manufacturing tolerances, the age of the pipeline and, in particular, the question whether the pipeline is laid in the ground or not, all play a part. A buried pipeline has considerably higher values of "a" than a pipe laid above ground.

Gas content % by volume	a m/s
0	1250
0,2	450
0,4	300
0,8	250
1	240

Table 4-1: "a" as a function of the gas content at a static water pressure of approximately 3 bar

5 Numerical simulation of water hammer

In current theory, the dependent model variables are the pressure p and the flow velocity v in the two partial differential equations (5.1) for every single pipe of a piping system; the time t and an unrolled reach of pipe x are independent variables.

Equations (5.1) are generally valid and cover the effects of both inertia and elasticity. Mathematically, the pipe ends serve as the boundary conditions of equations (5.1); different types of boundary conditions are introduced to include internal components such as pipe branches, vessels, pumps and valves in the model. For example, the creation of a complete piping system by connecting a number of individual pipes is done by taking a pipe node to be the boundary condition. The starting condition of equation (5.1) is the steady-state flow inside the pipe concerned before the onset of the disturbance. Equations (5.1) are solved by means of the characteristics method, which provides the basis for almost all surge analysis software available today.

The time frame covered by equations (5.1) is less appropriate for computing resonant vibrations. These can be calculated much more precisely using the impedance method, or, in other words, by looking at the frequency range.

$$\frac{\partial v}{\partial x} + \frac{1}{\rho \cdot a^2} \cdot \frac{\partial p}{\partial t} = 0 \qquad (5.1)$$

$$\frac{\partial v}{\partial t} + \frac{1}{\rho} \cdot \frac{\partial p}{\partial x} - g \cdot \sin(\alpha) + \frac{\lambda}{2 \cdot d} \cdot v \cdot |v| = 0$$

5.1. Accuracy of numerical surge analysis

Computer programs based on the characteristics method produce solutions whose accuracy by far exceeds that which is called for in practice. This is evidenced by numerous comparisons with actual measurements. Significant differences were only found for calculations aimed at predicting macro-cavitation or dampening of pressure waves inside a pipe.

For example, the pressures computed using the standard model of vapour cavitation derived from equations (5.1), i.e. the assumption of a simple cavity of low pressure following liquid column separation, are always higher than what they are in reality. However, the advantage of the conservative outcome is that one is always on the safe side.

The real energy losses due to friction, and the degree of warping of pipeline and pipe fixtures are somewhat larger than the forecast supplied by simulation. The first pressure peaks and valleys, therefore, tend to be simulated very precisely, whereas the pressures further down the line are on the whole depicted with an increasing lack of dampening. But imperfections of this kind are negligible compared with inaccuracies caused by entering wrong or insufficient input data.

Some of the potential sources of error are:

- Inaccurate valve and/or pump characteristics.
- Lack of knowledge about the actual wave propagation velocity inside the pipeline.
- Lack of information about tapping points in a main pipe.
- Unawareness of the degree of incrustation inside the pipes.

This shows that the quality of the surge analysis stands or falls with the accuracy of the input data.

A surge analysis can only be as accurate as the system data entered as inputs. Only if the input is accurate, and the computation model is a faithful reproduction of the real system conditions, will the analysis yield a high degree of accuracy.

In practice, it is often impossible to obtain exact data. If this is the case, one has to estimate the required inputs.

An example:

For a valve manufacturer, a small individual loss coefficient in the open condition of a valve is a powerful sales argument. By contrast, for a surge analysis the values obtained immediately preceeding total closure of a valve are of the essence, and measuring these is a time-consuming and complex affair. As a result of this, many individual loss characteristics available for valves do not extend far enough into the closing range. For cost reasons, the individual loss curves provided by most manufacturers are extrapolations, rather than curves plotted on the basis of original measurements.

When designing a plant with the aid of surge analyses, inaccuracies of this kind should be accounted for by designing the surge control equipment slightly on the conservative side.

5.2 Forces acting on pipelines as a result of water hammer

After computing the time-dependent pressure gradients, it takes a further separate step to calculate the forces acting on the elbows and connections of unsupported pipes. The interaction between fluid and pipe wall does not enter into the computation (separate calculation). Apart from the odd exception, which is of no relevance in the field of water supply and waste water disposal anyhow, this method tends to produce forces which are somewhat higher than what they are in reality, so that the conclusions drawn from the calculation results will definitely be on the safe side.

6 Computerised surge analysis

6.1 Technical procedure

A surge analysis will not provide direct solutions for the required parameters, such as, for example, the optimum air vessel size, compressor settings, valve closure characteristics, flywheel dimensions, etc. Instead, the surge analyst must specify the type of surge control to be employed and provide estimates of the relevant parameters. After checking the outcome of the surge analysis, the original parameters are suitably adjusted and a complete re-run of the surge analysis is made for the system. After several runs, the values supplied will come very close to the technical and economical optimum. As surge analyses necessarily need to be performed by surge specialists, they remain time and labour intensive despite the use of modern computer technology.

Considering that powerful surge analysis software is now commercially available, users may wonder whether they cannot do their own analysis just as well. As reliable[9] surge analysis software is far from a mass product, the low sales volume makes it expensive. Add to this the high cost of training and hands-on practice. Also if the software is not used for some time, operators usually have to brush up their skills. So, if users require fewer than, say, ten analyses per year, the cost involved in doing their own will probably not be worthwhile.

6.2 Interaction between ordering party and surge analyst

First of all, a distinction has to be made between the quotation phase and the calculation itself. During the quotation phase, the surge analyst requires the following information from the plant-engineering contractor to compute the cost involved:

1. A rough flow diagram of the installation indicating all important equipment, such as pumps, valves, additional inlet and outlet points, as well as any existing safety devices, such as aerators, air vessels, etc. The flow diagram can by all means be in the form of a quick sketch, which does not take more than a couple of minutes to draw.
2. A rough list of all main parameters, i.e. principal pipe lengths, diameters and flow rates.
3. A list of all major operation and downtime periods.
4. A list of all known incidents that could have been caused by water hammer.
5. Irregularities observed during operation.

If a surge analysis is to be performed, additional data to be specified by the surge analyst will have to be obtained. Some examples of additionally required data are:

- Piping elevation profile
- Lengths
- Diameters
- Wall thickness
- Materials of construction, lining material, pipe connections
- Pressure class, design pressure head curve
- Permissible internal pipe pressures (p_{min}, p_{max})
- Method used to lay the pipes: buried or placed on supports
- Modulus of elasticity of pipe materials
- Surface roughness coefficient
- Provision of air valves at the highest points of the piping
- Branch connections
- Zeta or flow factors as well as valve closing characteristices
- Characteristic curves or performance charts and characteristic data of all hydraulic equipment
- Mass moments of inertia of all hydroelectric generating sets
- Characteristic curves and data on surge control equipment already installed in the system
- Characteristic values of all aeration and deaeration equipment
- Settings of control equipment
- Water levels in tanks and reservoirs
- Rates of flow in the individual piping branches
- Degrees of opening of all shut-off and throttling valves
- Operating pressures

[9] *Users are in the uncomfortable position of not being able to verify the workings of surge analysis software. It is, therefore, important that a reputable manufacturer vouch for the quality of the product. Surge analysis software, as a rule, is developed by specialist university institutes. There are some examples of programs that were bought by commercial enterprises and provided with a sophisticated user interface, which makes them easier to handle for the user.*

7 Advantages of rules of thumb and manual calculations

A rough estimate can be a very useful tool to quickly assess the risk of water hammer. This leads us to the validity of rules of thumb and to the question whether the surge characteristics of one system can be applied to another, similar installation (scalability). To answer that question, we should start by pointing out that there is a great variety of water supply and waste water disposal plants, and that these are so different from each other that approximation formulas cannot be applied. Even if the characteristic values of different systems are very similar, i.e. same rates of flow, same pipe lengths, they cannot normally be scaled.

A simple example shows why: The only difference between two otherwise completely identical water supply systems are the elevation profiles of the main pipes; one system has a high point, the other does not. The system without the high point can be safely protected by an air vessel. A vessel of the same size will not adequately protect the second system, however, because the falling water level in the air vessel would cause the minimum dynamic pressure head to intersect the pipeline's high point. The low pressures thus created would pose a risk of dirty water being drawn into the system.

It takes lots of experience to be able to judge whether approximation formulas can be used to reliably calculate transient flow conditions. For every day engineering purposes, approximation formulas should be used exclusively to roughly estimate the potential risk in a system (examples, see section 4). Using them as a basis for a serious surge analysis or, even worse, for designing the surge control equipment, would have to be regarded as highly irresponsible. A brief description of all known processes of approximation and estimation formulas are detailed below:

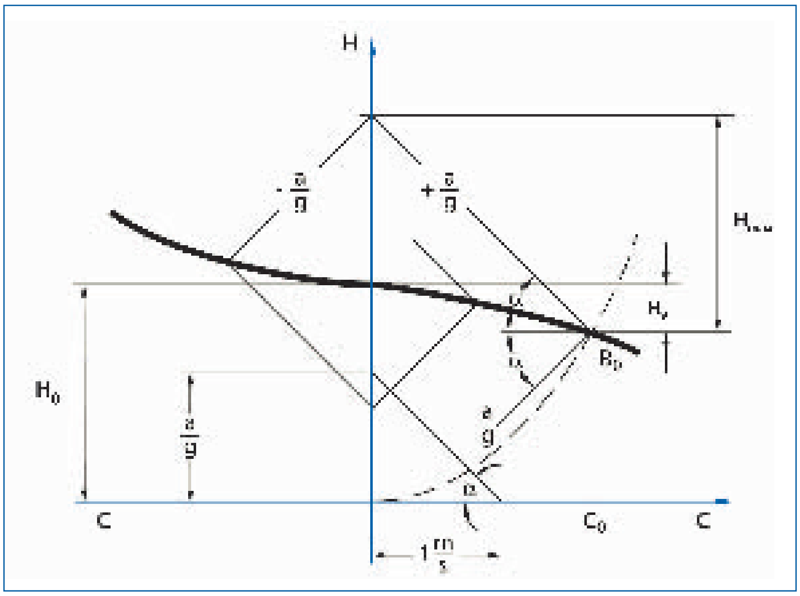


Fig. 7-1: Graphical method developed by Schnyder-Bergeron

- Before the days of modern computer software, the graphical Schnyder-Bergeron method was often employed and produced relatively reliable surge analysis. For practical reasons, use of this method is limited to systems comprising a single pipeline. Friction can only be taken into account by complex procedures. Besides, it takes a specialist to apply this method and obtain the desired results. **Fig. 7.1** is an example of a typical Schnyder-Bergeron diagram, which shows how the pressure wave propagation due to the closure of a valve is determined by graphical means.
- Application of the Joukowsky equation for rapid changes in flow velocity v (examples under 4).
- Graphical method to determine the required air vessel sizes.*)
- Graphical method used to estimate the condition of line packing.*)
- The largely ideal valve closing characteristics for the exceptional case of a single-conduit pipeline can be calculated by approximation.*)

These are the only manual calculation methods. This apparent lack is more easily understood if we take another look at the air vessel, our representative example of before. Reading the total volume of the vessel from a design curve is not all that is required. The way the air vessel works depends to a large extent on the ratio of water volume to air volume in the vessel, or, in other words, on the question whether pre-pressurisation of the vessel is "hard" or "soft". The pre-pressurisation level has an impact on the total vessel volume required. The pipeline profile also plays a significant part. For example, if it has a high point which should not be intersected by the minimum dynamic pressure head curve following pump trip (area of low pressure), the basic conditions for designing the vessel will be different, even if the plant parameters are otherwise the same. The vessel will have to be considerably larger. In many cases, the swing check valve and throttle installed in a bypass will keep the reverse pressure wave from causing an impermissible rise in pressure levels in the air vessel. It is impossible to determine these crucial variables using rules of thumb or graphical design methods.

*) *Expertise required.*

8 Main types of surge control

The purpose of surge control is to stop kinetic energy from being converted into elastic deformation energy. This can be done by the following basic methods:

– Energy storage

– One-way surge and venting facilities

– Optimization of valve closing characteristics[10]

– Optimisation of the strategy designed to control the piping system

8.1 Energy storage

With air vessels and standpipes, energy is stored as pressure energy; when a flywheel is installed, the energy stored takes the form of rotational energy. There is a sufficient amount of energy stored to maintain the steady-state flow for a relatively long time and to make sure the decrease in flow velocity due to dissipation will be slow to take full effect. A rapid pressure drop is thus prevented. If air vessels and standpipes are installed upstream of a pump in a long inlet pipe, they not only prevent a pressure transient by means of energy dissipation, but also the other way around, by absorbing energy.

8.1.1 Air vessels

Air vessels come in the form of compressor vessels (**Fig. 8.1.1-a**), [bag-type] accumulators (**Fig. 8.1.1-b**) and vessels with a vent pipe. Compressor- and accumulator-type air vessels basically work on the same operating principle. The reason for choosing one or the other is based on technical or commercial considerations. Because of their design, accumulators are only suitable for small volumes.

As explained earlier, the vessel volume is not the only important factor. If the water-to-air volume ratio is carefully chosen, a vessel with a substantially lower total volume may be used.

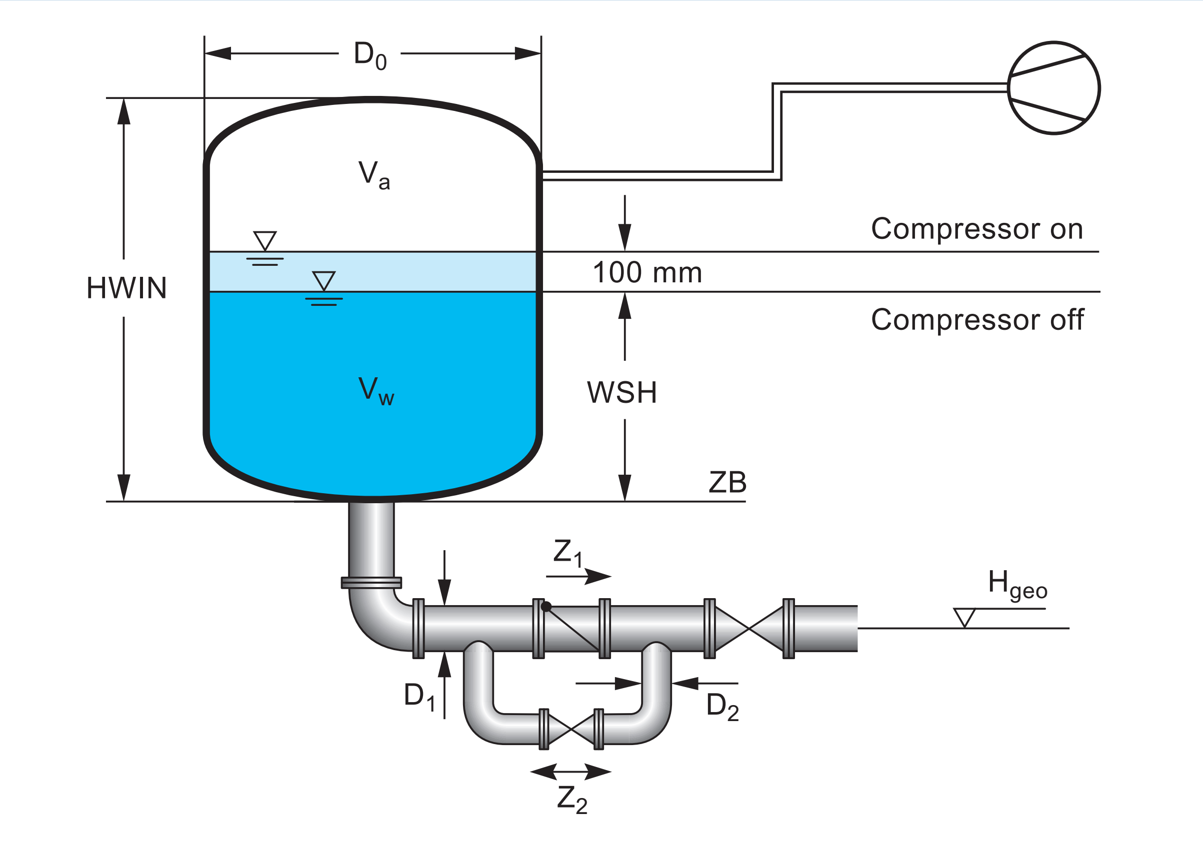


Fig. 8.1-a: Schematic layout of a compressor-type air vessel. To avoid excessive pressures on return of the vessel water, the connecting pipe may have to be fitted with a swing check valve with a throttled bypass.

[10] The valve closing charasteristics describe the closing angle of a valve as a function of time.

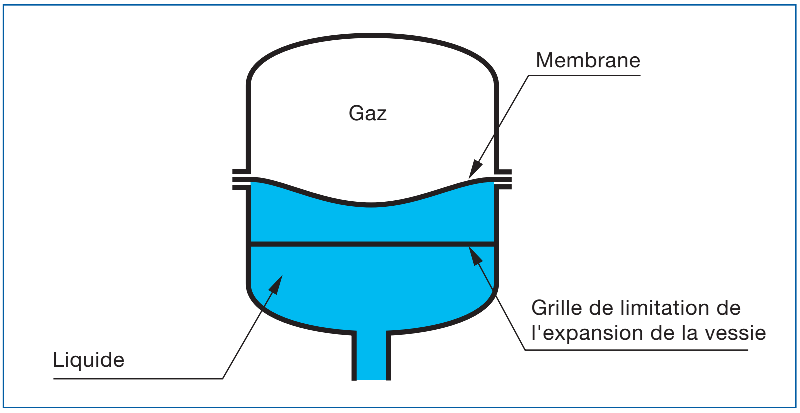


Fig. 8.1.1-b: Schematic of an accumulator

To make sure compressor vessels are always filled to the correct levels, they can be equipped with sensors which will switch the compressor on or off as required. Bag-type accumulators are typically adjusted by pre-pressurising the gas inside the bag or membrane enclosure to a certain initial pressure prior to installation.

Air vessels are not just installed at the pump discharge end to guard against the consequences of pump trips. They can also be installed in other suitable places in a piping system. For example in long inlet pipes, an additional air vessel at the inlet end of the pump provides effective surge control. If the pump fails or trips, an upstream vessel will absorb energy, while a downstream vessel will dissipate energy.

Air vessels or accumulators are not suitable for waste water disposal systems[11], because

- With waste water, it is not possible to measure the water level needed to set the compressor.
- The bag-like enclosure in an accumulator would be punctured by the sharp objects contained in the wastewater, such as razor blades, nails, etc.
- There is a major risk of incrustations, deposits and blockages.

Fig. 8.1.1-c: Accumulators

Provided they are adequately monitored, the operating reliability of air vessels is high. During their operation, attention has to be paid to the following:

- Monitoring of the water level in the vessel.
- For reasons of hygiene, the water volume must be continuously or regularly replaced.
- The compressed air must not contain any oil.
- To be able to take the air vessel out of service for an inspection, spare vessels should be available.
- It must be possible to lock the shut-off valves in the connecting pipeline against unintentional closure; the open position has to be monitored.
- Maintenance of the compressor (compressor vessel).

8.1.2 Standpipes, one-way surge tanks

Standpipes can only be installed at points of a piping system characterised by low-pressure heads. As a rule, a standpipe cannot replace a downstream air vessel. Fitted with a swing check valve in the direction of the flow and a filling mechanism (one-way surge tank), it is used to stop the pressure falling below atmospheric at the high points

[11] *An exception is a vessel fitted with a vent pipe; this arrangement, comprising an air vessel, a standpipe and a vent valve, is very rarely used in Germany.*

of long clean-water pipelines. Because of the possibility of malodorous fumes, standpipes are rarely found in wastewater installations. Standpipes and one-way surge tanks are highly reliable pieces of equipment provided the following points are observed:

- Continuous or regular changes of water (problem of hygiene).
- Filtering of air flow.
- Functional tests of the check valve on one-way surge tank arrangements.
- Monitoring of water level or filling device on one-way surge tank arrangement.

8.1.3 Flywheels

Mounted on the driver, a flywheel prolongs the rundown time of a pump to standstill by means of the stored rotational energy:

$$E_{kin} = \frac{1}{2} \cdot J \cdot \omega^2 \qquad (8.1)$$

J: Mass moment of inertia of flywheel in kgm^2

ω: Angular velocity s^{-1}

For a homogeneous solid disc with a radius r and a mass m, for example, the mass moment of inertia is

$$J = \frac{m \cdot r^2}{2}$$

Figs. 8.1.3-a and **8.1.3-b** show several practical applications. However, with a type of flywheel that is economically and technically feasible, one can only achieve a prolongation of the running down time of the kind which is suitable for a relatively short pipeline, or, put differently, with a short reflection time T_r. The limits for employing a flywheel are in the region of 1 to 2 km pipeline length. Example 3 in section 4 includes a rough estimate performed to check whether a flywheel can be used. For reasons of design, the flywheel solution is not suitable for submersible motor pumps. On other pump types, it must be checked in advance that the flywheel will not interfere with the starting procedure of the pump driver. Flywheels are probably the safest and most elegant types of surge control. Their reliability beats that of all other surge control methods. With the exception of the bearings of larger-scale systems, they do not require any in-operation monitoring.

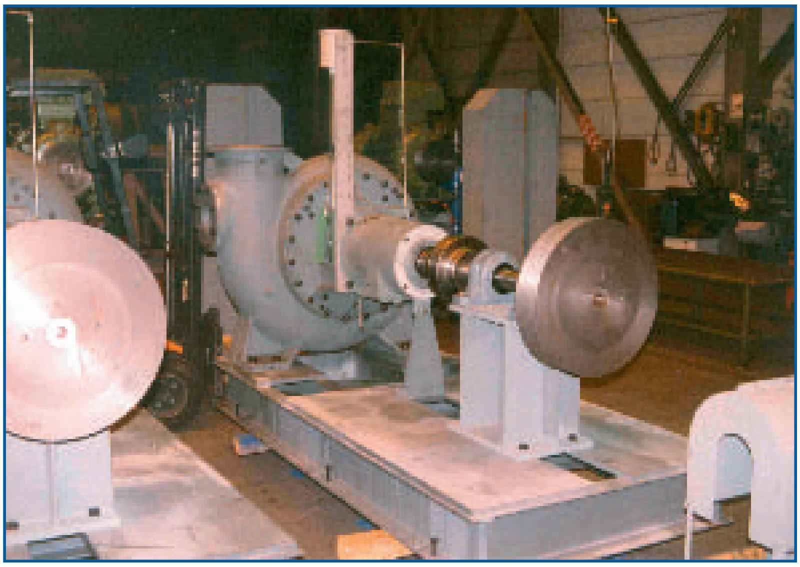


Fig. 8.1.3-a: The V-belt pulleys in this arrangement are solid discs

Fig. 8.1.3-b: Vertically mounted flywheel (driven by means of cardan shaft, D = 790 mm)

8.2 Air valves

Air valves should not be used until every other solution has been ruled out. Their drawbacks are:

- They require regular maintenance.
- If arranged in the wrong place or mounted incorrectly, they can aggravate pressure variations instead of alleviating them.
- Under certain circumstances, operation of the plant may be limited, because the air drawn into the system has to be removed again.
- The handling of wastewater calls for special designs.

Air valves (**Fig. 8.2-a**) have to be carefully designed. On large diameter pipelines, one has to arrange air outlet valves on top of domes, to make sure that the air drawn into the system will collect there. As long as the fluid flow has not reached the steady state, air drawn into pipes can, under unfavourable conditions, have a very negative effect. Air cushions normally have a dampening effect. However, the air drawn into the pipeline can also give rise to dangerous dynamic pressure increases. It has to be pressed out of the piping slowly; a large air outlet cross-section would lead to sudden pressure variations towards the end of the air outlet operation. For this reason, aerators and deaerators have different nominal diameters depending on which way the air flows. Air normally flows in through a large cross-section and out through a small cross-section.

Fig. 8.2-a: Duojet) two-way air valve with a medium-operated single-compartment valve.*

Large vent cross-section for drawing in and venting large amounts of air during start-up and shutdown of pumping systems.

Small cross-section for removing small amounts of air during operation against full internal pressure.

The reliability of aerators / deaerators depends on their design and is the lowest of all surge control equipment. They have to be tested for proper functioning in regular intervals and it may be necessary to filter the incoming air.

8.3 Actuated valves

Suitable actuation schedules for the opening and closing of valves are calculated and verified by means of a surge analysis on the basis of the valve characteristic.

The valves will give very reliable service if, on valves with electric actuator, adequate protection is provided for the actuating times and the break points of the actuation schedules or if, on valves with hydraulic actuators, adequate safety elements, such as orifice plates or flow control valves, are used. Proper valve functioning has to be checked at regular intervals with regard to the actuating times and closing characteristics.

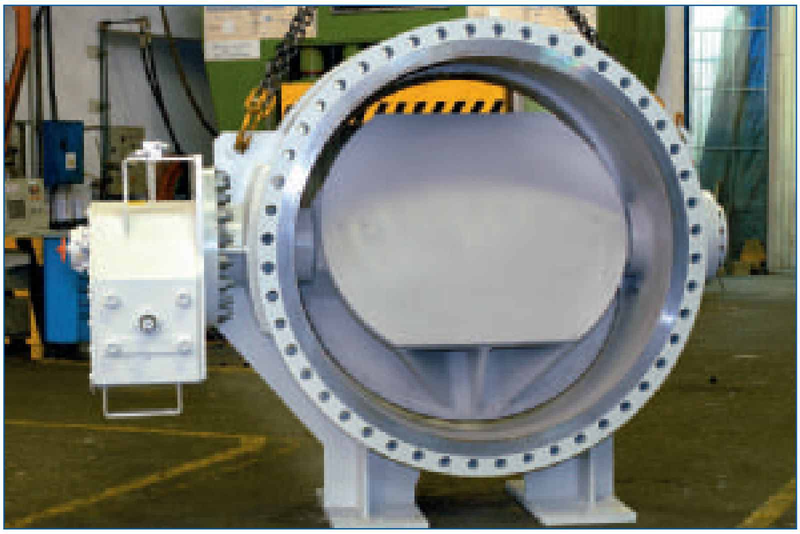


Fig. 8.3-a: Motorised shut-off butterfly valve

**) With the friendly permission of VAG-Armaturen GmbH.*

8.4 Swing check valves

The dynamics of swing check valves often have a major influence on the development of surge, because the valve's closure, after reversal of flow, generates velocity changes which, according to Joukowsky's equation (4.1), produce pressure variations.

Check valves generally have to meet the following two contradictory requirements:

- bring the reverse flow to a standstill as quickly as possible,
- keep the pressure surge generated during the process as small as possible.

Drinking water pumping installations protected by air vessels should ideally be equipped with nozzle check valves. Free-swinging valve discs can have a very unfavourable effect, because they take a long time to close, which means reverse flow sets in while they are still partly open, and the valve disc re-seats with considerable impact. The phenomenon is known by the term "check valve slam" and is much dreaded. Since the closing time is the main criterion for check valve slam, limit position dampers will improve the situation, but not eliminate the risk altogether. In waste water systems, nozzle check valves cannot be used because they tend to clog up. This means that valves with free-swinging discs and limit position dampers are the only remaining option, despite their drawbacks.

Pump check valves installed in the cooling pipes of a power station are designed to throttle the reverse flow in a controlled manner after the pump trips. This feature is important on pumps operated in parallel, when one pump fails whilst the remaining pumps continue to run and deliver flow against the tripped pump. In a case like this, controlled closing is achieved by adjustable hydraulic actuators without external supply but with a lever and counterweight, with the free-swinging valve disc opening in the direction of the flow and, upon actuation, closing in one or two stages according to a set closing characteristic.

The operating reliability of check valves is relatively high. In operation, they have to be checked for proper functioning at regular intervals.

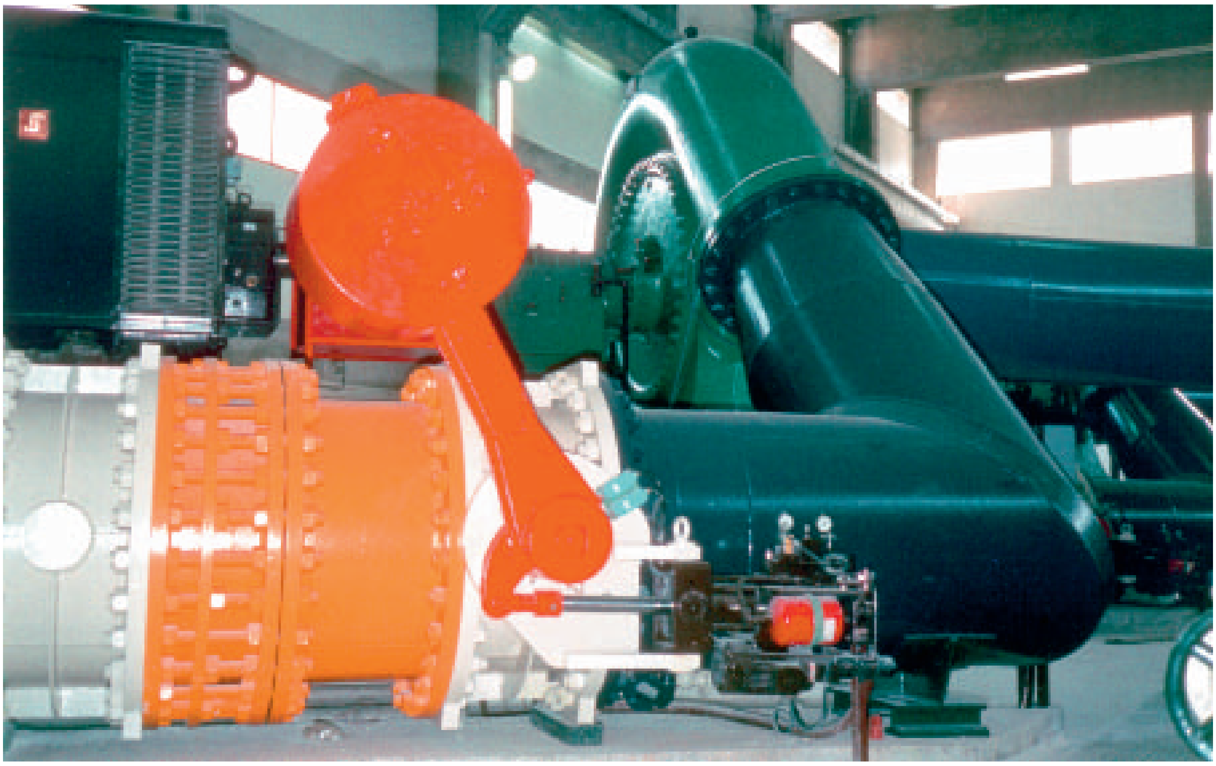


Fig. 8.4-a: Swing check valve equipped with a hydraulic actuator and counterweight

9 Case studies

The case studies below were taken from surge analyses performed by KSB. Although we have altered the system parameters, so that the installations concerned remain anonymous, the problems involved and the way these were resolved have not been altered.

9.1 Case study: long-distance water supply system

The system parameters are indicated in **Fig. 2.1-b**. A steady-state flow Q_{steady} = 500 l/s is pumped through a DN 600 pipeline of ductile cast material with a total length of L = 2624 by three centrifugal pumps operating in parallel at a total head of the pumps h_{steady} = 122.5 m into an overhead tank. The disturbance under investigation, which leads to excessive dynamic pressures, is the simultaneous failure of all three pumps. The dynamic pressure peaks produced by far exceed the permissible nominal pressure of PN 16 (see h_{max} curve) in **Fig. 2.1-b**; the minimum pressures drop to vapour pressure in wide areas of the system (see h_{min} curve) in **Fig. 2.1-b**. The system can be protected by installing an air vessel at the inlet of the long-distance pipeline. Although the vessel dimensioned as shown in **Fig. 2.1-b** will initially prevent the development of areas of low pressure, the water column in the pipeline swinging back will still produce dynamic pressure peaks in excess of 16 bar. Therefore, the reverse flow into the air vessel has to be additionally throttled; a schematic diagram of the operating principle is shown in **Fig. 8.1.1-a**. In the present case, the throttling action is achieved with the aid of a short length of DN 200 pipe fitted with a standard DN 80 orifice. **Fig. 2.1-b** shows the calculated pressure envelope with and without air vessel. The maximum head curve obtained with an air vessel h_{maxWK} is now only slightly above the steady-state head curve h_{steady} and the associated minimum head curve h_{minWK} runs at a wide safety margin above the peak point of the pipe.

Fig. 9.1 shows the head and flow curves of the system protected by an air vessel arrangement plotted against time (heads expressed in m above mean sea level)

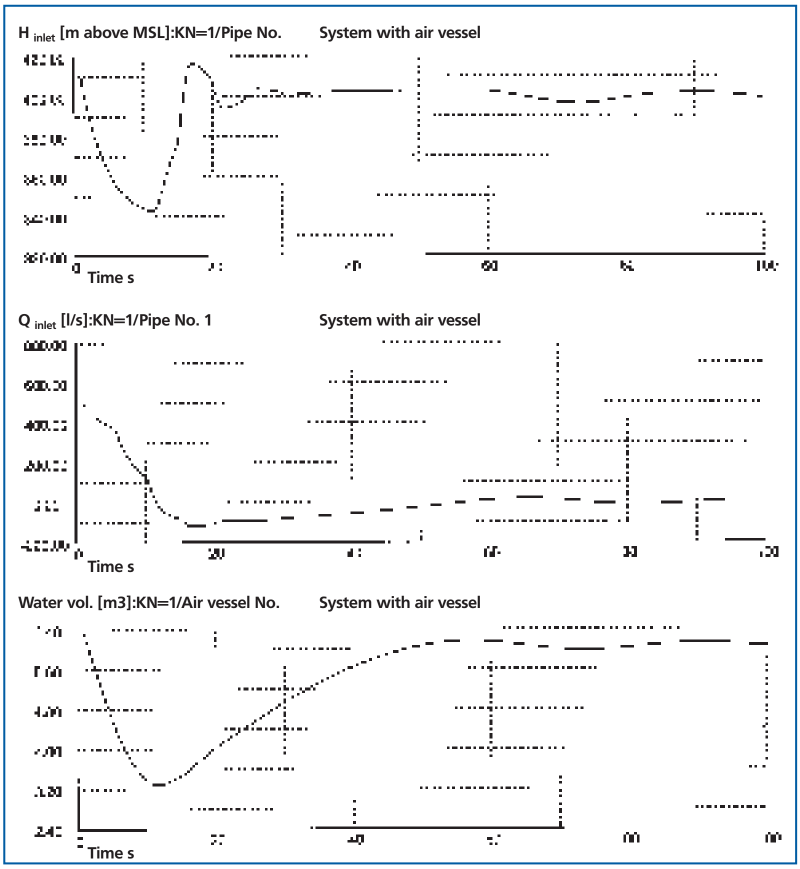


Fig. 9.1: Time plots for the long-distance water supply pipeline (Fig. 2.1-b); the example shows the head and flow curves of an air vessel-protected system as functions of time (heads expressed in m above mean sea level)

9.2 Case study: a stormwater conveyance pipeline

Starting from a wastewater pumping installation, a new DN 350 stormwater pipeline with a total length of L = 590 m was laid to an aeration structure. Pumping operation was by means of three identical pumps running in parallel, each equipped with a non-return valve and a motorised gate valve to control pump start-up and run-down. The first 100 m of pipe made of high-density polyethy lene were laid under ground, the remaining 490 m were of steel and laid above ground supported on pipe bridges. **Fig. 9.2-a** shows a schematic of the model installation. The nodes connecting the above-ground single pipes of the model, are 90° elbows. The engineering firm in charge of planning the plant neither performed ordered a surge analysis to accompany the project-planning phase.

During the first operating tests following the plant's completion, several incidents, among them a power failure which caused all three pumps to fail at the same time, caused the part of the piping laid above ground to shake considerably, damaging pipe fixtures and tearing off some pipes altogether.

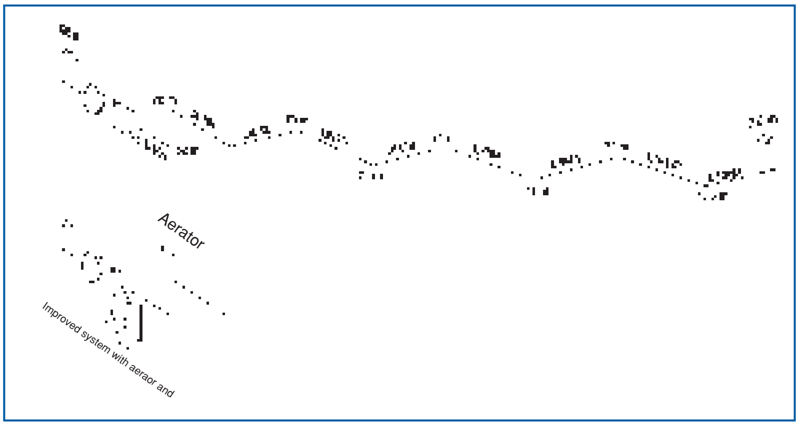


Fig. 9.2-a: Schematic diagram of the stormwater conveyance pipeline used in the example

When a surge analysis was finally ordered, its objective was:

- to determine what caused the surge pressures and forces that had been observed,
- to devise some protective measures or surge control equipment that would prevent the excessive dynamic pressures produced by a pump failure from occurring, and to prove their effectiveness mathematically.

Model parameters

Besides the parameters indicated in **Fig. 9.2-a**, the following system data were entered into the calculation:

Pump characteristic shown in Fig. 9.2-c

Model pipeline L1:
Material:
high-density polyethylene (HDPE)
D_{inside}: 354.6 mm
k: 0.1 mm
a: 600 m/s (estimated value)
Min. permissible pressure: vacuum
Pressure class: PN 6

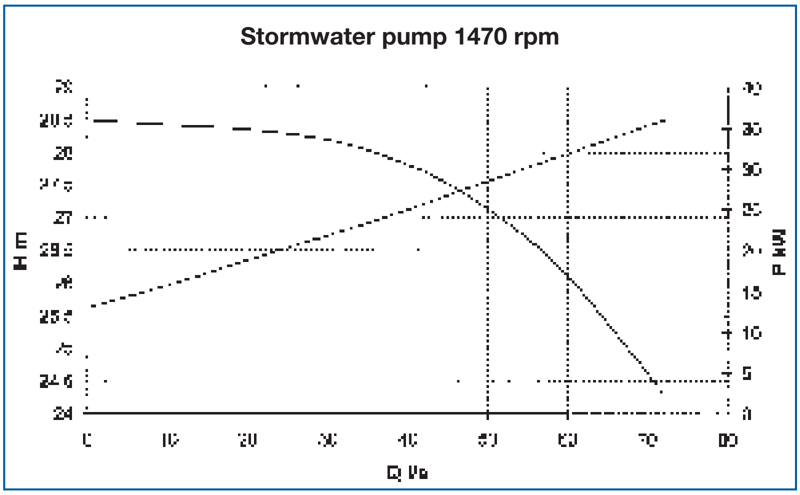


Fig. 9.2-c: Characteristic curve of the pump used in the stormwater conveyance system

Model pipeline L2 to L10:

Material: steel
D_{inside}: 349.2 mm
k: 0.1 mm
a: 1012 m/s
(from equation 4.1)
Min. permissible pressure: vacuum
Pressure class: PN 10

Nothing was known about the pump check valves. For the purpose of the model, it was therefore assumed – correctly so, as it turned out – that the valves would suddenly close upon reverse of the flow direction.

Calculation of actual duty data, first results

The steady-state flow calculated by the surge software for the parallel operation of three pumps amounted to Q_{steady} = 187 l/s. The first surge calculation of the simultaneous failure of all three pumps showed that macro-cavitation and, as a result of it, dynamic pressure peaks as high as 15 bar would occur inside the HDPE pipeline, i.e. considerably in excess of the given nominal pressure of the pipe of PN 6. The calculation showed that the pipe bridges between each pair of 90° elbows had to temporarily withstand longitudinal forces of just under 100 kN, or in terms of weight, the equivalent of a thrust somewhere in the region of 10 t. **Figs. 9.2-d** and **9.2-e** show some examples of the system behaviour without surge control plotted over time. **Fig. 9.2-d** shows the pump speed, head and flow at the entrance of model pipe L1 (head in m above pipe centreline); the curve in **Fig. 9.2-e** shows the axial forces acting on L8. This explained the violent shaking and resulting damage observed.

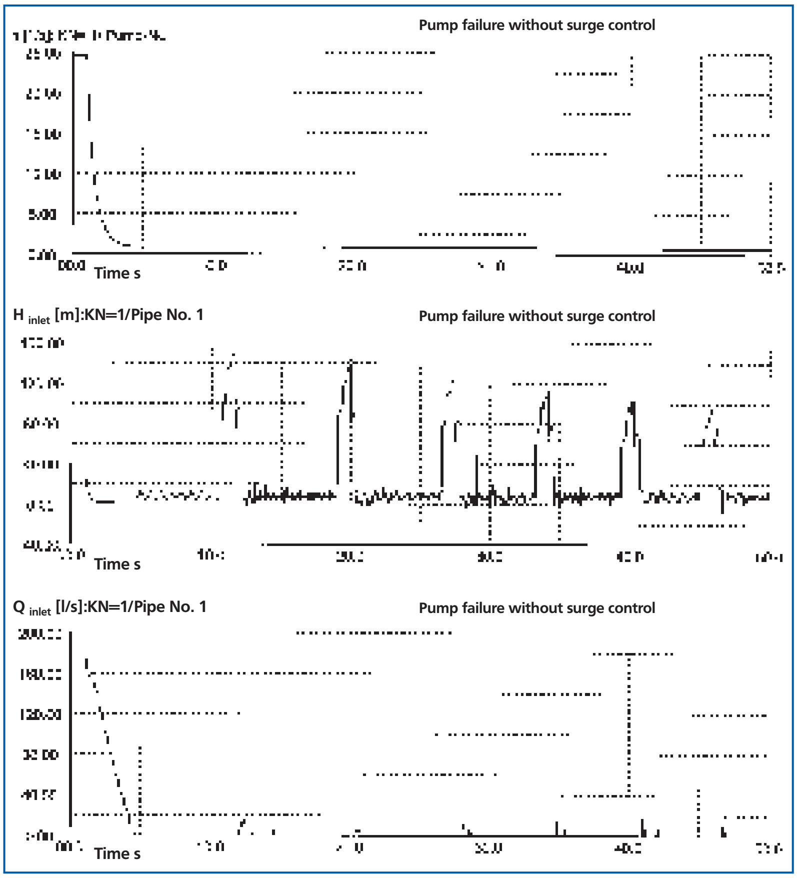


Fig. 9.2-d: Operating characteristics of the stormwater line without surge control plotted over time

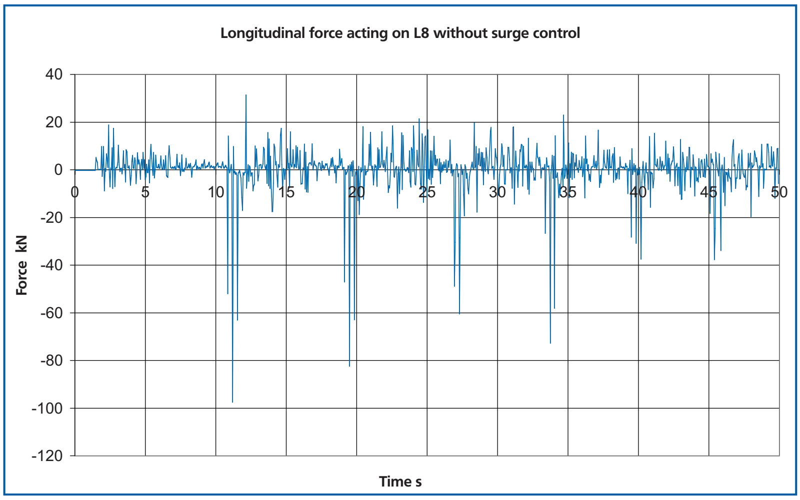


Fig. 9.2-e: Longitudinal force acting on L8 if the stormwater line is without surge control

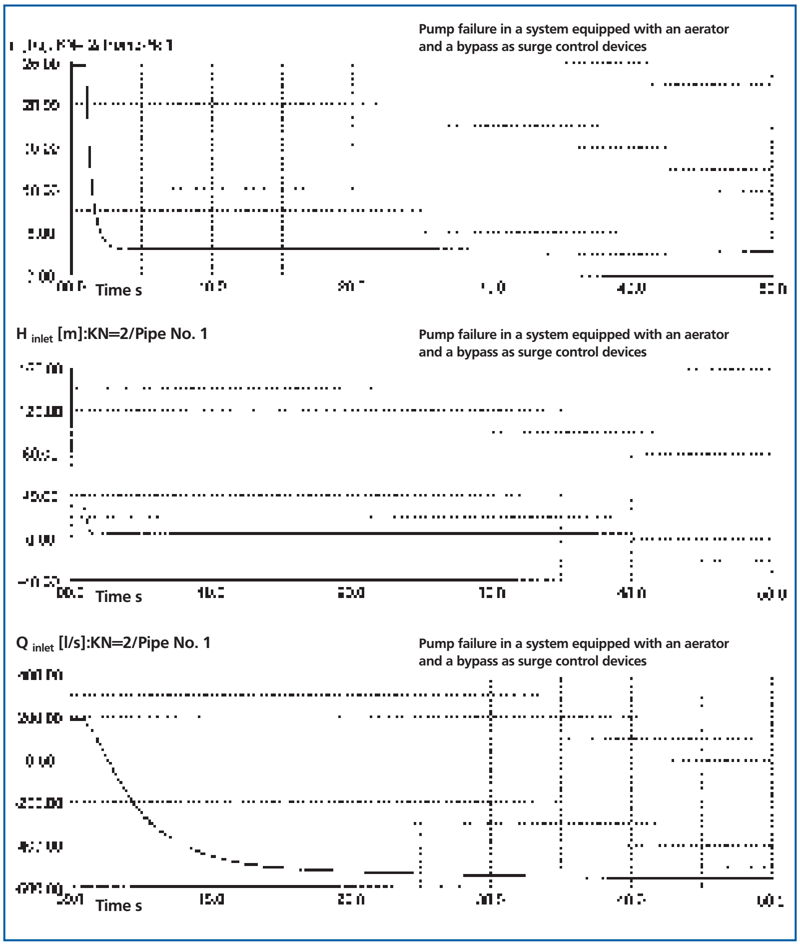


Fig. 9.2-f: Operating characteristics of the stormwater line with surge control plotted over time

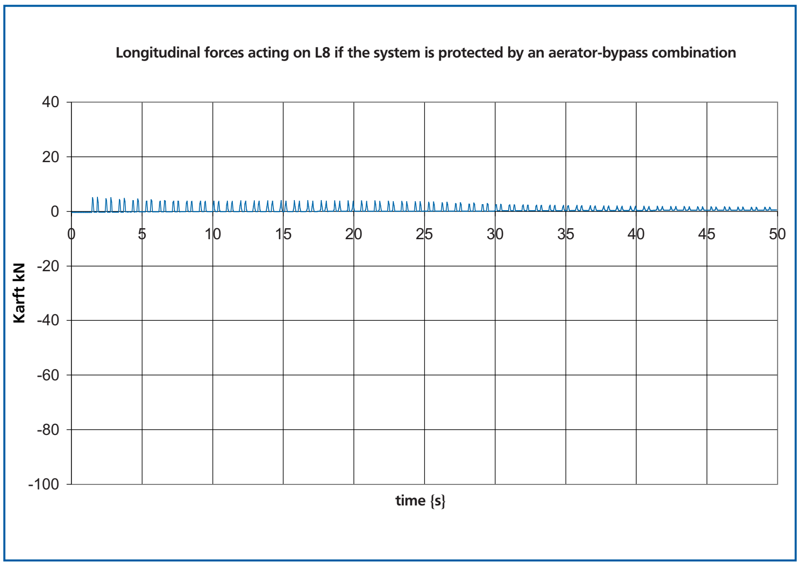


Fig. 9.2-g: Longitudinal force acting on L8 if the stormwater line is suitably protected

Surge control measures

To eliminate the macro-cavitation developing after pump failure, a second simulation calculation was run with a DN 150 aerator at the outlet of L2, the highest point of the piping. Despite the addition of a surge control device, the HD-PE pipe was still found mathematically to contain unacceptably high pressure increases a few seconds after pump failure. In order to eliminate these highly undesirable pressure peaks, it was eventually decided to add a shut-off valve with a bypass between the inlet of L1 and the pump suction tank which would be automatically opened by a maintenance-free electro-hydraulic lever and weight type actuator if all three pumps were to fail at once. To valve manufacturers today, systems like this are more or less part of their standard product range. After adding surge control devices, i.e. an aerator and a bypass fitted with an automatically opening shut-off valve, the simulation finally showed that the dynamic pressure peaks remained below the steady-state initial pressure, and that the longitudinal forces acting on the pipe bridge sections laid above ground had diminished to no more than 5% of the initial value. The calculation further revealed that the existing check valves could be dispensed with. **Fig. 9.2-f** shows – on the same scale as in **Figs. 9.2-d** and **9.2-e** to facilitate comparison – the n, H and Q curves of the surge-protected system plotted over time; **Fig. 9.2-g** shows the forces of the surge-protected system plotted over time. The global pressure envelope of the rehabilitated installation, as well as the curves of the system without surge control, are shown in **Fig. 9.2-h.**

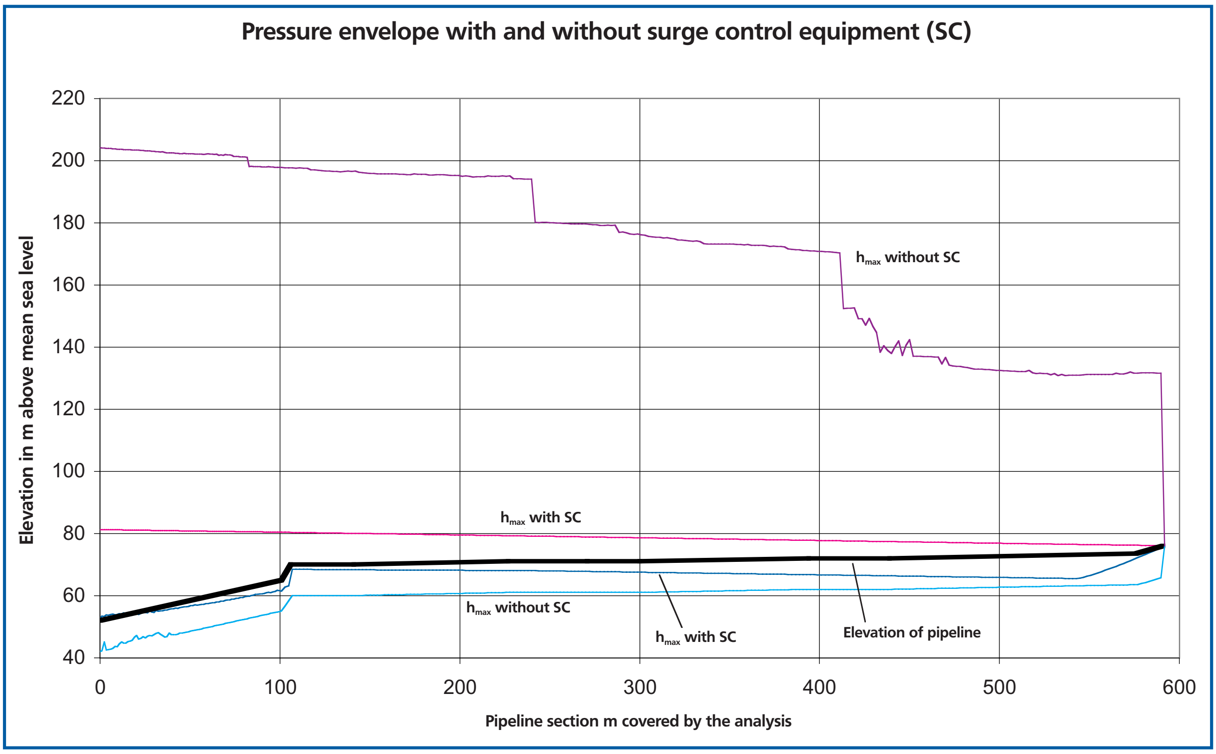


Fig. 9.2-h: Pressure envelope of the stormwater conveyance pipeline with and without surge control

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