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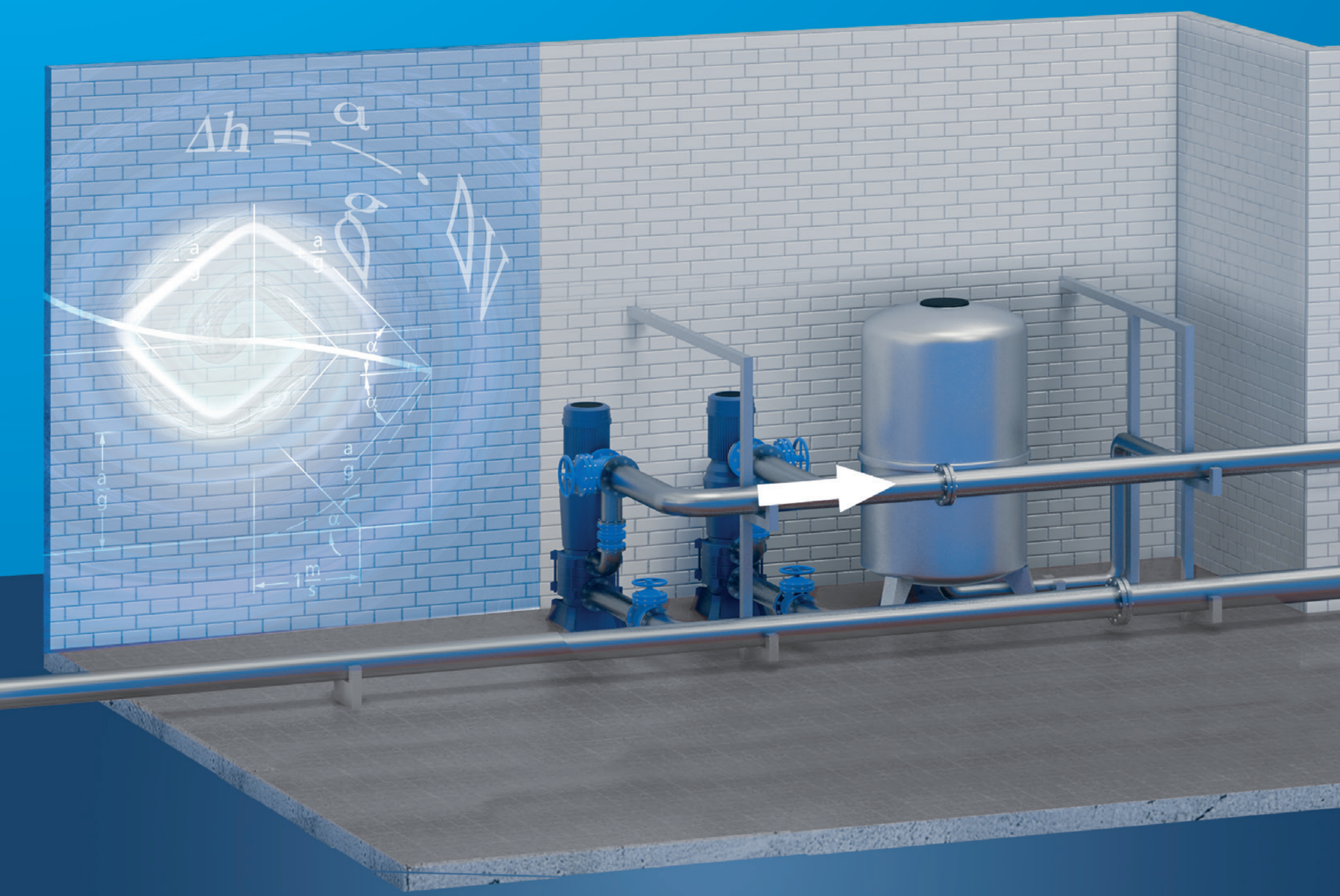
KSB Know-how: Planning Information
Surge Pressure



Dear Partners,

“Surge pressure” is a term most pumping system planners are familiar with. However, whether surge analysis is necessary in the planning phase is considered difficult to answer. In unfavourable conditions, surge pressure can cause damage in pipelines of only about one hundred metres with flow rates as low as a few decilitres per second. Sympathetic vibrations may even occur in pumping systems with very short, unsupported pipes that are not anchored correctly, although this 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. When surge pressure damage has occurred, system operators are usually reluctant to pass on information. And yet, the photos taken of some “accidents” (Figs. 1 - 3) clearly demonstrate: The damage caused by surge pressure by far exceeds the cost of preventive analysis and surge control measures.

Implementing 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 for the design and operation of 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 of surge pressure. 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 will answer in this brochure:

- How do we know whether there is a risk of surge pressure?
- How significant are approximation formulas for calculating surge pressure?
- 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 and economical is the surge control equipment available?
- How reliable is a computerised analysis?

System designer and surge analyst have to work together closely to save time and money. Surge pressure 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 line (wall thickness 12 mm)

Fig. 2: A destroyed support structure (double T section 200 mm, permanently deformed)

Fig. 3: A DN 800 check valve after surge pressure in the discharge line

1) Air vessels, sometimes also called “accumulators”, store potential energy by accumulating a quantity of pressurised hydraulic fluid in a suitable enclosed vessel.

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General Information on the Problem of Surge Pressure

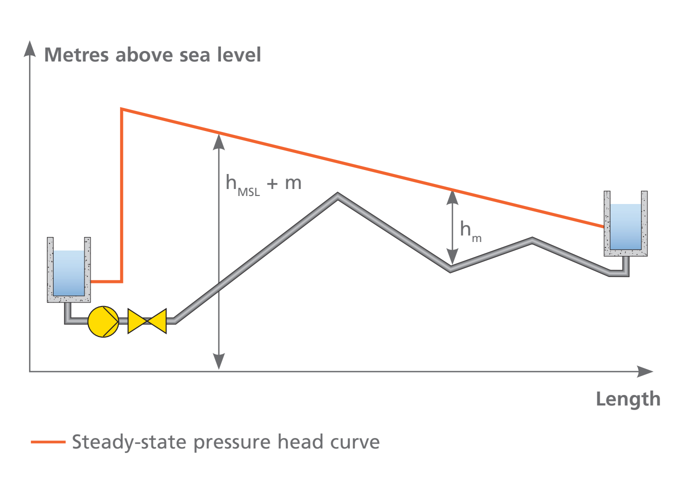
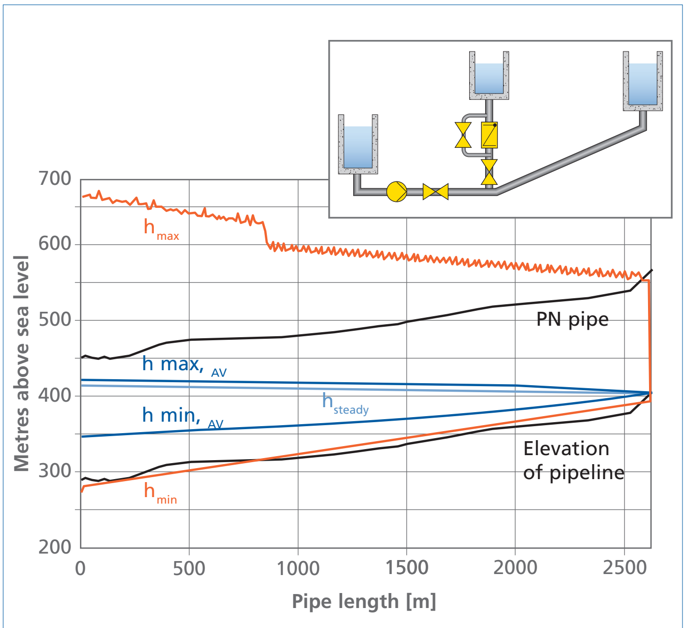


Fig. 4: Steady-state pressure head curve of a pumping system



Key:

Pipe length L:	2624 m
Inside pipe diameter D_i:	605.2 mm
Steady-state flow rate:	500 l/s
h pump sump:	287.5 m
H outlet:	400 m

Air vessel inlet pipe with bypass and non-return valve:
$V_{air} = 3.8\ m^3$, $V_{water} = 6.2\ m^3$

Fig. 5: Pressure envelope of dynamic pressure changes after the pump has cut out

Steady-state and transient flow in pipelines

When discussing the pressure of a fluid, a distinction has to be made between gauge pressure p [bar] or absolute pressure p_A [bar] and pressure head h_m. The 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, pipe centreline, pipe soffit, etc. System designers start by determining the steady-state operating pressures and volume flow rates in the piping of a pumping system. In this context, the term steady-state[2)] means that volume flow rates, pressures and pump speeds do not change with time.

Fig. 4 shows a typical steady-state 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 changes the operating conditions. Generally speaking, every change in operating condition 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:

- A pump cutting out as a result of power being switched off or a power failure
- One or more pumps starting up or stopping whilst other pumps are in operation
- Shut-off elements closing or opening in the piping
- Excitation of sympathetic vibrations by pumps with an unstable H/Q curve
- Changing inlet water levels

Fig. 5 serves as a representative example of the pressure envelope[3)] with and without an air vessel after the pump has cut out. h_{steady} in Fig. 5 is the steady-state pressure head curve. Pressure head envelopes $h_{min,AV}$ and $h_{max,AV}$ were obtained from an installation with air vessel, and h_{min} and h_{max} from an installation

2) Not to be confused with the term "static".
3) The term “pressure envelope” refers to the area defined by the minimum and maximum pressure head curves along the fixed datum line resulting from all dynamic pressures occurring within the time period monitored.

without air vessel. Whereas $h_{min,AV}$ and $h_{max,AV}$ 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 along 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
- Spalling of the pipe’s cement lining
- Dirty water being drawn into drinking water pipelines through leaking pipe sockets

We will come back to the topic of macro-cavitation at a later stage.

Surge Pressure

Pressure transients are also referred to as surge pressure, which is associated with a risk for pipes and system components.

Surge pressure causes dynamic mechanical loads on pipes, valves, pipe fixtures, supports, system components, etc.

“Surge pressure” can refer to either an increase or a decrease in pressure. In contrast to a force, pressure is non-directional. Occasionally, people speak of the “direction of pressure”, which is incorrect. 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 in the form a pipe rupture or flanges that have become loose or disconnected. The root cause of damage then tends to remain in the dark. Some representative incidents caused by surge pressure are listed in the following:

- 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
  – Spalling of the cement lining of pipes
  – Dirty water or air being drawn into pipelines through flanged or socket connections, gland packings or leaks
  – Water column separation followed by high increases in pressure when the separate liquid columns recombine (macro-cavitation)

Inertia

The sudden closure of a valve in a pipeline causes the inertia of the liquid column to exert a force on the valve's shut-off element. This increases the pressure upstream of the valve and decreases the pressure downstream of the valve.

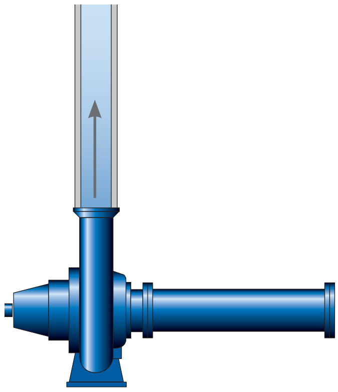
An example in figures:
DN 200 pipe, L = 900 m, v = 3 m/s.
The mass of water in the pipe equals
$m_{Water} = \frac{0.2^2 \cdot \pi}{4} \cdot 900 \cdot 1000 = 28{,}274$ kg
This is more or less the weight of a truck. v = 3 m/s equals 11 km/h. In other words, if the flow is suddenly shut off, our truck – to keep using the same image – runs into a wall (closed valve) at 11 km/h (water mass inside the pipe). In terms of our pipeline, the sequence of events taking place inside the pipe will result in high pressures as well as in high forces acting on the shut-off valve.

As a further example of inertia, Fig. 6 shows a pump discharge line. At a very small moment of inertia of pump and motor, the pump cutting out comes to a sudden standstill, which has the same effect as a gate valve closing suddenly, only this time on the downstream side of the gate valve. If inertia causes the fluid flow on the downstream side of the pump to separate into several columns, a cavity containing a mixture of water vapour and air coming out of solution will be formed. As the separate liquid columns subsequently flow backward and recombine with a hammerlike impact, high pressures develop. This phenomenon is referred to as macro-cavitation[4]).

Elasticity of fluid and pipe wall

The attempt at visualising surge pressure resulting from the inertia of a body of fluid made in the previous section 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 into harmless deformation heat[5]). Contrary to the body of a vehicle, however, water and pipe walls are elastic, even though they are so hard that this property is not noticeable in everyday use.

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. This kind of repetitive strain, or the bombarding of a sharply contoured area of the material surface, does not occur with macro-cavitation since the pressure rises are considerably lower.
5) To withstand minor damage that may be caused by other vehicles in tight car park spaces, car bodies must be elastic. To minimise the damage of a collision at high speed, however, car makers spend vast amounts of time and money to make their products as inelastic as possible.



Steady-state condition prior to pump cutting out

Formation of a vapour pocket (cavitation cavity) after a pump has cut out

Collision of separate liquid columns accompanied by high surge pressures

Fig. 6: Macro-cavitation after pump cutting out

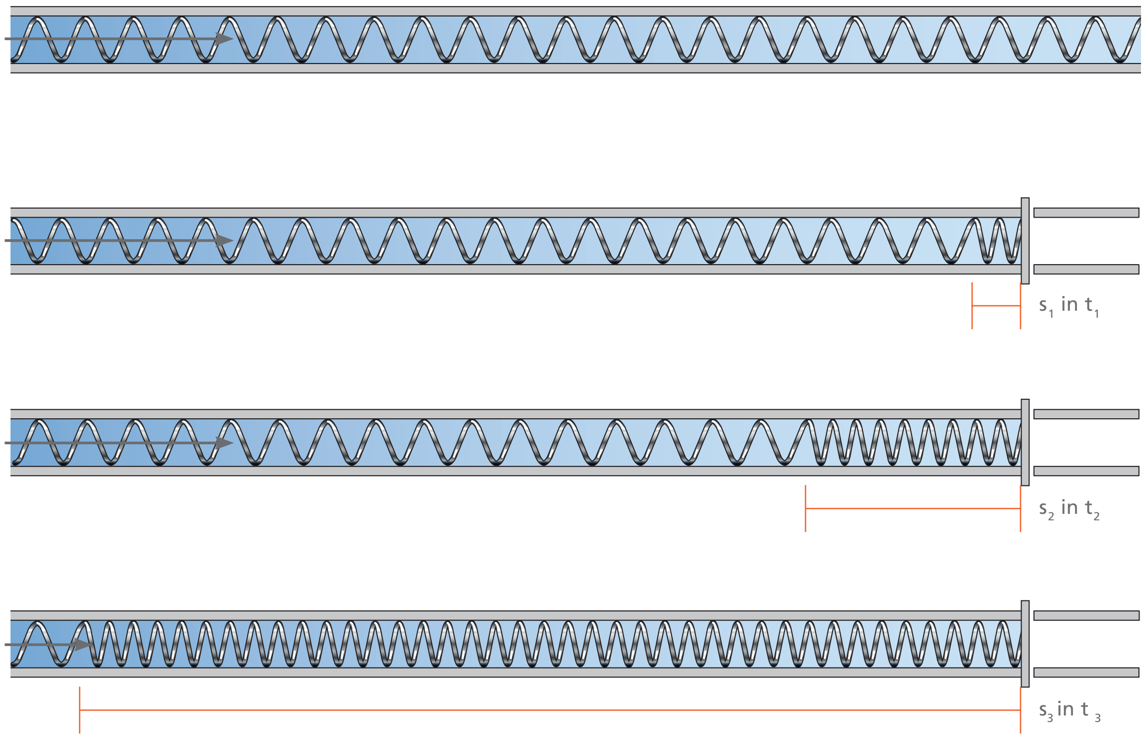


Fig. 7: Sudden closure of gate valve, visualised by a heavy steel spring

What actually goes on inside the pipe will be illustrated in the following using the example of a heavy steel spring sliding through a pipe. This spring deforms elastically when it is suddenly stopped (Fig. 7):
The deformation front runs in opposite direction to the original direction of movement at the speed typical for the steel spring, which is the wave propagation velocity a in m/s. In the compression zone, the sliding velocity of the steel spring v = 0 applies everywhere.

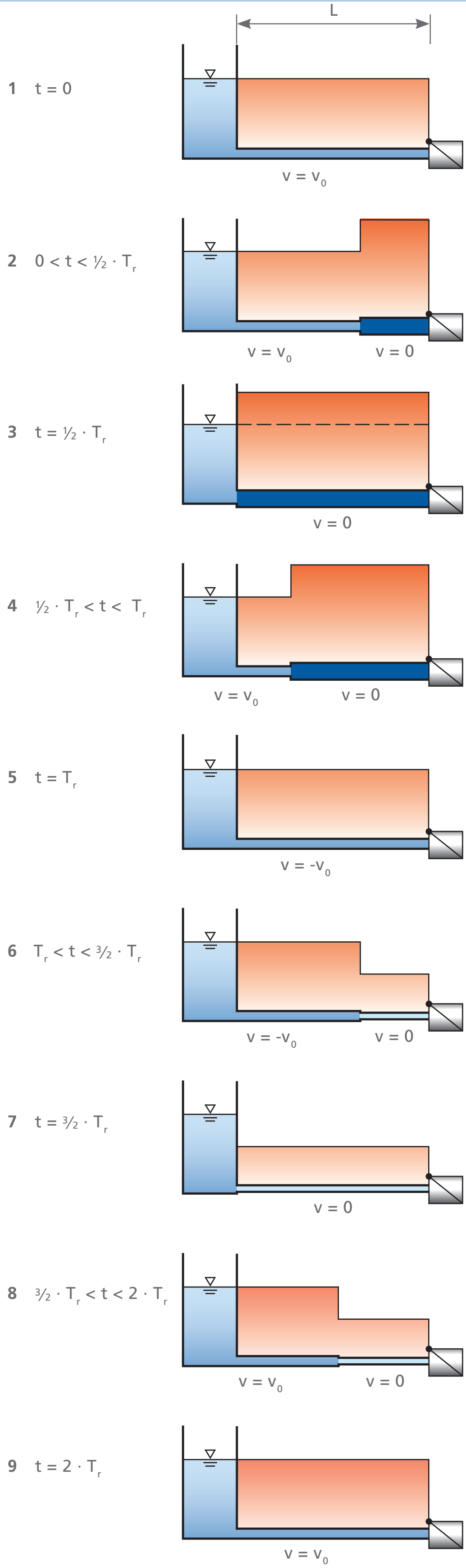
Following these simplified examples chosen to illustrate the subject, we will now return to the real situation inside the pipe, which is shown in Fig. 8, with friction being neglected. At the downstream end of a horizontal pipeline with a constant inside diameter, which is fed from a tank at constant pressure, a shut-off valve is closed abruptly:
So, one might ask, what has happened to the original steady-state kinetic energy of the fluid following the sudden closure of the gate valve? A closer look at Fig. 8 will reveal the answer. According to the law of the conservation of energy, it cannot have simply disappeared. First it is converted into elastic energy of the fluid and the pipe wall, then it changes into kinetic energy again as a result of reflection, then it becomes elastic energy again, and so forth. Let's look at Fig. 8 up to the point where $t = ½ \cdot T_r$.
The conversion into elastic energy takes place within this period of time. Immediately preceding the reflection of the wave at the tank, the velocity of the liquid column equals zero everywhere; 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 re-conversion of energy is also visible in the last condition at $t = 2 \cdot T_r$ in Fig. 8. If the gate valve were to be suddenly opened at this point, we would revert to the old steady-state condition at t = 0 without any 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 free from friction. However, 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 friction within the fluid and, last but not least, the deformation of pipe walls and fixtures is relatively small. To show the process in a less abstract manner, Fig. 9 provides the results of a computerised simulation of the example given in Fig. 8 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. 8, 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 end always equals zero.
- At the open end of a pipe with a constant total head (e.g. tank 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.

Surge pressures travel quickly, at about a = 1000 m/s in ductile or steel pipes (see: „Scope of the Joukowsky equation", page 12). They dampen out only gradually and, therefore, remain dangerous for a long time. The dampening time depends on the length of the pipelines. In urban water supply systems, surge pressures only last several seconds. In long pipelines, it can take a few minutes until the surge pressure has dampened out. From these facts, the basic working principle of all surge control equipment, such as air vessels/accumulators, flywheels, standpipes 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 energy. Without an air vessel, the dangerous conversion of kinetic energy into elastic deformation energy following cutting out of the pump would take place at the pump outlet, which could cause the liquid column to separate (Fig. 6). However, this does not happen, because the energy stored in the air cushion in the vessel takes over the work of the pump. Immediately after a pump has cut out, the air cushion starts expanding and takes over the pump's job of discharging water into the pipeline. Provided that the vessel is properly selected, it will prevent rapid changes in the flow velocity in the pipeline. Instead, the water level in the vessel and the undeformed 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 particular in longer pipelines.



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 sudden pressure head rise Δh; the related pipe wall expansion is roughly indicated in the graphic. The pressure wave generated runs in opposite direction to the steady-state direction of the flow at the wave propagation velocity 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. Its value equals $\frac{2L}{a}$.

3 At $t = \frac{1}{2} \cdot T_r$ the pressure wave has arrived at the tank. As the tank pressure p = constant, the pressure rise cannot continue. 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 tank.

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

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

Fig. 8: Pressure and velocity wave development in a single-conduit, frictionless pipeline following its sudden shut-off. Medium shading indicates zones of steady-state pressure, dark shading zones of pressure increase and light shading zones of pressure reduction. The expansion and contraction of the pipeline as a result of rising and falling pressure levels, respectively, are roughly illustrated. To illustrate the extent: If the pressure rises by 100 bar, the volume of water will decrease by about 0.5 %.

Surge pressure occurs when the kinetic energy of a fluid is converted into elastic energy. Only rapid[6] changes of the flow velocity will produce this effect, for example the rapid closure of a gate valve or the sudden cutting out of a pump. Due to the inertia of the fluid, the flow velocity of the liquid column as a whole can no longer adjust 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 almost undiminished speed of sound (roughly 1000 m/s for many pipe materials), reaching every part of the piping system with its damaging impact.

Resonance

Sympathetic vibrations are a special case. These occur when excitation frequencies of whatever origin happen to coincide with a natural frequency of the pipeline. Excitation frequencies can be generated by the pump drive or by flow separation phenomena in valves and pipe bends, for examples.

Improperly anchored unsupported pipeline sections in pump installations are particularly prone to sympathetic vibrations transmitted by the fluid pumped and by the piping structure. By contrast, resonance is all but negligible for buried pipelines. In order to select suitable anchoring for pump installations, an approximate structural dynamics analysis should always be conducted, with the pump speed serving as the excitation frequency.

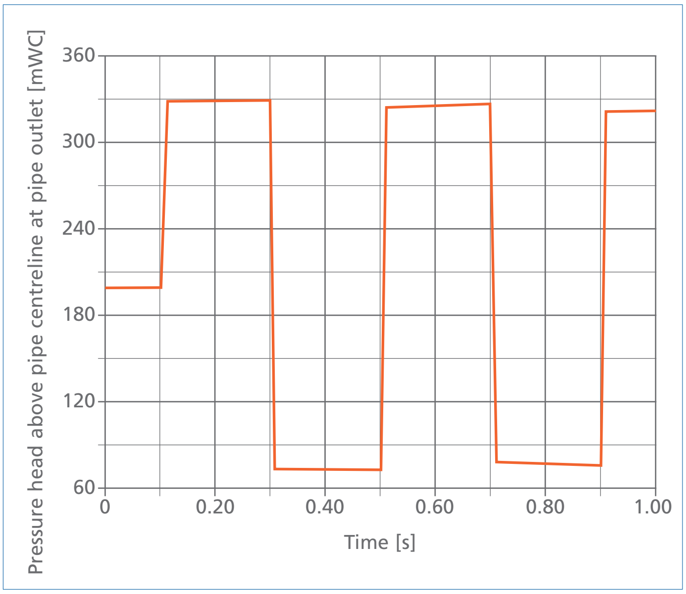


Fig. 9: Pressure head upstream of the gate valve. Compared with the situation shown in Fig. 8, minute differences are noticeable. For example, the pressure flanks are not perfectly perpendicular, because of the finite closing time of D_t = 0.01 s. As a result of friction, the pressure planes are not perfectly horizontal – this phenomenon will be discussed in greater detail in the section "Scope of the Joukowsky equation" on page 12.

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

The Joukowsky Surge

The pressure change Δp_{Jou} in a liquid caused by a change in flow velocity Δv is calculated as follows:

$$\Delta p_{Jou} = \rho \cdot a \cdot \Delta v$$ Formula (1)

Key:
Δp_{Jou} = Joukowsky surge = pressure change in a liquid [N/m²]
Δv = velocity change
ρ = density of the liquid [kg/m³]
a = wave propagation velocity in liquid-filled pipe [m/s]

Formula (1) contains Δv as well as the density ρ and the wave propagation velocity a. The relationship only applies to the period of time in which the velocity change Δv takes place.
If Δv runs in opposite direction to the flow, the pressure will rise, otherwise it will fall.

If the fluid handled is water[7]), i.e. ρ = 1000 kg/m3, formula (1) becomes:

$$\Delta h_{Jou} = \frac{a}{g} \cdot \Delta v \approx 100 \cdot \Delta v$$ Formula (2)

Key:
Δh_{Jou} = change in pressure head in m
g = acceleration due to gravity = 9.81 m/s²

In 1897, Joukowsky conducted extensive experiments on Moscow drinking water supply pipelines of the following lengths / diameters: 7620 m / 50 mm, 305 m / 101.5 mm and 305 m / 152.5 mm. He published the results of his various experiments and theoretical studies in 1898.
It may seem inconsistent that the Joukowsky surge Δp_{Jou} in formula (1) seems to have nothing to do with the mass of the fluid inside the pipeline. For example, if the surge pressure described in the “Inertia” section on page 06 of this brochure had been based on a pipe diameter twice that of the diameter used, $A = \frac{D^2 \cdot \pi}{4}$ 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 element, 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 surge pressure, although that does not seem necessary after a superficial glance at Joukwsky'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.

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 surge pressure. Calculate the force exerted on the gate valve disc.

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 gate valve is closed, Δv = 2 m/s. This gives us a pressure increase
$\Delta h = 100 \cdot 2 = 200$ m or approximately $\Delta p = 20 \cdot 105$ N/m², which is 20 bar. The gate valve cross-section measures
$A = D^2 \cdot 0.25 \cdot \pi \approx 0.2$ m². The force acting on the gate valve is
$p \cdot A = 0.2 \cdot 20 \cdot 105 = 4 \cdot 105$ N = 400 kN.

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 moment of inertia of pump and motor is negligible.
Is there a risk of liquid column separation, i.e. macro-cavitation, when the pump cuts out? If so, what is the anticipated pressure increase?

Answer:
Q = 300 l/s in a DN 400 pipeline roughly corresponds to v = 2.4 m/s. As a result of the pump cutting out and the loss of the moment of inertia, the pump comes to a sudden standstill, i.e. Δv = 2.4 m/s. According to the Joukowsky equation, this causes a pressure head drop of $\Delta h = -100 \cdot 2.4$ m = -240 m. Since the steady-state head is just 40 m, a vacuum is created, the liquid column separates and macro-cavitation sets in.

7) Despite the high flow velocities common in gas pipes, these do not experience surge pressure problems, because the product of $\rho \cdot a$ is several thousand times smaller for gas than it is for water.

Following 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 $D_h = 100 \cdot 2.4 = 240$ m, which is equivalent to 24 bar.

Example 3:

A pump transports water at a flow rate Q = 300 l/s and a head Δh = 40 m through a 2000 m long DN 400 pipeline, a = 1000 m/s. The moment of inertia[8)] of all rotating components (pump, motor, etc.) J = 20 kgm², the speed of rotation $n_0 = 24\ s^{-1}$ and the total efficiency h = 0.9, i.e. 90 %.

Is there a risk of liquid column separation, i.e. macro-cavitation, when the pump cuts out?

Answer:

For the instant of the pump cutting out, the change in speed $\dot{n}$ can be calculated with the inertia equation $M_\upsilon = I_p \frac{2\pi}{60}\frac{dn}{dt}$. Assuming as a (very rough) approximation a linear speed reduction $\dot{n} = \frac{n_0}{\Delta t}$, then, if $M_p = \frac{\Delta p \cdot Q}{2 \cdot \pi \cdot n_0 \cdot \eta}$ we obtain a time Dt in which the speed has dropped to zero. Further,

$$\Delta p = 1.000 \cdot 9.81 \cdot \Delta h, \text{ zu}$$

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

The reflection time of the pipeline $T_r = 3.4$ 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. Macro-cavitation has to be expected in this case.

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 applications

Reflection time T_r:

In Fig. 8 the pressure relief wave reflected by the tank arrives at the shut-off element after T_r has lapsed and compensates 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. 10 shows the pressure envelope which applies to a case of this kind.

Friction:

If the liquid pumped is highly viscous or if the pipeline is very long (say, 10 km and more), the work done by the pump only serves to overcome the friction of 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 pipe friction is commonly called line packing.

The following flow simulation calculation gives an example of this:

The gate valve in the example shown in Fig. 11 closes 20 s after the start of the calculation. The first steep increase by approx. 20 bar to approx. 55 bar is Δp_{Jou} as calculated by 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 fluids. It is unlikely to occur in urban water supply and waste water disposal systems.

8) Moment of inertia J: J expressed in kgm² is the correct physical quantity. The moment of gyration GD², which was used in the past, should no longer be used, because it can easily be confused with J.

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 surge pressure events. It is calculated using formula (3).

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

Formula (3)

Key:
a = wave propagation velocity [m/s]
ρ = density of the fluid [kg/m³]
E_F = modulus of elasticity of the fluid [N/m²]
d_i = inside pipe diameter [mm]
μ = transverse contraction number
E_P = modulus of elasticity of the pipe wall [N/m²]
s = pipe wall thickness [mm]

Formula (3) produces a range of values from approximately 1400 m/s for steel pipes to around 300 m/s for ductile plastic pipes. The wave propagation velocity “a” in an unconfined body of water is approximately 1440 m/s. To all intents and purposes, the validity of formula (3) should not be overestimated; surge analyses are often performed without it, in which case the value of "a" is estimated.

The volume of gas contained in the fluid, which is not taken into account by formula (3), can have a strong impact on “a”, as is shown by some examples in Table 1:

Gas content [volume %]	a [m/s]
0	1250
0.2	450
0.4	300
0.8	250
1	240

Table 1: “a” as a function of gas content at a static water pressure of approximately 3 bar

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, ageing 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.

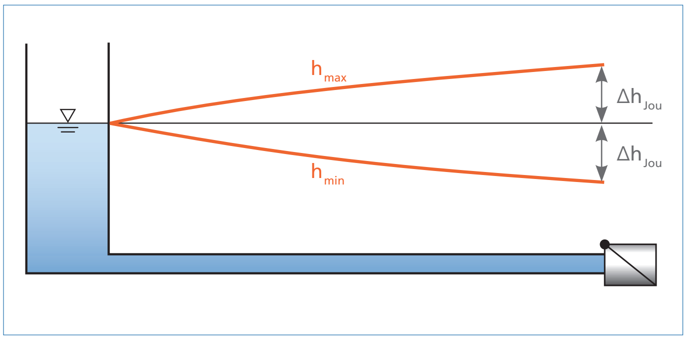


Fig. 10: Pressure head envelope for closing times > reflection time T_r

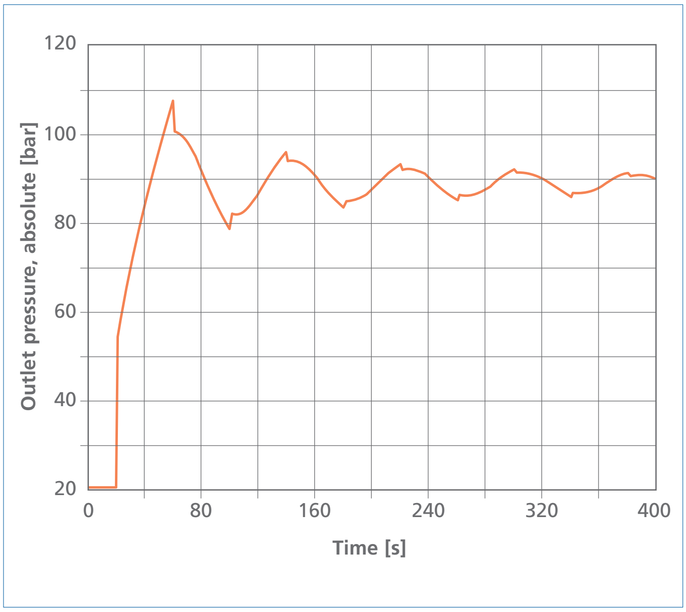


Fig. 11: Pressure curve at the outlet of a 20 km long crude oil pipeline following the sudden shut-off of a gate valve.
Calculation parameters:
DN 300, k = 0.02 mm, inlet pressure 88 bar constant,
Q = 250 l/s, fluid handled: crude oil, ρ = 900 kg/m³

Numerical Simulation of Surge Pressure

In current theory, the dependent model variables are the pressure p and the flow velocity v in the two partial differential equations of formula (4) for every single pipe of a piping system with time t and pipe section x.

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

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

Formula (4)

The equations of formula (4) are generally valid and cover the effects of both inertia and elasticity. Mathematically, the pipe ends serve as the boundary conditions of the equations of formula (4); different types of boundary conditions considered by the model can be internal components such as pipe nodes, tanks, pumps and valves. 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 the equation of formula (4) is the steady-state flow inside the pipe concerned before the onset of the disturbance. The equations of formula (4) 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 the equations of formula (4) is less appropriate for computing sympathetic vibrations. These can be calculated much more precisely using the impedance method, or, in other words, by looking at the frequency range.

Accuracy of simulation calculations

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 measurements performed. 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 formula (4), i.e. the assumption of a simple vacuum cavity following liquid column separation, are always above real-life values. However, you will be on the safe side with these conservative calculations. The actual energy losses due to friction and the degree of warping of pipeline and pipe fixtures are somewhat larger than predicted by the simulation. The first pressure peaks and valleys tend to be simulated very accurately, whereas the pressures further down the line are less accurate as the degree of dampening factored in tends to be too low. However, imperfections of this kind are negligible compared to inaccuracies caused by entering wrong or insufficient input data. Possible data errors are:

- Inaccurate valve characteristics and pump characteristics
- Lack of knowledge about the actual wave propagation velocity inside the pipeline
- Lack of information about tapping points in a main pipe
- Unknown 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. In practice, it is often impossible to obtain exact data. If this is the case, the required inputs have to be estimated.

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 preceding total shut-off 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 a somewhat conservative selection of surge control equipment.

A surge analysis can only be as accurate as the system data entered. 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.

Forces acting on pipelines as a result of surge pressure

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 is not taken into account (this would be a separate calculation). Apart from the odd exception which is of no relevance in the field of water supply and waste water disposal 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.

Practical Computerised Surge Analyses

Technical procedure

A surge analysis will not provide direct solutions for the required parameters, such as optimum air vessel sizes, 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. Given the necessary work performed by surge specialists, surge analyses 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 it would make sense to do the analyses themselves. As reliable[9)] surge analysis software is far from a mass product, the low sales volume makes it expensive. This is further added to by the cost of training and hands-on practice. And 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, doing their own analyses will probably not be worthwhile.

Cooperation between customer and analysing specialists

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

- 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 surge control devices, such as air valves, 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.
- A rough list of all main parameters, i.e. principal pipe lengths, diameters and flow rates
- A list of all major operation and downtime periods
- A record of all known incidents that could have been caused by surge pressure
- Any 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
- Diameter
- Wall thickness
- Construction materials, 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 patterns
- Characteristic curves or performance charts and characteristic data of all hydraulic equipment
- Moment of inertia of the machine train systems
- Characteristic curves and data of surge control equipment already installed in the system
- Air valve characteristics
- 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 universities.
Some of these software programs were later bought by commercial enterprises and provided with an advanced user-friendly interface.

Use of Rules of Thumb and Manual Calculations

A rough estimate can be very useful to quickly assess the risk of surge pressure. 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 (scaleability). To answer that question, we should start by pointing out that there is a great variety of water supply and waste water disposal systems, 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. they have got the same flow rates and the 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 negative 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 everyday engineering purposes, approximation formulas should be used exclusively to roughly estimate the potential risk in a system (examples 1 - 3 in the "Joukowsky Surge" section, page 11 ff). Using them as a basis for a serious surge analysis or, even worse, for selecting surge control equipment, would have to be regarded as highly irresponsible. A brief description of all known processes of approximation and estimation formulas follows below:

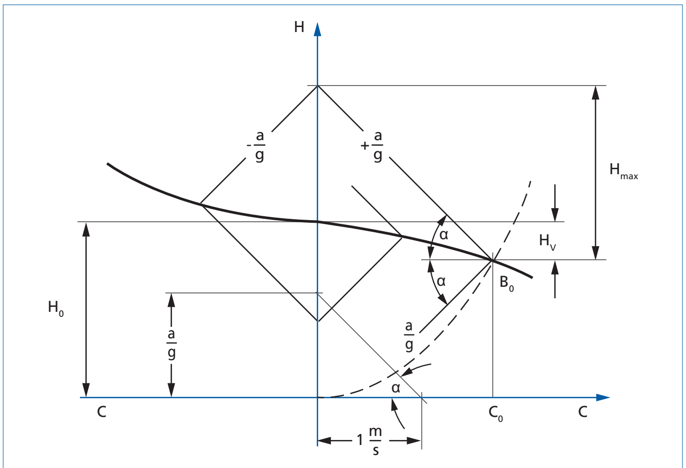


Fig. 12: Graphical method developed by Schnyder-Bergeron

Schnyder-Bergeron method

Before the days of modern computer software, the graphical Schnyder-Bergeron method was often employed; it produced relatively reliable surge analyses. For practical reasons, 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. 12 is an example of a typical Schnyder-Bergeron diagram, which shows how the pressure wave propagation due to the shut-off of a valve is determined by graphical means.

Application of the Joukowsky equation

for rapid changes in flow velocity v (examples 1 - 3 in the "Joukowsky Surge" section, page 11 ff).

Graphical method

- for determining the size of air vessels*)
- to estimate the line packing*)

Closing pattern

The largely ideal valve closing pattern for the exceptional case of a single-conduit pipeline can be calculated by approximation.*)

*)Expertise required

These are the only manual calculation methods. This seeming shortcoming is more easily understood if we take another look at the air vessel as a representative example. 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 vessel pre-precharging is "hard" or "soft". The pre-charge pressure 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 after a pump has cut out (area of negative 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 pressure rise in the air vessel. It is impossible to determine these crucial variables using rules of thumb or a graphical design method.

Main Surge Control Systems

The purpose of surge control is to stop kinetic energy from being converted into deformation energy. The following basic methods can be used:

- Energy storage
- One-way surge devices and air valves
- Optimisation of valve closing patterns[10])
- Optimisation of the strategy employed to control the piping system

Energy storage

With air vessels and standpipes, energy is stored as pressure energy; when a flywheel is installed, the energy is stored as rotational energy. The amount of energy stored is sufficient to maintain the steady-state flow for a relatively long time and to make sure its flow velocity dissipates only slowly. A rapid pressure drop is thus prevented. If air vessels and standpipes are installed upstream of a pump in a long inlet pipe, they can prevent surge pressure not only by means of releasing energy but also by absorbing energy.

Air vessels

Air vessels come in the form of compressor vessels (Fig. 13), accumulators (Fig. 14) 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. In the case of compressor vessels, sensors make sure the vessels are always filled to the correct levels, switching the air compressor on or off as required. Accumulators are typically adjusted by pre-charging the gas inside their bladder 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 a pump cutting out. They can also be installed in other suitable places in a piping system. For example in long inlet pipes, an additional air vessel can be fitted at the inlet of the pump as a surge control device. If the pump cuts out, the upstream vessel will absorb energy, while the downstream vessel will release energy.

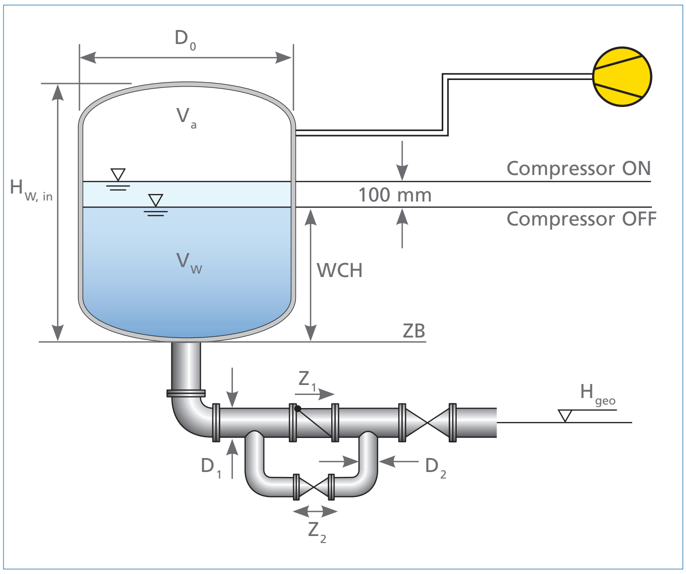


Fig. 13: Schematic of a compressor-type air vessel
To avoid excessive pressures on return of the vessel water, the connecting pipe may have to be fitted with a swing check valve with a throttled bypass.

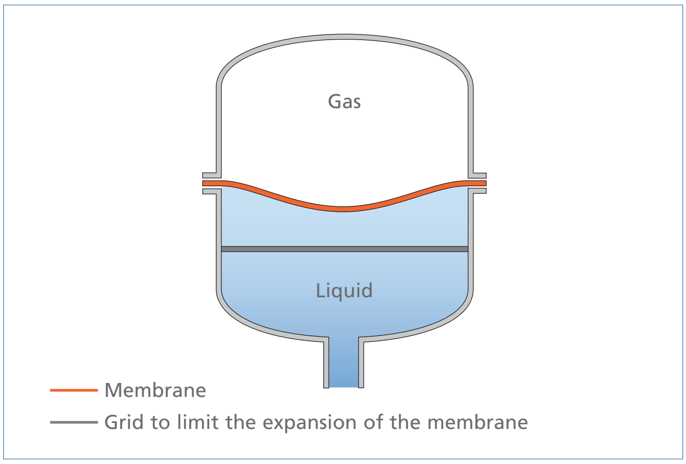


Fig. 14: Schematic of an accumulator

Video : Surge pressure problem in waterworks solved by KSB expert.
Click to play or scan QR code

10) The valve closing pattern describes the closing angle of a valve as a function of time.

Fig. 15: Accumulators

For waste water pumping stations the use of air vessels is somewhat limited. The reasons are:

- With waste water, it is not possible to measure the water level as needed to set the compressor.
- The bladder in an accumulator would be punctured by sharp objects contained in the waste water, such as razor blades, nails, etc.
- Risk of incrustations, deposits and clogging

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

- The water level in the vessel must be monitored.
- 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; their open position has to be monitored.
- Maintenance of the compressor (of compressor-type vessels)

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 flow and a filling mechanism (one-way surge tank), it is used to stop the pressure falling below atmospheric at the high points of long clean-water pipelines. Because of potential odour nuisance, standpipes are rarely found in waste water installations.

Standpipes and one-way surge tanks are highly reliable pieces of equipment provided the following points are observed:

- Continuously or regularly replacing the water (for hygiene reasons)
- 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 arrangements

Flywheels

Mounted on the drive, a flywheel prolongs the run-down time of a pump to standstill by means of the stored rotational energy E_{kin}:

$$E_{kin} = \frac{1}{2} \cdot J \cdot \varpi^2 \quad \text{Formula (5)}$$

Key:
J = moment of inertia of the flywheel [kgm^2]
ϖ = angular velocity [s^{-1}]

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

$$J = \frac{m \cdot r^2}{2} \quad \text{Formula (6)}$$

Figs. 16 and 17 show some practical applications. However, flywheels that are economically and technically feasible can only achieve a suitable prolongation of the running down time for a relatively short pipeline, or, put differently, for a short reflection time T_r. The limits for employing a flywheel are in the region of pipeline lengths of 1 to 2 km. Example 3 in section 4 includes a rough estimate performed to check whether a flywheel can be used. For design reasons, 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 motor.

Flywheels are probably the safest and most elegant type 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.

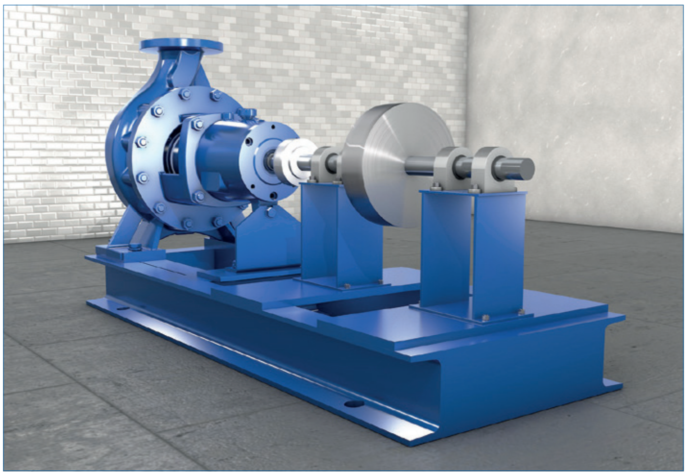


Fig. 16: The V-belt pulleys in this arrangement are solid discs.

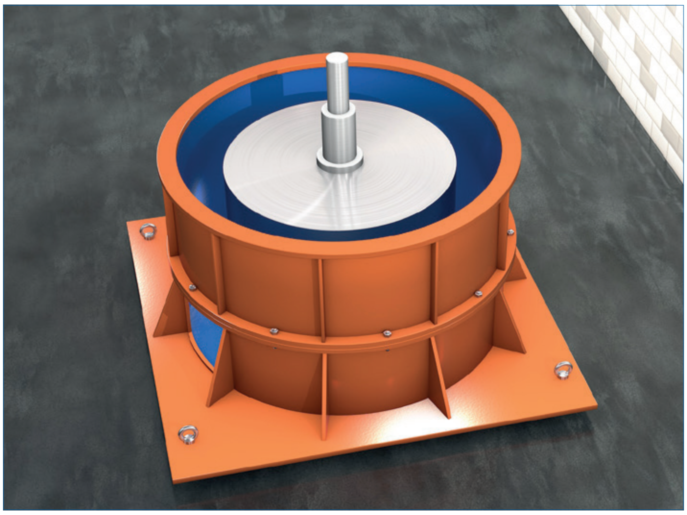


Fig. 17: Vertically mounted flywheel (driven via a universal-joint shaft, D = 790 mm)

Fig. 18: Duojet fluid-operated single-chamber air valve. Large orifice cross-section for drawing in and discharging large amounts of air during start-up and shutdown of pumping systems. Small orifice cross-section for releasing small amounts of air during operation from a fully pressurised system. (With kind permission of VAG-Armaturen GmbH)

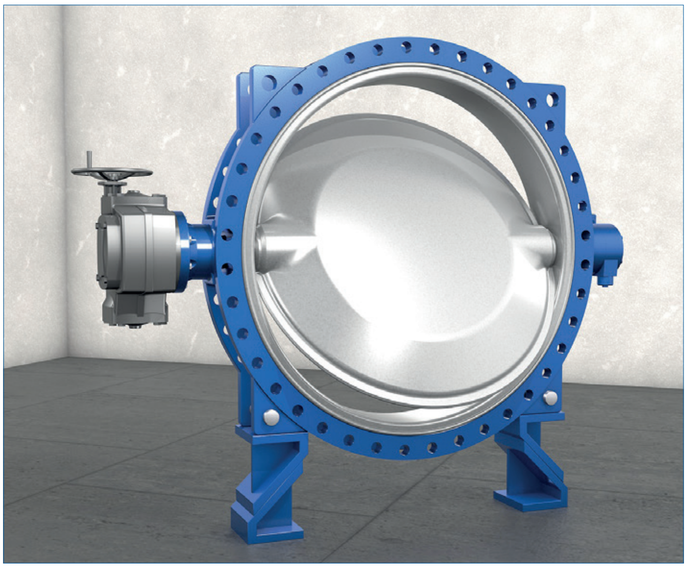


Fig. 19: Motorised Mammouth butterfly valve by KSB

Air valves

Air valves should only be used if every other solution has been ruled out. Their drawbacks are:

- They require regular maintenance.
- If mounted in the wrong place or incorrectly, they can aggravate surge pressure rather than prevent it.
- Under certain circumstances, operation of the plant may be limited, because the air drawn into the system has to be removed again.
- Special designs are required for waste water systems.

Air valves (Fig. 18) have to be selected carefully. On large-diameter pipelines, air valves have to be arranged 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 impact. Air cushions normally have a dampening effect. In addition, the air drawn into the pipeline can also give rise to dangerous dynamic pressure increases. The air has to be pressed out of the piping slowly; a large air outlet cross-section would lead to abrupt pressure variations towards the end of the air discharge operation. For this reason, air valves 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. The reliability of air valves depends on their design. They are the least reliable types of surge control equipment. Their correct function has to be tested at regular intervals and the incoming air may need to be filtered.

Actuated valves

Suitable valve opening and closing patterns 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 actuating times and closing pattern.

Swing check valves

The dynamics of swing check valves often have a major influence on the development of surge pressure, because closing of the valve after flow has reversed generates velocity changes which, according to Joukowsky's equation, formula (1), produce pressure variations.

With a few exceptions from the rule, check valves generally have to meet the following two contradictory requirements:

- Bring the reverse flow to a standstill as quickly as possible,
- Keep the surge pressure 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 slams shut with considerable impact. This dreaded phenomenon is known by the term "check valve slam". Since the closing time is the main criterion for check valve slam, limit position dampeners will improve the situation but they will 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 dampeners 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 a pump has cut out. This feature is important on pumps operated in parallel, when one pump cuts out whilst the remaining pumps continue to run and deliver flow against the pump that has cut out. In a case like this, controlled closing is achieved by adjustable hydraulic actuators without external power supply but with a lever and weight, 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 pattern. The operating reliability of swing check valves is relatively high. In operation, they have to be checked for proper functioning at regular intervals.

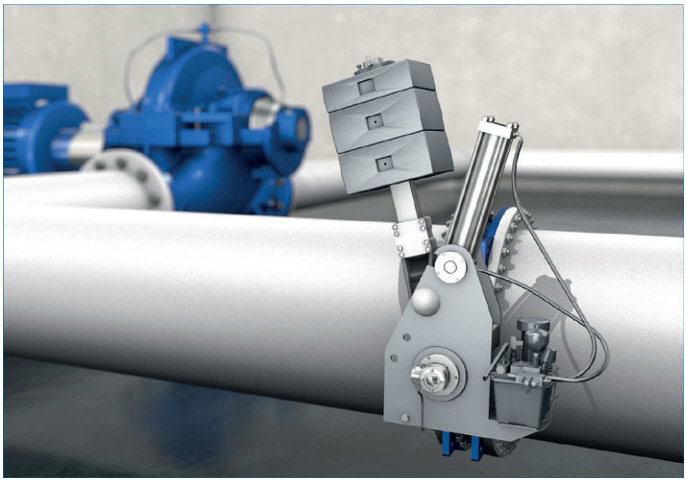


Fig. 20: KSB's DUALIS combined butterfly/check valve with hydraulic actuator

Case Studies

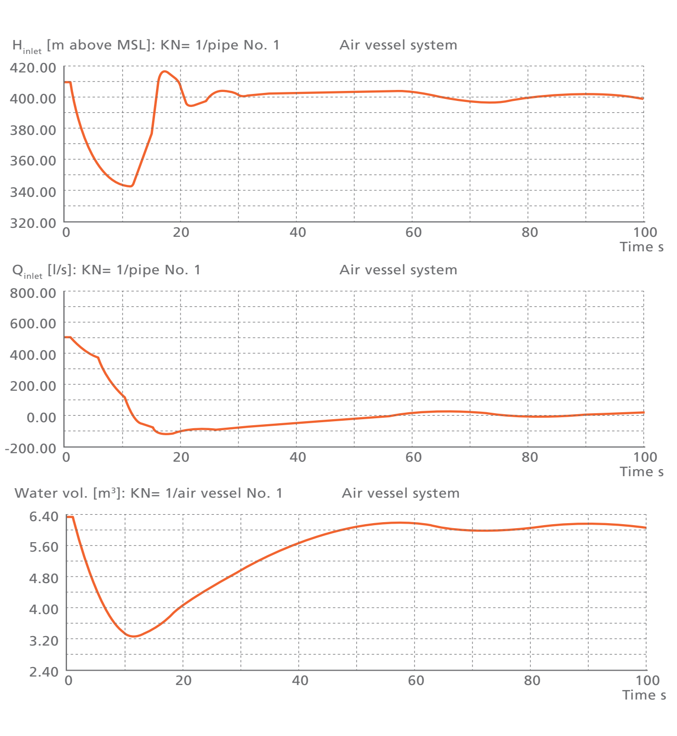


Fig. 21: Time plots for the long-distance water supply pipeline (Fig. 5); the example shows the pressure head and flow curves of an air vessel protected system as functions of time (pressure heads expressed in m above mean sea level)

Example of a long-distance water supply pipeline

The system parameters are indicated in Fig. 4. A steady-state flow Q_{steady} = 500 l/s is pumped through a DN 600 pipeline of ductile cast material and a total length L = 2624 m by three centrifugal pumps operating in parallel at a pump head Δh_{steady} = 122.5 m into an elevated water reservoir.

The operating incident under investigation, which leads to excessive dynamic pressures, is that of all three pumps cutting out simultaneously. The dynamic pressure peaks produced by far exceed the permissible nominal pressure of PN 16 (see h_{max} curve) in Fig. 5, and the minimum pressures drop to vapour pressure in long sections of the system (see h_{min} curve) in Fig. 5.

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. 5 will initially prevent the development of areas of negative 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. 13. 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. 5 shows the calculated pressure envelope with and without air vessel. The maximum head curve obtained with an air vessel $h_{max,AV}$ is now only slightly above the steady-state head curve h_{steady} and the associated minimum head curve $h_{min,AV}$ runs at a wide safety margin above the pipe soffit.

Fig. 21 shows the pressure head and flow curves of the system protected by an air vessel arrangement plotted against time (pressure heads expressed in m above mean sea level).

Example of a pumped stormwater pipeline

Starting from a waste water pumping station, a new DN 350 stormwater pipeline with a total length L = 590 m was laid to an aeration structure. System operation was by means of three identical pumps running in parallel, each equipped with a check valve and a motorised gate valve to control pump start-up and run-down. The first 100 m of pipe made of high-density polyethylene were laid underground, the remaining 490 m were made of steel and laid above ground on pipe bridges. Fig. 22 shows a schematic of the model installation. The nodes connecting the single pipes of the model, which are laid above ground, are 90° elbows. The plant engineering consultants in charge of planning the plant neither ordered a surge analysis to accompany the project planning phase, nor did they perform one themselves. During the first test runs following the plant's completion, several incidents were simulated, among them a power failure causing all three pumps to cut out at the same time. This made the piping section laid above ground shake considerably, damaging and partly tearing off pipe fixtures.

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 suitable protective measures or surge control equipment that would prevent impermissible dynamic pressures produced by pumps cutting out, and to prove their effectiveness mathematically.

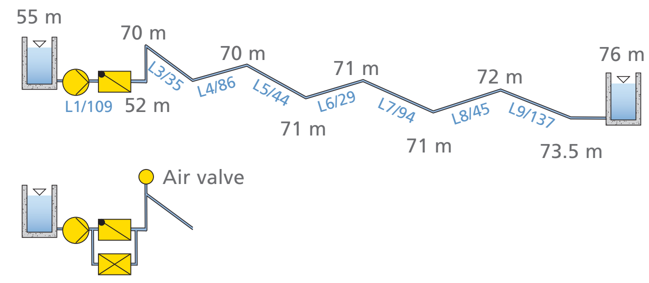


Fig. 22: Schematic of the stormwater pipeline

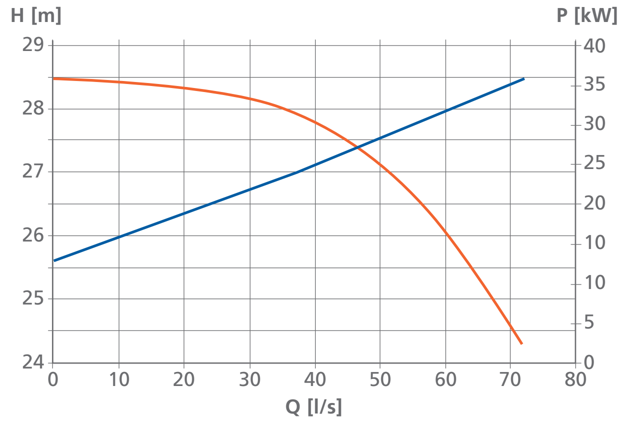


Fig. 23: Characteristic curve of a pump in the stormwater pipeline example

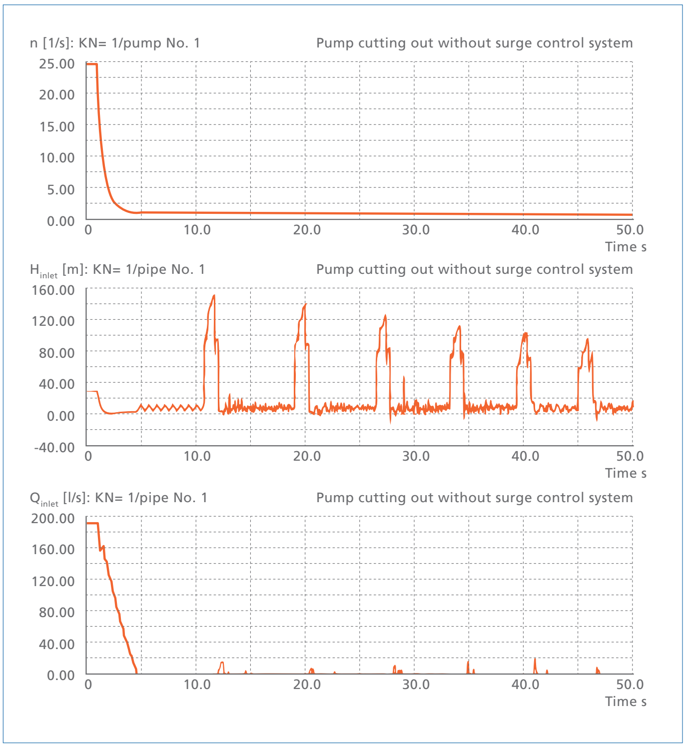


Fig. 24: Operating characteristics of the stormwater pipeline without surge control, plotted over time

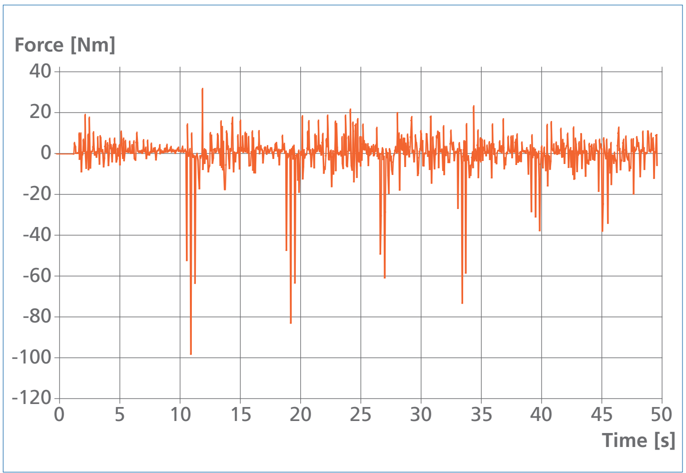


Fig. 25: Longitudinal force acting on L8 if the stormwater pipeline is not protected by surge control

Model data

Besides the parameters indicated in Fig. 23, the following system data were entered into the calculation:

- Pump characteristic curve shown in Fig. 23
- Model pipeline L1:
  Material: HDPE
  D_{inside}: 354.6 mm
  k: 0.1 mm
  a: 600 m/s (estimated value)
  Min. permissible pressure: vacuum
  Pressure class: PN 6
- Model pipeline L2 to L10:
  Material: steel
  D_{inside}: 349.2 mm
  k: 0.1 mm
  a: 1012 m/s (from formula (3))
  Min. permissible pressure: vacuum
  Pressure class: PN 10

None of the parameters of the pump check valves were known. For the purpose of the model, it was therefore assumed – largely correctly as it turned out – that the valves would close abruptly upon reversal 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 analysis of all three pumps cutting out simultaneously showed that macro-cavitation and, as a result of it, dynamic pressure peaks as high as 15 bar would occur inside the HDPE pipeline, exceeding the nominal pressure of the PN 6 pipe to an impermissible extent. 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. Fig. 24 and Fig. 25 show some examples of the system behaviour without surge control, plotted over time. Fig. 24 shows the pump speed, pressure head and flow rate at the inlet of model pipe L1 (pressure heads in m above pipe centreline); the curve in Fig. 25 shows the axial forces acting on L8. This explains the violent shaking and resulting damage observed earlier.

Surge control measures

To eliminate the macro-cavitation developing after the pumps have cut out, a second simulation calculation was run with a DN 150 air valve at the outlet of L2, the highest point of the piping. Despite the addition of this surge control device, the HDPE pipe was still found mathematically to contain unacceptably high pressure increases a few seconds after the pumps had cut out. In order to eliminate these highly undesirable pressure peaks, it was eventually decided to add a bypass with a shut-off valve between the inlet of L1 and the pump inlet tank. The valve would be automatically opened by a maintenance-free electro-hydraulic lever and weight type actuator if all three pumps were to cut out at once. Valve manufacturers today offer systems like this more or less as part of their standard product range. After adding these two surge control devices, i.e. an air valve 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. 26 shows – on the same scale as in Fig. 24 and Fig. 25 to facilitate comparison – the n, H and Q curves of the surge-protected system, plotted over time; Fig. 27 illustrates the forces of the surge-protected system, plotted over time. The global pressure envelope of the newly protected system as well as the curves of the system without surge control are shown in Fig. 28.

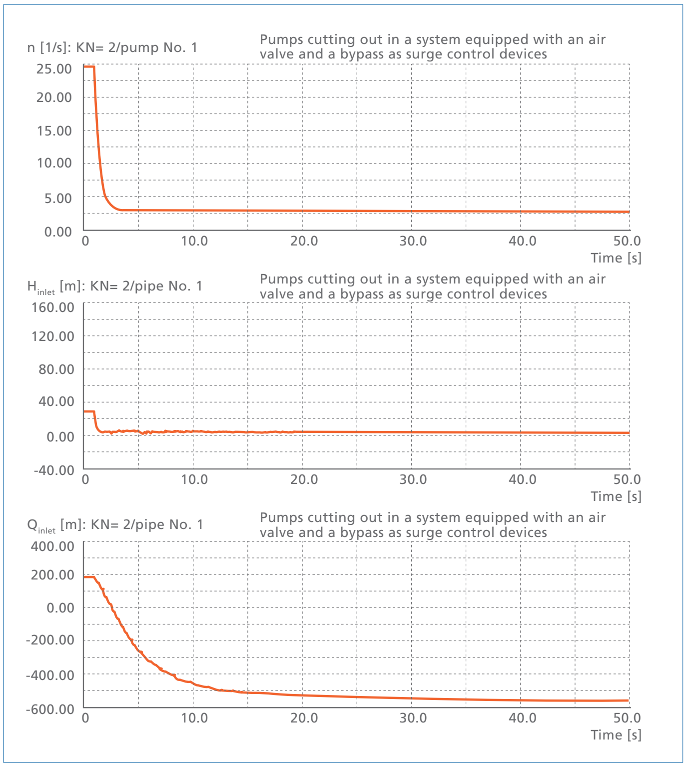


Fig. 26: Operating characteristics of the stormwater pipeline with surge control, plotted over time

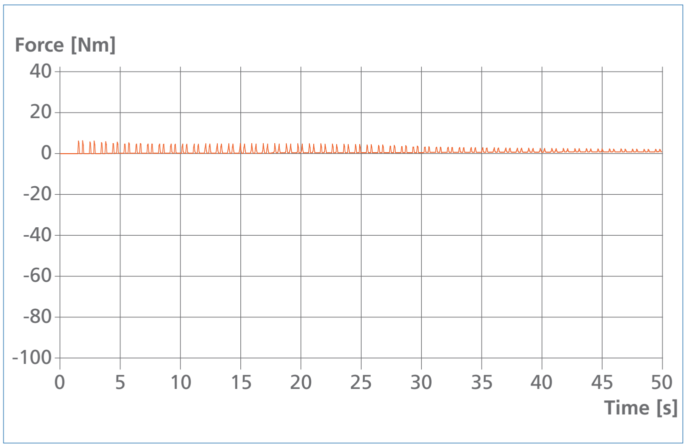


Fig. 27: Longitudinal force acting on L8 if the stormwater pipeline is protected by surge control

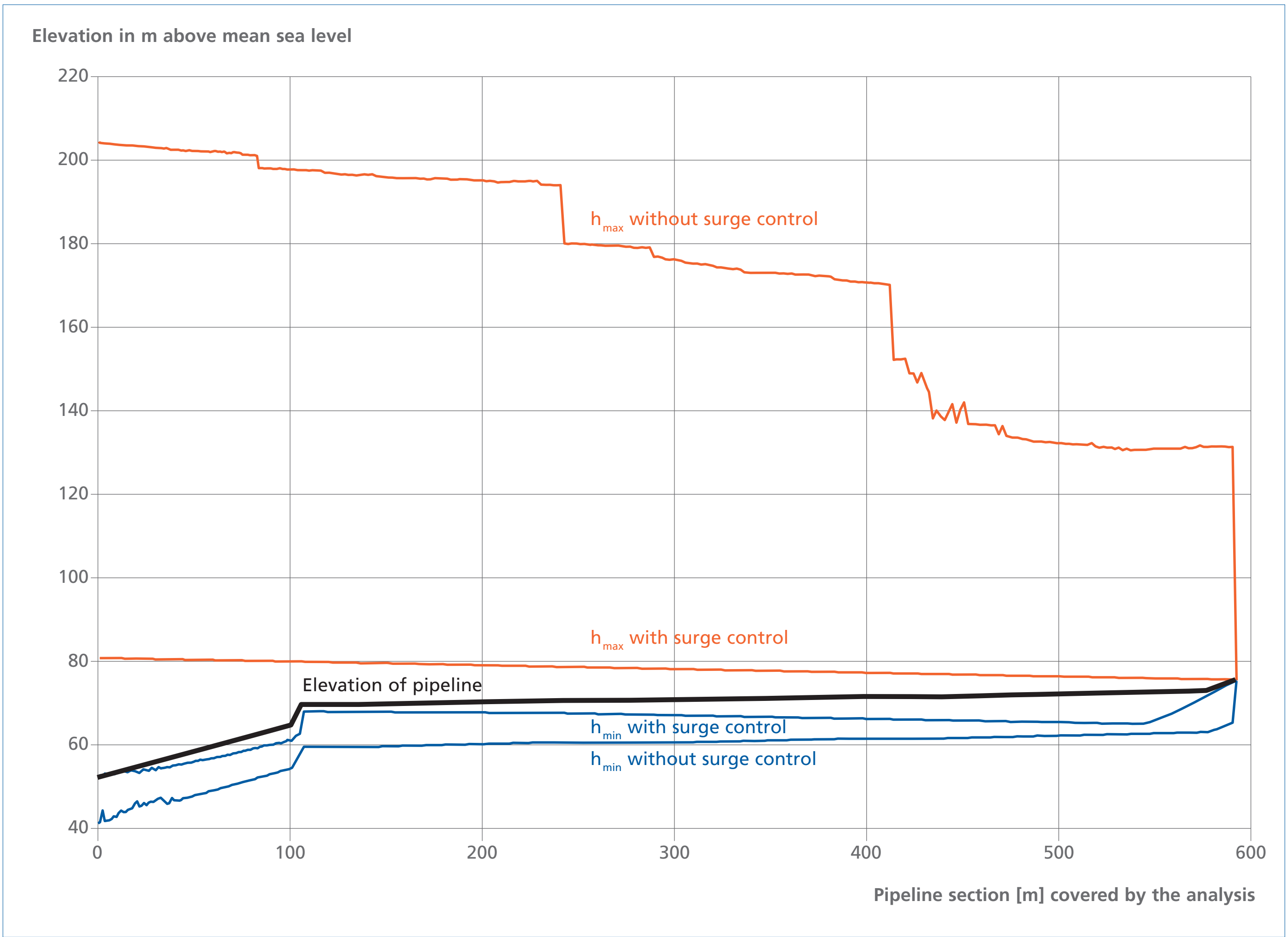


Fig. 28: Pressure envelope of the stormwater pipeline with and without surge control

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