WATER HAMMER – DO WE NEED TO PROTECT AGAINST IT?

HOW TO PREDICT IT AND PREVENT IT DAMAGING PIPELINES AND EQUIPMENT

S.C. Ord
Stopford Projects Ltd, UK

INTRODUCTION

Water hammer is also known as surge or transient flow, and is a phenomenon that is always present – it's just not obvious most of the time. When it does get large enough to notice, it could be as little as a banging noise, but it could be big enough to move pipes on their racks, damage pipe supports, cause leaks or even cause pipes and vessels to burst.

Many engineers have only outline knowledge of water hammer. This paper is aimed at making readers more aware of the dangers, what can be done to avoid the problem, and what can be done to control the phenomenon if avoidance is not possible. It also will make readers aware of the danger signs that signal the possibility of dangerous levels of water hammer. Due to the brevity of the paper, some topics are only discussed in outline, and for more detail, standard texts should be consulted (e.g. Reference 8).

WHAT IS WATER HAMMER?

Water hammer was so called because it is usually observed as banging sounds in pipes. It is caused by sudden changes in fluid velocity. Domestically, it can be caused sometimes by quickly closing a tap, especially if it is a tap that only requires a 90° turn to close it. The result is that the pipes move due to the pressure waves in the water, and thus might strike floor joists or walls, creating the banging noise. However to get the worst type of water hammer, a domestic tap would need to be closed very quickly, as pipe lengths are usually small in houses. So, water hammer in the home is rarely more than an irritation.

Industrially however, water hammer can be much more than a minor irritation. The following sections describe how it occurs and hence how it manifests itself, and consequently how to prevent it or guard against it.

A CAUSE OF HIGH PRESSURES

Changes in pressure are responsible for maintaining and for changing the velocity of a liquid. These changes can result in forces on the pipe itself. Normally, they are small, such that their presence is not noticed. However if the rate of change of flowrate is high enough, the forces can get high enough to make the pipe move or exert high forces on its supports. If the supports are not fixed, the pipe can come loose, or collide with the limit stops, sometimes violently.

For example, consider a valve that is reducing the flowrate of water that is flowing by gravity from a reservoir (Figure 1). As the valve closes, the pressure in front of the valve increases, and that causes a deceleration of the liquid column in the pipe. As the flowrate decreases, the pressure drop across the valve increases until the flow is at a new lower value. Continuously closing the valve causes the flow to continuously decrease. Eventually the valve will be fully closed and the flow will have stopped.

Liquids are not incompressible. Compared to gases, their compressibility is very small, but they typically have a compressibility of 2.2 GPa (water, 20°C). As a result, the pressure increase in front of the valve that we have been talking about is compressed, and so the valve movement does not cause the whole column of liquid that is approaching it to slow down instantaneously. Rather, a pressure wave travels at speed, usually close to the speed of sound, up the pipe, conveying this change of pressure upstream against the flow. When the wave front meets the moving liquid that is still moving, it slows it to rest. The liquid that is now stationary between the wave front and the valve is compressed and at elevated pressure.

If the pipe is short and the speed of the wave is high, then this process is rapid, and the column does slow down more or less evenly. This is the case of water hammer being imperceptible.

However if the converse is true, and especially if the closure time of the valve is similar to the time the wave takes to travel the length of the pipe, then this compression effect is pronounced. This causes high pressures to be produced.

The first person to describe this effect was the Russian scientist Joukowski. He showed both theoretically and experimentally that there is a maximum pressure that can be produced, known now as the "Joukowski head" or "Joukowski pressure" depending on the units of its expression. It is given by the following formula

$$h = \frac{v * c}{g} \tag{1}$$

Where

$c =$ speed of the sound wave in the pipe, known as the "wave celeric", m/s
$v =$ initial velocity of the liquid, m/s
$g = 9.81\ \mathrm{m/s^2}$
$h =$ Joukowski head, m

Thus for water in a rigid pipe, where the wave celeric is (for example) 1500 m/s, for an initial water velocity of 2 m/s, then suddenly stopping the flow with a rapid valve action will result in a head of about 300 m, or 30 bar.

The maximum closure time that will give this head is given by

$$t = \frac{2 * L}{c} \tag{2}$$

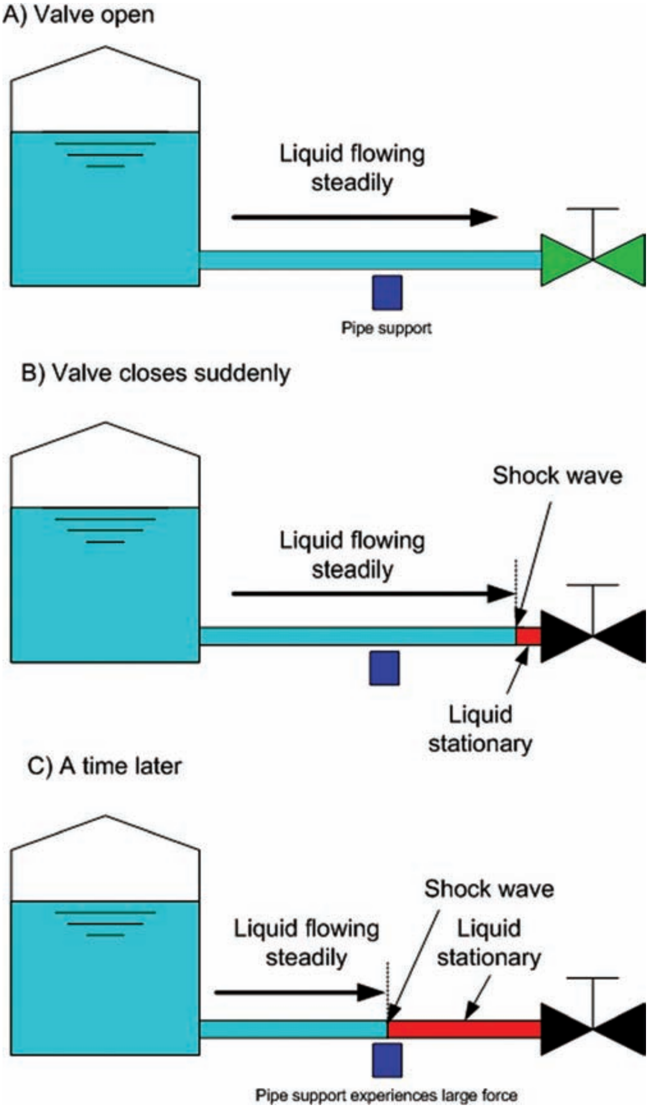


Figure 1. Liquid flowing through a value

Where

$L =$ length of pipe
$t =$ that time, which is the time it takes for a pressure wave to travel the pipe length and back again.

Hence if the pipe is 500 m long, the maximum closure time is about 0.67 s, ie a closure in this time or less produces a Joukowski head. For closure times up to 10 times this, ie 6.7 s, the pressure is reduced in about inverse proportion. So for a valve closing in 6.7 s, a maximum pressure of 3 bar over and above the static pressure might be expected. Thus depending on the closure time, this can be a high pressure that could damage pipes. Pipes that are not ruptured can of course suffer gradual deterioration if the pressure surge is a frequent occurrence. For example, it is common to see gaskets weeping – are pressure surges causing some leaks on your plant?

A CAUSE OF HIGH FORCES

To understand the cause of high forces, consider a pipe as shown in Figure 2. This shows a pressure wave travelling down a pipe which has bends in it.

Due to the pressure discontinuity, the passing of the wavefront produces a force on the pipe. Thus, for a 0.1 m diameter pipe, the passage of the 30 bar wave described above produces a force of

$$F = \frac{\pi}{4} * D^2 * P \tag{3}$$

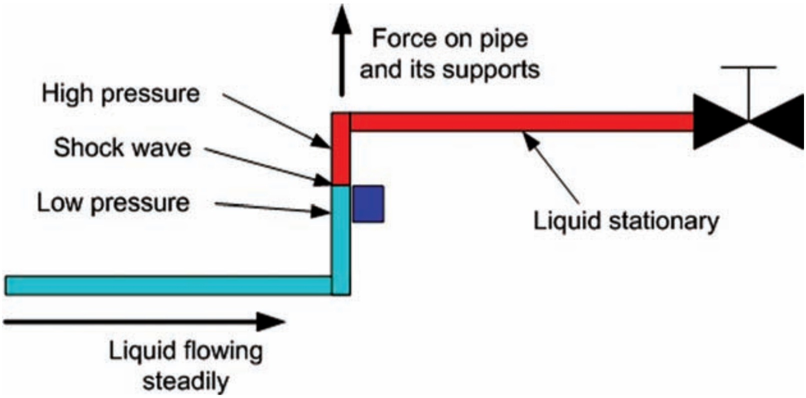


Figure 2. Pressure wave passing through a pipe with bends

Where

$F =$ force, N
$P =$ the pressure equivalent of the Joukowski head.

It is well known that

$$P = h * \rho * g \tag{4}$$

So the Joukowski pressure is

$$P = v * c * \rho \tag{5}$$

Where $\rho =$ fluid density.

In this case, the force F would be about 2.4 Te. This is clearly a very significant force and threatens to move the pipe, damage its supports, and possibly damage pipe bridges that carry it. Had the pipe been a 0.3 m diameter one, the force would have been about 22 Te.

BEYOND THE SIMPLE THEORY

Sadly that's not the full story. The forces and pressures calculated above are for specific, fast events. There are lots of shades of grey between a Joukowski event and a gradual change in fluid speed that gives no perceptible surge forces. So in many cases, a more complex analysis has to be done to ascertain if a pipe is at risk (see section 7). This will usually involve transient modelling of the pipe in greater or lesser detail to see how large the surge pressures will be. This leads to the ability to calculate the maximum and minimum pressures the pipe will experience during the transient event.

The situation for forces on pipes and their supports is more complex. Even once the analysis of transients is complete, the dynamic response of the pipework has to be considered. This comes from vibration theory and consideration of the time for which the forces act.

With rigid pipework, vibration theory shows that the maximum force can reach twice the value calculated in equation 3. For flexible pipework, the force transmitted to anchors can be small. If the pipe is short and its period of vibration is long, then the pressure wave passes through that section of pipe quickly. Hence the transient can be gone before the pipe has time to move enough to sustain damage. However if the converse is true, then the harmonic motion of the pipe is such that the forces calculated previously need to be doubled to get the correct result. Thus if harmonic motion of the pipe is not to be studied, then the result of equation (3) should be doubled to be safe. This factor modifying the force result is known as the Dynamic Load Factor (DLF), so the full equation for force is:

$$F = \frac{\pi}{4} * D^2 * v * c * \rho * DLF \tag{6}$$

It is safe to assume that $DLF = 2$. If this assumption gives an unacceptable result, then more detailed analysis of the pipe can be carried out to find if a lower DLF can be justified.

WHAT ELSE CAN GIVE RISE TO WATER HAMMER?

DOWNSTREAM OF VALVES

The foregoing discussion considers water hammer due to positive pressure increases in front of a valve. However for pipes with valves not close to the pipe end, it is more likely that problems will be experienced downstream of the valve. Here, the pressure can fall to the vapour pressure of the liquid, and boiling can occur. If boiling does occur, a cavity forms. In a typical situation such at that shown in Figure 3, the liquid downstream of the cavity returns and builds up almost the same velocity as the liquid had when the cavity started to form. However when the liquid returns to the valve or to the liquid remaining before the cavity, the collision is a severe one, equivalent to a valve closing in a fraction of a second. This can result in severe water hammer.

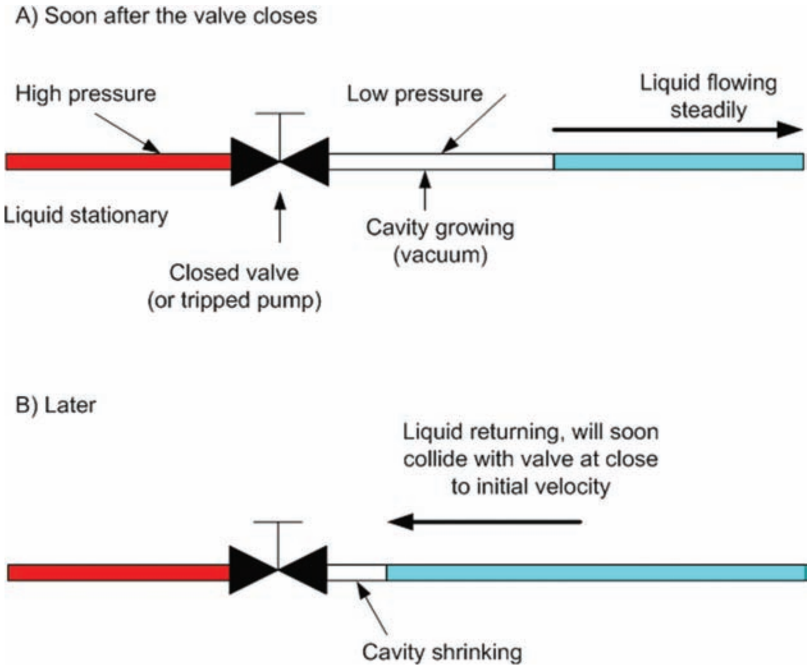


Figure 3. Cavity growing downstream of the value

PUMP TRIPS

The same sort of situation occurs downstream of a pump that has tripped. If the pump slows the liquid sufficiently quickly, then cavitation will occur downstream of the pump, and the vapour collapse gives rise to severe water hammer (Figure 3). Immediately after a trip, the fluid drives a centrifugal pump like a turbine, and the dynamic behaviour is complex. For positive displacement pumps, the run-down is largely independent of the fluid conditions. In either case, a cavity can form in the low pressure area and lead to a cavity collapse and subsequent pressure shock.

HIGH POINTS

Cavitation can also happen in elevated places such as on pipe bridges. In the situation shown in Figure 4, the liquid is vulnerable to cavitation on pump trip as the downcoming liquid can pull suction on the elevated section. This can cause a problem even when the elevated section is less than a barometric leg above the destination tank. However if the pipe high point is more than a barometric leg above the destination tank, then unless valves are used to prevent it, there will always be a cavity formed. As a result, when the pump restarts, the liquid columns will collide, again creating severe water hammer if the velocity is high.

If non-condensable gases come out of solution in the cavity, the subsequent collapse will be cushioned, reducing the surge pressure and forces. However this cannot be relied on for surge protection.

RESTRICTOR ORIFICE

When a flow is restricted by a restrictor orifice that is close to the end of a pipe, a problem can occur on startup (Figure 5). The system depicted transported a boiling liquid from one storage tank to another. A control valve was needed in the pipe to regulate the flow, but to prevent flashing across it a restrictor was placed in the pipe downstream, close to the inlet of the destination tank.

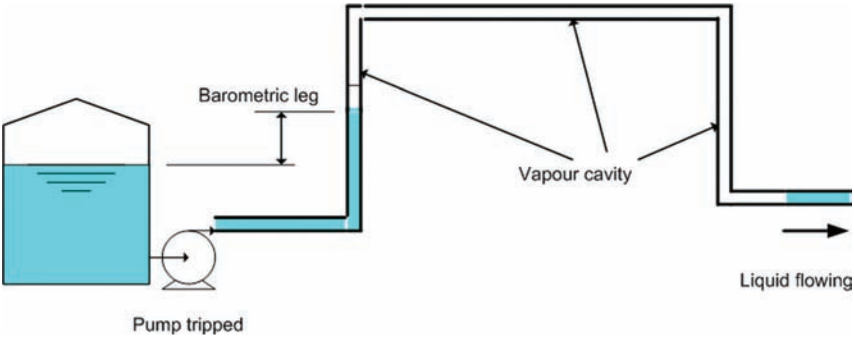


Figure 4. Cavity forming in a pipe bridge

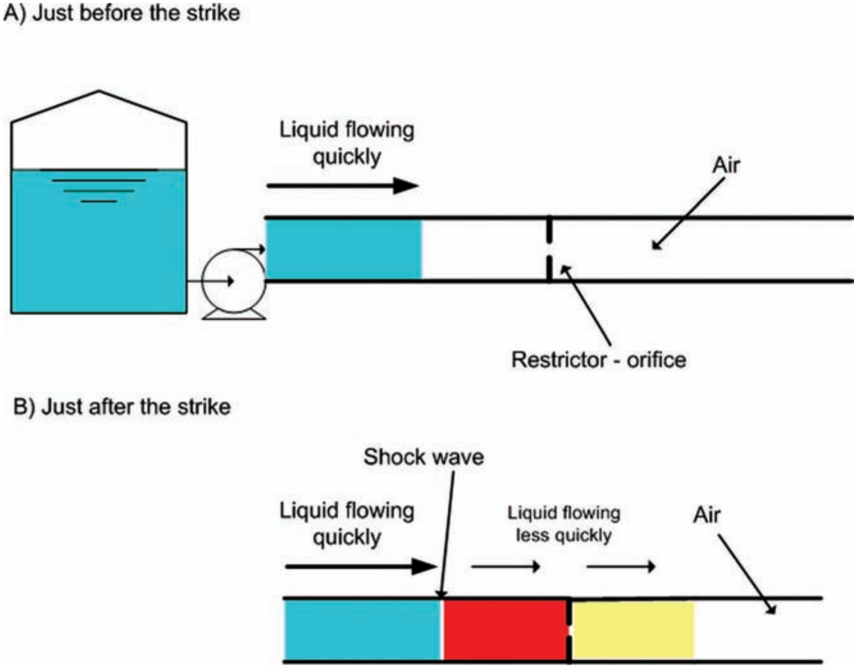


Figure 5. Flow hits an RO

Under normal running conditions, the pressure drop across the restrictor causes a back-pressure that prevents flashing across the control valve and hence allows it to operate correctly. However on startup, when the pipe is empty, the liquid runs at high rate until is reaches the restrictor, when its flow suddenly falls. This causes a surge event with the velocity being the change in velocity caused by the passage of liquid through the restrictor. In the case studied, this was quite significant, and the pipework had to be designed to withstand the forces produced.

NON-RETURN VALVES

Non-return valves can be the pressure surge engineer's nightmare. Normally, such valves are held open by the fluid flow, and are closed either by a spring, or by the drag of the fluid as it flows in a reverse direction, e.g. after a pump trip.

The problem is that unless correctly chosen, these valves can have unfavourable characteristics, in that they suddenly slow the liquid as they reach the closed position. So, if the valve is slow to operate, a significant reverse flowrate can build up, and then the valve slams shut – exactly the conditions to get a pressure surge.

Fortunately some manufacturers have recognized this problem and make non-return valves that close quickly and prevent the reverse flow building up. These should be used whenever reverse flow is likely in long pipelines. Selection of a suitable valve can be based on the deceleration of the liquid expected. This is described in Reference 4.

WHAT ARE THE DANGER SIGNS?

LONG PIPES WITH ISOLATION VALVES

How long is long? This depends on the pipe material and how fast the event triggering the flow change takes place. It also depends on the type of valve in use, and its characteristics.

For a valve operated by hand requiring only a 90° turn to close it completely, such as a butterfly valve or a ball valve, then a closure time of well under a second is possible. Assume a closure time of 1/4 second, with a wave celeric of 1000 m/s (typical for a water in a thin wall steel pipe). Then equation (2) shows that a pipe of 125 m length is long enough to experience a Joukowski head, and even a pipe of 12.5 m will see a significant pressure rise from the closure.

For an automatic valve closing in 5 seconds, the pipe lengths increase to 2500 m and 250 m respectively. Thus pipes longer than 1000 m should be regarded with suspicion until pressure surge effects have been considered.

So far it has been assumed that the valve has a completely linear effect on the liquid, ie closing the valve 50% causes a 50% reduction in flowrate, and so on. In practice this is rarely the case, and valves typically cause a much greater rate of deceleration in the liquid as they approach the closed position. Therefore the figures above are best cases, and for safety a factor of (say) 4 should be applied until the real valve characteristics can be allowed for. This is discussed more in section 5.3.

So, in practice, pipes longer than 250 m should be considered as potentially at risk from surge.

LONG PIPES WITH PUMPS

Pumps can run down in any length of time from less than a second to tens of seconds, according to their inertia and internal friction. The rate of slow-down decreases as the speed decreases. However a typical pump will run down to under 50% of its initial speed in 2 to 3 seconds, causing a similar decrease in flowrate.

The effect of this can be estimated as if it were a closing valve, which was discussed above. If the pump is a positive displacement pump located at the end of a long suction pipe, it is similar to a valve closing at the end of a long pipe. However this situation is not typical. The common problem with pumps is cavitation in the delivery line, if the cavity can subsequently collapse. It is not possible to give typical figures here, because whilst there is only 1 bara of atmospheric pressure, frequently the static pressure in the pipe system can be high and thus prevents cavitation. Hence the situation needs careful analysis. Changes in elevation, particularly pipe bridges, are danger areas as it is here that static heads can be lower and cavitation is more likely.

LARGE CHANGES IN ELEVATION

If there are large changes in elevation, it is possible to form cavities that will collapse suddenly when the liquid flow is started. In this case, long pipes are not necessary, as the collapse can effectively be a very fast event, causing water hammer even in relatively short pipes.

HOW CAN WATER HAMMER BE AVOIDED?

LOW VELOCITIES

It is sometimes said that if the liquid velocity is low enough and the pipe can cope with the expected Joukowski head, then there is no need for concern.

This is a simple approach to avoiding pressure surge problems, but it does limit the system to low velocities in most pipes. So for a pipe rated to 10 barg, from equation 5, the maximum allowed velocity is 1 m/s. This is not likely to be economic. At the same time, if this pipe were 0.2 m diameter, from equation 6 it must be capable of withstanding a surge force of up to 6.4 Te.

Even then, it is dangerous territory. It is possible to have networks that can cause a number of reflected waves which might meet in an unfortunate way and cause a pressure higher than Joukowski. It would be unfortunate to suffer a loss from this, but a loss is likely to be suffered in time from a pipe subjected repeatedly to a limiting pressure and force.

ELASTIC PIPES?

The calculation of the wave celeric has not yet been discussed. If the pipe were infinitely rigid and fully supported, then the pressure wave would travel at the speed of sound in the liquid. However real pipes aren't so, and they bulge slightly under the influence of pressure. This is equivalent to the fluid being more compressible, and so the wave travels less quickly. The final wave speed after this effect is known as the wave celeric. It can be calculated for thin pipes using the following equation (Reference 9):

$$C = \frac{C_0}{\sqrt{[1 + (K * \mathrm{d}/\mathrm{E} * \mathrm{e}) * \psi]}} \tag{7a}$$

And

$$C_0 = \sqrt{\frac{K}{V}} \tag{7b}$$

Where

C = wave celeric, m/s
C_0 = speed of sound in the liquid, m/s
K = bulk modulus of elasticity of the liquid, Pa
d = pipe internal diameter, m

E = Young's modulus for the pipe material, Pa
e = wall thickness, m
ρ = liquid density, kg/m^3
Ψ = pipeline support factor
Ψ = 1 for pipes with expansion joints
$\psi = 1 - \mu^2$ for pipes that are completely constrained against axial movement
$\psi = 1.25 - \mu$ for pipes supported at one end only
μ = Poisson's ratio for the pipe material

A spreadsheet that will calculate these can be downloaded from Reference 1.

From equation 1, the magnitude of pressure surge is directly related to the wave celeric, so it is possible that by making the modulus of the pipe material sufficiently small, or the wall sufficiently thin, then the wave celeric will also be small and hence surge problems will be reduced. However there are practical problems with making pipes like rubber! GRP pipes usually have a low wave celeric, typically half or less than that of a steel pipe, but their ability to withstand pressure, vacuum and surge forces is usually limited. Hence elasticity is not normally seen as a practical solution. However credit should be taken for the reduction in wave celeric in GRP pipes when calculating the effects of pressure surge.

GOOD VALVE PERFORMANCE

Slow operation

Slowing down valve operation by increasing their stroke time is a good way to prevent water hammer. The only accurate way to calculate the minimum stroke time is by making a transient model of the system, including the correct valve characteristics (Cv against angle of closure) and then model different stroke times until a limit is found. Actuators can be slowed down to achieve the safe closure time calculated.

Hand valves can be fitted with gearboxes so that it is not possible for an operator to close them suddenly. This should always be done in vulnerable situations rather than rely on "good operating practice".

Good characteristics

Valves can be made with a number of different characteristic curves. Many valves have poor curves from a surge point of view, and they slow the liquid little during the major part of their travel, but then cause a rapid change in flow as they approach closure, e.g. the "rapid opening" type. Surge protection requires smooth changes in velocity, and so such valves are not welcome in systems susceptible to surge problems.

The ideal curve of Cv against valve position depends on the proportion of the pressure drop that occurs across the valve when the flow is normal. This is shown in Figure 6. For a typical design, where the valve pressure drop is about 10% to 20% of the total frictional pressure drop, a curved characteristic (the lower curve) gives the

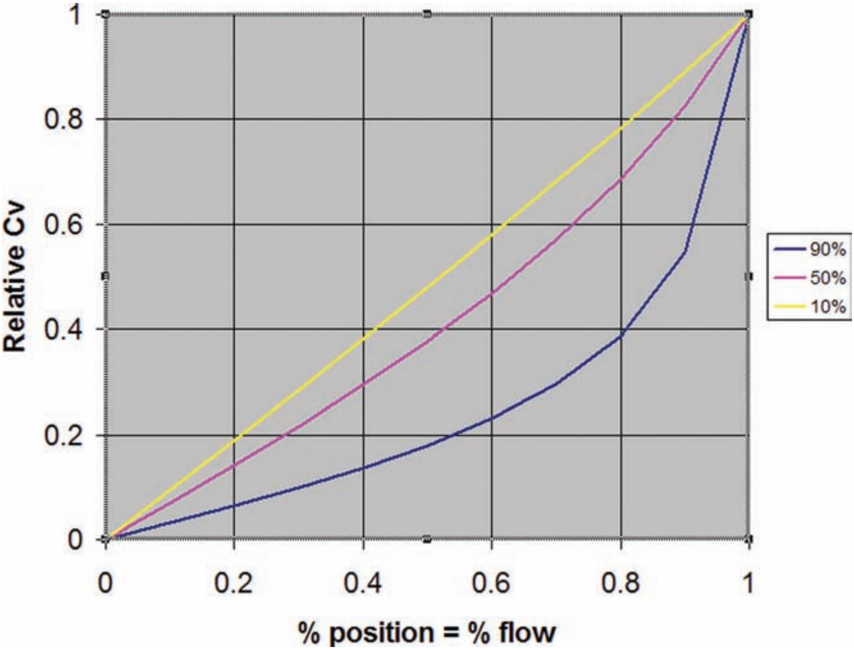


Figure 6. Relative CV vs position, such that flowrate is proportional to value position, for three values of the % of the total pressure drop across the pipe with the value fully open

desired flow vs valve position pattern. The valve trim known as the "equal percentage" type matches this best. This therefore gives a fairly linear decrease in flowrate as the valve closes, and so the slowing of the liquid is shared more evenly across the valve stroke. This is ideal for surge prevention, and the valve stroke time can be minimized by using such a valve characteristic.

PUMP INERTIA

Sudden shutdown of a pump, due to a failure of the pump or a failure of the electricity supply, is not possible to prevent absolutely. Depending on the consequences of a trip, it might be desirable to increase the inertia of the pump so that its rate of slowdown is reduced. This can sometimes be done by fitting an over-sized motor, but if this in not possible it is necessary to fit a flywheel between the motor and the pump. This might seem to be an unusual arrangement, but it has been used on many occasions. A typical flywheel pump is shown in Figure 9.

COMPRESSIBLE MATERIAL IN THE PIPE

A further way to prevent water hammer is to increase the compressibility of the liquid by injecting a gas into the flow. Gas bubbles effectively increase the compressibility, hence decrease the wave velocity, and decrease the size of any surge problems. The wave celeric can be reduced by as much as 90% by adding as little as 1% by volume of air in a pipe. However, it may be costly to arrange reliably, and is not a usual solution.

It should be remembered that whenever the wave velocity is decreased, the benefit of the reduced Joukowski head (equation 1) has to be offset against the reduced pipe length that is allowable without incurring surge problems (re-arrange equation 2).

HOW CAN THE CONSEQUENCES OF WATER HAMMER BE AVOIDED?

STRONGER PIPES AND SUPPORTS

Long pipes are where water hammer problems are most likely. Making such pipes sufficiently strong to withstand Joukowski pressures and forces is unlikely to be economic.

Most pipes in good condition can withstand pressures well in excess of their design pressure for short periods of time. Given also the subtlety of the dynamics of pipe responses to transients (section 2.3) it is often true that bursting the pipe is not the main concern of the pressure surge engineer. Hence if a pipe is just on the limit of pressure surge problems, it is often more important to be concerned about the pipe supports rather than the pipe itself.

If avoidance of contamination of the material in the pipe is important, such as with buried potable water pipes, then it might be mandatory to prevent the internal pressure from falling below atmospheric pressure. In this case, strong pipes and supports are not considered adequate.

CONTROL DEVICES

There are many devices that have been created to deal with the dynamics of surge. A full discussion of the various types is beyond the scope of this paper, but three types are discussed below.

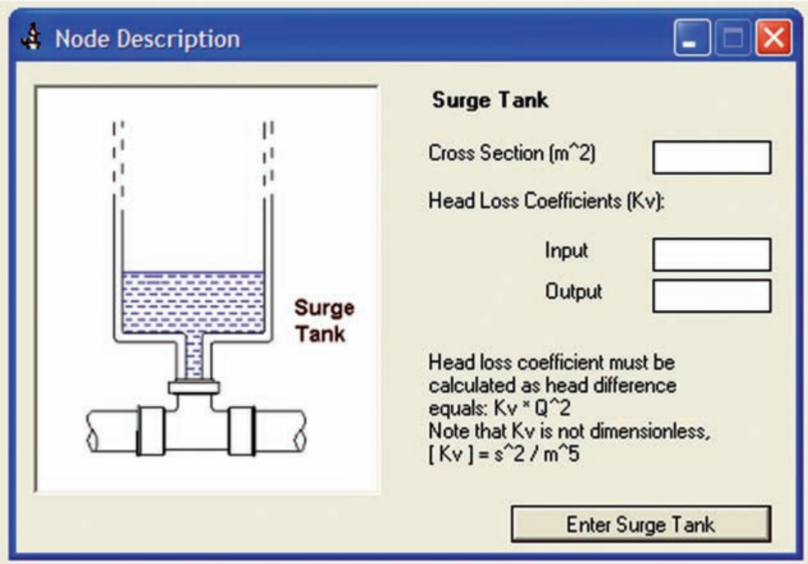
Surge vessel

This is a vessel into which, and out of which, liquid can flow. (Figure 7.) Sometimes it is arranged with air or other gas trapped in it, either in contact with the liquid or in a bladder, but for water pipes it is often an open vessel. It can be regarded as both a potential sink and a potential supplier of energy to the pipe at the point where it is fitted. It contains the process liquid and (sometimes) a gas at pressure. If the gas is likely to dissolve in the process liquid, then it is usually placed in the bladder, which might be pre-pressurised.

Alternatively, the process fluid might be placed into the bladder if it is corrosive, allowing the vessel to be made from carbon steel. In this case, the pressurizing air or gas is between the vessel and the bladder.

When a surge event comes along, liquid can enter or leave the surge vessel, thus interfering with the pressure wave. It can be regarded as a local high flexibility of the

Input dialogue box – courtesy of Istec Ingenieria



Surge Tank

Cross Section (m^2)

Head Loss Coefficients (Kv):

Input

Output

Head loss coefficient must be calculated as head difference equals: Kv * Q^2
Note that Kv is not dimensionless,
[Kv] = s^2 / m^5

Figure 7. Surge tank

pipe, or a high local compressibility of the liquid. The effect is to reduce the size of the wave passing the connection point. Sizing of such a vessel is fairly straightforward with transient flow software designed for this purpose (see section 7.1).

Non-return valve
In section 3.5, warnings were given about the dangers of using the wrong types of non-return valves when significant water hammer is possible. However, sometimes a non-return valve is needed to prevent the formation of cavities due to reverse flow. The appropriate type of valve should be chosen according to the situation under consideration.

High inertia pump
The potential danger from a low inertia pump was described in section 4.2. Hence a normal pump feeding a long pipeline is a surge risk. If the inertia is increased, this stores energy and so the pump and the liquid will slow down more gradually. The increase required might be small, such as a large or magnetic coupling. In other cases, a flywheel can be placed in the pump drive (Figure 9).

COSTS OF VARIOUS METHODS OF SURGE CONTROL

Surge control options will vary with each project, and the engineer will have to examine several to obtain the most cost effective solution to a problem. For example, in the case of a long caustic soda line, bladder type alleviators were initially favoured over flywheel pumps, since their technical risk was considered to be lower. However, examination of reliability and testing requirements shifted favour towards the flywheel pump. Costs strengthened this argument, as the two alleviators would have to cost £98,000 each, whereas the two pumps with suitable 10 kg·m^2 flywheels were £21,000 each. Flywheel pumps were installed, and have operated well. (Data courtesy of Ineos Chlor.)

RELIABILITY OF SURGE CONTROL EQUIPMENT

Strong pipes and supports are reliable provided they are strong enough to withstand the pressures and forces they might experience, and are correctly maintained.

Similarly, a high inertia pump will always protect against water hammer on pump trip. Given a typical layout of a flywheel pump, the coupling between the pump and flywheel should be made stronger than that between the flywheel and motor, so that in the event of a mechanical failure, the flywheel will still protect the pump.

Trapped gas type protectors can be checked by depressurizing the system (e.g. stopping the pump) and confirming that the pre-charge pressure is still within the expected range. However if the system has a high static head, this might not be possible without draining the pipeline, which can be inconvenient. It is not usually possible to assess the serviceability of a bladder by inspection without withdrawing it, so in safety critical situations such a device might not be acceptable.

HOW TO ANALYSE A PIPING SYSTEM FOR RISK OF WATER HAMMER

Having performed the simple calculations from the foregoing equations and found that there is a risk of water hammer, it is necessary to do some more detailed computation.

Dynamic analysis of pipe systems is often a complicated, lengthy and expensive procedure, and so the steps shown in Figure 10 are recommended before embarking on full dynamic computer models.

Due to the specialized nature of computer modelling and the costs involved, it is sensible to attempt cheaper solutions first. Reference 8 presents a logical procedure to examine a system without computer simulation, and a spreadsheet has been developed helps distinguish between problematic and problem-free systems without dynamic modelling (Reference 1).

If this indicates a need for further analysis, then a detailed computer model needs to be considered. Again there are several options for the engineer.

SOFTWARE WITH COMPLEX MODELLING CAPABILITIES

There are several excellent programs that are capable of modelling very detailed and complex piping systems, including

- Flowmaster (Flowmaster Ltd)
- Wanda (Delft University)

- Hammer (Haestad Systems)
- Pipenet (Sunrise Systems)

These are expensive programs and require skilled use. It is often possible for the casual user to make mistakes due to the versatility and variety of the models and facilities present in the package. However for complex situations such as branched networks, partially full pipes, emptying and filling of pipes, and complex rheology, these programs give the best solutions available. Fees are in the order of £10,000 per year for these packages.

The author is not fully familiar with all of these packages, but is aware that they use the same mathematical technique, and are capable of highly accurate modelling. Some vendors have used practical apparatus set up specifically to produce results that can be compared with the results coming from the software. There are also “standard” results available against which software developers can test their product.

SOFTWARE WITH LIMITED MODELLING CAPABILITIES

There are other packages cheaper than those mentioned above that are less comprehensive in their capabilities. However they still use the mathematical method that is recognized as best for solving this type of problem (the “method of characteristics”) and as such can be relied upon to do safe analyses of simpler problems. These include:

- HiTrans (Reference 3)
- Hytran ((Reference 7)

These programs tend to come from smaller organizations that do not have the resources to conduct their own practical tests. However the author has run problems on both Flowmaster and HiTrans and found similar results. Flowmaster can deal with more complex situations than HiTrans, but he found that every problem he has dealt with in chemical factories to date could be modelled on both programs. Fees for these are from a few hundred pounds for outright purchase.

These programs all describe the pipe system as a series of nodes with pipes in between them. The modeller is free to choose the nodes. Ideally the system would be described by as many nodes as there are bends in the pipe. However sometimes this is impractical and unnecessary, and shorter sections of pipe can be amalgamated in the model.

INTERPRETATION OF RESULTS

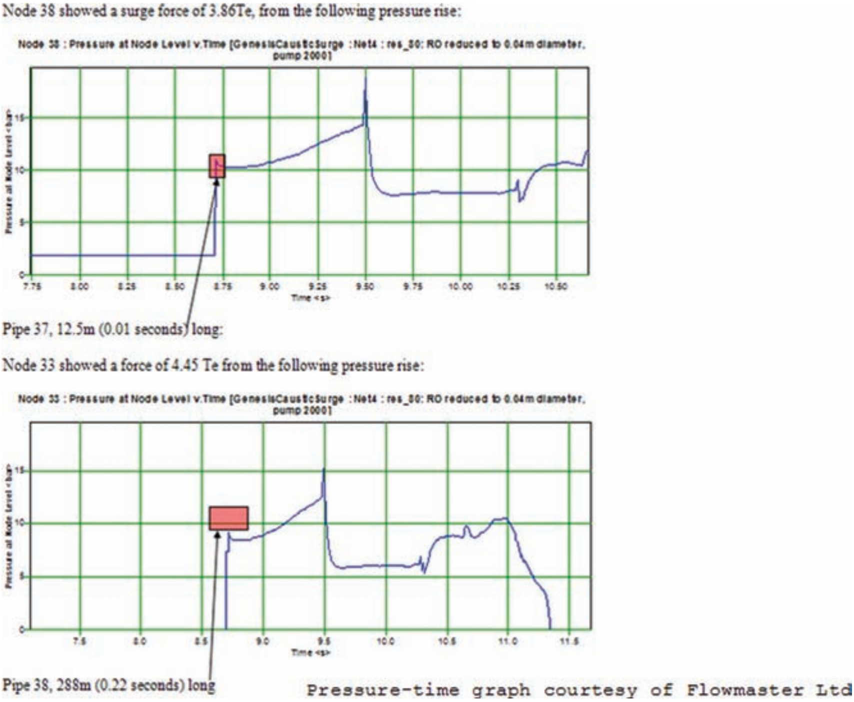
Models have to be carefully written to give the results needed. They must be written to represent the system being studied so that all important events such as startup, shutdown, emergency shutdown, power failure etc., can be studied. Once the model has been written, scenarios representing these events are run. All runs then have to be analysed. The first steps are:

1. Look for high and low pressures in the system
2. Look for the formation of cavities (or pressure falling below the liquid vapour pressure).

These need to be compared with practical limitations, such as the maximum and minimum allowed pressure in the pipe, and the maximum force allowed on the pipe supports.

The pressure-time history produced by modelling software is usually complex due to reflection and interference of waves. At any point in the pipe, the change in pressure with time can be plotted, and examined to determine what the maximum and minimum pressures were, and to deduce the forces produced by them at that point. The whole pipe length has to be examined this way.

To calculate the force that might appear in a section of the pipe, it is necessary to look for the maximum pressure step that can "fit" in the pipe. The software operates by calculating the pressures and flowrates in the system at each node at many timesteps over the period of time that the modeller wants to study. It is possible to look at the sections of pipe and the pressure-time history of the nodes next to it, and look for times when the pressure has been changing rapidly. Instantaneous events take place over one timestep, and will fit into any length of pipe (Figure 8, first diagram). The coloured bar represents a certain pipe in units of time taken for the wave celeric to traverse it. It is a short pipe, and


Node 38 showed a surge force of 3.86Te, from the following pressure rise:

Pipe 37, 12.5m (0.01 seconds) long:

Node 33 showed a force of 4.45 Te from the following pressure rise:

Pipe 38, 288m (0.22 seconds) long

Pressure-time graph courtesy of Flowmaster Ltd

Figure 8. Filling pressure-time history into a pipe to calculate pipe forces

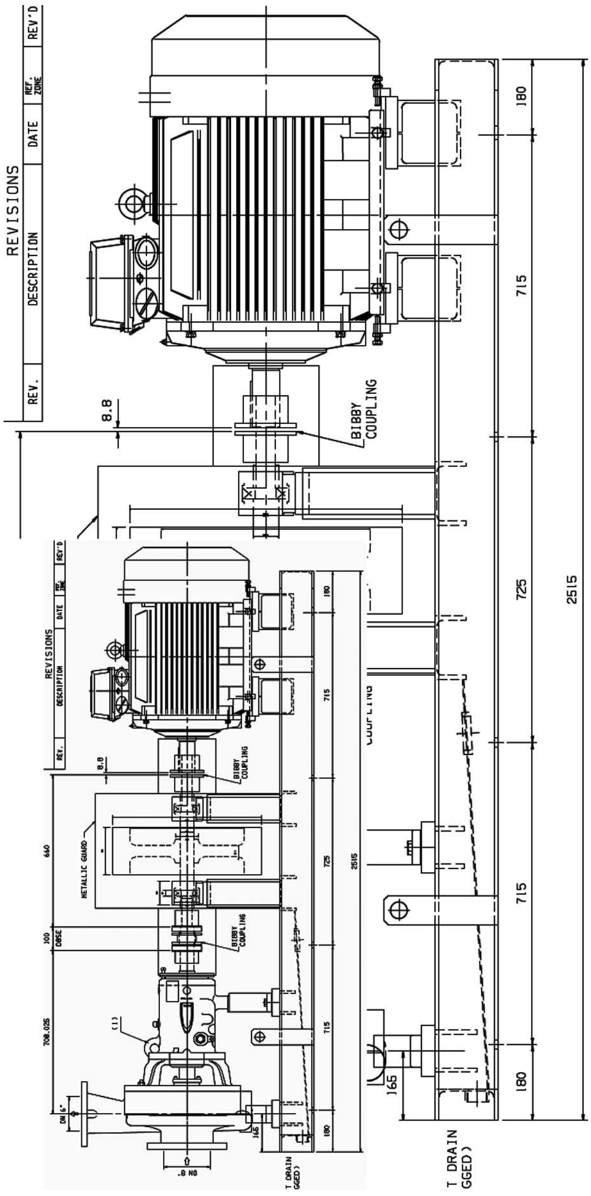
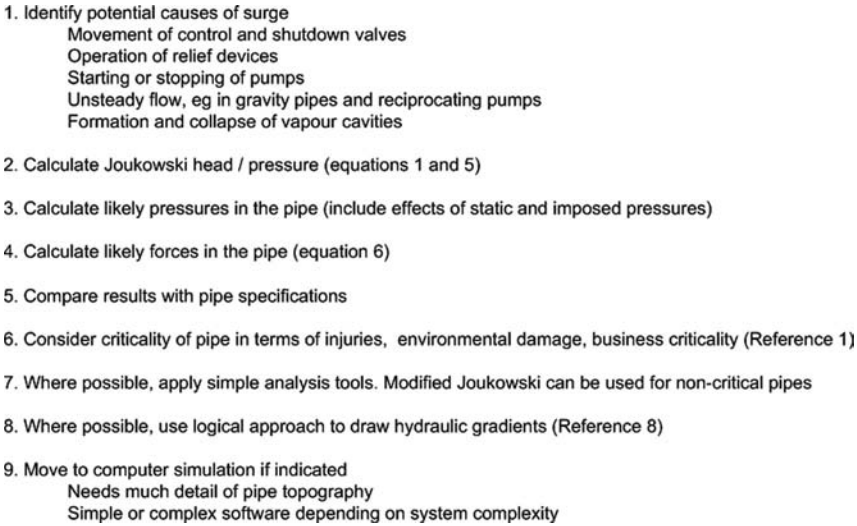


Figure 9. Flywheel pump

Courtesy of Flowserve



1. Identify potential causes of surge
    - Movement of control and shutdown valves
    - Operation of relief devices
    - Starting or stopping of pumps
    - Unsteady flow, eg in gravity pipes and reciprocating pumps
    - Formation and collapse of vapour cavities
2. Calculate Joukowski head / pressure (equations 1 and 5)
3. Calculate likely pressures in the pipe (include effects of static and imposed pressures)
4. Calculate likely forces in the pipe (equation 6)
5. Compare results with pipe specifications
6. Consider criticality of pipe in terms of injuries, environmental damage, business criticality (Reference 1)
7. Where possible, apply simple analysis tools. Modified Joukowski can be used for non-critical pipes
8. Where possible, use logical approach to draw hydraulic gradients (Reference 8)
9. Move to computer simulation if indicated
    - Needs much detail of pipe topography
    - Simple or complex software depending on system complexity

Figure 10. Steps to resolve possible surge problems

only a rapid pressure change can fit into it. Slower changes in pressure will not fit into short pipes as the leading edge of the pressure wave has turned the next corner before the trailing edge gets into the pipe. In this case, it is only the pressure change that can fit into a pipe that produces an out-of-balance force in the pipe. For a longer pipe (Figure 8, second diagram), not only an instantaneous event, but also part or all of a more gradual change, can result in a force in the pipe.

Given that the pressure change that fits into the pipe has been evaluated, the force is simply calculated from:

$$F = \frac{\pi}{4} * D^2 * \Delta P * DLF \tag{8}$$

Where ΔP = maximum sudden pressure change fitting into the pipe.

Finding the worst events in a pressure-time history comprising many thousands of data points can be arduous. Some software gives help by offering charts showing maximum and minimum pressures at each node and graphs of pressure-time history for each node (Reference 2). Other software allows the pressure-time history to be copied (Reference 3) and pasted into a spreadsheet such as Excel. Here, functionality can be created to search out the most significant regions of the pressure-time history and calculate the forces on that section of pipe.

CONCLUSIONS

Water hammer is a real phenomenon that can cause spectacular damage to pipes. The reasons for its low profile are that few people are aware of its widespread presence and problems have to be acute enough to notice readily. Perhaps more significant is the chronic effect of repeated pressure transients on pipes, leading to early failure of joints rather than failure of the pipes themselves. Damage to pipe supports, with possible collateral damage to other pipes on the same pipe racks or bridges, is usually a more significant worry than burst pipes.

It is possible to rapidly assess if there is a risk of water hammer damage. Engineers should be aware of this risk and make proper use of modern techniques and software to ensure that water hammer problems are dealt with appropriately.

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