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Waterhammer

practice - criteria - provisions - mathematical description - examples

J. Wijdieks

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CONTENTS

1 Introduction

Waterhammer may be best defined as the appearance of pressure changes in closed conduits caused by velocity changes with time of the flow. Therefore waterhammer may occur in all kinds of pipelines in which flow changes with time occur. For instance in long (or short) pipelines for oil, sewage, drinking water, cooling water, slurry, coal slurry, chemicals and in fresh water or city heating networks. The greater and faster the velocity changes, the stronger the pressure changes. In such cases "rapid" must be related to the critical time of pressure waves; that is the time needed for a pressure wave to travel from one end of a single pipe to the other and back (see Fig. 1).

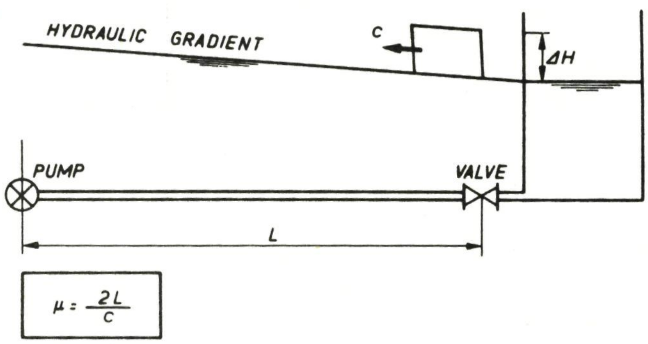


Fig. 1 Travel time of pressure waves

The pressure wave speed c depends on the pipe material, the type of fluid in the pipe, the diameter and wall thickness of the pipe, the amount of free gas in the fluid and the way the pipe is anchored [1].

For gas-free fluids c normally ranges from 300-1000 m/s (see Fig. 2), while with small amounts of free gas c will have a much lower value (see Fig. 3) than for the gas-free fluid, since in such a case the bulk modulus of the fluid-gas mixture is lowered drastically. In plastic pipes c depends also on the underpressure in the pipe, if any, due to the important influence of bending moments in the pipewall at such pressures [2, 3].
A relation between velocity changes and pressure changes was first found by Joukowsky in 1900 [4] when measuring in the Moscow drinking water line. Disregarding friction losses compared with static head in the pipeline, and assuming that the velocity changes in a time shorter than the critical time ("rapid" change of velocity change), he found (see Fig. 4) for gas-free flow:

$$\Delta H = \frac{c}{g}\,\Delta v \tag{1}$$

or

$$\Delta p = \rho c \Delta v \tag{2}$$

in which: Δh = pressure rise in m fluid column

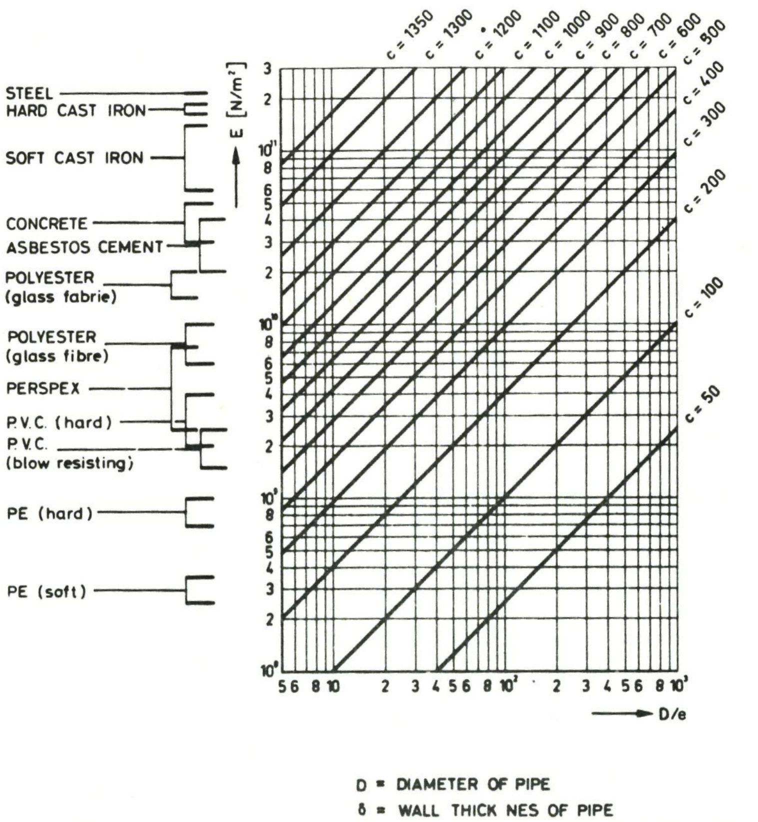
c = wave celerity in m/s

g = gravity acceleration in m/s^2

Δv = velocity change in m/s

Δp = pressure rise in N/m^2

ρ = density of the fluid in kg/m^3.



$$c = \frac{1}{\sqrt{\rho\left(\frac{1}{K} + \frac{D}{Ee}\right)}}$$

Fig. 2 Pressure wave speed in pipelines filled with pure water for $c_1 = 1$

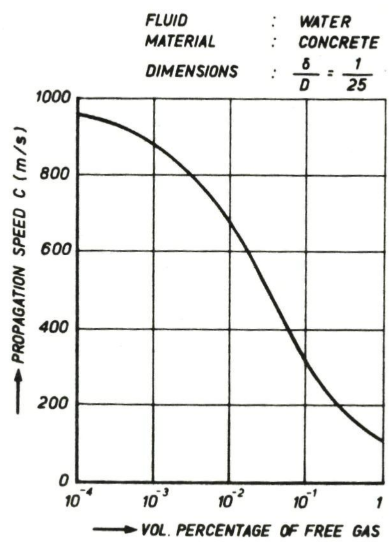


Fig. 3 Relation between pressure wave speed and free gas content

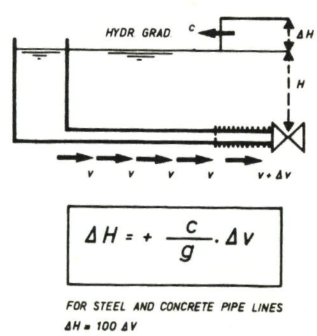


Fig. 4 Joukowsky's law

This means that a relatively small velocity change leads to a relatively high pressure change. For instance a 0,1 m/s velocity change in a steel water pipeline leads to a pressure change of 10 m water column = 1 bar = 10^4 N/m^2 (= 10^4 Pascal).

The foregoing is only useful in a qualitative way, since in practice Joukowsky's assumptions are not often valid:

1. The time in which a velocity change occurs is longer than the critical time μ. This leads in general to pressure changes smaller than those in Joukowsky's law, due to reflections of the pressure waves at the pipe end.
2. The friction losses cannot be disregarded compared with the static head in the pipeline. This leads in general to pressure changes higher than those - in Joukowsky's law.

In the past waterhammer pressures were determined in a quantative way by means of the graphical method developed by Bergeron and others [5]. Today this is done in an efficient way by means of computer programmes.
Both methods are based on the same basic physical laws combined with the necessary boundary conditions. When using computer programmes more complex pipeline systems as well as more complex boundary conditions, such as the occurrence of extended cavitation during the transient condition (see Chapter 2), can be built into the computer programme. The graphical method usually leads to infinitely long calculating times. On the other hand the graphical method is still a good tool in obtaining an insight into the phenomenon of waterhammer.

2 Practice

In practice waterhammer occurs due to controlled or emergency valve operation, controlled stopping or starting of pumps, uncontrolled pump-stopping due to failure in the energy supply, closing of check valves, etc. Then overpressures as well as underpressures occur in the pipeline (see Fig. 5). The underpressure may be as low as the vapour pressure of the fluid. In such a case so-called cavities ("empty" volumes in the pipe, filled with gas or fluid vapour) occur. These cavities may occur in fixed places in the pipeline, whereby over a length of pipe the greater part of the pipe's cross-section is filled with gas or fluid vapour. (concentrated cavitation). Concentrated cavitation may occur just downstream of a closing valve, in high points of the pipeline scheme or at the pressure side of a tripping pump (see Fig. 6). After some time the cavities may disappear (collapse) due to overpressure at the pipe ends, producing a pressure rise at the moment of collapse.

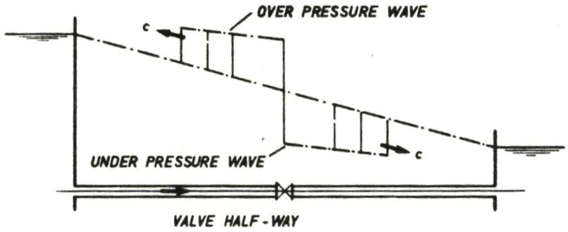


Fig. 5 Pressure waves introduced by valve closure

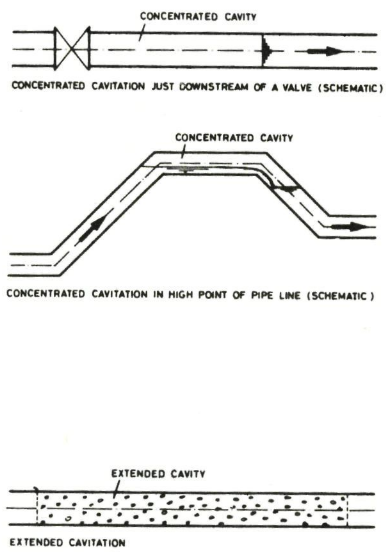


Fig. 6 Cavitation types

It is also possible for vapour pressures to occur in extended parts of the pipeline (see Fig. 6), whereby over a relatively long length of the pipeline only a small part of the cross-section of the pipe is filled with gas or fluid vapour. (extended cavitation). Extended cavitation may occur due to a pump stop, pump failure or the closure of an upstream valve in relatively long transmission lines. During such an operation an underpressure wave is initiated at the upstream end of the pipeline (see Fig. 7). This underpressure travels as a wave through the line. Also extended cavitation, possibly accompanied with concentrated cavitation in high points, occurs and disappears (collapse) after some time due to the overpressure at the pipeline ends (see Fig. 8). This collapse may lead again to overpressure waves.

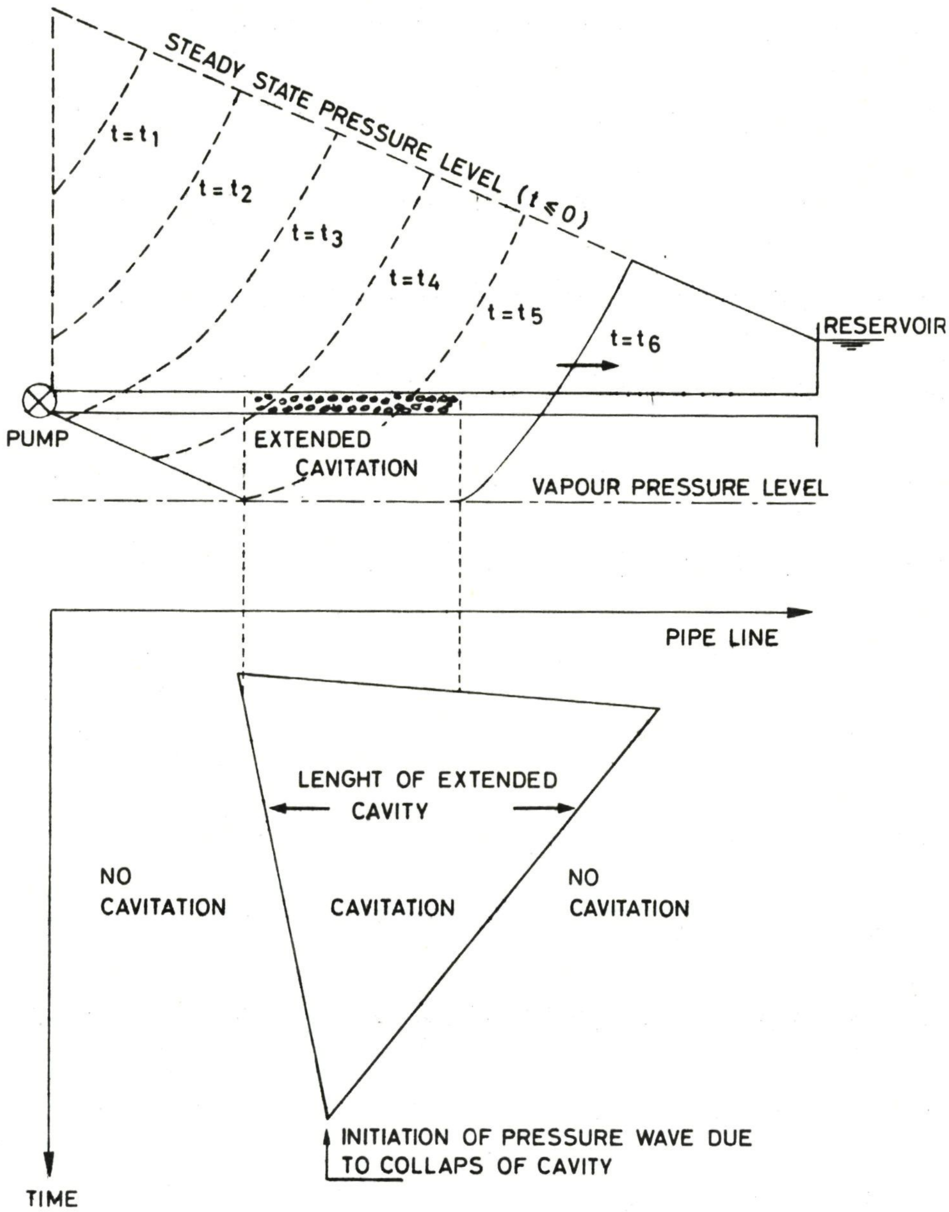
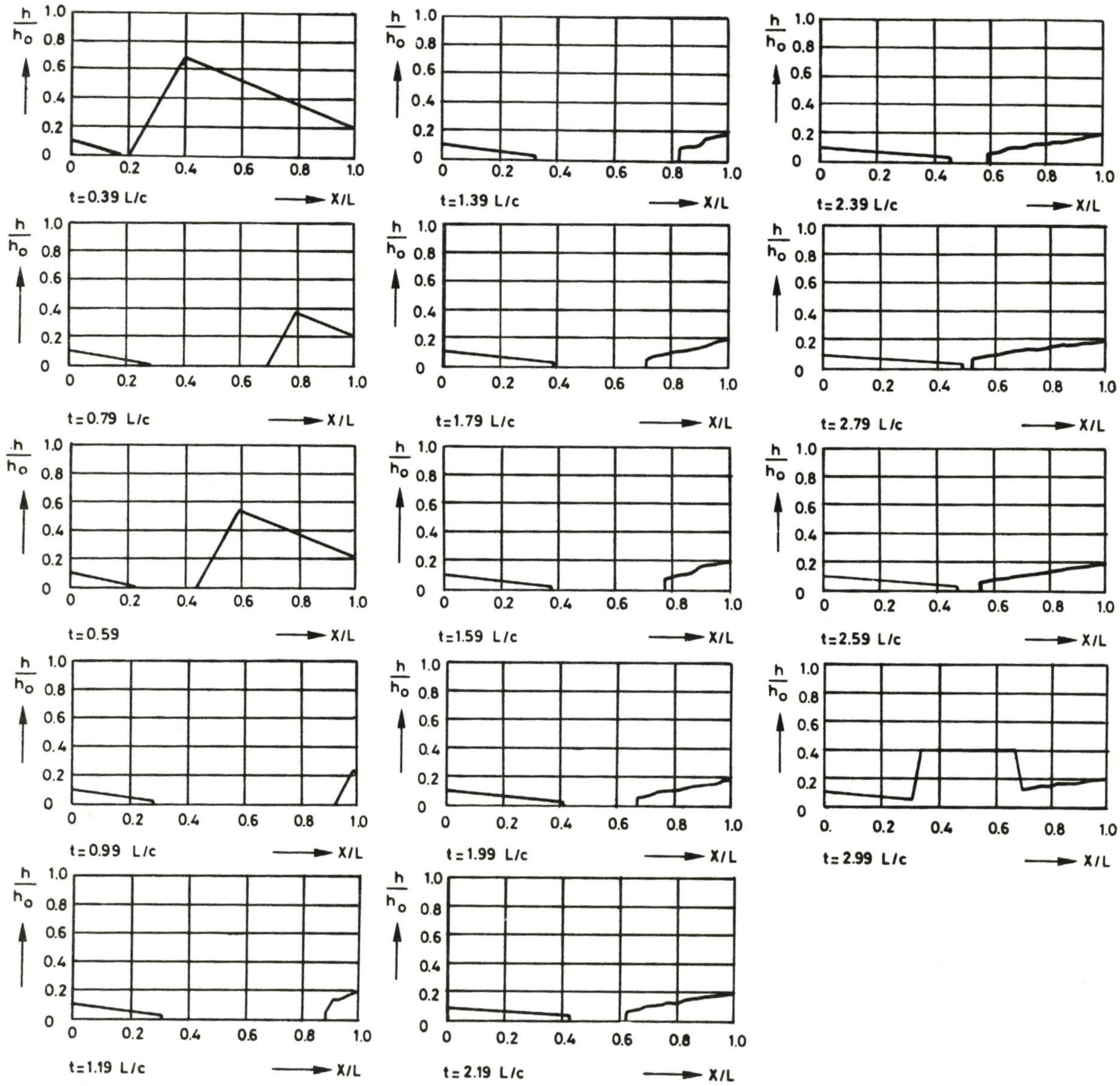


Fig. 7 Pump failure (at t = 0) with extended cavitation



AT TIME t=0 BEGINNING OF PRESSURE DROP AT UPSTREAM PIPE END

Fig. 8 Subsequent pressure levels due to pressure level drop at upstream end of pipeline

It is common practice to determine maximum and minimum pressures in simple pipelines as well as in complex pipe systems with the computer programmes developed over the past 15 years. These programmes are based on the formulas and boundary conditions described in Chapter 5. The consequences of normal operations as well as emergency operations are examined.
Usually the resulting pressures p in the pipeline are presented as pressure heads H as a function of time and place referred to some datum level. The actual pressure in the pipeline is then easily found from:

$p = \rho g H - z$

in which: p = pressure in the pipeline in N/m^2

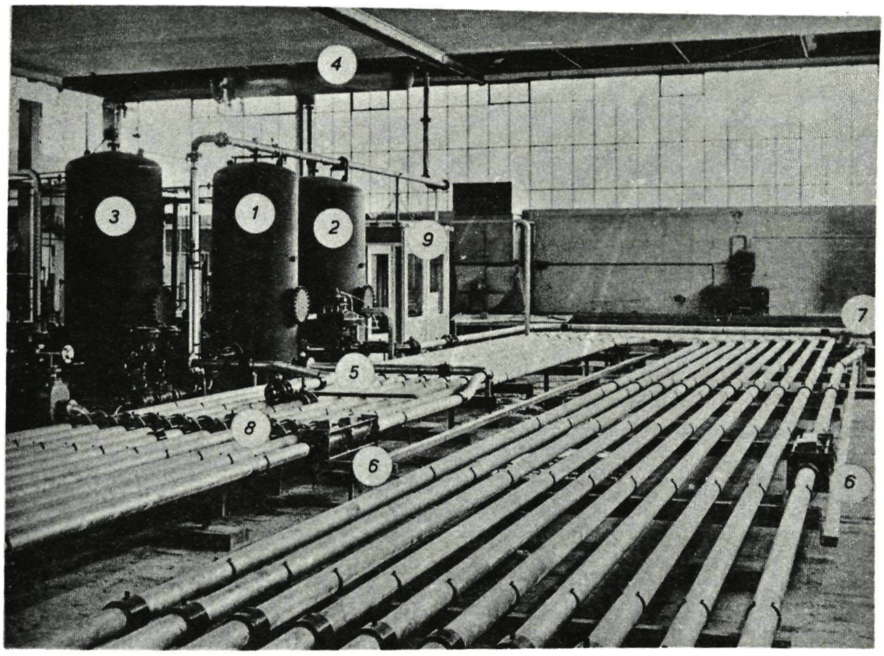
ρ = density of the fluid in kg/m^3

g = acceleration of gravity 9,81 m/s^2

H = pressure head with reference to some datum level in m "fluid column"

z = height of the relevant place in the pipeline with reference to the datum level in m.

In some of these computer programmes the initiation, growth and collapse of concentrated and extended cavities is built in. Both kinds of cavities are then concentrated in the grid points. Furthermore it is assumed that the pressure inside a cavity remains at vapour pressure until its collapse occurs, while no gas release into the cavity takes place.



Test rig for investigation of waterhammer accompanied with concentrated and extended cavitation

1) High pressure upstream tank
2) Low pressure upstream tank
3) Low pressure downstream tank
4) Deaeration tank
5) Coupled ball valves to introduce sudden pressure drop
6) Electromagnetic flowmeters for transient flow
7) Electromagnetic flowmeters for steady flow
8) Plexiglass pipes with pressure cells
9) Instrumentation and control room

This method of waterhammer calculation including cavities was checked in a model pipeline (see photo, l = 1450 m, d = 0,10 m) in which the upstream pressure could be lowered in a very short time from one constant value to another [6], [7]. In this model well defined underpressure waves could be initiated. Also checking took place in two prototype pipelines with l = 27,9 km, D = 1,80 m [8] respectively l = 0,8 km, D = 0,4 m. The resulting maximum and minimum pressures as calculated for a power failure of the pumps in the 27,9 km line are shown in Figure 9. In this figure it can be clearly seen that

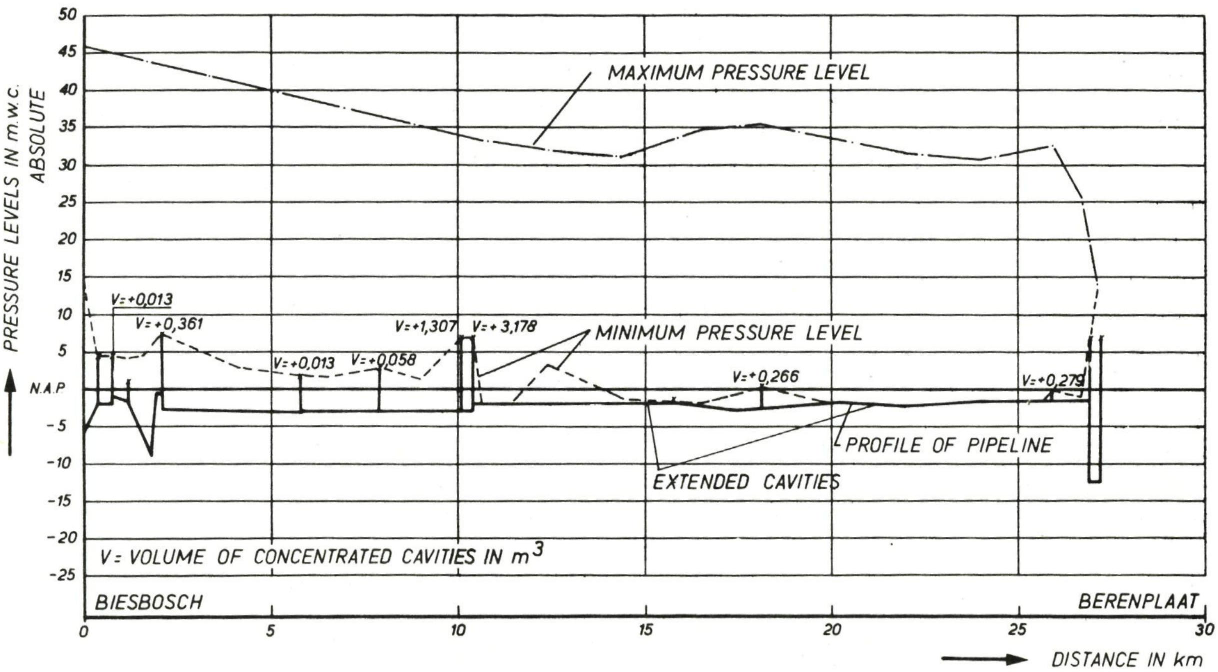


Fig. 9 Extreme pressure heads along the pipeline due to power failure of pumps

concentrated cavities occur in the 'high points" of the pipeline, whereby two extended cavities occur in the downstream part of the pipeline. In this case maximum pressures due to the collapse of the cavities remain below the maximum steady state pressure at the upstream end of the pipeline. Comparison of calculated pressures as a function of time in some relevant points along the line with measured pressures at these points showed good agreement [8]. Some of the results for calculated and measured pressures as a function of time for the power failure of one pump in the 0,8 km-line are shown in Figure 10. In this figure it can be clearly seen that a concentrated cavity occurs just downstream of the failing pump for some 2,5 s. Furthermore calculation and measurement are in good agreement.

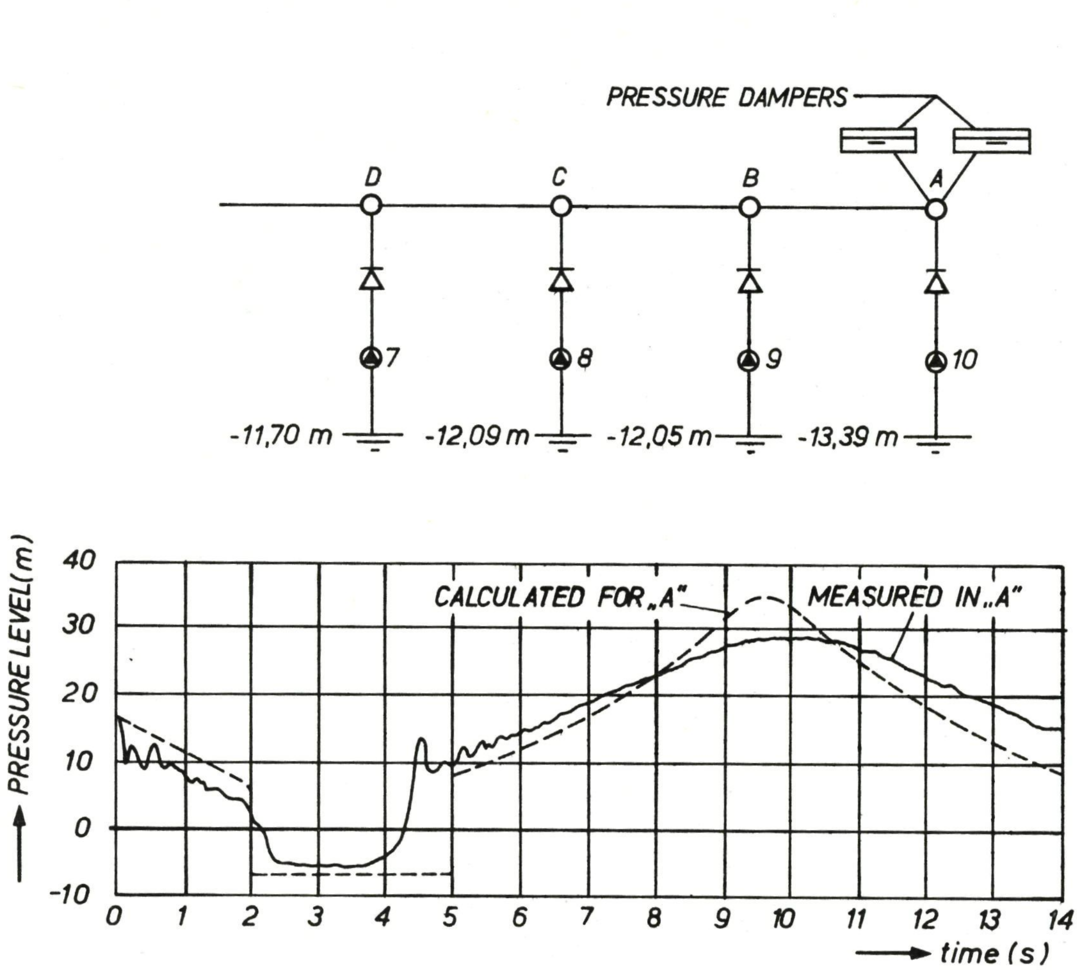


Fig. 10 Pressure level in a point of a pipeline as function of time due to pump failure

From the foregoing it may be concluded that the results of waterhammer calculations with the above mentioned assumptions are on the safe side, whereby the dampening effect is expected from the release of dissolved gases.

The results of waterhammer calculations may indicate that special waterhammer provisions like valve control, air vessel, speed control of pumps etc. are necessary.

The application of waterhammer programmes requires knowledge of the waterhammer phenomenon itself as well as the hydraulic behaviour of pumps, valves, surge towers, air vessels, bypasses, pressure relief valves, etc. during steady state flow and during transient flow. Incidental use of such programmes by non-hydraulic engineers may lead to "strange" results.

3 Criteria

The accepted overpressures and underpressures depend in the first place on the strength of the pipe (safety margins included) and pipe joints. Moreover the acceptable underpressure depends on the type of transported fluid and the way the pipeline is operated. Normal as well as emergency operations such as pump failure and uncontrolled closing of valves have to be taken into account.
The following normal operations can be distinguished:

- continuous flow with incidental gradual start and stop procedures
- gradual changing flow between a minimum and maximum flow with gradual changing and start/stop procedures
- discontinuous flow with regular, rather sudden start and stop procedures.

During normal operations regular cavitation should be avoided since the amount of damage it causes to the pipe wall is not yet predictable.

Fresh and raw water transmission lines are normally operated with continuous flow and incidental gradual start and stop procedures, or with gradual changing flow between some minimum and maximum flow with incidental gradual stop and start procedures. During such normal operating procedures, pressures down to vapour pressure are not accepted, since air which may come out of solution during such low pressures may lead to air pockets in the pipeline and so to higher hydraulic losses, respectively higher energy costs. Moreover these air pockets may be compressed to high pressures during a too fast starting procedure. Sometimes automatic air vents at strategic points of the pipeline scheme are used to prevent this. During emergency stops however (e.g. pump stop due to power failure) pressures down to vapour pressure are accepted*. In some parts of the pipeline air pockets will then occur. Before restarting the pumps this air should be released as gradually as possible through hand-operated or automatic air vents. During the release of air two water columns are flowing with different velocities (see Fig. 11). At the instant that the last air is released and the vent is closed these two columns "hit each other" resulting in two pressure waves.
A too fast release of air through the air vents may lead to very high pressures. Hereafter pumps should be restarted gradually or one by one with ade-

* It should be examined whether overpressures caused by the collapsing cavities are acceptable or not.

quate time intervals, to prevent high pressures due to the compression of any remaining air pockets.

In fresh water lines underpressures are ususally not accepted as the sucking in of water from outside the pipe has to be prevented.

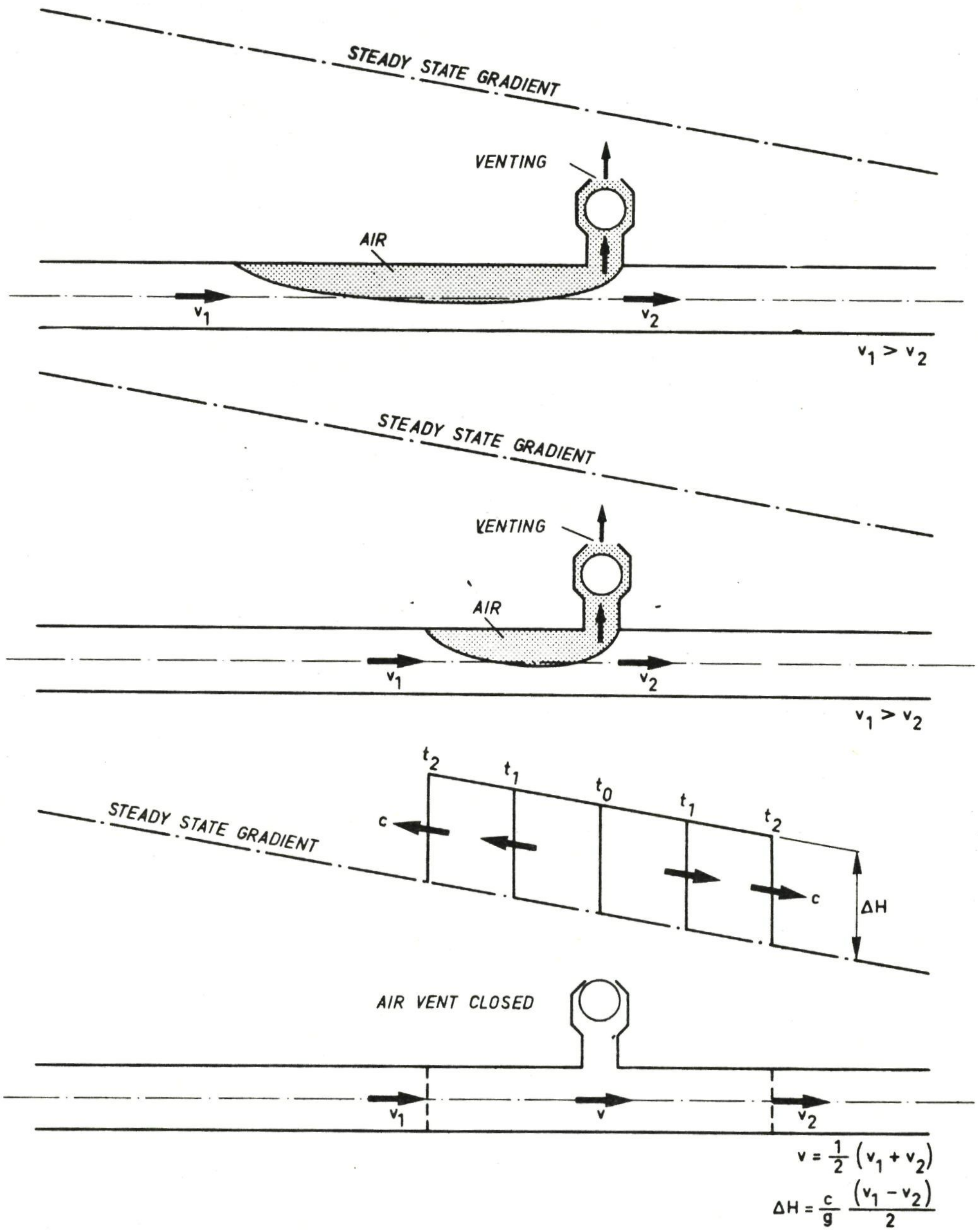


Fig. 11 Pressure waves introduced by air vents

<u>Fresh water networks are normally operated with discontinuous flow</u> with regular, rather sudden, pump start and stop procedures. During such operating procedures pressures below atmospheric pressure are not accepted at all, or only for some seconds. This criterium is used to prevent the sucking-in of water from outside the pipe(s).

Sewage pipelines normally have discontinuous flow with regular, rather sudden start and stop procedures: Pumping action changes stepwise depending on the water level in the suction sump. The maximum accepted underpressure is rather arbitrarily restricted to -7 water column (3.10^5 Pa abs pressure), since it is not predictable how much gas will come out of solution at lower pressures nor at which place of the pipeline scheme important gas pockets will be formed. With this "-7 m" restriction it is expected that cavitation and important gas pockets will not occur, nor that the starting up of pumps will lead to unacceptable overpressures due to the compression of any small gas pockets.

Main cooling-water pipelines in electric power plants are normally operated with continuous flow with incidental, rather sudden, changing, starting and stopping procedures. Such pipelines (see Fig. 12) contain condensors in which top underpressures of up to -8 m water column (2.10^5 Pa abs. pressure) may often occur during the steady state [9]. Air which is coming out of solution due to such low pressures is continuously released at the top of the condensor by means of the turbine vacuum of the power plant. During stopping and changing procedures or emergency pump failure vapour pressure is reached very easily at the top of the condensor, resulting in a concentrated cavity there.

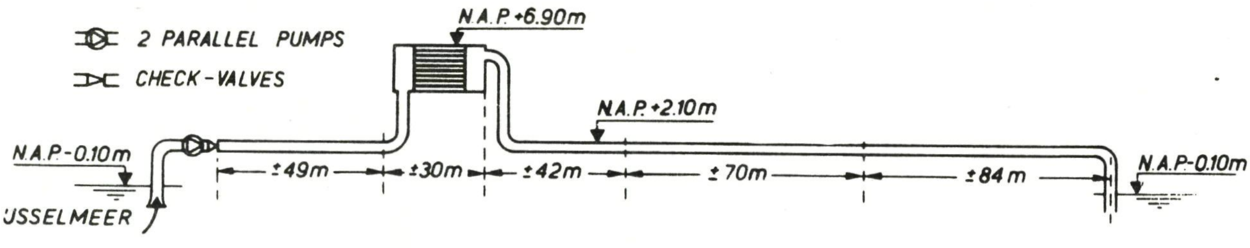


Fig. 12 Scheme of a typical cooling water circuit of an electrical power plant

Depending on the position of the condensor in the pipeline scheme, the length of upstream and downstream piping and the amount of free air in the fluid, the collapse of such a cavity may lead to unacceptable overpressures. If so special provisions have to be taken (see Chapter 4). In general the pumps should then be restarted or flow should be raised again after complete filling of the condensor (e.g. by means of vacuum pumps or the vacuum of another running turbine). Gradual rise of the pumps or gradual rise of the flow without preceding deaeration may also be possible in practice on a trial and error basis.

District heating networks and their eventual feeding transmission lines from heatpower-producing plants are normally operated with continuous flow and gradual changing flow at more or less regular intervals (winter peak demand, daily peak demand). In such cases the gradual change in flow is generated by means of variable speed-operated pumps. During such operations pressures in the network and transmission lines should remain above the vapour pressure of the water at the prevailing temperature. This is to prevent vibration and noise due to collapsing cavities. In most cases vapour pressures will not occur due to such operations, since the steady state pressures in all parts of the network and transmission lines are normally well above some 4 bar. On the other hand emergency closure of valves near the pumps may lead to vapour pressures in the network or transmission lines. This will happen when the connecting lines between expansion tower or expansion vessel, which act as a kind of surge provision, are too long or the diameter too small. However, such cavities often collapse very gradually and do not lead to unacceptable high pressures.

Crude oil transmission lines and crude oil pipelines respectively chemical pipelines for loading and unloading ships are normally operated with continuous or gradual changing flow. Normal start and stop procedures of pumps and valves are chosen in such a way that overpressures remain acceptable, while vapour pressures are not allowed to occur since not much is known about the amount of gas which is coming out of solution during such low pressures. During an emergency pump stop because of power failure or an emergency quick closure of upstream valves, vapour pressures are acceptable when based on the results of waterhammer calculations which include the initiation, growth and collapse of the cavities which occur.
Restarting of the pumps should be done gradually in order to prevent too high pressures due to the compression of the gas pockets which occur. In most cases emergency valve closure at the downstream end of the pipeline leads to unacceptable overpressures. Special measures or provisions are then needed to avoid this [10].

Most of the above-mentioned pipelines are equipped with one or more non-return valves. Such valves prevent reverse flow and are often used in combination with pumps. When pumps are switched off the flow through them will decelerate to zero. If the non-return valves are not closed at the instant of zero flow, the flow will reverse. The non-return valve will than close during reverse flow, possibly resulting in very high pressures and slamming or even damage to the valve.

4 Provisions

Based on waterhammer calculations it can be decided whether "provisions" are necessary. These provisions are applied because of one or more of the following reasons:

1) To prevent unacceptable overpressures, based on the strength of the pipe or anchorage.
2) To prevent underpressures, to avoid the suction of unwanted fluid from outside the pipeline into the pipeline.
3) To prevent unacceptable underpressures based on the buckling strength of the pipe.
4) To prevent unacceptable underpressures to avoid the growth of gas pockets.
5) To prevent regular occurrence of concentrated or extended cavitation, to avoid cavitation erosion.
6) To prevent concentrated or extended cavitation, regular as well as incidental, where the collapse of the cavities leads to unacceptable overpressures.

The provisions are all based on the principle of obtaining a more gradual velocity change in the pipeline (which leads to the desired lower pressures). In other words, to obtain a more gradual flow change.

For normal valve and pump operations this often means control of valves and pumps. This control can be derived from the waterhammer calculations. That is to say checking whether a certain chosen control satisfies the requirements. Too fast velocity changes due to quick emergency closure of valves or tripping of pumps due to power failure, are made more gradual by means of a provision which "takes over" or "supplies" part of the pipeline flow. Depending on the type of pipeline the following provisions may be used: surge tower, air vessel (with air valve only or with air valve and compressor) bypass around pump, flyweel on pump, pressure relief valve, pump and valve control initiated by means of a "near by" pressure switch or by means of a radio signal from a "far off" pressure switch.

<u>The working principle of a surge tower</u> is shown in Figure 13. After a power failure of the pumps, pump discharge stops at once (order of 1 s). The flow into the pipeline is taken over from the pumps by the surge tower. The discharge from it into the pipeline (and so the flow in the pipeline itself) gradually decreases due to the pipe wall friction and the gradual lowering of

the level in the surge tower. When quick closure of a valve occurs at the downstream end of a line, fluid flows from the pipeline into the surge tower instead of through the valve. Gradual decrease of the flow into the surge tower (and so the flow in the pipeline itself) occurs due to the gradual rise of the level in the surge tower.

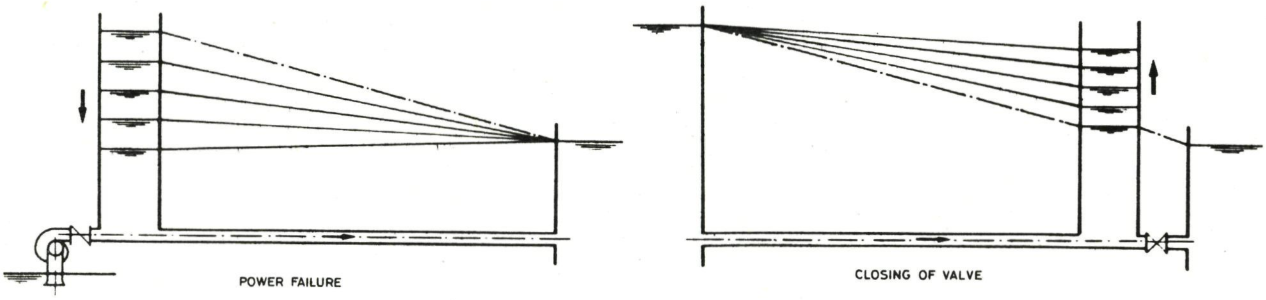


Fig. 13 Effect of a surge tower

Some of the results of a waterhammer study performed in relation to a pump failure in a sewage transmission line with surge tower are shown in Figures 14-16. Because of the relatively small diameter of the tower (2 m) a large reservoir (diameter 4 m, length 25 m) is necessary to prevent complete emptying of the tower. In this way sucking of air into the transmission line is prevented. The lowest pressure in the transmission line is an underpressure of 0,5 m water column ($9{,}5.10^5$ Pa absolute pressure) at the upstream end of the pipeline. Also it can be clearly seen that flow in the transmission line is very gradually lowered in 200 s, while fluid is delivered by the surge tower (see Fig. 15). The flow through the pumps and non-return valves is lowered to zero in 1,3 s (see Fig. 16). In such a case an ordinary flapper, non-return valve with counterweight can be used. However, for bigger pipe diameters and shorter pipe lengths between the pumps and the surge tower, the flow will be reduced to zero in a much shorter time. Ordinary flapper non-return valves will then close in reverse flow accompanied by slamming and damage. To prevent this such flapper non-return valves should be equipped with an oildashpot, which retards the last part of the closing reach of the valve, or another type of valve should be used.

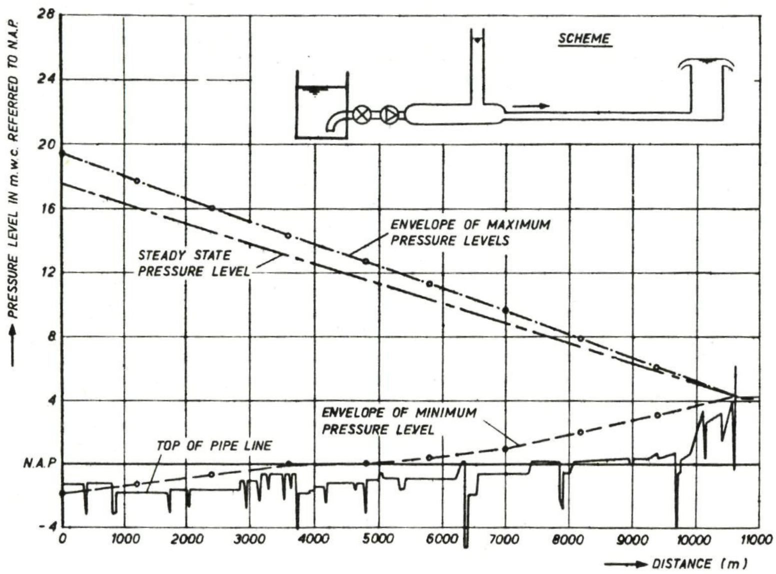


Fig. 14 Extreme pressure levels in a pipeline protected with a surge tower

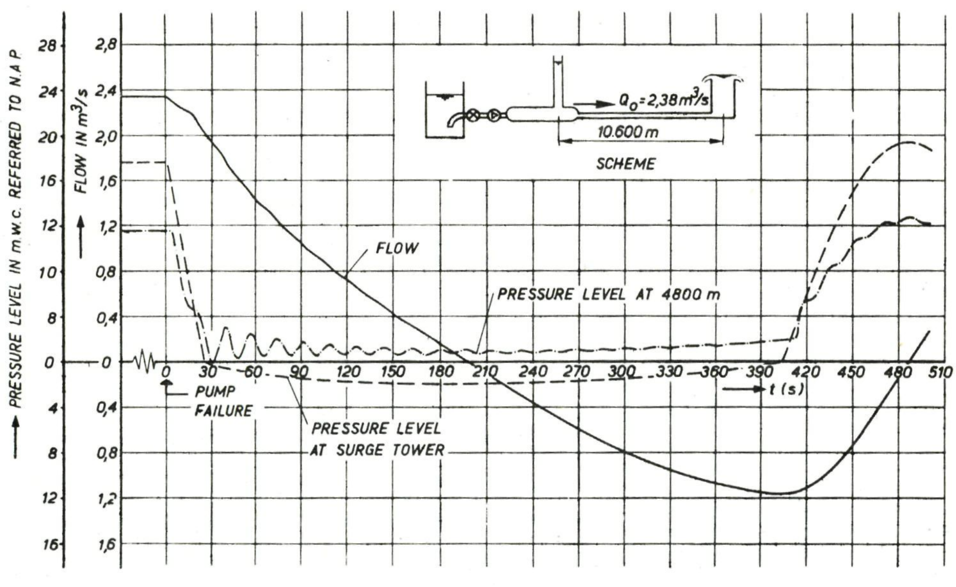


Fig. 15 Flow and pressure level history in a pipeline protected with a surge tower for power failure of pumps

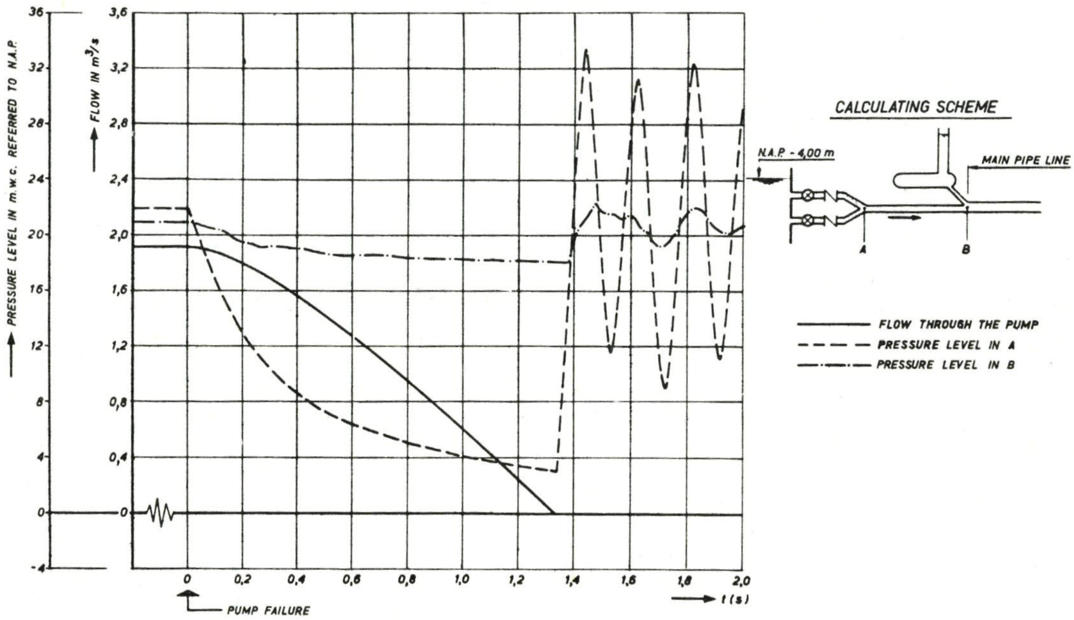


Fig. 16 Flow and pressure level history in pipeline between pumps and surge tower, for power failure of both pumps

An air vessel has the same working principle as a surge tower. In this case the fluid which has to be delivered to the pipeline, or has to be stored from the pipeline, remains at a low level covered with an aircushion (see Fig. 17).

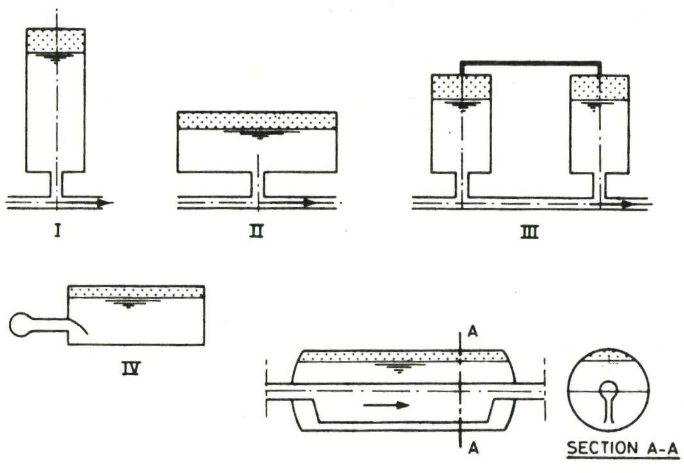


Fig. 17 Some types of air vessels

This air cushion is compressed or decompressed during the transient flow in the pipeline. Some results of a waterhammer study performed in relation to pump failure in a sewage transmission line with two air vessels (L = 7 m, D = 3 m) are shown in Figures 18 and 19. The lowest pressure in the line (2,5 m.w.c. below atmospheric pressure) is reached for the adiabatic expansion of the air in the air vessel. The lowest water level in the air vessels (0,75 m above outlet opening) is reached for the isothermic expansion of the air in the air vessel. Furthermore it can be clearly seen that the pressure level in the line gradually reaches its lowest level in some 2,5 min.

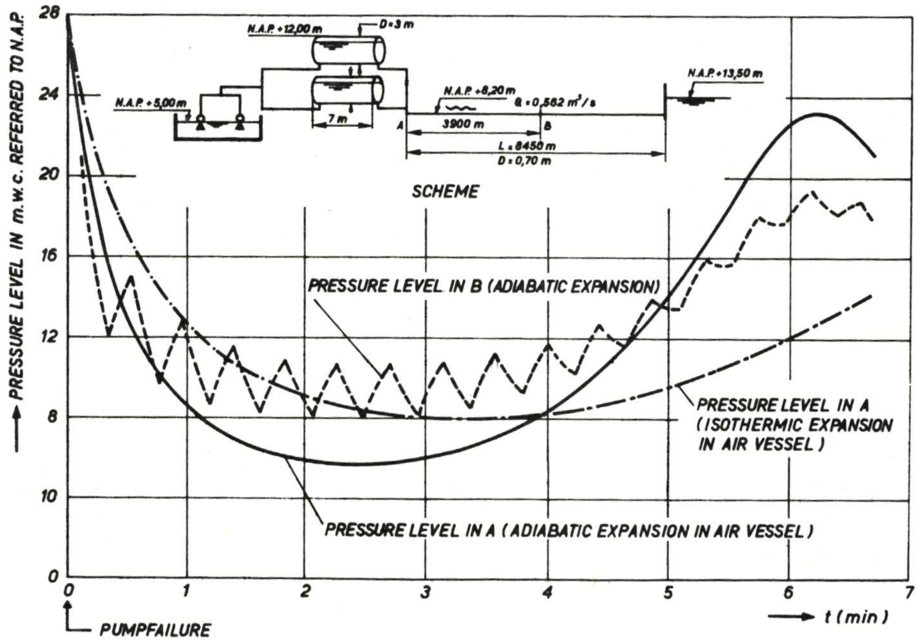


Fig. 18 Pressure level history after pump failure in a pipeline equipped with air vessel

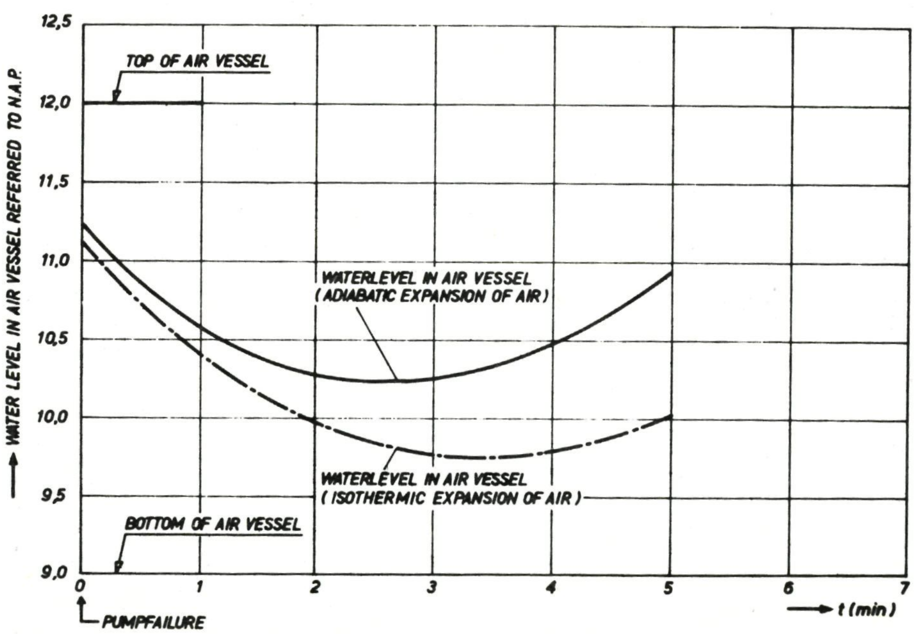


Fig. 19 Water-level history after pump failure in an air vessel

Sometimes a flyweel on the pumps or a bypass around the pumps are used to prevent too low underpressures during pump failure. Due to the inertia of pump and flywheel the energy, which the pump puts into the fluid, is gradually lowered. This results in the gradual decrease of the flow in the pipeline. When a bypass around the pumps is used, the pressure level at the upstream end of the pipeline will not fall below the suction level of the pumps. Hereby fluid is sucked from the suction reservoir into the pipeline. In this way again the flow in the pipeline gradually decreases.

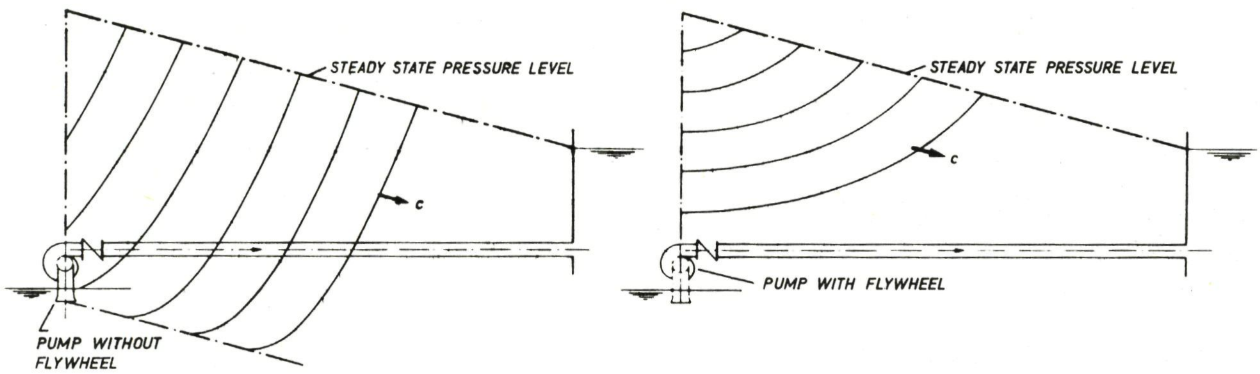


Fig. 20 Effect of flywheel

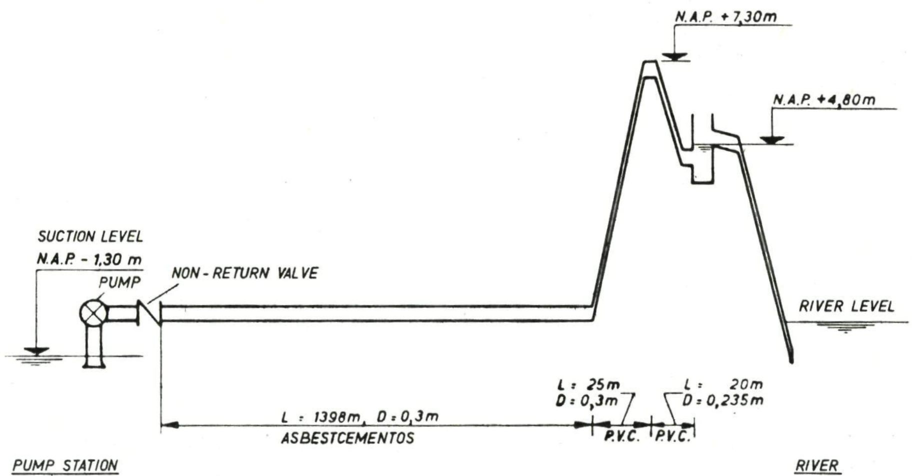


Fig. 21 Flywheel use in practice

The effect of a flywheel is shown in Figure 20. Without a flywheel vapour pressures may occur in the pipeline. With a flywheel the flow decreases more gradually. Only small underpressures then occur due to the reflection of the underpressure wave against the downstream reservoir. An example of a pipeline equipped with a flywheel on the pump is shown in Figure 21. Some of the results of waterhammer calculations carried out in relation to the power failure of a pump in a pipeline are shown in Figures 22 and 23. It can be clearly seen that the flow decreases to zero in about 25 s. A simple flapper type non-return valve can be used here. Flywheels are economic and feasible for pipelines up to a length of 3 km.

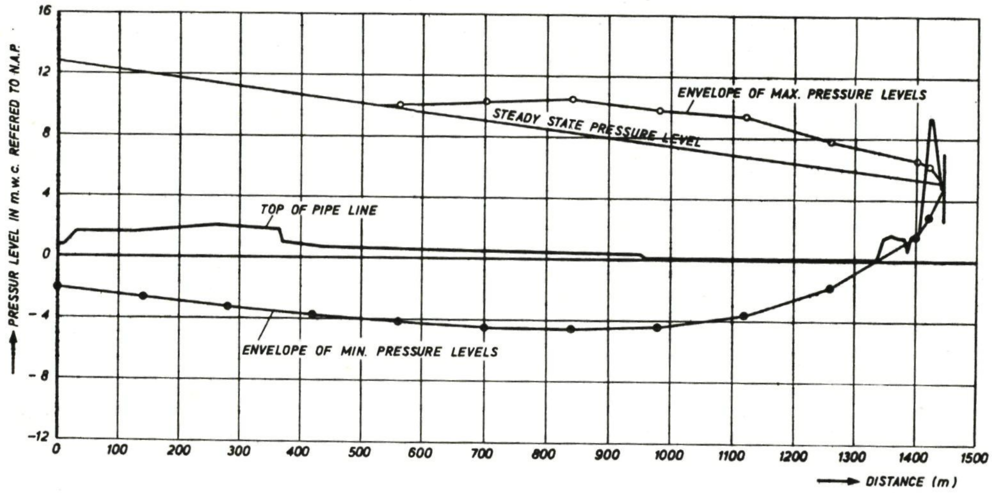


Fig. 22 Envelopes of maximum and minimum pressure levels in the pipeline with flywheel provision 0,75 kgm^2

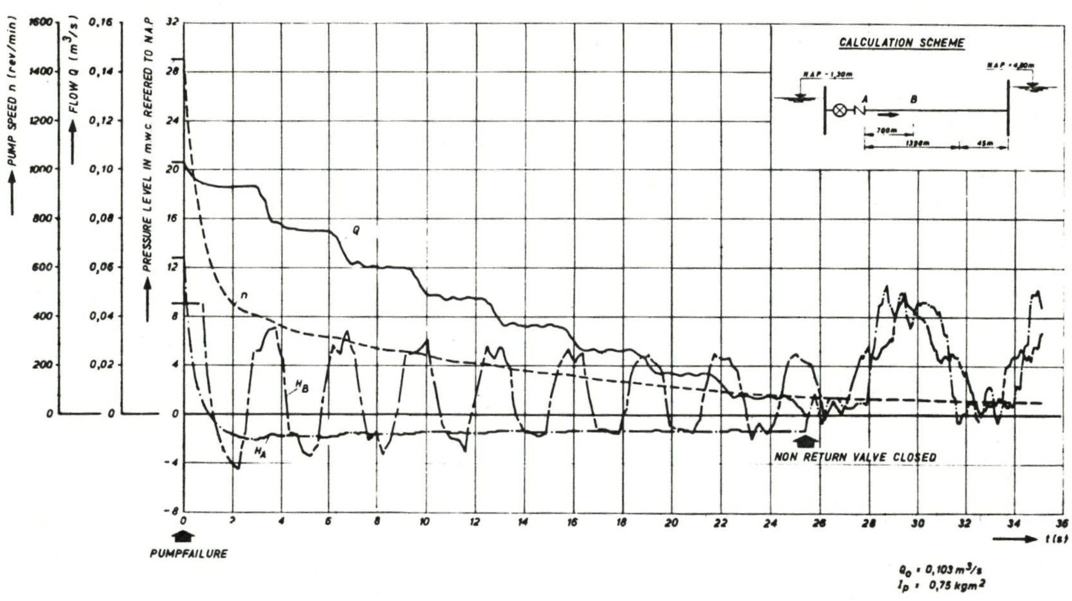


Fig. 23 Time history of flow, pressure level and pump speed for pipeline with flywheel provision 0,75 kgm^2

Pressure relief valves provided with storage tanks can be used to protect crude oil and chemical pipelines against too high overpressures [10]. If the pressure in the pipeline has reached the "setpressure" of the relief valve, the relief valve opens and fluid flows from the pipeline through the relief valve into the storage tank. After that the pressure in the line will rise over the setpressure depending on the hydraulic resistance of the relief valve and the number of relief valves. Two principles for relief valves are shown in Figure 24. It is very important that such provisions are able to discharge the required amount of fluid at the selected pressure limit. Moreover they have to react quickly and their connecting pipe with the main pipeline has to be short. In fact this provision has the same working principle as the surge tower and the air vessel: fluid is flowing from the pipeline into a storage tank and the flow in the pipeline is decreasing gradually due to the (accepted) pressure rise in the line up to the pressure at the relief valve. The required type of valve and the total number required are checked by waterhammer calculations.

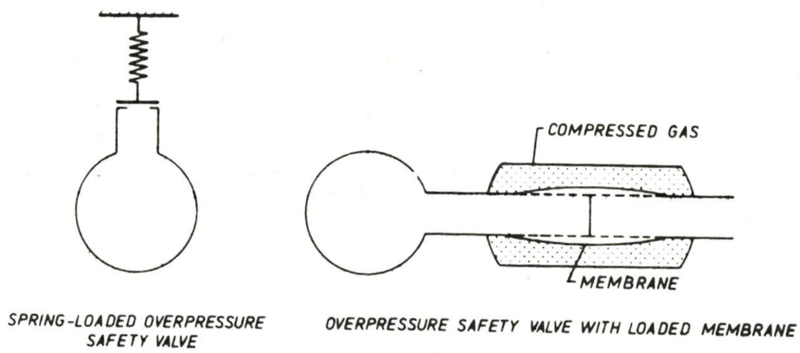


Fig. 24 Pressure relief valves

Some of the results of a waterhammer study carried out for a crude oil transport and loading system are shown in Figures 25-30. The line (see Fig. 25) has a length of 25 km, diameter 1,2 m and maximum flow of 5 m^3/s. The effects of an emergency closure of the ship's valves in 15 s, resp. 40 s and 60 s, was studied [10].

Where necessary feasible provisions had to be chosen and checked by calculations. Without any provisions it was calculated that with a 15 s-ship valve closure an overpressure of 660 m oil column (60 bar) would occur. Moreover extended caviation in the pipelines from the North and South Tankfarm would occur. To prevent this the following provisions were chosen and checked (see

Fig. 26): Pressure relief valves with storage tank on the sea island with a relief pressure level of +232 m oil column (25 bar)*, pressure switch and radio signal in the pipeline at the sea island to shut down the pumps at a pressure level of +212 m oil column (23 bar) at the sea island, and pressure relief valves on the suction side of the pumps with a relief pressure level of +175 m oil column (19 bar).

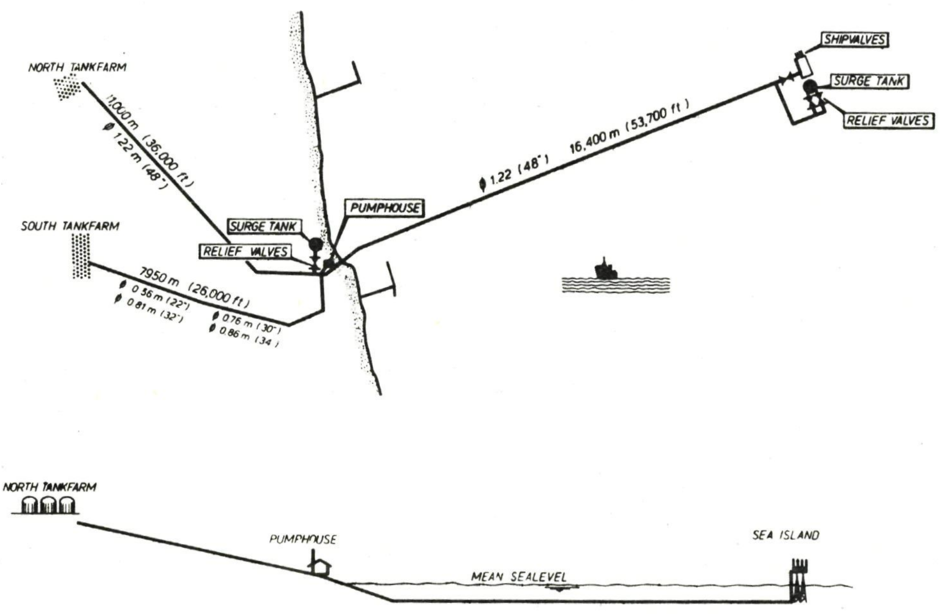


Fig. 25 Scheme of the line system

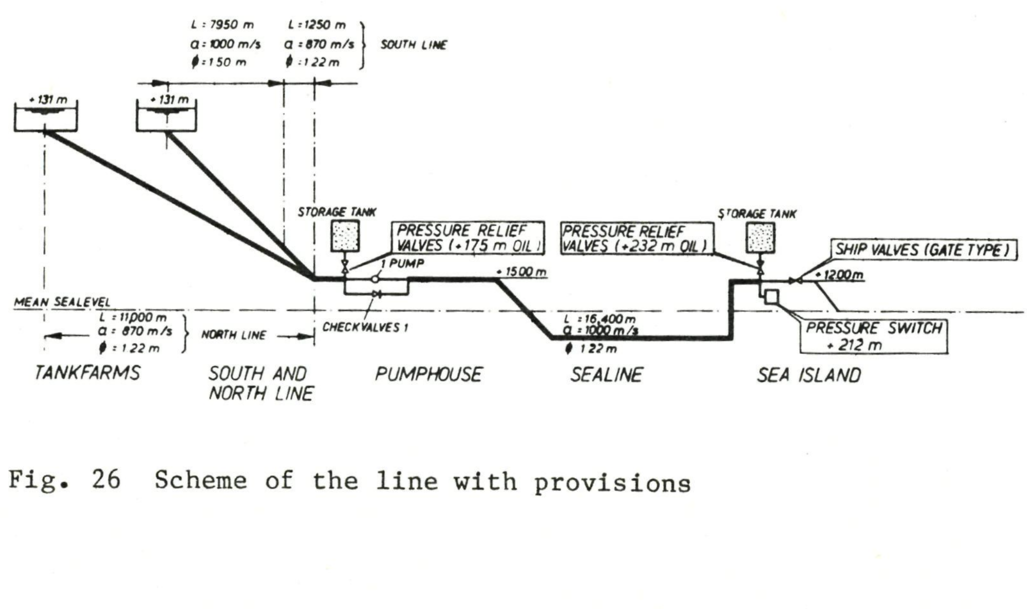


Fig. 26 Scheme of the line with provisions

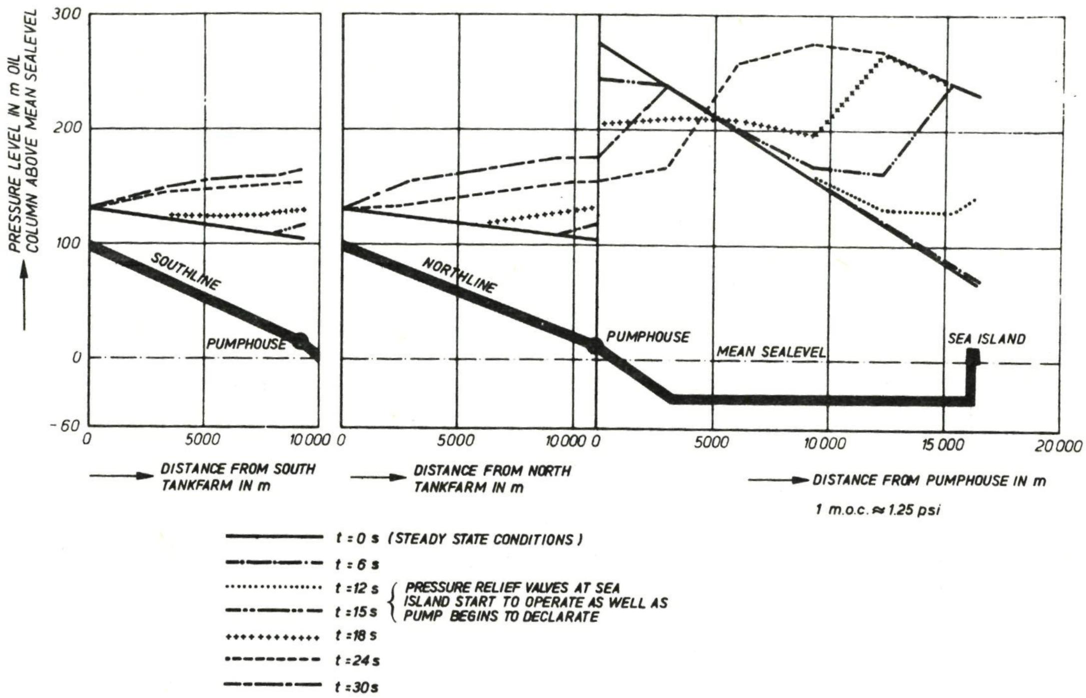


Fig. 27 Pressure levels along the lines during first 30 s after closure of ship valves for t_v = 15 s and t_p = 15 s

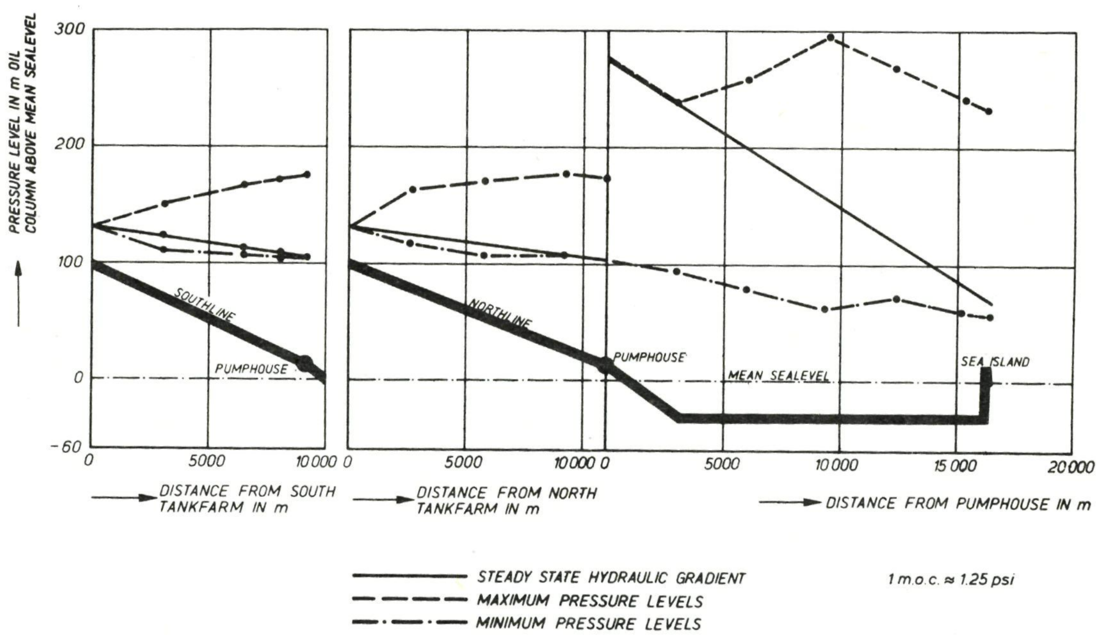


Fig. 28 Envelope of maximum and minimum pressure level for t_v = 15 s and t_p = 15 s

Subsequent pressure levels in the pipe system for valve closure and pump stop both in 15 s are shown in Figure 27. The evelope of maximum and minimum pressure levels are shown in Figure 28. From Figure 29 it can be clearly seen that some 12 s after the beginning of valve closure, the pressure relief valves on the sea island start to operate for about 1 min. Also the pressure switch signals the pumps to shut down. Figure 30 shows clearly that the flow through the ship's valves is taken over by the relief valves. Note that this commences when the ship's valves are closed for appr. 75%. The dashed area has to be stored in the storage tanks.

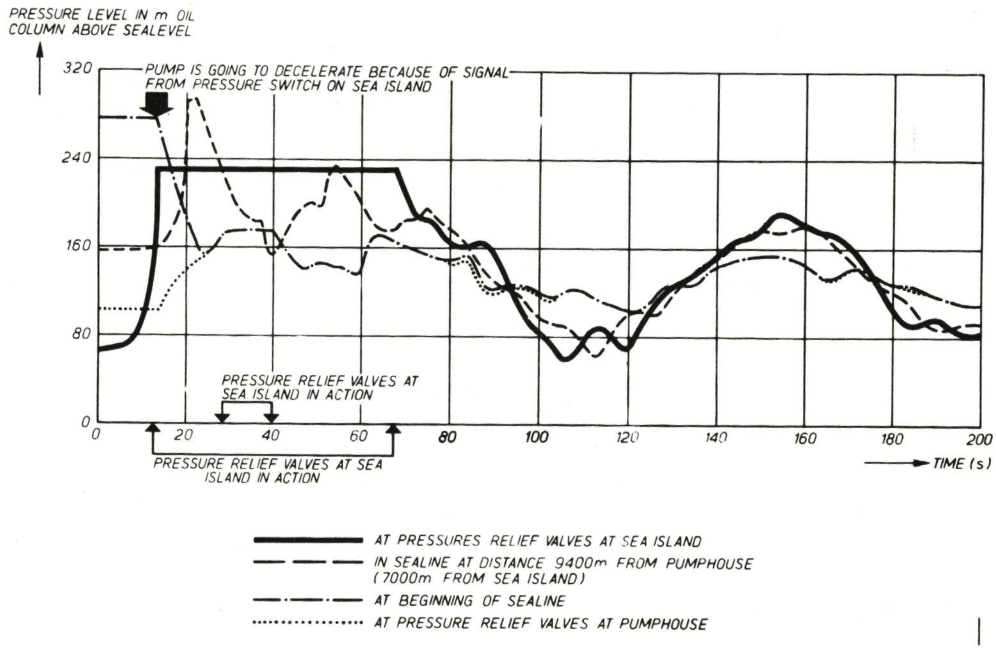


Fig. 29 Pressure level time histories in some points of the lines for t_v = 15 s and t_p = 15 s

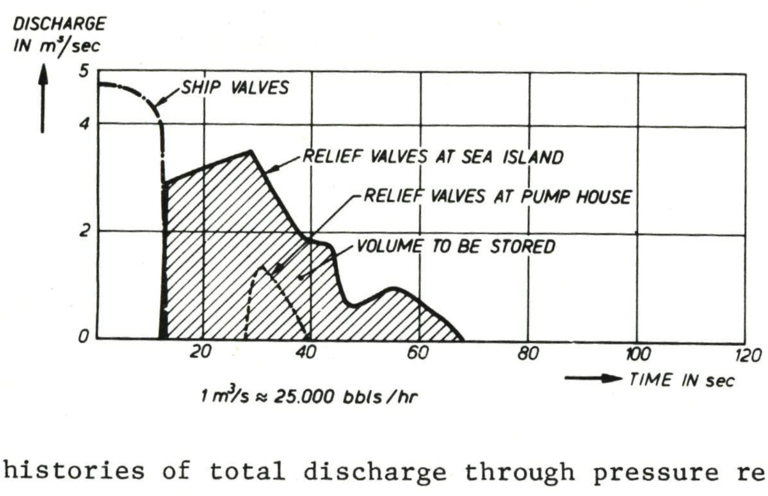


Fig. 30 Time histories of total discharge through pressure relief valves and ship valves for t_v = 15 s and t_p = 15 s

* In the calculations hydraulic losses of the relief valve were disregarded. Therefore when the relief valve is operating the pressure level in the pipeline is considered to be constant at the value of 232 m oil column.

In all the above mentioned types of pipelines slamming or damage of non-return valves may occur due to closure in fast reverse flow following from a pump stop or power failure. This can be prevented by the application of fast-acting valves which are almost completely closed at the instant of zero flow, or by application of non-return valves with special devices which prevent fast closure of the valve during reverse flow. In the latter case some reverse flow through the valve occurs and must be accepted. Different types of valves are shown in Figures 31-33.

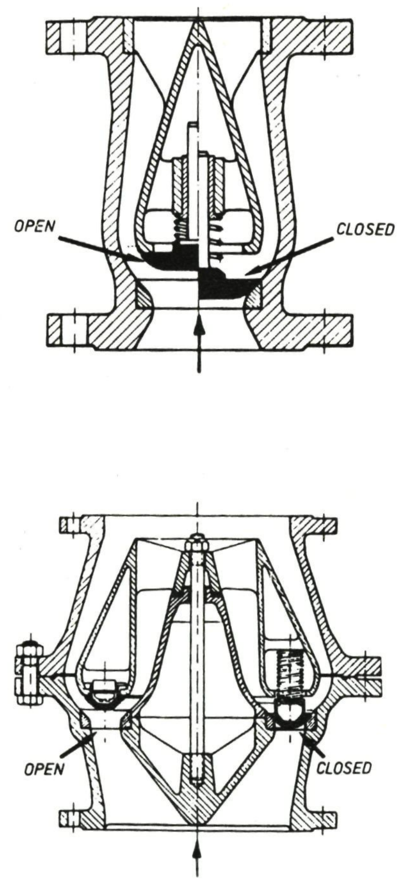
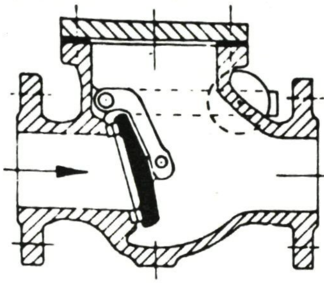
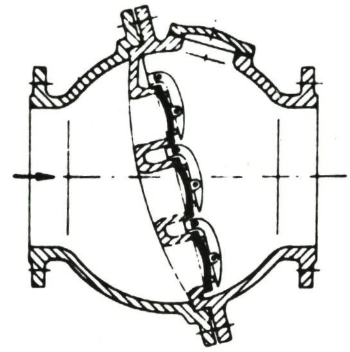


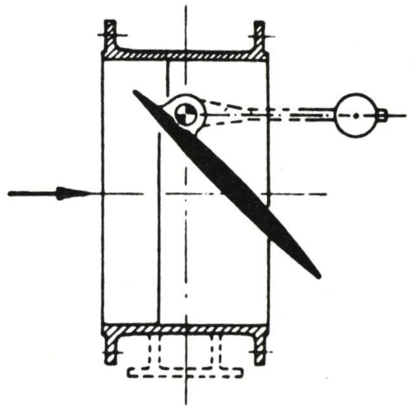
Fig. 31 Examples of fast-acting, non-return valves



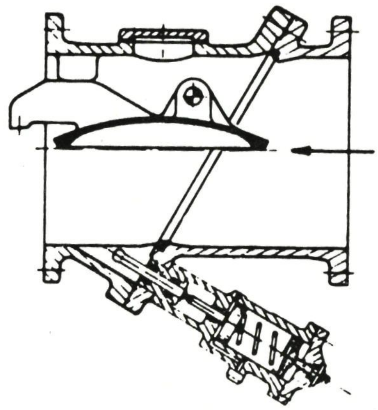
FLAPPER TYPE WITH WEIGHT
(MAX. ca. Ø 1,00 m)



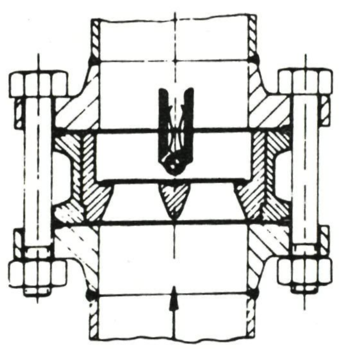
FLAPPER TYPE FOR LARGE
PIPE DIAMETER



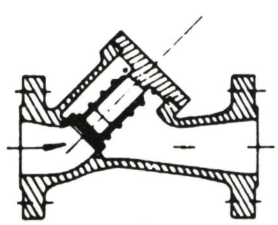
EXCENTRIC BUTTERFLY TYPE
WITH WEIGHT



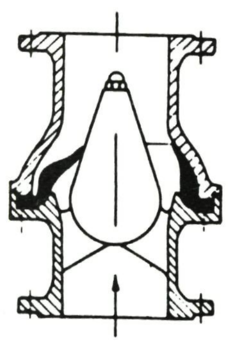
EXCENTRIC BUTTERFLY PIPE
WITH OIL BRAKE SYSTEM



SPRING LOADED FLAPPER TYPE



SPRING LOADED PISTON TYPE



RUBBER MEMBRANE TYPE

Fig. 32 Different types of non-return valves

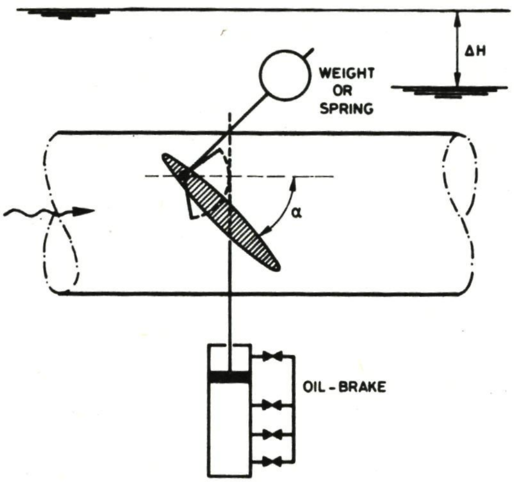


Fig. 33 Excentric butterfly type non-return valve with weight and oil-brake system

By applying Newton's Law to the fluid in pipeline L where the pumps and non-return valve is situated (see Fig. 34) an impression can be obtained regarding the amount of time in which the flow decelerates.

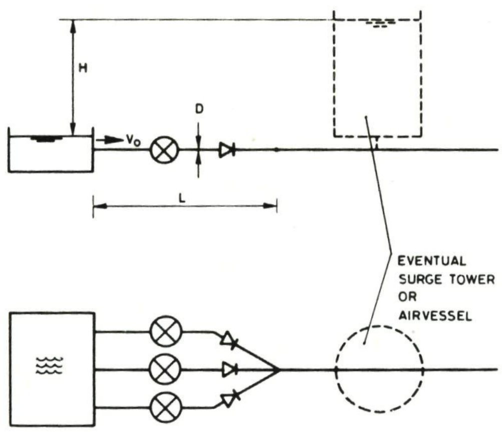


Fig. 34 Situation of non-return valves in pipelines

If friction in this line is disregarded, the differential pressure head "H" is kept constant and the pumps and check valves are omitted, a deceleration of the flow results in:

$$\frac{dv}{dt} = \frac{gH}{L} \tag{3}$$

in which: v = velocity in the line L

t = time

g = acceleration of gravity

H = differential pressure head

L = length of the line

or to a time in which the flow decelerates from the original velocity to zero:

$$t_o = \frac{v_o L}{gH} \tag{4}$$

in which: t_o = time in which flow decelerates to zero

v_o = original velocity.

These formulas lead to very high decelerations dv/dt and consequently to very short times t_o. For instance with H = 40 m water column, v_o = 3 m/s and L = 10 m the value of $t_o = \frac{3.10}{10.40} \simeq 0,1$ s.
In reality the pressure head difference H diminishes and the tripping pumps deliver still energy to the fluid by the inertia of the rotating mass of the pumps. Both effects lead to somewhat lower values of dv/dt and so to somewhat longer times t_o (see further Chapter 4).

Ordinary flapper type and butterfly type non-return valves can be made "faster" by replacing the closing weight by a spring; or they can be made "slow" by an oil-brake during the last part of the closing reach. Such oil-brakes consist of an oil-filled cylinder with one or more bypasses around the plunger (see Fig. 33). Adequate design and adjustment of such oil-brakes is very important. Faulty design or adjustment may lead to negative effects (louder slamming, more damage). Unfortunately until now such adjustments have to carried out in practice on a trial and error basis.
The following two "methods" aid in the better selection of the type of non-return valve to be used [11]:

1. <u>Detailed waterhammer calculations</u> are made for a pump stop for the pipeline part pump - non-return valve - surge tower (or pump - non-return valve - meeting point with other pump lines, pump - non-return valve - air vessel etc. see Fig. 34). For this purpose the hydraulic characteristics of the pump, non-return valve and pipeline have to be known for forward as well as

for reverse flow. The hydraulic characteristics for the pumps are estimated by interpolation from literature [12]. Unfortunately such data are often unavailable for non-return valves. They can be measured in a special valve test rig at the factory or in an independant hydraulics laboratory [13]. Furthermore the mechanical characteristics of the non-return valve such as weight, spring force or oil-brake force acting on the valve body have to be known in the function of its travel rate and travel speed.

The results of the calculations give information about the deceleration of the flow, the pressures occurring and the motion and closing speed of the non-return valve. With these results it is possible to decide whether the selected non-return valve will satisfy or not.

This method is expected to be most suitable for relatively slow closing non-return valves. Some reverse flow will occur in such cases.

2. Simple waterhammer calculations are made for a pump stop for the pipeline part pump-surge tower (or pump-meeting point with other pump lines, pump-air vessel etc. see Fig. 34). For this purpose the hydraulic characteristics of the pump and pipeline have to be known for forward as well as for reverse flow. The non-return valve is omitted in the calculations. The results of the calculations give information about the rate of deceleration of the flow in the pipeline. These results are compared with the results of measurements done with the relevant non-return valve in a special non-return test rig [14], [15].

   In such a test rig fast decelerating flow can be introduced. From the measurements a relation can be found between the rate of deceleration of the flow and the maximum reverse flow which occurs (see Fig. 35). Using Joukowsky's law (2a or 2b), in which Δv stands for the maximum reverse velocity belonging to the calculated rate of deceleration, the maximum pressure in the pipeline can be deduced. As such the maximum value of reverse flow gives an indication of the speed at which the non-return valve will hit its seat.

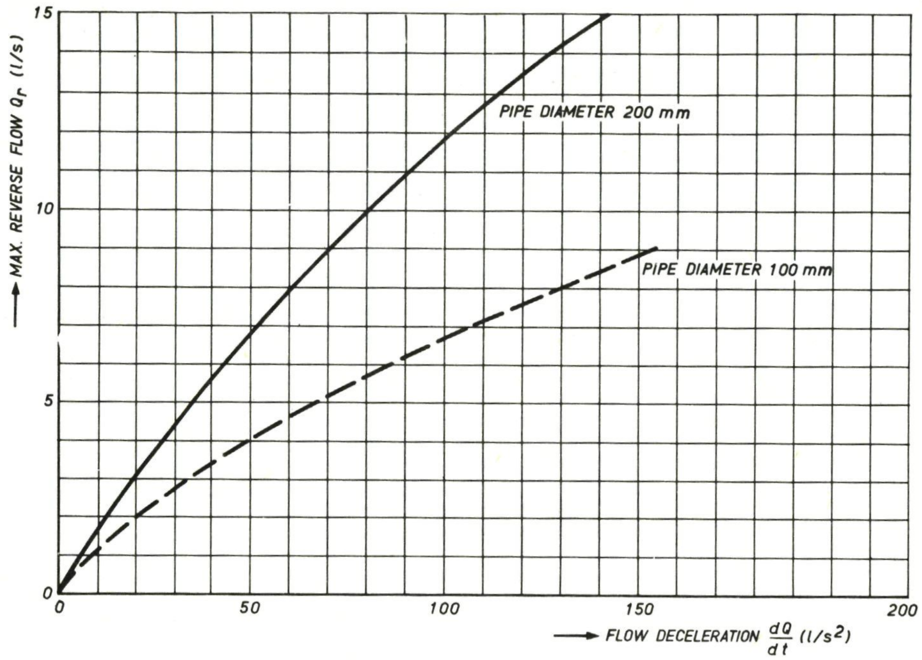


Fig. 35 Typical dynamic characteristic of a non-return valve

This method is expected to be most suitable for relatively fast closing non-return valves.
A summary of considerations when carrying out a waterhammer study is given in Table 1.

Type of pipeline	Way of operation		Acceptable overpressure based on....	Acceptable underpressure based on.....	Vapour pressure with cavitation accepted	Provisions	Remarks
clean and raw water trans- mission lines	continuous flow or gradual changing flow between min. and max.	incidental pump stop/ start and valve closure	pipe strength joint strength	pipe and joint strength (buckling)	not accepted because of occurrence of gas pockets	air vessel/surge tower/flywheel (for short lines only)/speed control of pumps and valve timing/ air inlet valves in industrial water lines/by- pass across pumps	In drinking water- lines underpressures are not accepted avoid sucking in of unknown water. Special attention should be paid to type of non-return valve.
		emergency stop due to power failure of pump. emergency valve closure	pipe and joint strength	pipe and joint strength (buckling)	allowed, if pressure rise due to collapse of cavity is below all. overpressure. Special restarting procedure required.		
sewage pipelines	dis- continuous flow	regular rather sudden start, stop	pipe and joint strength	pipe and joint strength (buckling) and not lower than -7 m.w.c	not allowed because of occurrence of gas pockets and unknown cavitation damage	air vessel surge tower flywheel (for short lines only) bypass across pumps	Special attention should be paid with respect to settlement in air vessels and surge towers and to odours. Surge tower for this reason less attractive. Special attention should also be paid to type of non-return valve
		emergency stop due to power failure of pumps	pipe and joint strength	pipe and joint strength (buckling) and not lower than -7 m.w.c (3.10^5 Pa.abs.)	not allowed because of occurrence of gas pockets		
main cooling water pipelines in electrical power plants	continuous flow	incidental rather sudden change, stop, start	pipe and joint strength	pipe and joint strength (buckling)	allowed, if pressure rise due to collapse of cavity is below accept- able overpressure. Restarting of pumps or flow rise after complete filling of condensor, or gradual restarting of pumps resp. flow rise without deaeration of condensor	surge tower air inlet valves	Often contains condensor in which top already under- pressure up to -8 m.w.c (2.10^5 Pa abs. pressure) during continuous flow. Special attention should also be paid to type of non- return valve
		emergency stop due to power- failure of pumps	pipe and joint strength	pipe and joint strength (buckling)			
district heating networks and fee- ding transmis- sion lines	continuous flow	regular gra- dual changing flow	pipe and joint strength	pipe and joint strength (buckling)	not likely to occur	expansion tower or expansion vessel acts also as surge provi- sion	connecting pipe between expansion vessel should be short and of suf- ficient diameter
		emergency stop by means of valves near pumps			accepted if collapse pressure rise is below allowable overpressure		
crude oil trans- mission lines and crude oil resp. chemical pipelines for loading and unloading ships	continuous flow or gradual changing flow between min. and max.	regular start, stop	pipe and joint strength	pipe and joint strength (buckling) restricted to vapour pressure of fluid	not allowed because of unknown degasing of fluid	pump and valve control, by radio signal or pressure switch, pressure relief valves . storage tanks bypass across pumps rupture disc	special attention should be paid to type of non- return valve
		emergency stop of pumps or quick closure of upstream valve emergency quick closure of downstream valve	pipe and joint strength	pipe and joint strength (buckling)	allowed if pressure rise due to collapse of cavity is below allow- able overpressure Special restarting procedure required.	pump or valve control by radio signal or pressure switch pressure relief valves storage tanks bypass across pumps rupture disc	

Table 1: Summary of considerations when carrying out a waterhammer study

5 Mathematical description of waterhammer

The flow and pressure in a pipeline can be described by two differential equations following from Newtons Law and the law of conservation of mass [1]. Solving these equations with given boundary conditions leads to the desired pressure as a function of time at any place in the pipeline. Boundary conditions for these equations are the hydraulic characteristics of pumps, valves, surge towers, pressure relief valves, pipe friction, formation of cavities etc. (see Fig. 36).

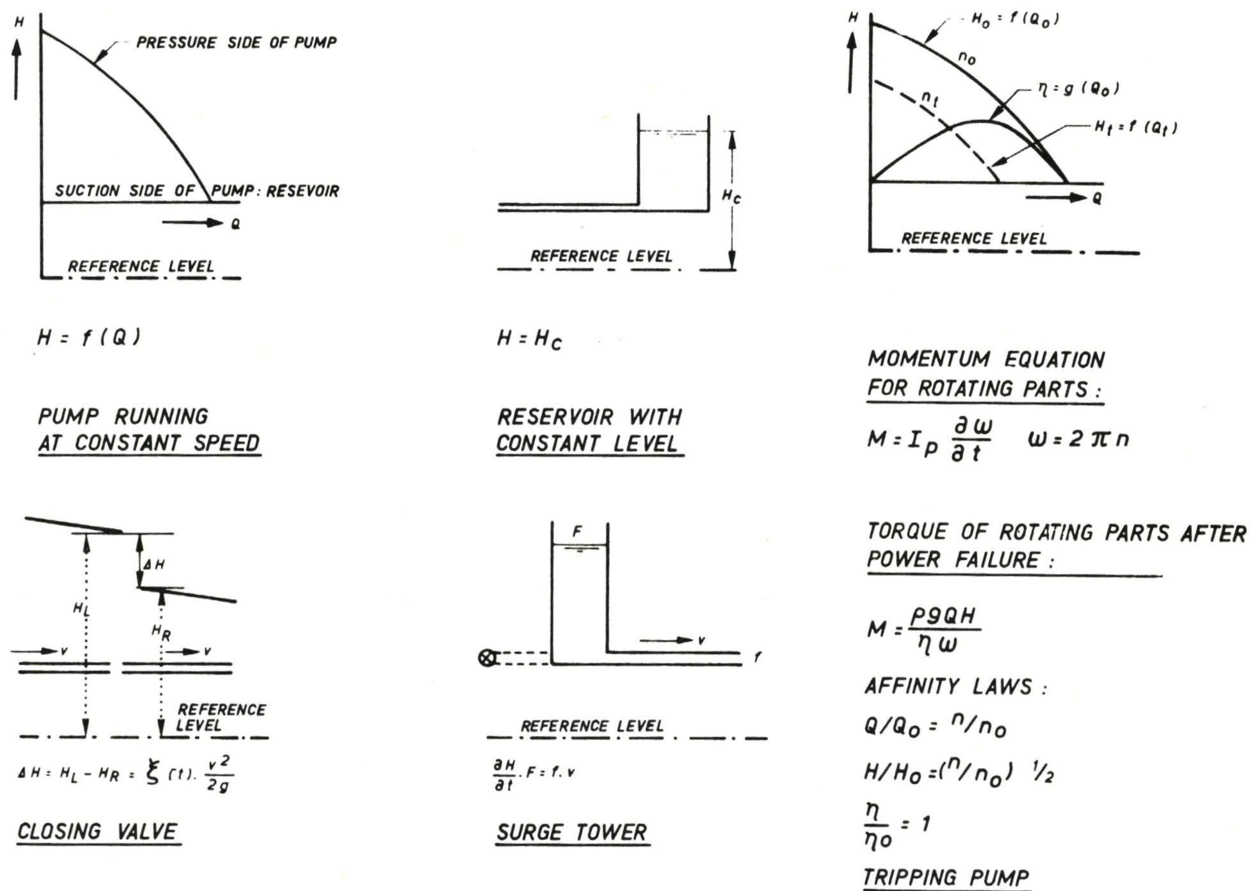


Fig. 36 Some boundary conditions

Newtons Law (see also Fig. 37)

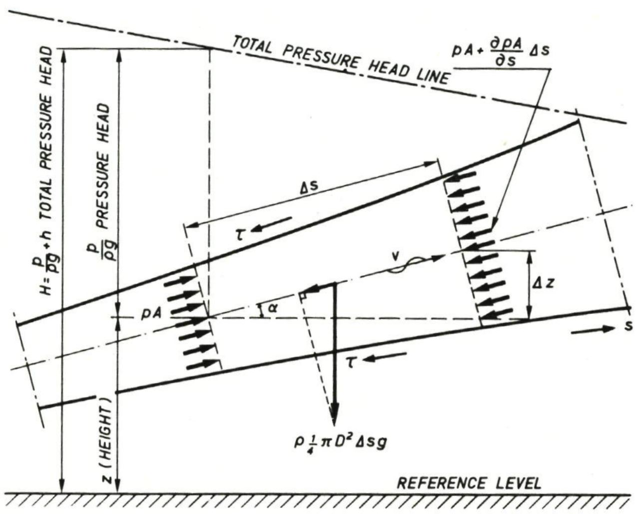


Fig. 37 Momentum equation

In the flow direction the following forces are acting on a fluid element:

Pressure force $\quad pA - (pA + \frac{\partial pA}{\partial s} \Delta s)$

Component of weight of fluid element $\quad \rho g \, \Delta s (\frac{A + A \frac{\partial A}{\partial s} \Delta s}{2}) \sin\alpha$

Friction force $\quad -\tau 0 \, \Delta s$

Component of pressure force on side wall of fluid element $\quad (p + \frac{1}{2} \frac{\partial p}{\partial s} \Delta s) \frac{\partial A}{\partial s} \Delta s$

Mass of the fluid element $\quad \rho \Delta s (\frac{A + A \frac{\partial A}{\partial s} \Delta s}{2})$

Acceleration of fluid element $\quad \frac{dv}{dt}$

Applying Newtons law:

$$pA - (pA + \frac{\partial pA}{\partial s} \Delta s) - \rho g \, \Delta s (\frac{A + A \frac{\partial A}{\partial s} \Delta s}{2}) \sin\alpha - \tau 0 \, \Delta s + (p + \frac{1}{2} \frac{\partial p}{\partial s} \Delta s) \frac{\partial A}{\partial s} \Delta s =$$

$$\rho Ds\left(\frac{A + A + \frac{\partial A}{\partial s}\Delta s}{2}\right)\frac{dv}{dt}$$

Disregarding lower order terms and dividing both sides of the equation by $\rho A.\Delta s$:

$$\frac{1}{\rho}\frac{\partial p}{\partial s} + g\ \sin\alpha + \frac{\tau O}{\rho A} + \frac{dv}{dt} = 0 \qquad (5a)$$

Assuming wall friction τ during transient flow does not differ substantially from wall friction during continuous flow (see also Fig. 38):

$$\tau O\ \Delta s = \rho g\ \Delta H\ A$$

$$\tau O\ \Delta s = \rho g\ \lambda\ \frac{\Delta s}{4A/O} \cdot \frac{v/v/}{2g}\ A$$

$$\tau = \rho\ \frac{\lambda}{8}\ v/v/$$

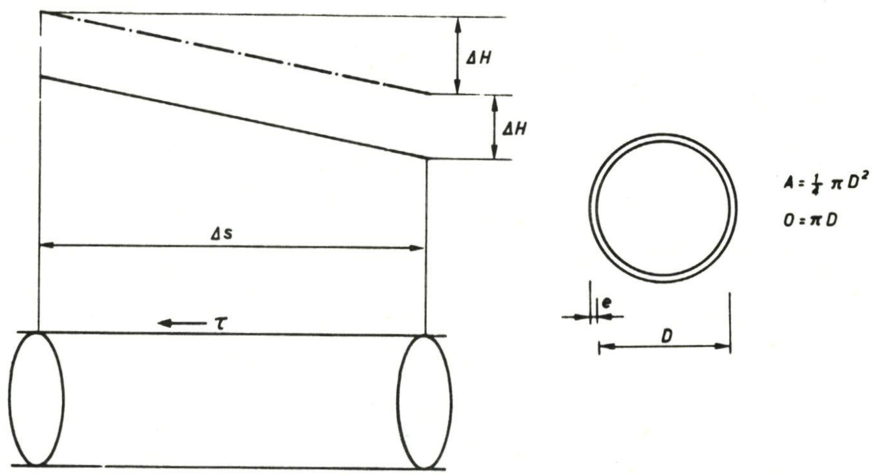


Fig. 38 Assumption for wall friction during transient flow

(5) may then be written as:

$$\frac{1}{\rho}\frac{\partial p}{\partial s} + g\ \sin\alpha + \frac{\lambda}{8A/O}\ v/v/ + \frac{dv}{dt} = 0 \qquad (5b)$$

<u>Law of conservation of mass</u> (see also Fig. 39)

This law means that the difference between the amount of fluid which has flown during a time interval Δt into and from a <u>pipe element Δs</u> equals the amount of

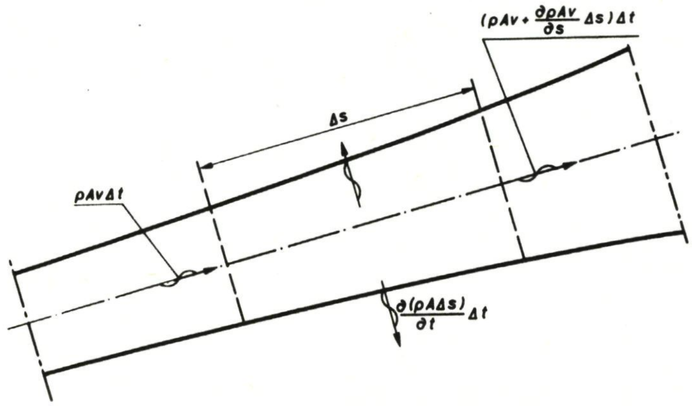


Fig. 39 Conservation of mass

fluid which has been stored <u>inside</u> the pipe element due to the elasticity of the fluid and pipe wall. Since the pipeline length also changes, one should keep in mind that Δs is a function of time.

Flown into the pipe element during Δt: $\rho A v\ \Delta t$

Flown from the pipe element during Δt: $(\rho A v + \frac{\partial A \rho v}{\partial s}\Delta s)\Delta t$

Stored in the pipe element during Δt: $\frac{\partial \rho A \Delta s}{\partial t}\Delta t$

This leads to:

$$\rho A v \Delta t - (\rho A v + \frac{\partial \rho A v}{\partial s}\Delta s)\Delta t = \frac{\partial(\rho A \Delta s)}{\partial t}\Delta t \tag{6}$$

With $d\rho = \frac{\partial \rho}{\partial s} ds + \frac{\partial \rho}{\partial t} dt$, $dA = \frac{\partial A}{\partial s} ds + \frac{\partial A}{\partial t} dt$, $d\Delta s = \frac{\partial \Delta s}{\partial t} dt + \frac{\partial \Delta s}{\partial s} ds = \frac{d\Delta s}{dt} dt$,

$v = \frac{ds}{dt}$, and dividing both sides of the equation by $\rho A \Delta s \Delta t$:

$$\frac{\partial v}{\partial s} + \frac{1}{\rho}\frac{d\rho}{dt} + \frac{1}{A}\frac{dA}{dt} + \frac{1}{\Delta s}\frac{d\Delta s}{dt} = 0 \tag{6a}$$

Defining the bulk modulus for a constant mass of fluid by:

$$\frac{dV}{V} = \frac{1}{K} dp \quad \text{and} \quad \frac{dV}{V} = -\frac{d\rho}{\rho}$$

(6a) can be written as:

$$\frac{\partial v}{\partial s} + \frac{1}{K}\frac{dp}{dt} + \frac{1}{A}\frac{dA}{dt} + \frac{1}{\Delta s}\frac{d\Delta s}{dt} = 0 \tag{6b}$$

After multiplying $\frac{1}{A}\frac{dA}{dt}$ and $\frac{1}{\Delta s}\frac{d\Delta s}{dt}$ with $\frac{dp}{dp}$, and multiplying both sides of the equation with ρ:

$$\rho\frac{\partial v}{\partial s} + \frac{dp}{dt}\,\rho\left(\frac{1}{K} + \frac{1}{A}\frac{dA}{dp} + \frac{1}{\Delta s}\frac{d\Delta s}{dp}\right) = 0 \qquad (6c)$$

In this equation $\frac{1}{K}$ stands for the change in the amount of fluid inside the pipe element Δs, due to a pressure change. As such $\frac{1}{A}\frac{dA}{dp} + \frac{1}{\Delta s}\frac{d\Delta s}{dp}$ stands for the change of the dimensions of the pipe element Δs due to a pressure change. In the latter case the pipe properties and the way of anchoring play a role.

After substitution of

$$c = \frac{1}{\sqrt{\rho\left(\frac{1}{K} + \frac{1}{A}\frac{dA}{dp} + \frac{1}{\Delta s}\frac{d\Delta s}{dp}\right)}} \qquad (7)$$

(6c) becomes:

$$\rho\frac{\partial v}{\partial s} + \frac{1}{c^2}\frac{dp}{dt} = 0 \qquad (6d)$$

For pipelines with a linear change of cross-section A and length of the elements Δs ($\frac{dA}{dp}$ = constant and $\frac{d\Delta s}{dp}$ = constant) "c" will be also constant. This holds in general for all circular pipes with overpressure and thick-walled pipes with underpressure.

In formula then:

$$c = \frac{1}{\sqrt{\left(\frac{c_1 D}{eE} + \frac{1}{K}\right)\rho}} \qquad (7a)$$

in which c = constant

c_1 = a constant depending on the pipe anchoring*

D = pipe diameter in m

e = wall thickness of the pipe in m

E = modulus of elasticity of the pipe material in N/m^2

K = bulk modulus of the fluid in N/m^2

ρ = density of the fluid in kg/m^3

---

* $c_1 = 5/4 - \mu$ for upstream anchoring only, $c_1 = 1 - \mu^2$ for complete anchoring and $c_1 = 1$ for pipelines with expansion joints. μ = Poissons ratio = 0,3 - 0,4, hence c_1 = 0,9 - 1,0. This means influence on c1-value is small

For thin-walled, relatively slack pipes with varying underpressure, "c" is not constant but depends on the pressure p [2], [3].
The "constant" c in (6d) and (7a) is in fact the speed of pressure waves in pipelines, as is proved below:

1. After substitution of $H = \frac{p}{\rho g} + z$ and $\frac{\partial p}{\partial s} = \rho g(\frac{\partial H}{\partial s} - \frac{\partial z}{\partial s}) = \rho g \frac{\partial H}{\partial s} - \sin\alpha$ (5b) becomes:

$$g \frac{\partial H}{\partial s} + \frac{A}{8A/O} v/v/ + \frac{dv}{dt} = 0 \tag{5c}$$

while (6d) becomes:

$$\frac{\partial v}{\partial s} + \frac{g}{"c"^2} \frac{\partial H}{\partial t} = 0 \tag{6e}$$

and (5c) becomes with $dv = \frac{\partial v}{\partial s} ds + \frac{\partial v}{\partial t} dt$

$$g \frac{\partial H}{\partial s} + \frac{A}{8A/O} v/v/ + \frac{\partial v}{\partial t} + v \frac{\partial v}{\partial s} = 0 \tag{5d}$$

2. The value of $v \frac{\partial v}{\partial s}$ can generally be disregarded with respect to $\frac{\partial v}{\partial t}$ (see Fig. 40)
3. Consider frictionless pipeline: $\frac{\lambda}{8A/O} v/v/ = 0$
4. Consider "c" is constant

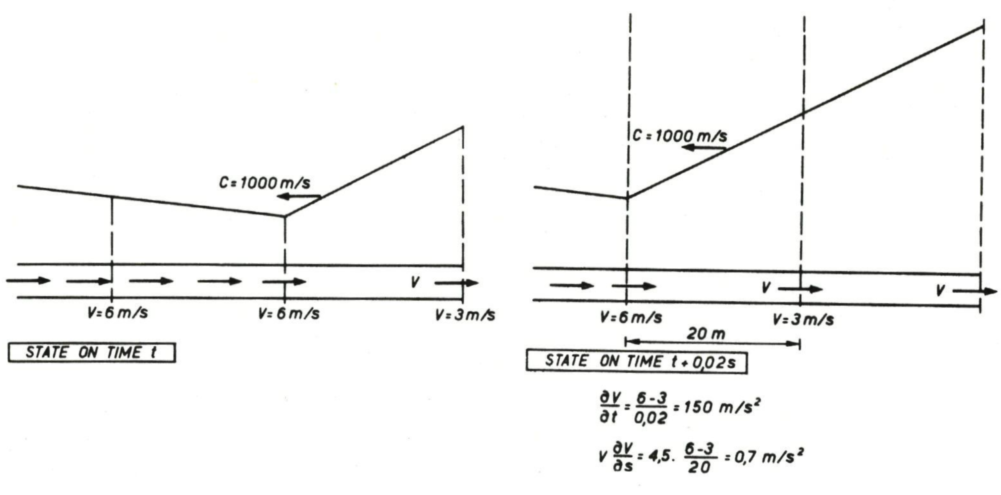


Fig. 40 Estimation of $v \frac{\partial v}{\partial S}$

This leads for (5d) and (6d) to:

$$\frac{\partial H}{\partial s} + \frac{1}{g}\frac{\partial v}{\partial t} = 0 \tag{5e}$$

$$\frac{\partial H}{\partial t} + \frac{c^2}{g}\frac{\partial v}{\partial s} = 0 \tag{6f}$$

The general solution of this couple of equations is:

$$h = F(s + ct) + f(s - ct) + H_o \tag{7}$$

$$v = -\frac{g}{c}\left\{F(s + ct) - f(s - ct)\right\} + v_o \tag{8}$$

The meaning of (7) and (8) follows from a further consideration of H = f(s - ct) in a place-time diagram (see Fig. 41):

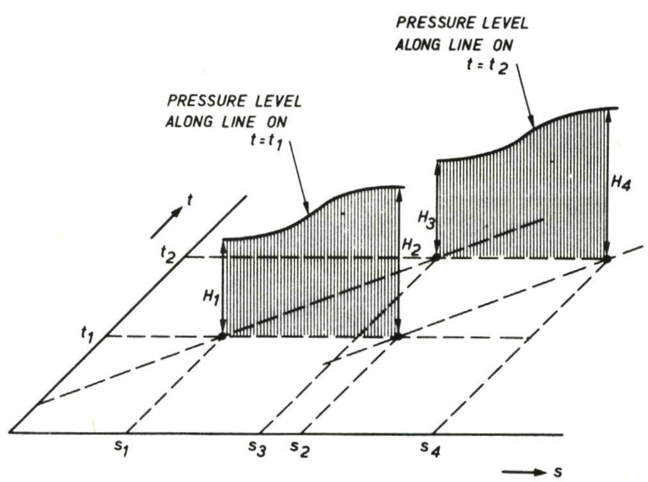


Fig. 41 s-t-H diagram

- In place s_1 of the pipeline, a pressure head H_1 then prevails at time t_1.
- The same pressure head H_1 prevails in the place s of the s-t diagram for values of $s - ct$ = constant = $s_1 - ct_1$. This means $H_3 = H_1$ at time t_2 in place s_3 of the pipeline.

- In place s_2 a pressure head H_2 prevails at time t_2.
- Analogue $H_4 = H_2$ at time t_2 in place s_4 of the pipeline.

So the pressure head "function" has been shifted along the pipeline in a time interval $t_2 - t_1$ over a distance of $c(t_2 - t_1)$ (Fig. 42). This is a propagating pressure head wave with propagation speed c in a positive s-direction. Analogue it can be shown that $H = F(s + ct)$ means a pressure head wave propagating with speed c in a negative s-direction.

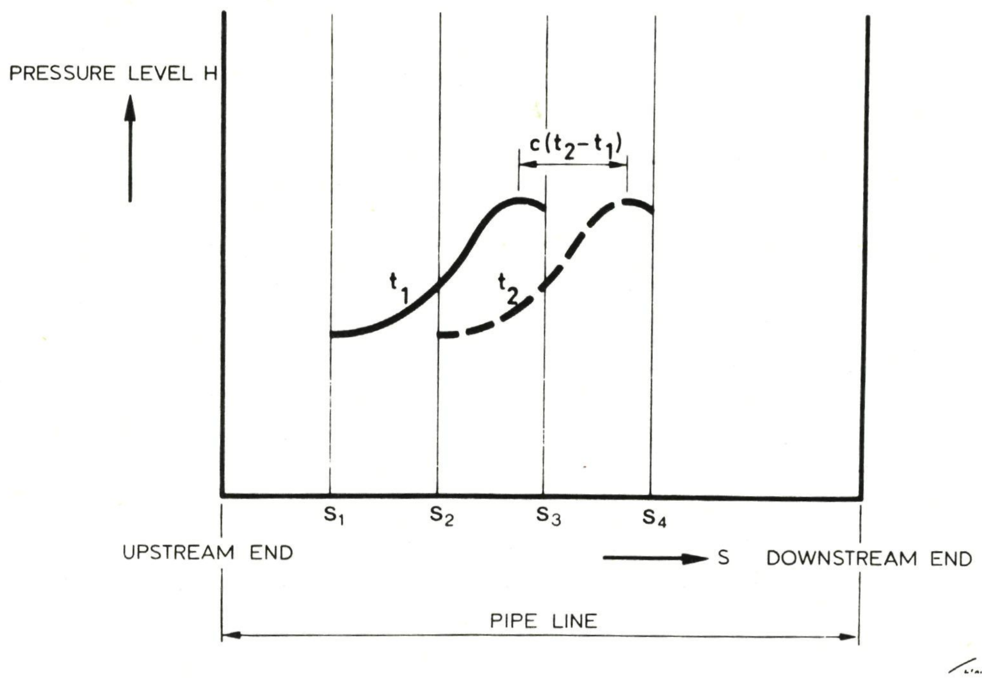


Fig. 42 Travelling pressure wave

<u>Hence the pressure head and velocity history in the pipeline is composed of propagating waves defined by the boundary conditions in the left and right end of the pipeline</u>. Due to friction these waves will become smaller (dampened) during their propagation through the pipeline.

Some boundary conditions are shown in Figure 36.

As already stated at the beginning of this chapter, the determination of pressure heads and velocities in a pipeline or pipe system amounts in fact to solving both the following equations with the relevant boundary conditions (see Fig. 36):

$$\frac{\partial H}{\partial s} + \frac{1}{g}\frac{\partial v}{\partial t} + \frac{\lambda}{8A/O} v/v/ = 0 \qquad (9)$$

$$\frac{\partial H}{\partial t} + \frac{c^2}{g}\frac{\partial v}{\partial s} = 0 \qquad (10)$$

The accuracy of the results of waterhammer calculations depends largely on the accuracy of the boundary conditions.
The mathematical description of two common boundary conditions in waterhammer studies are described below:

1. Pump failure

During the steady state ($t < 0$), when the pump is running at constant speed, the total torque exerted by the drive and fluid on the pump shaft is zero:

$$M_{ao} + M_{vo} = 0 \qquad (11)$$

In which M_{ao} = steady state torque exerted by the drive on the pump shaft [Nm]
M_{vo} = steady state torque exerted by the fluid on the pump shaft [Nm]

At the instant of pump failure ($t = 0$) it is assumed that the driving torque M_{ao} becomes instantaneously zero. At that instant the fluid torque M_{vo} is the only torque exerted on the pump shaft.

$$M_{vo} = - \frac{\rho g\ Q_o\ H_o}{\eta_o \cdot \frac{2\pi\ n_o}{60}} \qquad (12)$$

in which ρ	= density of the fluid	[kg/m^3]
Q_o	= flow at steady state	[m^3/s]
H_o	= head at steady state	[m.f.c.]*
η_o	= efficiency at steady state	[-]
n_o	= speed at steady state	[rev./min]

* m.f.c. = m fluid column

Due to this torque the pump speed is reduced and so the flow and head of the pump. According to Newton's law, at time t after pump failure, the following formula holds:

$$M_v = I_p \frac{d\omega}{dt} = I_p \frac{d \frac{2\pi\ n}{60}}{dt} \tag{13}$$

in which M_v = fluid torque exerted on the pump shaft at $t \geqslant 0$ [Nm]

I_p = polar moment of inertia of the rotating parts of the pump [Nm]

First of all it is assumed that when the pump slows down the working points at each pump speed keep their Q-H, respectively Q-M, characteristics which are valid for the steady state flow through the pump at speed n. Using pump affinity laws the following boundary conditions can be derived:

$$H = \left(\frac{n}{n_o}\right)^2 f_1 \left(\frac{n_o}{n} Q,\ n_o\right)^* \tag{14}$$

$$M = \left(\frac{n}{n_o}\right)^2 f_3 \left(\frac{n_o}{n} Q,\ n_o\right)^* \tag{15}$$

Furthermore it is assumed for transient flow that:

$$M_v \frac{2\pi\ n}{60} = - \frac{\rho g\ Q\ H}{\eta} \tag{16}$$

in which M_v = fluid torque exerted in the pump shaft (function of t) [Nm]

n = pump speed (function of t) [rev/min]

Q = flow (function of t) [m^3/s]

η = efficiency (function of t) [-]

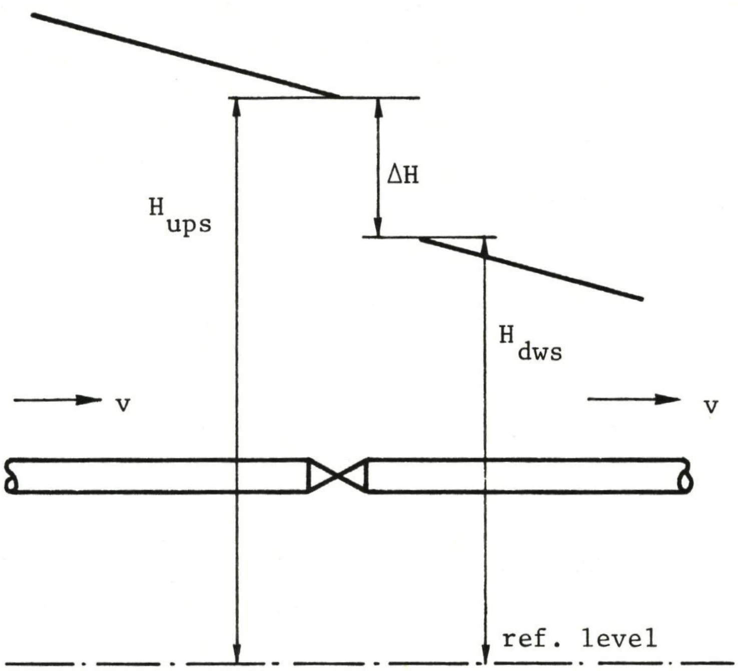
ρ = density of fluid [kg/m^3]

g = gravity acceleration [m/s^2]

*(14) and (15) have to be known completely in the 1° and 4° quadrant.

The boundary conditions as described by (12), (13), (14), (15) and (16) and the basic equations as described by (9) and (10) are sufficient to calculate the transient behaviour of a failing pump, defined by Q, H and n as a function of time t.

2. Closing valve



During the steady state (t < 0) the following holds for a partly or completely open valve:

$$\Delta H_o = \xi_{valve.o} \frac{v_o^2}{2g} \tag{17}$$

or

$$H_{ups} - H_{dws} = \xi_{valve.o} \frac{v_o^2}{2g} \tag{18}$$

in which	ΔH_o	= head loss across valve at $t < 0$	[m.f.c.]
	$\xi_{valve.o}$	= head loss factor for valve at $t < 0$	[-]
	v_o	= velocity in the connecting pipes	[m/s]
	H_{ups}	= head in upstream connecting pipe	[m.f.c.]
	H_{dws}	= head in downstream connecting pipe	[m.f.c.]

During closure of the valve (transient situation $t \geq 0$) the following holds:

$$\Delta H = \xi_{valve} \cdot \frac{v^2}{2g} \tag{19}$$

or $$H_{ups} - H_{dws} = \xi_{valve} \cdot \frac{v^2}{2g} \tag{20}$$

in which ΔH = head loss across valve (function of time)

ξ_{valve} = head loss factor of the valve (known function of time)

H_{ups} = head in upstream connecting pipe (function of time) [m.f.c.]

H_{dws} = head in downstream connecting pipe (function of time) [m.f.c.]

v = velocity in the connecting pipes (function of time) [m/s]

If the valve is situated at the downstream end of a pipeline discharging at a constant level into a reservoir then:

$$H_{dws} = H_{res} = \text{constant} \tag{21}$$

in which H_{res} = fluid level in reservoir [m.f.c.]

The boundary condition as described by (20) and the basic equations as described by (9) and (10) are sufficient to calculate the transient behaviour of a closing valve, defined by H_{ups}, H_{dws} and v as a function of time t.

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