Cranfield
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SYSTEMS DESIGN

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PIPELINE AND PUMPING SYSTEMS DESIGN

1 REMINDER OF BASIC CONCEPTS AND DEFINITIONS

Figure 1.1 shows the basic elements of a simple pumping system. It comprises a sourve tank or sump, a length of pipeline to the pumping station, a pump and non-return valve (essential on most systems to prevent reverse flow when the pumps stop) and delivery pipeline to the discharge tank or reservoir.

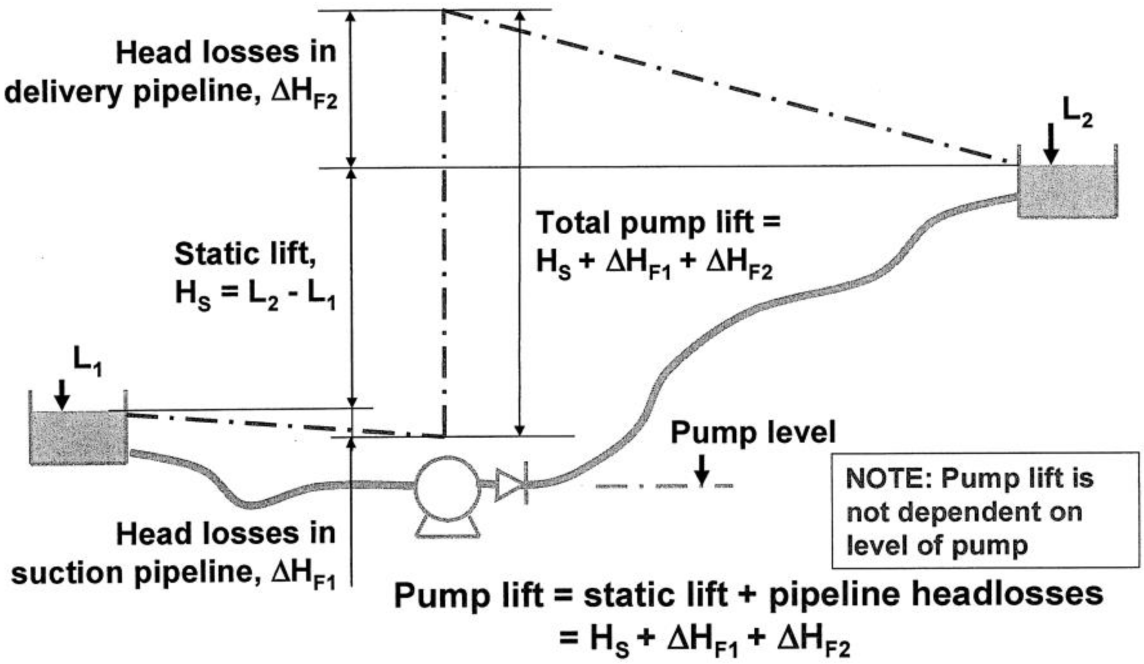


Figure 1.1 Elements of a simple pumping system.

The static lift is the height, or equivalent head of fluid, between the water level in the source tank and that in the delivery tank (or if the flow is discharged with a high level inlet above the water level then that controlling level). This is the head across the pump, or more strictly across the non-return valve when there is no flow. The static lift is not dependent on the level of the pump.

The pump must provide enough energy to overcome the static head before any flow can be delivered but in addition it must provide head to overcome the friction and fittings losses in the pipeline. These are both a function of the pipeline velocity, V (and approximately a function of V^2) so the more flow required to be delivered the higher the pump lift must be.

The total lift required from the pump is the static lift, H_S, plus the friction and fittings losses in both the suction and delivery main, ΔH_{F1} and ΔH_{F2}.

i.e. the pump lift, $\Delta H = H_S + \Delta H_{F1} + \Delta H_{F2}$

In a pipeline of constant diameter, the rate of loss of energy, the 'friction' loss is constant so, ignoring the fittings losses for the moment, the ***hydraulic grade line*** is at a constant slope. This line represents the change in total energy in the system. Strictly, of course, the fitting losses should be included as abrupt steps in the total energy line but it is usually acceptable to include them as part of the total energy loss and lump them in with the friction losses as spread out over the whole line.

The ***pressure head*** at any point in the system is the height of the hydraulic grade line above the local level of the pipe. Strictly, it is the height of the pressure level or piezometric line above the pipeline as illustrated in Figure 1.2. However in most systems the velocity head is very small in comparison with the pressure head and it can be ignored. (BUT should not be forgotten – sometimes it can be important!). If the pressure or piezometric line is below the pipeline level then that indicates that there are sub-atmospheric pressures in the pipe. In extreme cases, if the pressure drops as low as about -10 m head then it reaches the vapour pressure of water and vapour cavities can form.

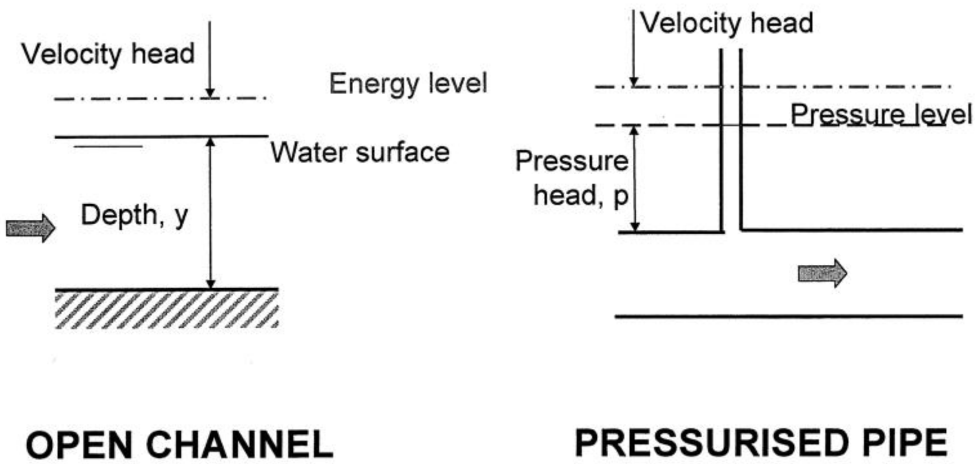


Figure 1.2 Definition of pressure and energy levels

Figure 1.3 shows a more detailed look at energy losses and pressure and energy levels in a system.

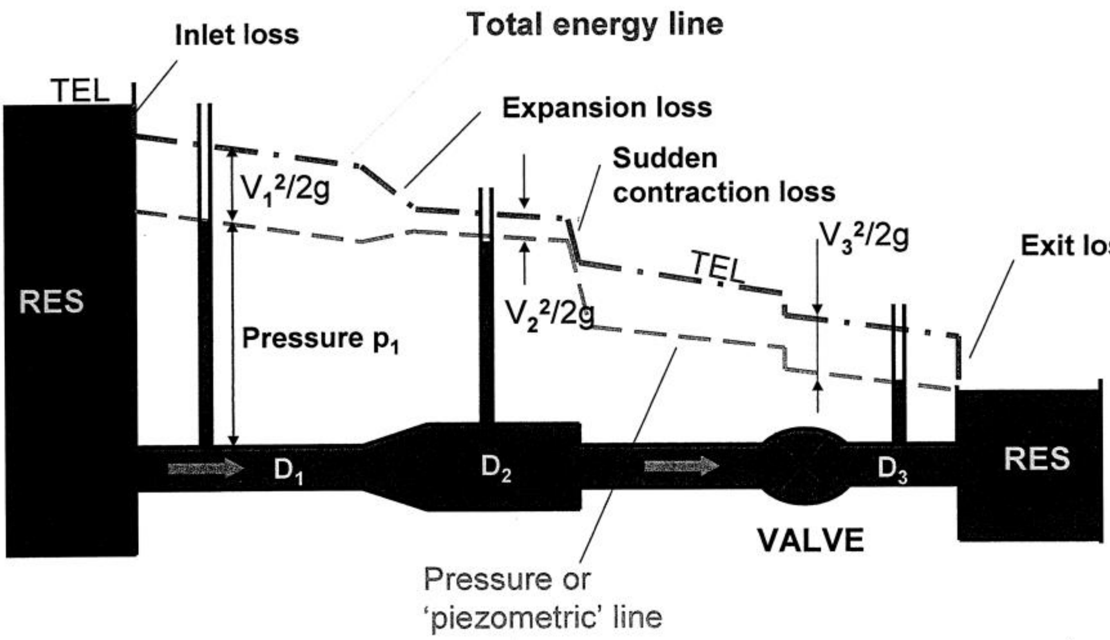
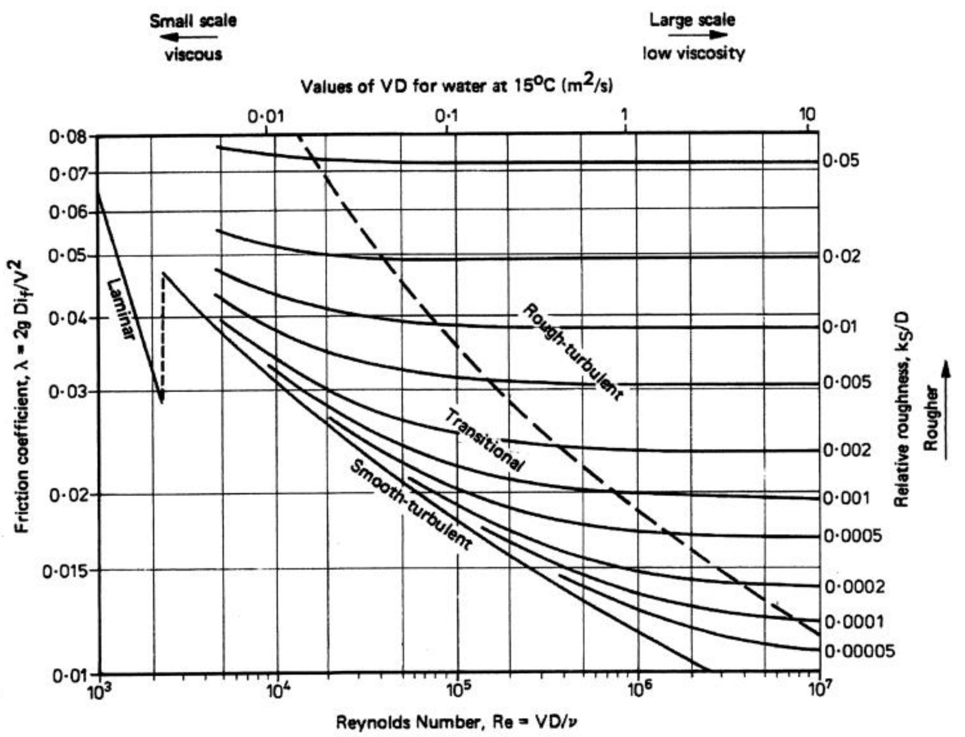


Figure 1.3 Energy losses in a simple gravity system

Friction losses are calculated using the Darcy Weisbach formula with the friction factor, f, defined by the Colebrook-White equation or one of the directly solvable versions of it. They can also be assessed using the Moody diagram.



Fi

$$\Delta H = K.V^2/2g$$

where K is an empirical coefficient depending on the type of fitting and its geometry.

For any particular flow it is thus possible to calculate the static lift and friction/fittings losses and this can be done simply for the design flow. However, a system curve should always be developed as it is necessary to consider a wider range of parameters than just the design condition.

2 SYSTEM CURVES AND PUMP DUTIES

2.1 Single fixed-speed pump systems

A ***system curve*** is simply the relationship between the flow in a system and the pump lift required to deliver that flow. Figure 2.1 shows a typical system curve. It should be noted that the system curve is solely a function of the pipeline system and includes the losses in both the suction and delivery lines. It is not dependent on the pump but it does define what the required pump duty may be.

The reason for drawing the system curve even for a simple pumping system with a single pump, is that it is rarely the case that a single point defines the range of operating duties for the pump.

For a start, the static lift varies as the sump and delivery tank levels change. In addition there is uncertainty over the pipeline roughness and the fittings losses. Initially when the system is new the pipeline roughness will be low but it is likely to deteriorate with time – and that is certainly the case in a sewage pumping system where biological growth on the pipe walls will develop. It is also the case that the fittings losses are not defined with great accuracy. Accurate values are given in references 1 and 2 but for design a safety factor is often applied and other references (3 and 4 for example) will give higher values.

Thus there is an envelope of potential operating conditions between:-

- A system curve representing the highest pump lifts – maximum static lift, maximum likely roughness and design fittings losses
- A system curve representing minimum pump lifts – minimum static lift, 'as new' pipe roughness and minimum fittings losses.

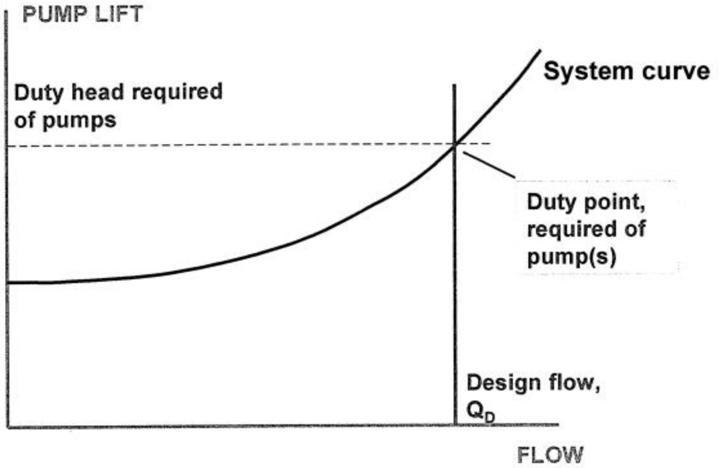


Figure 2.1 Simple system curve

Figure 2.2 illustrates this range and the choice of pump duty now becomes a little more complicated.

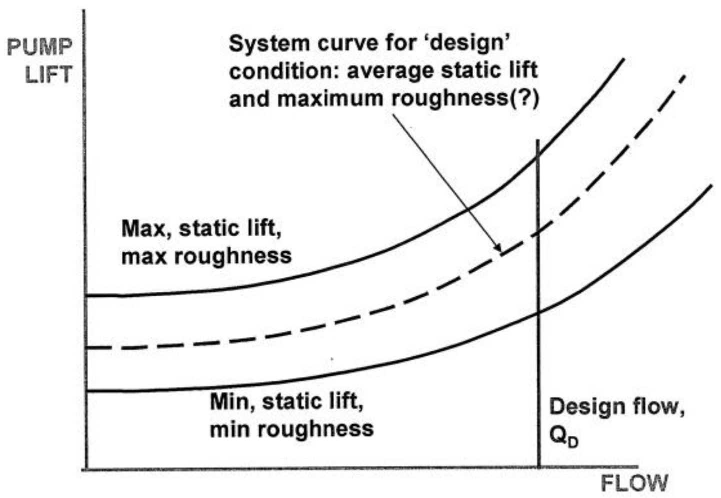


Figure 2.2 Range of potential operation of a single pump

Figure 2.3 illustrates the problem for a fixed-speed pump. In theory any pump lift between points A and C could meet the required duty. In practice though a duty based on point C would mean the pump would be operating at its required flow only under the most beneficial combination of parameters – minimum static lift, minimum roughness. As the sump level dropped and the static lift increased the pump would not deliver its required output and if the roughness deteriorated at all then even at maximum sump and minimum delivery level the output would not be achieved. This is clearly no acceptable.

On the other hand a duty point at A would mean that under all conditions, even the moist adverse, the required flow would be met and under most conditions the pump would be delivering more than the necessary flow. This may be a necessary requirement and it would not be wrong to define the pump duty as point A, however it would possibly entail a larger pump than necessary with the possible need for a larger motor and increased pipework size. Consideration should therefore be given to a duty based on some intermediate point such as B. This might be defined as the average static lift combined with the maximum roughness condition. During an operating cycle with the sump and delivery tank levels varying, the pump would operate on its curve between points B1 and B2. On average, even under maximum roughness conditions, it would deliver the required volume of water. This might provide a more economic solution than a duty based on point A.

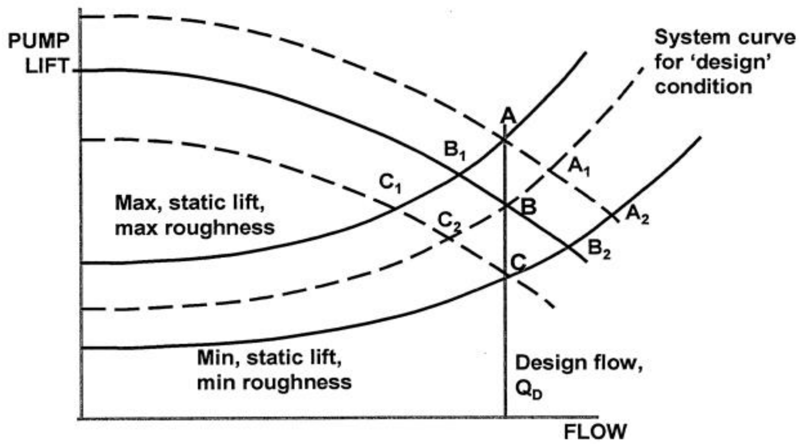


Figure 2.3 Potential range of pump duties

2.2 Multiple- pump systems

Figure 2.3 show the complications in defining the pump duty for a single pump. It gets more complicated with multiple pumps. Figure 2.4 shows the range of system curves and typical superimposed pump curves for a two operating pump station. The defined duty point has been chose as Point 1. This represents the total output required of the installation so with two pumps operating in parallel each pump will be required to deliver half the design flow. Thus the duty point for each pump is Point 1'.

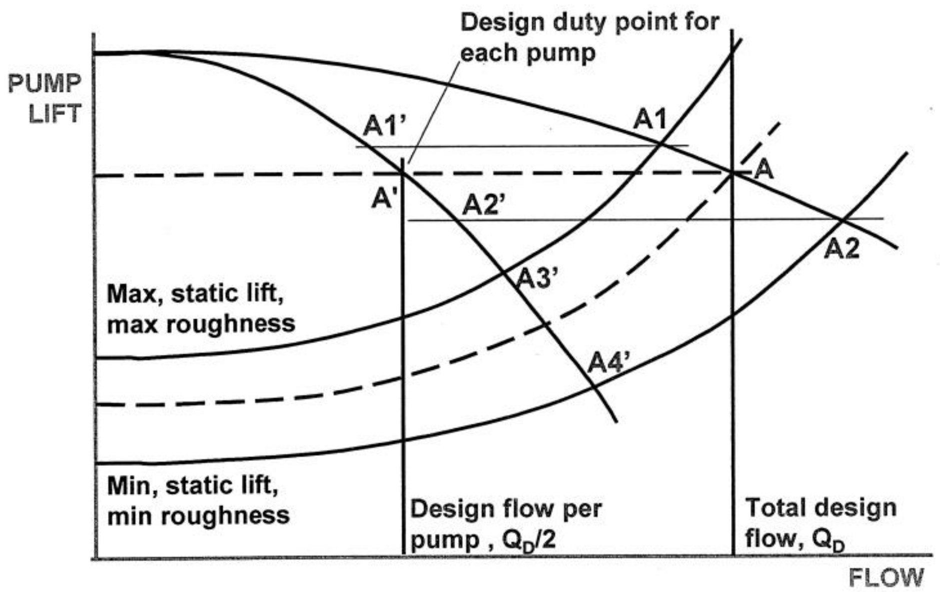


Figure 2.4 Two pump installation

With two pumps running in parallel the system could thus operate between points A1 and A2 with each pump individually operating between points A1' and A2'.

However it would also be possible to operate that system with just one of those pumps running. The pump would now be operating between points A3' and A4' so the full range of potential pump operation would be between points A1' and A4'.

For a single pump installation it would be normal to choose a pump with a best efficiency point close to the defined duty. However for a two pump installation consideration must be given to the most likely operating duty. For example if this were a drainage pumping station it might be the more common mode of operation for 1 pump to operate with the second being called on only in major storm events. If that were the case then it might be better to consider a pump with best efficiency under one pump operating conditions.

It is also worth pointing out that adding a third similar pump would not increase the station output by 50%. Because of the upward curvature of the system curves, the addition of a third pump would add less than $Q_D/2$. There is an effect of diminishing return by adding extra pumps unless the major proportion of the lift is static.

However the station could be designed with three pump operation to meet the duty flow Q_D – i.e each pump delivering $Q_D/3$. Figure 2.5 shows such a design with three pumps in parallel and illustrates another potential problem that reinforces the need to develop system curves for all pumping installations. In this case the duty point for a single pump is Pont 1'. However if the system were to be operated with just one pump running then there is a risk that under some conditions the pump could operate beyond its run-out point. If this were the case then there would need to be some throttling of the pump to increase the system head loss.

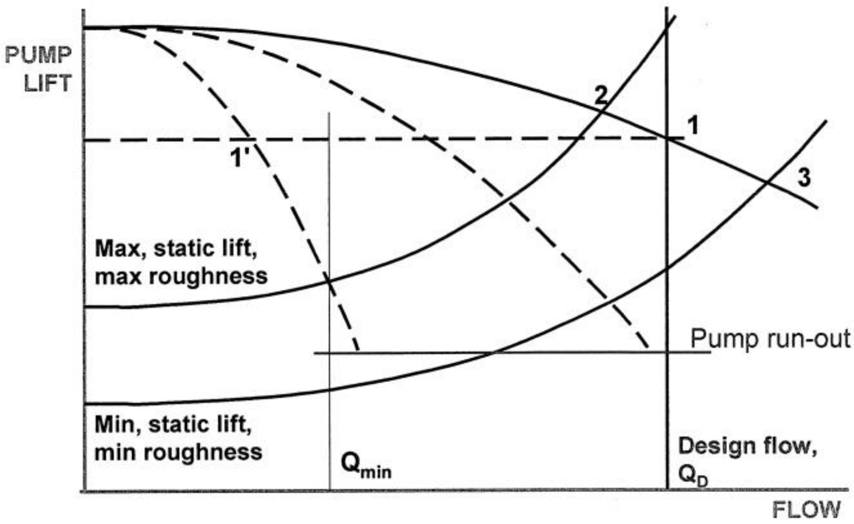


Figure 2.5 Three pump installation with potential for run–out if operated with single pump

3 THE LONGITUDINAL PROFILE

3.1 The influence of air valves

Figure 3.1 shows a simple gravity system with flow passing from a higher tank to a lower one. The pipeline profile shows a high point in along the line. Asuming a constant pipe diameter then the hydraulic grade line is straight and, as can be seen in this example, it drops below the pipe level.

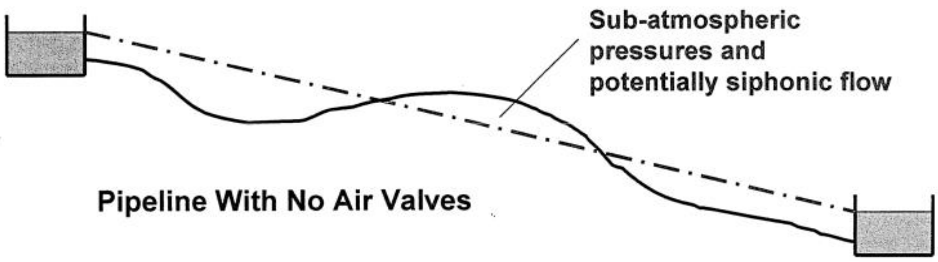
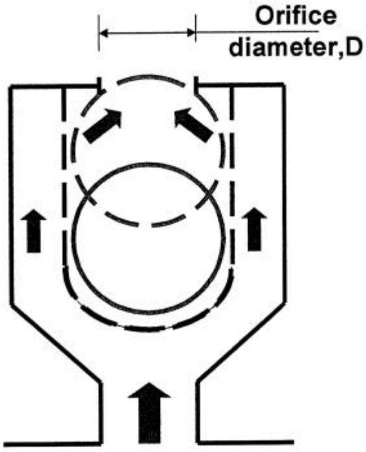


Figure 3.1 Simple gravity system with high point

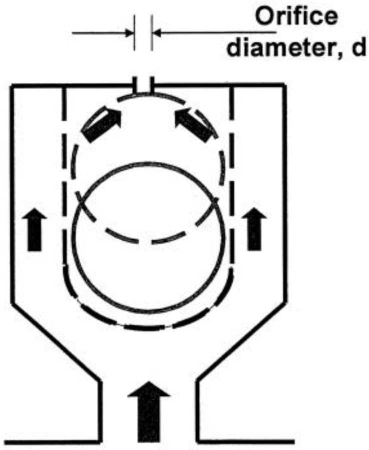
As discussed earlier this indicates sub-atmospheric pressures at this high point. In theory, provided the pressure does not drop down to the fluid vapour pressure the system would still work but with siphonic flow over this length.

In practice siphonic flow should be avoided. It is difficult to maintain unless the pipeline is welded and in potable water systems any sub-atmospheric pressures should be avoided because of the risk of contamination being pulled in. More particularly, there will almost certainly be an air valve located at the high point which will open and relieve the sub-atmospheric pressures.

There are effectively two types of air valve as illustrated in Figure 3.2 which show the essential elements of the two types, although, of course, the manufacturer's details vary considerably.



'Large-orifice', LO, air valve



'Small-orifice', SO, air valve

Figure 3.2 Air valves

They both work in a similar way but the size of the orifice means they act differently and have different uses. If the pipe is empty then the float in both types falls away for the orifice and air can freely pass into or out of the line. On filling the line air is discharged mainly through the large orifice valves and as the air is evacuated so in both valves the float rises up and seals the opening. If while the pipeline is pressurised and in use air collects in the body of the LO valve the pressure difference across the large opening generates sufficient force to hold the float in place, but in the small orifice valve the net force is less than the weight of the float and this can fall away from the opening allowing air to escape.

Thus the Large-Orifice valve only opens if the line pressure is at or below atmospheric and is used primarily to allow air into and out of the line on emptying or filling.

The Small-Orifice valve is used for bleeding of air that may collect in the valve during operation.

A Double-Orifice, DO, valve is merely one with both types of valve in a single body.

The locations at which valves are positioned is illustrated in Figure 3.3. Large orifice air valves are located at high points in the system to prevent air being trapped on filling. Small orifice air valves are located at those similar high points – hence the use of double-orifice valves – and at downward changes of pipe gradient and on long lengths of pipeline.

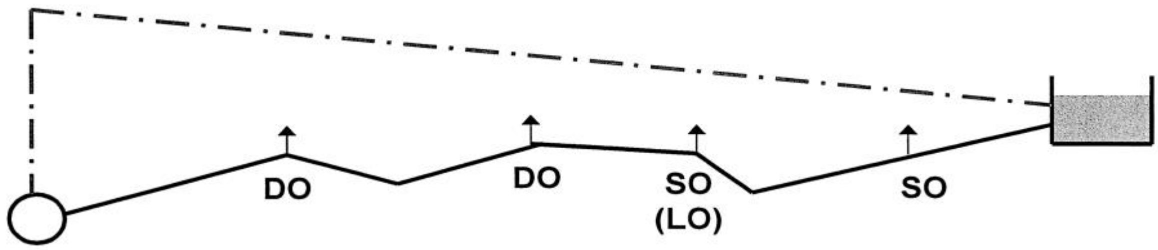


Figure 3.3 Air valve locations

If we now return to the simple system illustrated in Figure 3.1, then it is probable that a double-orifice air valve would be located at the high point. Both valves would open if the pressure dropped below atmospheric so a siphon could not be generated and instead the hydraulic gradient would now be between the upstream tank and the high point. This would be flatter than the hydraulic gradient shown in Figure 3.1 and hence the flow would be less.

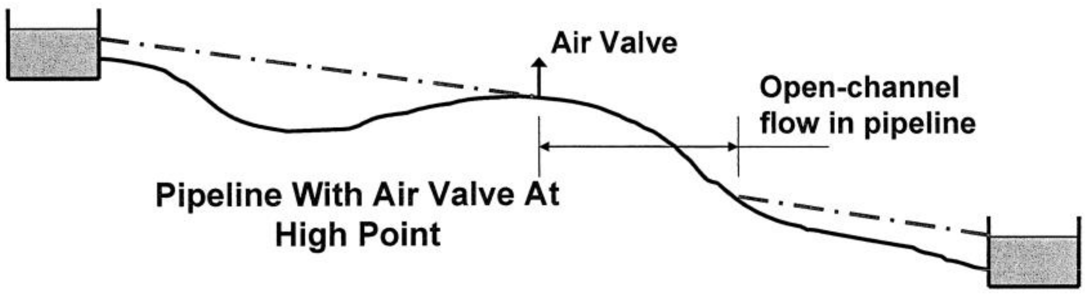


Figure 3.4 Effect of Air Valve at high point

Downstream of the summit the pipeline would flow as open channel until the pipe ran full again in its lower section with the same hydraulic gradient as the first section.

3.2 Influence of Pipeline Profile on Pumping System

Figure 3.5 shows a simple pumping system with a profile rising from the pump to the delivery tank .The hydraulic grade lines for no flow, low flow and maximum flow are shown.

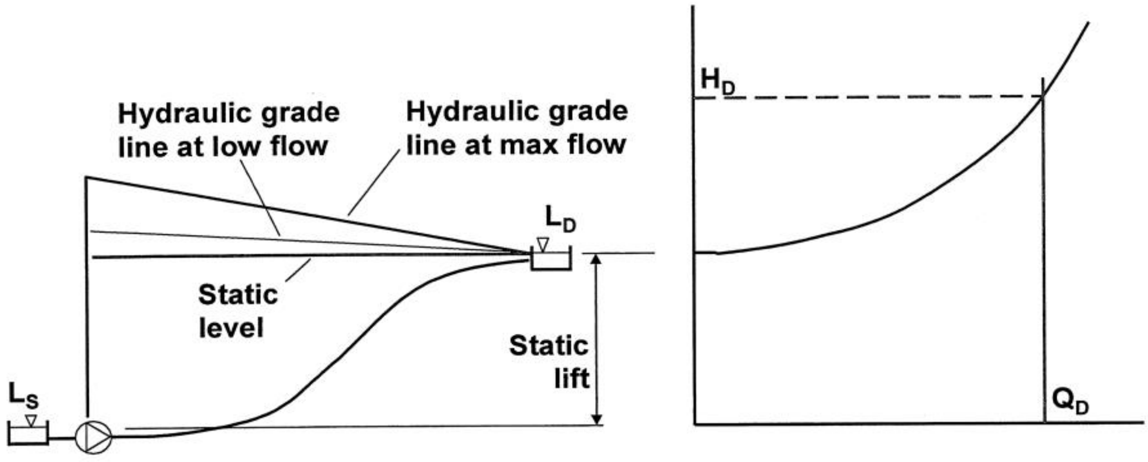


Figure 3.5 System curve for simple rising profile

The system curve is straightforward with the static lift being the difference in water levels between the delivery tank and the sump, ($L_D - L_S$), and the dynamic losses following a near parabolic curve, all as discussed before.

Consider now the pipeline profile as shown in Figure 3.6. There is a highpoint on the route and there will be an air valve at that location.

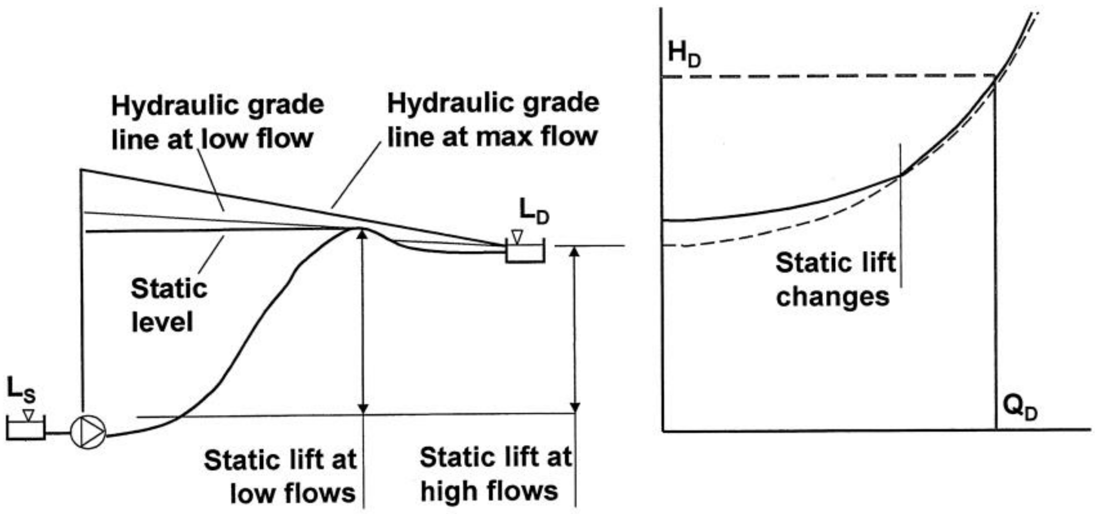


Figure 3.6 System curve for pipeline with high point along route

The difference in the tank levels ($L_D - L_S$) remains the same and at high flows the hydraulic grade line is above the summit so the static lift remains ($L_D - L_S$) as before.

When the pump(s) are turned off the pressure in the system upstream of the high point is defined by the high point in the pipeline and the static lift is now the difference between the pipeline level at the summit and the sump level, ($L_H - L_S$). This remains the case at low flows until the hydraulic grade line is just above that summit when the static lift changes as indicated for the system curve. For flows below that point the pump is pumping to the high point and the dynamic losses are those in that upstream section of pipe only.

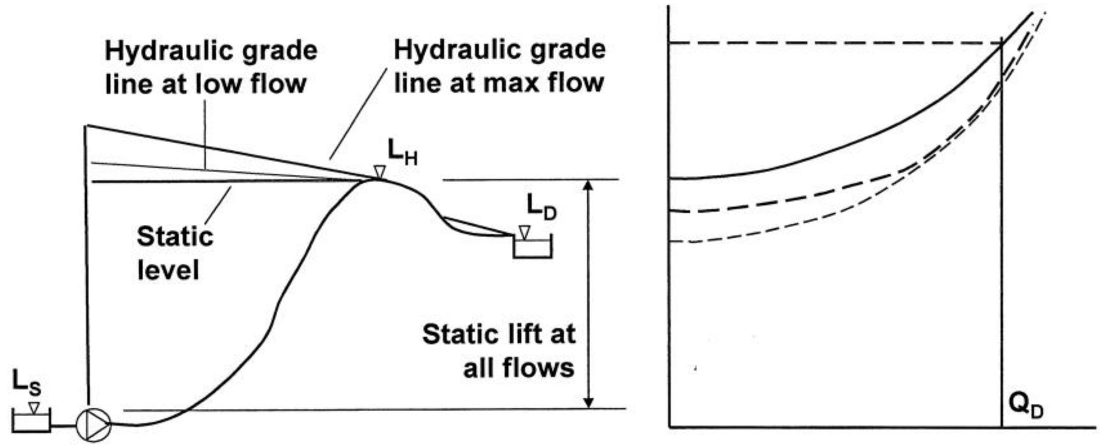


Figure 3.7 System curve for pipeline with very high point along route

Figure 3.7 shows the same system but with a very high point on route such that even at maximum flow the pumps are delivering to the high point. The static lift remains at LH – LS for all flows and the dynamic losses apply only to the pipeline upstream of the summit. In both cases (Figures 3.6 and 3.7) the pipe downstream of the summit will flow with open channel conditions at low flows and, in the latter case, at all flows. There is a hydraulic control at the summit.

This may be acceptable in raw and wastewater systems but would not be acceptable in a potable water supply system because the open-air valve at the summit would potentially allow the ingress of contamination. On a potable water supply system therefore a solution must be sought that ensures the pipe remains pressurised throughout. Such solutions might involve:-

- Re-routing the pipeline to avoid the high point
- Including a pressure-sustaining valve at the discharge end of the line so that the effective delivery level is high enough to ensure the pipeline is pressurised at the summit
- Providing a break pressure tank at the summit with a control valve at the discharge point, operating in response to the water level in the tank, or operating the pumps to ensure the tank never empties.

4 MORE COMPLICATED SYSTEMS

All the above discussions have been directed at simple systems with a single pumping installation. Many pumping systems, of course, are much more complicated, such as urban distribution systems.

The principles behind the analysis and design of such systems are exactly the same as outlined above but the calculations are more difficult and these days are almost always done using computer programs. For a large distribution system it is not possible to define individual system curves as there is such a range in the potential demands and their distribution. Storage is almost always provided on such a system and flow will be passed into or drawn from storage depending on the demands which vary daily, weekly and seasonally. What can be done, however, is to carry out a series of system analyses for various demand patterns and to plot these on a pump lift/total flow graph as individual points rather than full curves. It is possible to build up a zone of potential operation and hence to define suitable pump duties. Full operating scenarios can then be run over periods of days or longer to check the storage requirements.

5 SUMMARY OF PIPELINE SYSTEM DESIGN

1. Think in terms of the energy line - the hydraulic gradient
2. Unless energy is put into the flow (eg pumping) energy must be lost and the total energy line must reduce in the direction of flow.
3. The pipeline pressure head is the difference between the piezometric level and the pipeline invert. I.e. it is a function of the pipeline profile as well as the hydraulic gradient.
4. Generally the velocity head is small relative to the pipeline pressure. Thus the pressure in the pipe can usually be taken as the difference between the total energy line and the pipeline level.

   BUT this is not true if
   - the pipeline velocities are high
   - the pressures are low

5. Draw the pipeline profile and total energy lines for the full range of flows and pipeline roughnesses.

   Consider the cases of maximum and minimum pumping conditions and zero flow.

6. Draw the system curves and pump curves over the full range of possible operation. Consider appropriate duty requirements for pump(s)
7. Consider potential run-out, need for throttling, NPSH requirements
8. At an early stage consider how the system is going to be controlled.

References

1. Miller, DS (1990) *Internal Flow Systems*; 2nd edition BHr Group Ltd

2. Idelchik

3. Barr DH & HR Wallingford (2006) *Tables for the Hydraulic Design of Pipes , Sewers and Channels*, 8th edition . Thomas Telford.

4. Manual of British Water Engineering Practice

PIPELINE AND PUMPING SYSTEMS DESIGN

1 GENERAL

A fluid moving through a system loses energy. To move the fluid that energy has to be available from gravity (potential energy) or imparted to the flow through pumping.

Figure 1 shows the basic concepts of a pumped system.

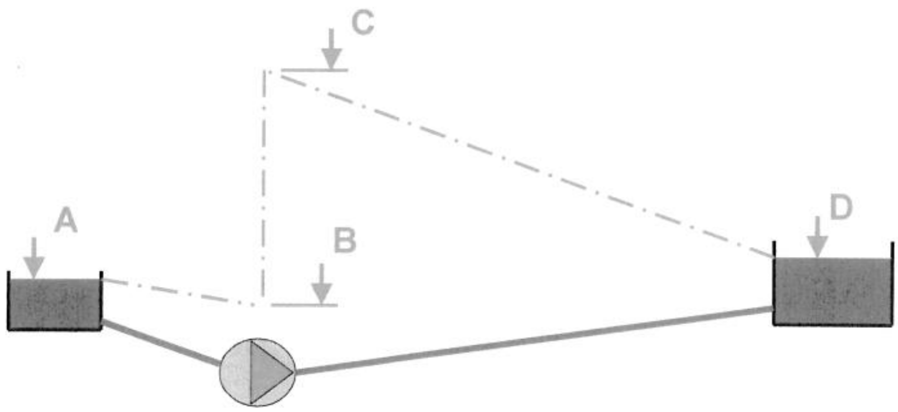


Figure 1 A basic pumping system

Flow is being delivered from a suction tank to a higher delivery tank. Energy is lost along the pipelines as indicated by the sloping lines from A to B and C to D. The energy available at B is less than that required to deliver the flow to D. So a pump is introduced to impart that shortfall

The energy head put into the system by the pump is thus the loss in the suction pipeline, the loss in the delivery main and the height difference between the suction tank water level and the delivery level.

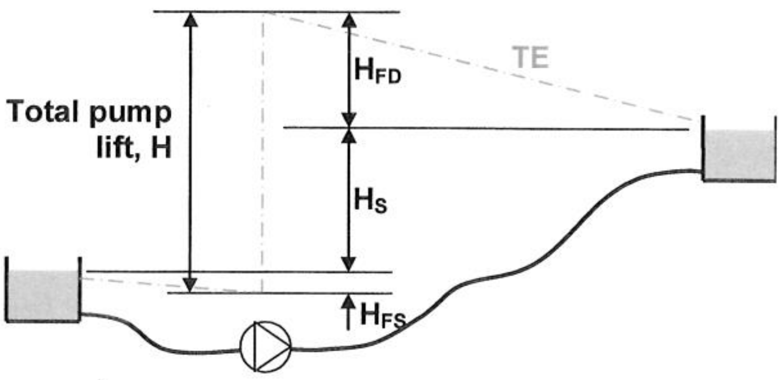


Figure 2 Definition of pump lift

Thus as shown in Figure 2, the total head lift of the pump, H

$$H = H_{FD} + H_{FS} + H_S$$

Where H_{FD} is the friction and fittings loss in the delivery main, H_{FS} is the similar loss in the suction main and H_S is the static lift between the suction and delivery tank levels.

(Note that if the pipe enters the delivery tank at the bottom then the relevant delivery level is the tank water level but if it is a top entry then it is the level of the entry pipe that represents the delivery level).

The energy at a point in the system comprises two elements – potential energy and kinetic energy. For a pipeline this is expressed as

$$E = p + z + V^2/2g$$

where p is the pressure head in the pipe (see Figure 3 below), z is height of the pipe above a datum and V is the velocity of the flow. E is thus the energy height above the datum, with p + z representing the potential energy and $V^2/2g$ the kinetic energy terms.

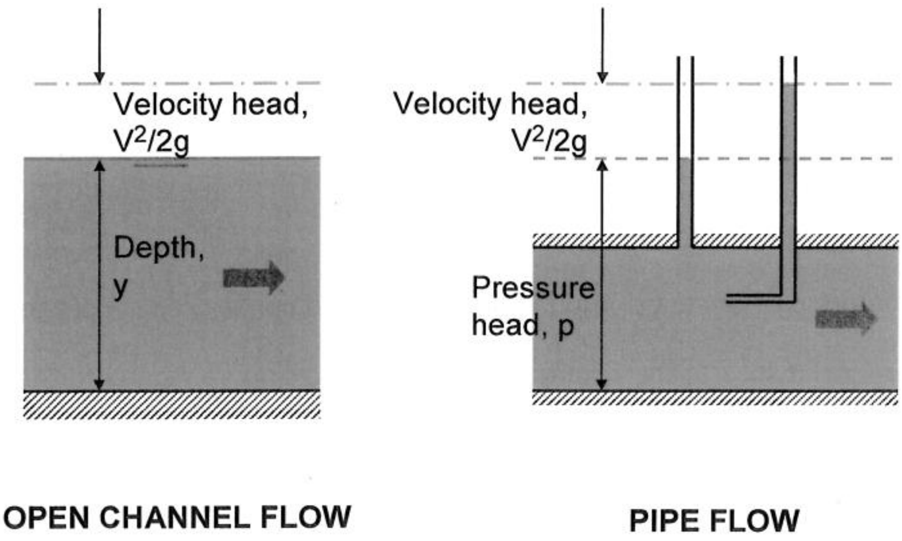


Figure 3 The reality of the energy equation

Energy loss or 'Head' loss in the flow of fluids in pipes arises from turbulence caused by

- the hydraulic resistance — or 'friction' — between the fluid and the pipe; and
- separation of the flow and the resulting turbulence at bends, valves, changes in cross section and the like.

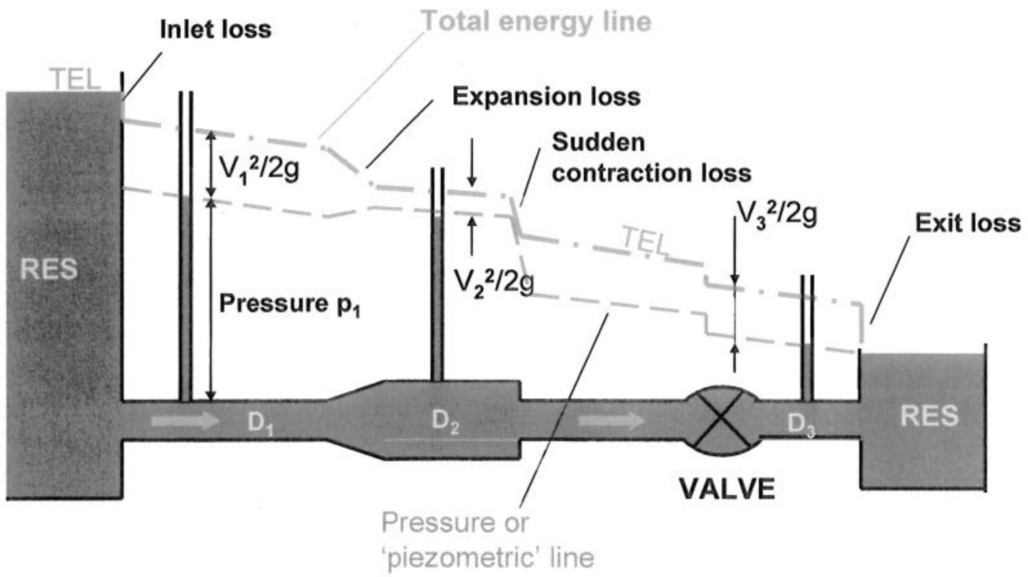


Figure 4 Head losses in a piped system

Figure 4 shows the head losses in a piped system. Note the following:-

- The energy level at the upstream end is given by the water level in the tank – there is no significant velocity and hence no velocity head
- There is a local loss at the inlet which is shown by the abrupt drop in the Total Energy Line.
- There is a friction loss at constant rate down the first length of pipe of diameter D_1, followed by a local loss at the expansion into the larger section of pipe.
- Velocities in that larger pipe are lower and the friction loss and hence gradient of the TEL are therefore less
- There are further local losses at the contraction and the throttling valve and further friction losses in the pipework of diameter D_3.
- Note in particular the loss at the outlet into the receiving tank. Here all the velocity is lost as the flow expand into the large body of water and hence the exit loss is 1.0 $V^2/2g$.
- The energy level in the receiving tank or reservoir cannot be higher than the water surface.

Total losses through a gravity system like this equal the difference in water levels between the two tanks. If uncontrolled the flow will adjust to give losses equal to that total head difference. If we control the flow using the valve then we increase (or decrease) the head loss at the valve such that the total head loss matches that required to pass the correct flow through the system.

Losses are calculated in terms of energy loss and thus apply to the total energy level. Unless energy is put into the system by pumping, the TEL must drop along the line as energy is lost.

However, the 'piezometric' or 'pressure' level is below the TEL by the height of the velocity head. Knowing the flow and pipe diameter at each point in the system, the velocity head and pressure level can be calculated. **NOTE** that the pressure level can rise in the downstream direction as, in the above example, at the gradual expansion into the large pipe.

In most pipeline work the difference between the TEL and the pressure level is small and can be ignored. For example flow in a pipe with velocity 1.5 m/s has a velocity head of about 0.1 m. Even for a low head situation the pipe pressure might be 10 m or more, so the velocity head component is a very small fraction of the total. BUT don't forget it and in very low head situations, such as pipework within treatment plant, and high velocity conditions, such as in the low-level release systems from dams, the velocity head may become significant.

Friction losses

Friction losses are calculated using one of the pipe loss formulae – preferably the Colebrook-White equation (or an explicit version of it) or from the Moody diagram.

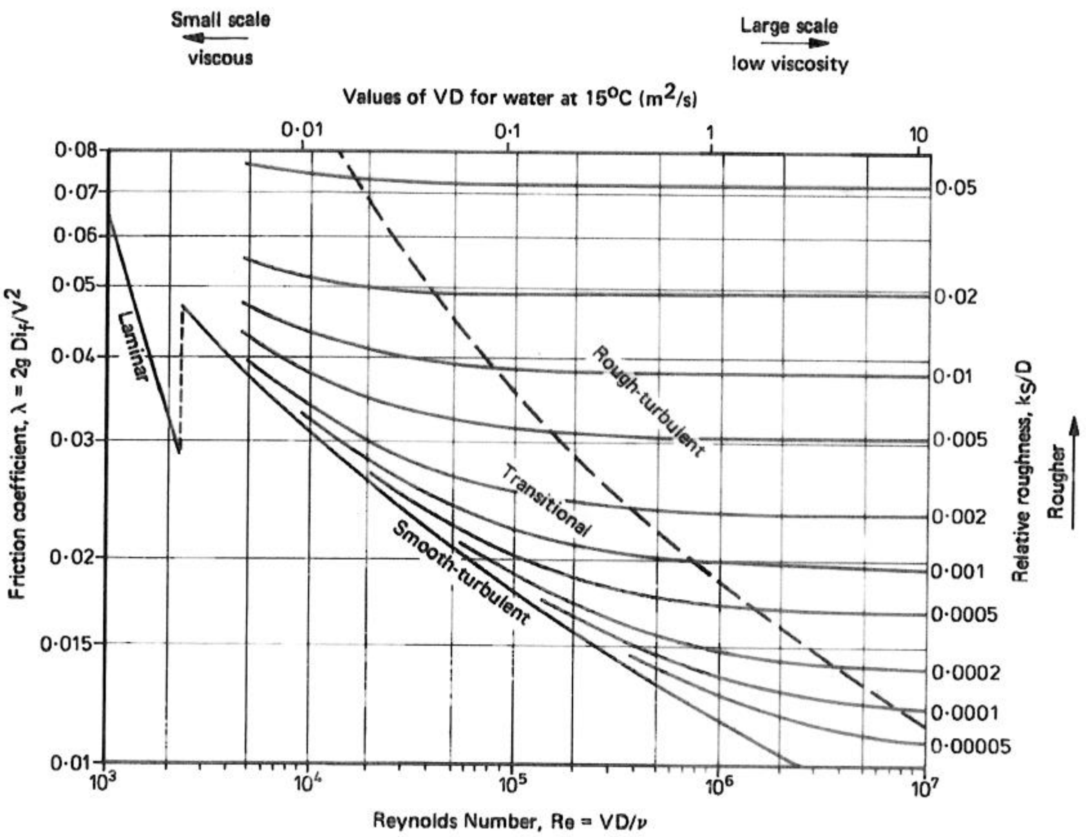


Figure 5 Moody diagram

The Colebrook-White equation:-

$$\frac{V}{\sqrt{2i.g.D}} = \frac{0.900Q}{\sqrt{(2i.g.)}D^{2.5}} = -2\log_{10}\left\{\left(\frac{k_S}{3.71D}\right) + \left(\frac{2.51.\nu}{D.(2g.D.i)^{0.5}}\right)\right\}$$

The Barr equation (an explicit version of the Colebrook-White equation):-

$$\frac{0.900Q}{\sqrt{(2i.g)}D^{2.5}} = -1.9\log_{10}\left\{\left(\frac{k_S}{3.71D}\right)^{1.053} + \left(\frac{4.932v.D}{Q}\right)^{0.937}\right\}$$

Where D is the pipe diameter, k_S is the surface roughness of the pipe, Q is the flow, v is the kinematic viscosity of the fluid and i is the gradient of the total energy line.

There is extensive data on roughness data for pipes – the most comprehensive reference being:

HR Wallingford and DIH Barr "*Tables for the design of pipes, sewers and channels*" 8th edition, pub. Thomas Telford 2006.

Local losses

Local or 'fittings' losses arise fro the turbulence caused by the change of geometry at fixtures and fittings in a pipeline system. These include bends, tees, expansions and contractions, valves and other devices that cause separation of the flow from the pipe surface and result in a turbulent wake.

Local losses are usually defined in terms of the velocity head of the flow:-

$\Delta H = K. V^2/2g$ where K is an empirical coefficient.

The best reference for head loss coefficients is

DS Miller "*Internal flow Systems*" 2nd edition pub. BHRGroup, 1990.

Miller's data is extensive and the most accurate but for design it may be prudent to include a safety factor and many other references will give higher values.

2 PIPELINE DESIGN ISSUES

Design process

Almost always the process starts with certain defined parameters:-

- Flow to be delivered

In the case of a main transfer pipeline this would probably be a max daily volume. If pumped then there may be constraints on the pumping hours.

For a potable distribution system, you need to know the peak demands, the variation of those demands daily, weekly and seasonally.

In the case of a sewage collection system, average and peak dry weather and peak storm flows at entry to the system.

- Source levels and possible variations in those levels
- Delivery locations and levels or, in the case of a distribution system, the minimum pressure to be maintained at supply points

The design process then covers some or all of the following activities but rarely in a nice logical linear sequence!

1. Decide route (approximately);
2. Initial sizing of pipe - ideally carry out optimisation of capital and operating costs. Consider pipe material.
3. Consider:
    - pipeline longitudinal profile and its influence on the hydraulic operation
    - overall system operation
    - need for pumping and number of stations
    - the system control philosophy and the range of potential operating conditions

4. Carry out initial hydraulic design and assess maximum working pressures.

5. Make an initial assessment of surge problems and consider need for surge protection.

6. If necessary, reconsider route and longitudinal profile.

7. Consider security and safety.

8. Finalise pipe size(s) and carry out detailed hydraulic analysis.

9. Define pump duties, number of pumps and range of operation.

10. Consider air valve and washout locations.

11. Finalise route and depths of cover.

12. Carry out structural design:-

    - consider soil loadings
    - vehicular and other live loadings
    - potential for internal sub-atmospheric pressures
    - temperature induced loads

13. Consider corrosion protection requirements

14. Consider need for pigging and/or swabbing

15. Finalise design of surge protection, valving requirements, thrust blocks etc.

16. Produce construction drawings, specifications etc

Further Design Issues For Potable Water Systems

- Security of supply - reticulation requirements?

- Deterioration of pipe lining in service - eg hard water? The roughness may increase with time

- Velocity limitations? No sediment to transport but may be restrictions on high velocity to prevent re-suspension of fine material.

- Potential for contamination entry at air valves, pipe joints. Negative pressures must be avoided

- Disinfection requirements? Residual chlorine levels? Possible need for re-chlorination in extensive system or long pipeline.

Further Design Issues For Raw Water And Sewage Pumping Systems:

- Security of supply - for raw water main to treatment works may need to consider twin pipes or local storage

- May need to transport sediment - possibly higher velocities

- Organic slimes likely to develop on pipe walls:

    - roughness may increase with time
    - disinfection may be required
    - provision for pigging/swabbing may be required

- Contamination probably not an issue - negative pressure may be acceptable under some conditions.

Simple gravity system

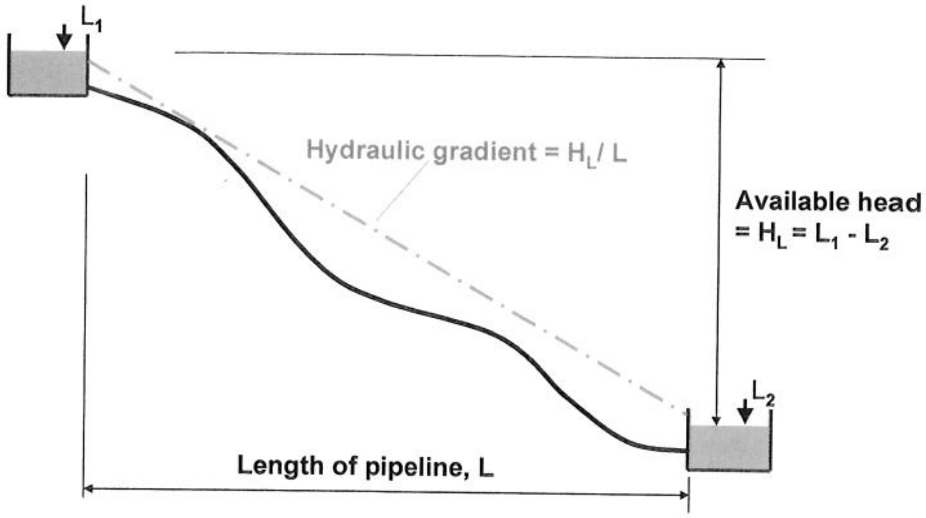


Figure 6 Simple gravity system

Figure 6 shows a simple gravity pipeline with flow passing from a high level tank (level L_1) to a lower tank (level L_2). As noted above, the flow in an uncontrolled system such as this is determined by the balancing of head losses with the available head.

In practice if we wanted to pass a given flow then we would try different pipe sizes to achieve that required flow – the larger the pipe, the lower the velocity and the lower the head losses. It is also highly likely that we would want to be able to control the flow – for instance when the downstream tank is full we would want to stop the flow.

A control is therefore needed at the downstream end of the system. If provided at the upstream end then closing the valve would result in negative pressures in the line:

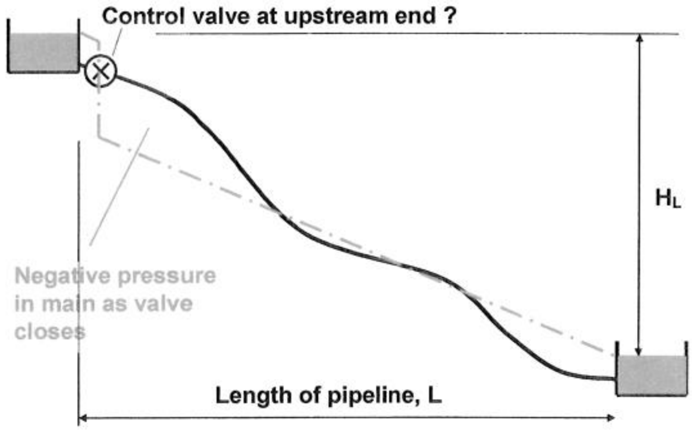


Figure 7 **Control valve at upstream end of system?**

It is necessary therefore to provide the control valve at the downstream end of the pipeline so that closing valve maintains and increases the positive pressures

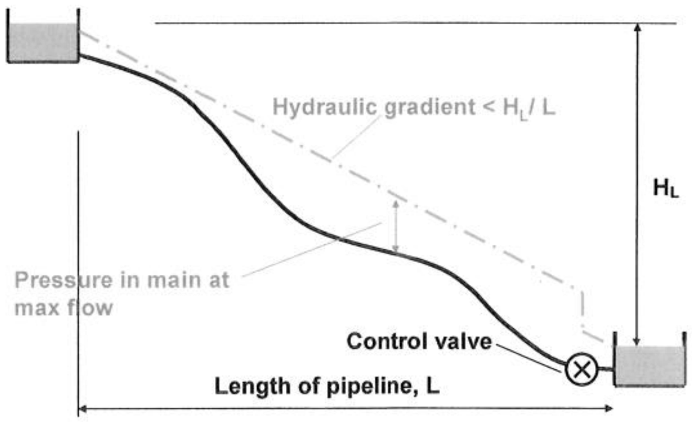


Figure 8 **Control valve at downstream end of system.**

Maximum pressures occur when the valve is fully closed and static pressures occur throughout the line.

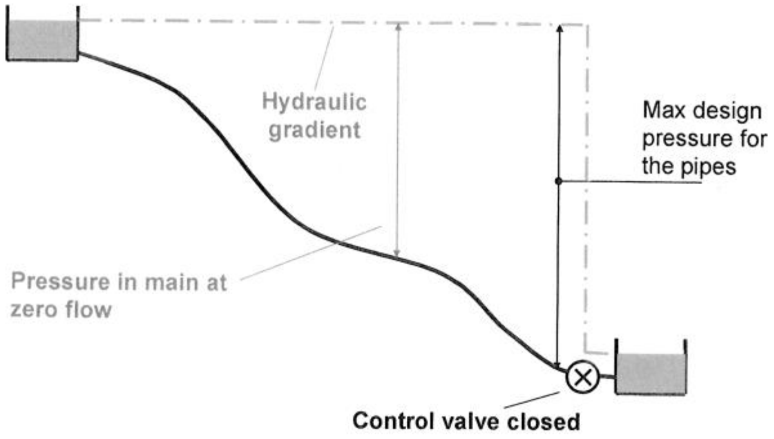


Figure 9 Maximum pressures in system

Figure 10 shows a similar system but with a more adverse profile. Two problems:-

- How do we achieve a design that maintains positive pressures through out?
- How do we deal with the maximum pressures that could occur?

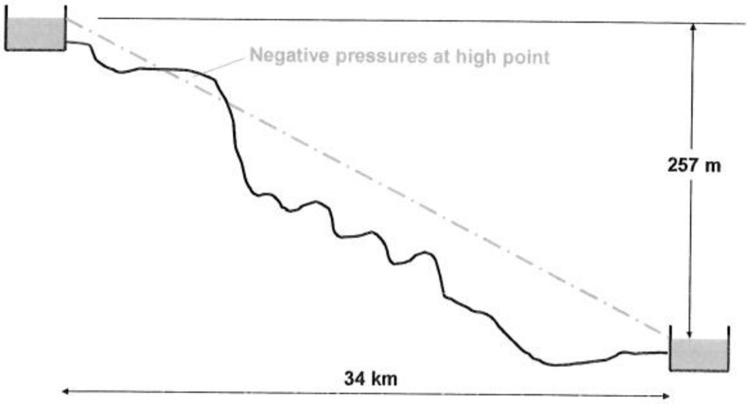


Figure 10 Simple gravity system but with potential negative pressures

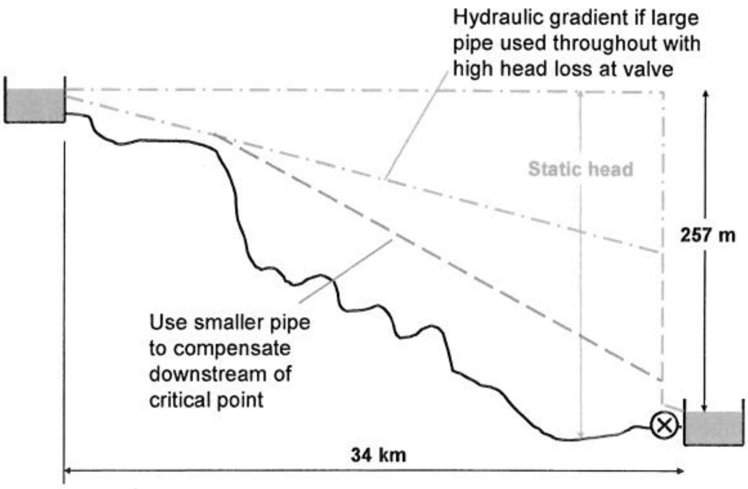


Figure 11 Use of larger pipe to ensure flow gets over the high point

Figure 11 shows one option to prevent negative pressures which is to increase the pipe size. This decreases the velocity for the design flow and flattens the hydraulic gradient but if this same size pipe is used throughout then it results in a high head loss if the flow is to be controlled at the valve.

That available head however can be used to size a smaller pipe downstream of the critical point thus not only reducing the energy thrown away but also recovering some of the cost of the larger pipe used upstream.

However there is still the question about maximum pressures in the system. Two points to consider:-

- The control valve has to be able to control the flow over the full range and at low flows the head difference across the valve is well over 200 m. This is extremely high and although we are not covering valves in this lecture it is worth remembering that a special high head control valve would be needed.

- The maximum pressure that the pipe has to withstand will be under static conditions with the valve closed. The maximum pressure head in the pipe is in excess of 260 m. Most types of pipe come in standard pressure classes – Ductile Iron (DI) pipes, for example, come in standard pressure classes of 6, 10, 16, 24 and 36 bar. Obviously the higher the pressure rating, the thicker the pipe wall and the greater the cost. Thus it is inherent on the designer to try and minimise maximum pressures and hence the cost of the pipes. 26 bar pressures would require a fairly expensive pipe and may rule out certain materials such as Polyethylene (PE)

Figure 12 shows one possible way of reducing the pipe pressures – the inclusion of a break-pressure tank. Maximum pressures are now reduced to less than 16 bar (160 m) so a much less onerous condition for the pipes. Note that there is still a large head loss at the entry to the break-pressure tank and a smaller pipe might be used downstream of the high point.

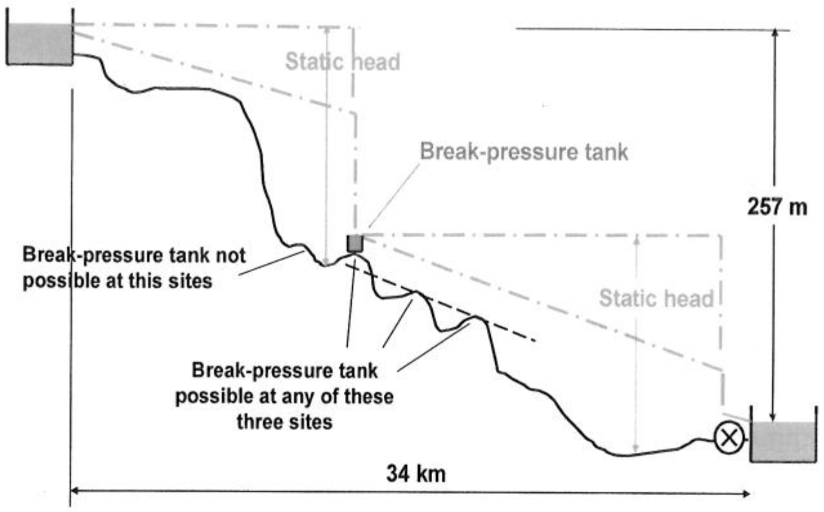


Figure 12 Use of a break-pressure tank

Control of the flow into the break-pressure tank needs to be considered but this could be simply achieved by a float valve. The tank fills up until the valve starts to close and if the tank fills – if no demand downstream – then the float valve just closes. Demand downstream is controlled by the downstream valve so as that valve is opened, the water level in the break-tank drops and the float opens as necessary.

Note too, that a pressure-reducing valve cannot be used to protect the downstream pipeline from high static pressures. To work effectively a PRV needs a flow through it and it cannot be guaranteed to be drop tight at no flow. Hence the use of a break-pressure tank in this instance. PRVs are widely used, for example, in distribution systems, particularly to reduce zonal pressures and leakage at night when demand is low but they do rely on there being some flow to operate properly.

Air valves

Before going on to look at pumping systems we need to aware of the impact of air valves on the operation of pipeline system whether gravity or pumped.

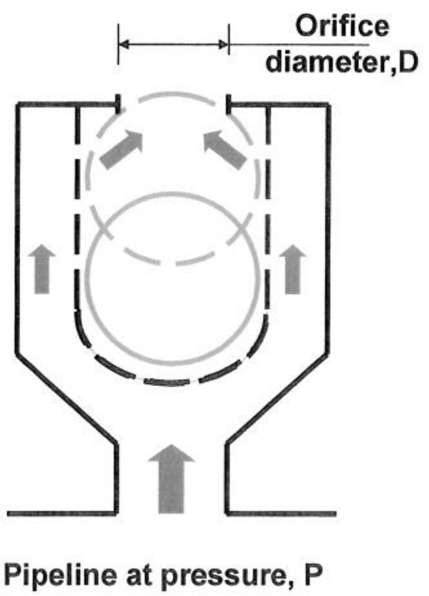
Air valves are used for at least three purposes:-

- To allow air to be exhausted from the pipeline when filling the line and to allow air back in if the pipeline is being emptied

- To discharge air that may collect in the top of the pipeline during operation – i.e. when the line is under pressure

- As surge protection measures. (But, beware, air valves can also cause severe surge problems!)

There are two basic types of air valve developed primarily to deal with the first two requirements



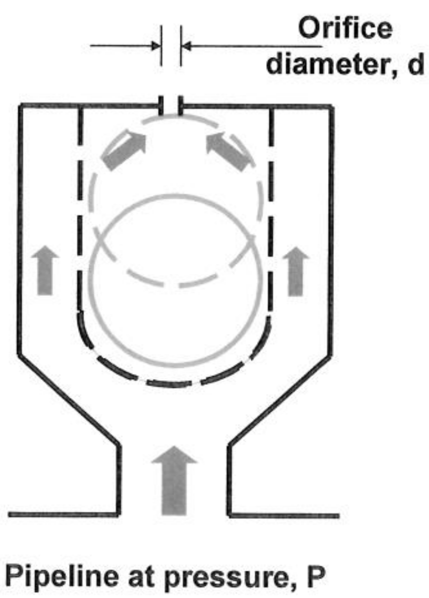


Figure 13 Large-orifice air valve **Figure 14 Small-orifice air valve**

The difference between the two types is only in the size of the orifice but that has a marked difference on the way each valve works.

The **'large-orifice' (LO) air valve** is used for emptying and filling a pipeline.

- When the pipeline is empty the float is held in a cage and air can escape around float as pipeline fills.
- When water enters the valve the float rises up to seal the orifice. The closing force generated by the difference in pressure between the pipeline pressure and the external atmospheric pressure is greater than the weight of the float so the float is held shut against orifice even if air collects in valve.
- Only when the pipeline pressure drops down to or below atmospheric (e.g. on emptying) can the float drop down and the valve re-open.

The **'small-orifice' (SO) air valve** is used for bleeding off air when the line is under pressure. It works in exactly the same way as the large-orifice valve but the closing force generated by the pressure difference across the small size of opening is less than the weight of the float. Now if air collects in the valve during pipeline operation the float can fall away from the hole and allow air to be bled off, closing again only when the water level rises and the float is lifted against the opening.

A **double-orifice (DO) air valve** is imply the two types of valve in a single unit, which is useful as the obvious location for both types of valve is at high points along the pipeline route

There are many variations on these basic concepts and each valve manufacturer has his own designs. There are designs developed for sewage applications and especially for surge applications where the problem is that LO valves prevent low pressures in a pipeline by opening and letting in air but when the pressures rise and the air is exhausted they can slam shut generating high shock pressures. So there are valves that let air in but not out and vice versa and special valves that claim to be anti-slam.

Generally air valves are a necessary evil though as they are mechanical units with moving parts which need regular maintenance. Usually located in chambers along the pipeline route they are very rarely maintained or even thought about. So if you can avoid their use do so.

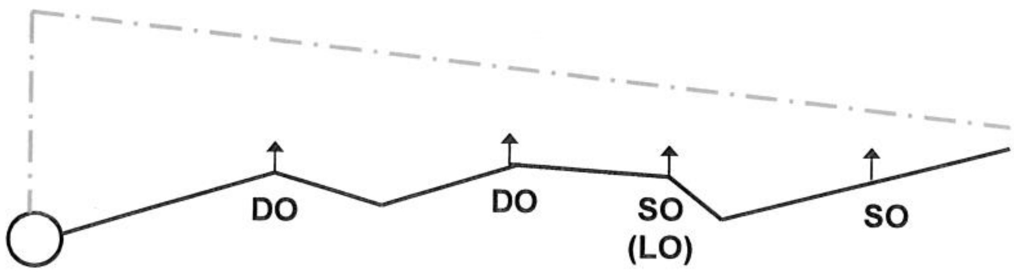


Figure 15 Siting of air valves

Figure 15 shows the typical locations where air valve should be sited. Generally LO valves are required at high points and should be considered at downward increases of slope where air might get trapped. SO valves are also required at high points, relative both to datum and to the hydraulic gradient; at changes of gradient, particularly where pipeline steepens in downward direction, and at regular intervals (< 1000 m) on long rising or falling lengths of pipeline.

The reason that air valves have been discussed in some detail is that they can have a significant effect on the way a system operates. Figure 16 shows a simple gravity system but with a high point on route. The hydraulic gradient with a single optimised pipe size drops below that high point indicating negative pressures and potentially siphonic flow. In theory the magnitude of the flow is dictated by the head difference between the two tanks (assuming that the negative pressure is not down to full vacuum)

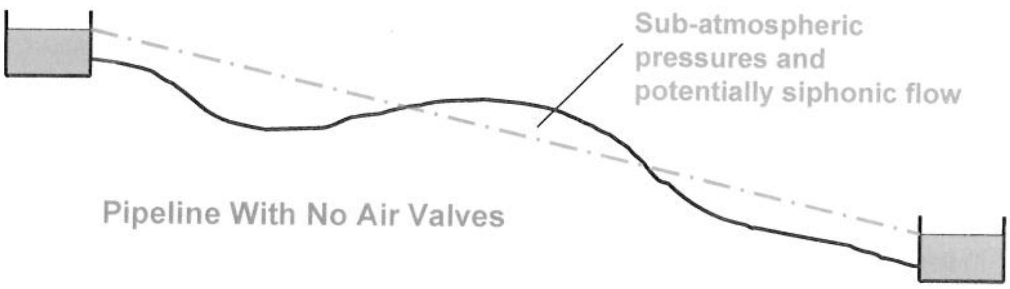


Figure 16 Gravity system with no air valves

Whether or not siphonic flow would be acceptable, in practice there would almost certainly be a Large-Orifice air valve at the high point. (Probably an DO valve containing an SO valve as well). Negative line pressures at this point mean that the LO valve would open, air would enter the line and atmospheric pressure would occur.

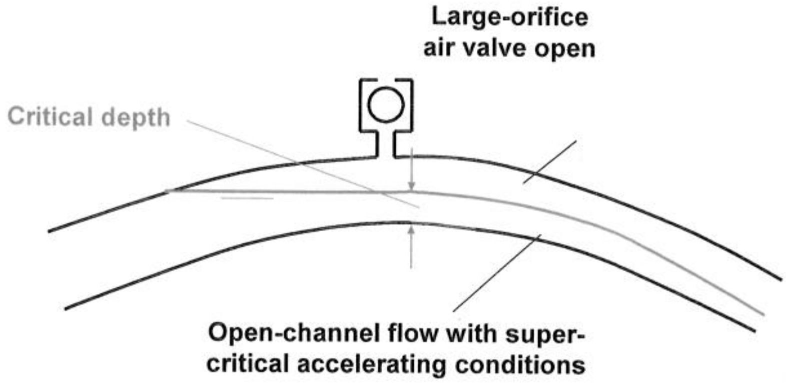


Figure 17 Detail of conditions at the summit

Figure 17 shows the details of what would actually occur. There would, in effect, be weiring flow over the summit with fast supercritical flow in the downstream pipe. Open-channel flow will continue down the pipeline until the it reaches the point at which the pipe full flow re-occurs. The magnitude of the flow would now be controlled by the head difference between the upstream tank level and the energy level at this high point. Figure 18 illustrates that the hydraulic gradient is now flattened. This can only result in a lower flow. So the presence of the high point and the inclusion of the necessary air valve affects the hydraulics of the whole system.

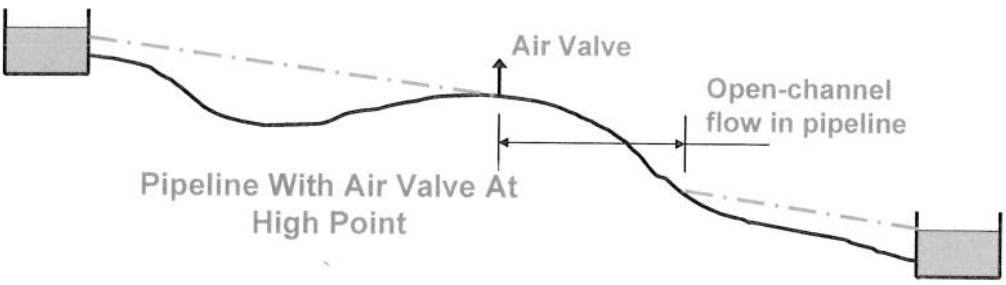


Figure 18 Pipeline with air valve

Both this and the previous example show how the profile of the pipeline affects the hydraulics of the system and it is absolutely essential in all cases to draw out the profile and superimpose the hydraulic gradient(s). This will tell you more than any other information whether the system will work as you think it will or whether there are problems. We will return to this in the System Design discussions.

3 PUMPING SYSTEMS

Definition of terms

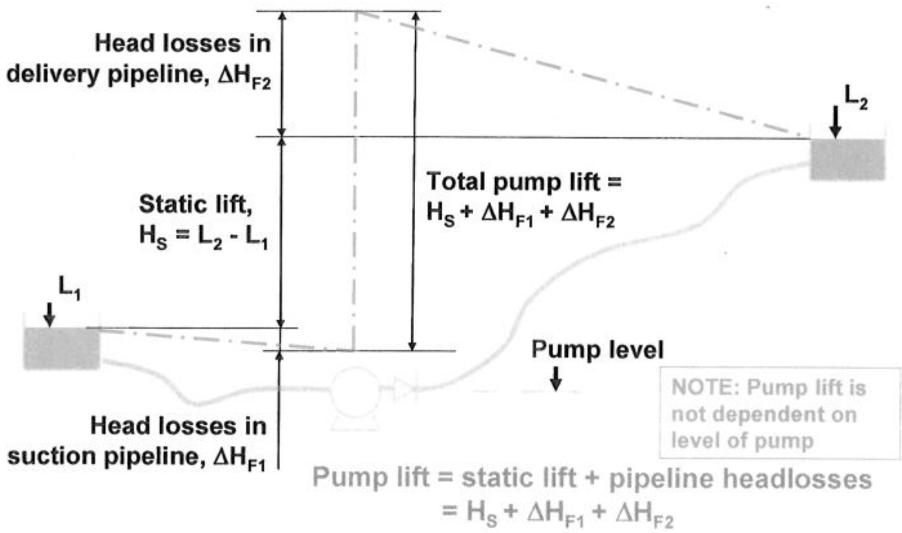


Figure 19 Definition of terms in a pumping system

We started of with a similar diagram but it is worth just stressing some of the points again because this diagram is fundamental to understanding of how a pumping system works.

The **static lift, H_S** of the pumps is the height between the water level in the suction tank or sump and the delivery level. If the delivery pipe is connected into the receiving tank at low level then the water level in that tank is the delivery level. If the pipe comes into that tank at high level above the water surface then it is that inlet pipe level that determines the delivery level.

The static lift can be thought of as the pump lift required before any flow is delivered down the line.

The **dynamic losses** in the system are the sum of the losses in the suction and delivery mains. Those losses comprise the friction and fittings losses, $\mathbf{H_{F1}}$ and $\mathbf{H_{F2}}$. Both the friction and fittings losses are functions of the velocity and hence the flow and in both cases quite close to being proportional to V^2 or Q^2.

The **pump lift** or '**pump head**', **H**, is the sum of the static and dynamic components. i.e.

$$\mathbf{H = H_S + H_{F1} + H_{F2}}$$

If we now plot the pump lift, H, against flow we get a curve as shown in Figure 20. It is known as the system curve and represents the required pump lift to deliver a given flow through that system.

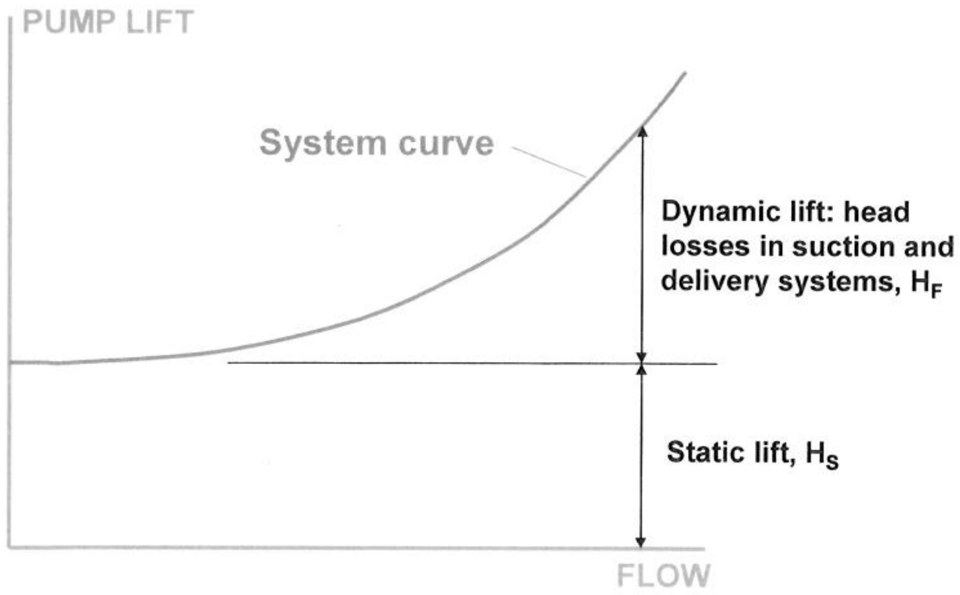


Figure 20 A typical system curve

The curve cuts the pump lift axis at the value of the static lift – the head necessary to deliver any flow at all. Note that if pumping downhill H_S would be negative. As noted above the dynamic losses are nearly a function of Q^2 so the shape of the curve is parabolic.

NOTE: The system curve is not a function of the pump level nor of the type of pump nor its characteristics. It is only a function of the system pipework and fittings.

Pump duty

Figure 21 shows how the curve is used to define the required pump lift at the design flow rate – the **pump duty.** We can now choose a pump to meet that duty

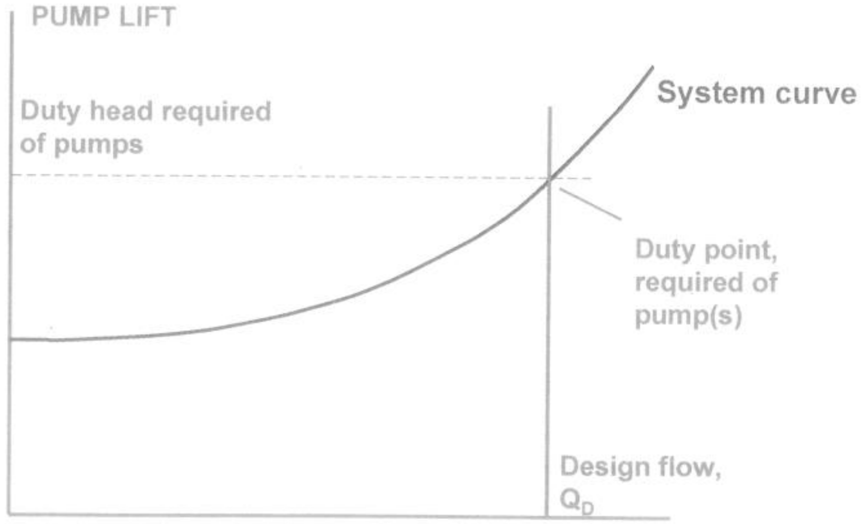


Figure 21 Determination of pump duty

However it is not quite as easy as that. For a start there is never just a single system curve. A number of parameters are subject to uncertainty and/or variation:-

- The static lift will change as the levels in the sump and delivery tank change. It will be a maximum when the sump is at its lowest level and the delivery tank is full and a minimum when the reverse is true.

- There is uncertainty over the pipe roughness and consideration must be given to the deterioration of the pipe in service – the 'as-new' surface roughness of the pipework is unlikely to be maintained over time, and certainly not if pumping sewage as slime and biological fouling affects the pipe.

- There is even uncertainty over the fittings losses (though this is not usually considered unless the fittings losses are a major part of the total loss – as may be the case in the short pipe runs within a treatment works for example).

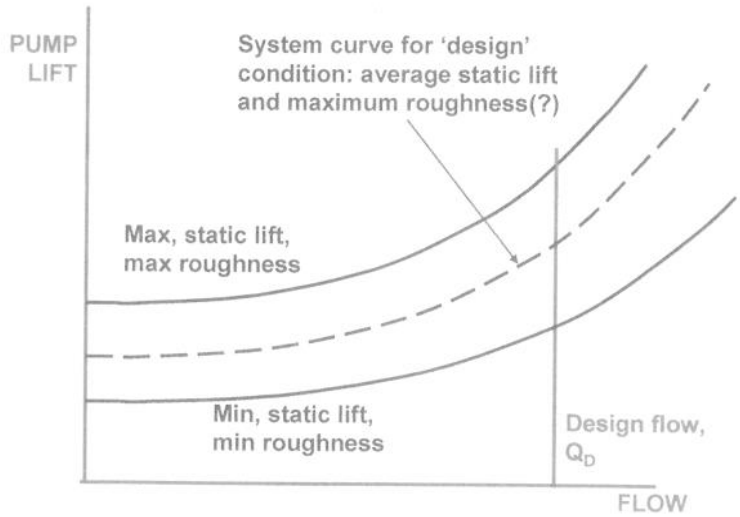


Figure 22 Envelope of system curves

Figure 22 illustrates the potential envelope of pumping conditions with system curves drawn for the conditions of

- Maximum static lift and maximum roughness
- Minimum static lift and minimum roughness, and
- an intermediate condition of average static lift and maximum roughness

We now have a decision to make in defining the pump duty as at the design flow rate there is a range of pump lifts that meet that flow under different conditions. Figure 23 illustrates the problem with illustrative pump curves added. We could define the pump duty at any point between A and C.

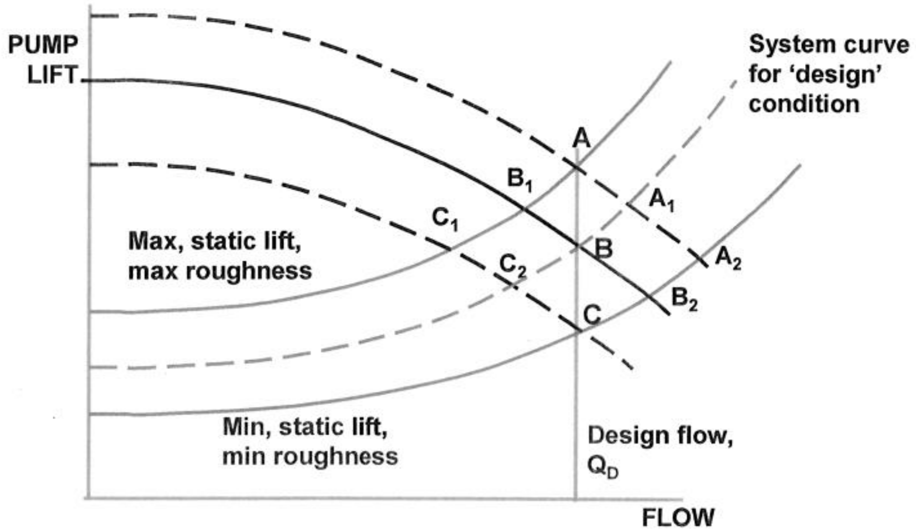


Figure 23 Potential operating range of a single fixed-speed pump

The easiest answer is to require the pump to meet the maximum duty lift – i.e. point A. We know then that whatever the roughness or static lift the pump will provide the design flow, Q_D. This might well be required when pumping into a distribution system for example – if you have no storage in that system then you must be able ot meet the maximum demand at all times. However at any condition other than the most onerous the pump will provide more flow than necessary working along its curve between A and A_2. It means possibly providing a larger pump and a bigger motor than may be needed. Remember too that for the maximum static lift the sump is empty and the delivery tank full – the pump is about to be turned off!

Clearly we cannot choose a duty at point C. Here the pump meets the design flow requirement only under the most advantageous conditions and will normally be operating at points along its curve between C_1 and C.

Point B however might be a more economic duty. It represents the pump head requirement at the design flow under conditions of average static lift and high roughness. As the static lift changes the pump will be operating at flows between that at B_1 and that at B_2. On a time-averaged basis that may be acceptable. Certainly for a system transferring a daily volume of water between storage reservoirs, the instantaneous flow is unimportant provided that daily rate can be met.

Ultimately the design duty for the pump is a decision the system designer must take but remember it is not always in the client's best interest to define the maximum possible duty.

The above discussion was based on the premise of a single fixed-speed pump required to match the duty. With variable-speed pumps there is more flexibility in defining the pump duty but even so the full range of possible pumping conditions needs to be considered. Moreover in most systems two or more pumps in parallel are used to meet the design duty as even with fixed-speed pumps there is greater flexibility in their operation and the stand-by pump provision is not so onerous. (e.g. with a single pump 100% stand-by capacity is likely to be needed; with two pumps only 50% stand-by provision may be required. With 3 pumps probably only 33% stand-by capacity is likely to be required).

Figure 24 shows the system curves for the same system as before but now with the duty being met by two pumps operating in parallel. With pumps in parallel the head to which they must deliver is the same and the flows are additive so the right hand higher pump curve in Figure 24 represents the combined output fro two pumps whilst the left hand lower curve represents a single pump.

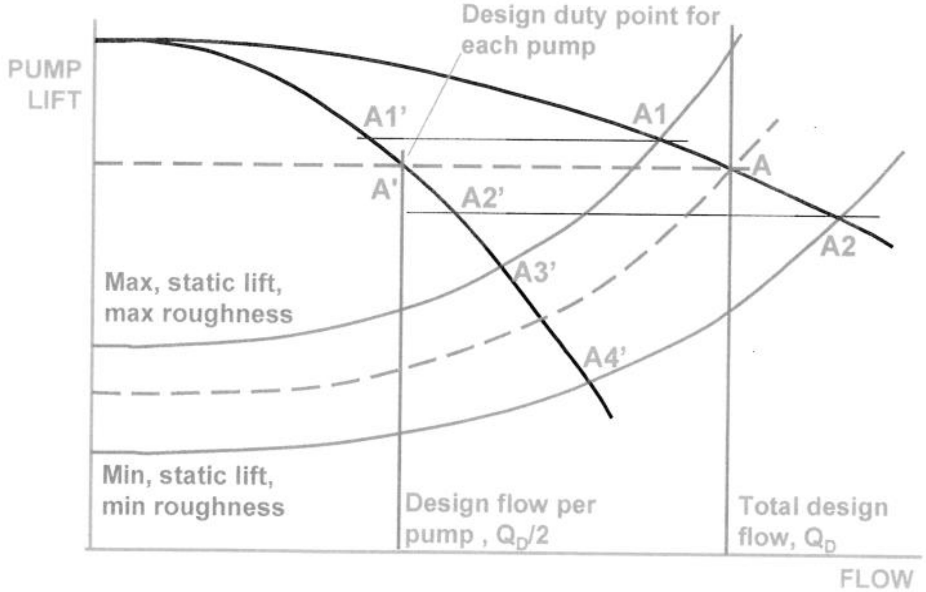


Figure 24 Two pumps in parallel

Defining the total pump duty point as point A (average static lift, maximum roughness) then the combined flow from the two pumps is Q_D and the design duty for one pump is $Q_D/2$ at the same pump lift as at point A. The duty for one pump is therefore point A'. With two pumps operating together their total output varies along the combined pump curve between points A1 and A2. The equivalent variation in output for each pump is between A1' and A2'.

However it would also be possible to operate the system with a single pump on its own. Now the output would vary, depending on the static lift, between points A3' and A4'. Thus depending on whether one or two pumps wer operating at the same time, the output from a single pump could vary between points A1' and A4'.

This has implications on the specification of the best efficiency point for the pumps. With a single fixed-speed pump it would normally be the case that the best pump efficiency should be at or close to the defined duty point. However in a two pump system that need not be the case. If for example it were expected that operation would normally be with a single pump then it may be better to require the best efficiency to be in the range between A3' and A4'.

Again this is a decision of the design engineer.

With three pumps operating together in parallel the system and pump curves are illustrated in Figure 5. The dashed line represent the output from a single pump and two pumps combined; the solid line is the combined output from three pumps. The duty point for the total pump output is point 1 and with each pump providing a third of that flow at the same head the design duty for each pump is given by point 1'.

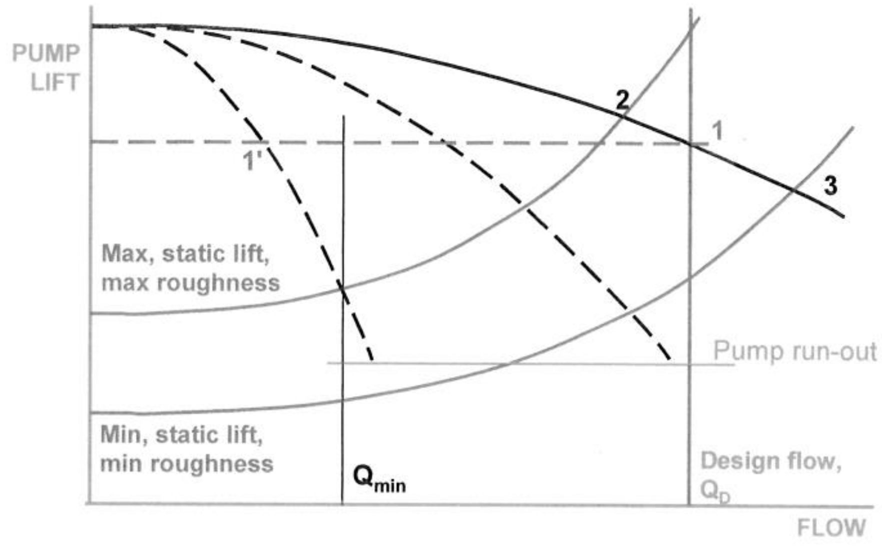


Figure 25 Multiple pumps in parallel

One further point needs to be checked, again illustrating why it so important to plot the system and pump curves. As shown in this figure, if one pump were to operated alone there is a risk that it would operate outside the acceptable conditions for that pump – i.e. beyond the run-out point of the pump. The run-out point is tied up with the NPSH requirement and represents the conditions at which cavitation starts to become unacceptable around the pump impeller. If it were possible to operate on eof the pumps in this condition then it would be necessary to provide throttling facilities to increase the pressure downstream of the pump.

4 PUMPING SYSTEM DESIGN

Figure 26 shows a simple pumping system with flow drawn from a sump close to the pump and delivering to a tank at higher level. The pipeline profile is rising all the way with no high points along the route.

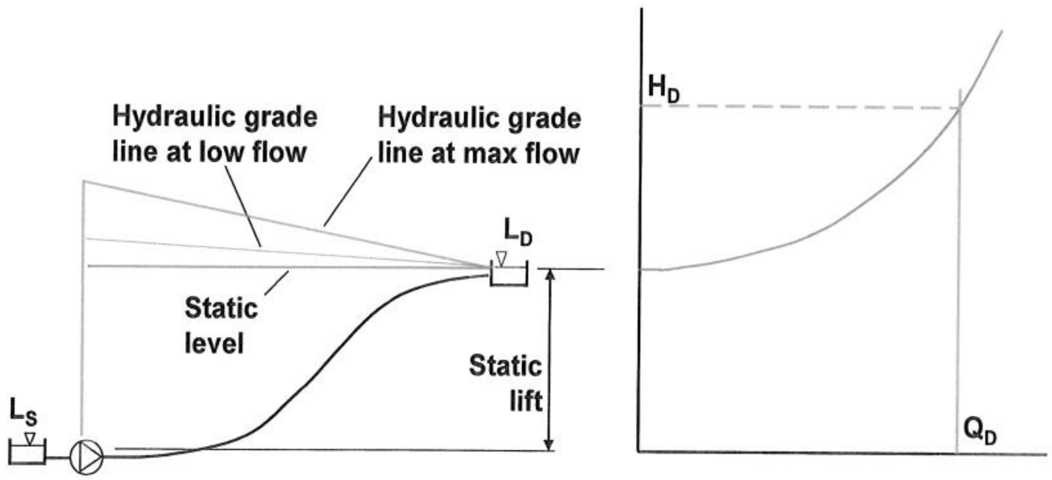


Figure 26 Simple pumping system

On the left hand diagram are shown the hydraulic gradients at different flow rates and the diagram on the right shows the system curve. (For simplicity we will assume a single system curve for this illustration). The static lift at all flows is the lift from sump level L_S to delivery level L_D.

Figure 27 shows the same system but with a profile which has a high point between the pumps and delivery tank.

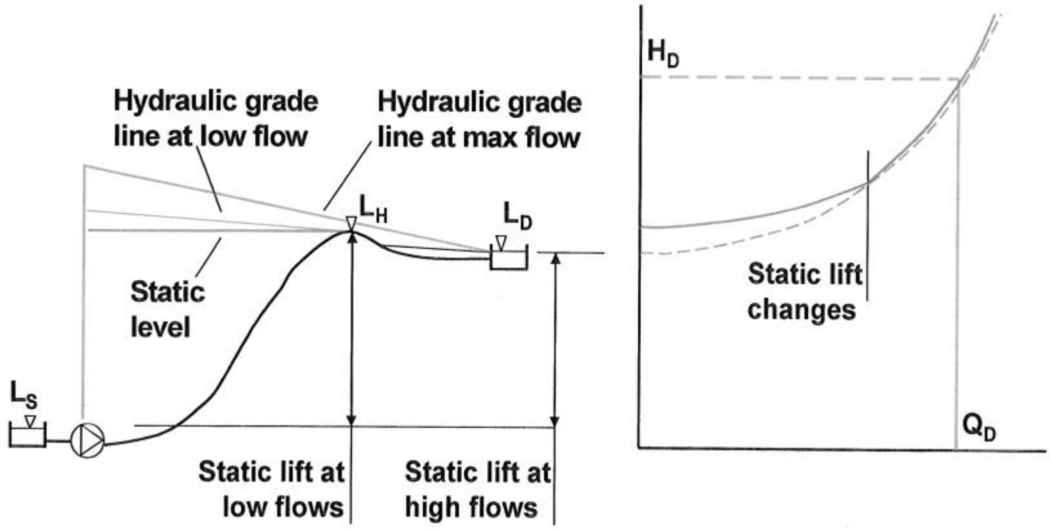


Figure 27 Simple pumping system with high point on profile

Under static conditions the section of pipe upstream of the high point will remain full (the non-return valve prevents back flow) but the section downstream of the high point will drain down to the discharge level L_D. The static lift at low flows is thus the lift necessary to reach the summit. The flow will be delivered to that point and, as there would almost certainly be a large orifice air valve at the summit the flow will run down the downstream section as open-channel flow until the point is reached at which the hydraulic gradient for pipe full flow dictates that the pipe runs full. As the flow is increased eventually the hydraulic gradient in the downstream section of pipe will be steep enough to require that section of pipe to run full. The air valve will then close and at all higher discharges the pipe will run full throughout and the pumps will deliver direct to the discharge tank. Thus at some point the static lift will change from the lift to the summit (at low flows) to the lift to the delivery tank (at higher flows). The losses in the pumped system at low flows will be only those in the upstream section of pipe so although the static lift is greater the slope of the system curve will be flatter as illustrated in the right hand diagram in Figure 27. The system curve is now a combination of the two system curves.

Figure 28 shows the same system but with a yet higher summit along the route. Now, under all flows the pump is delivering to the high point and the static lift remains constant but a higher value. System losses as seen by the pump are in the upstream section of pipe only so the system curve is illustrated by the higher line in the right hand diagram in Figure 28.

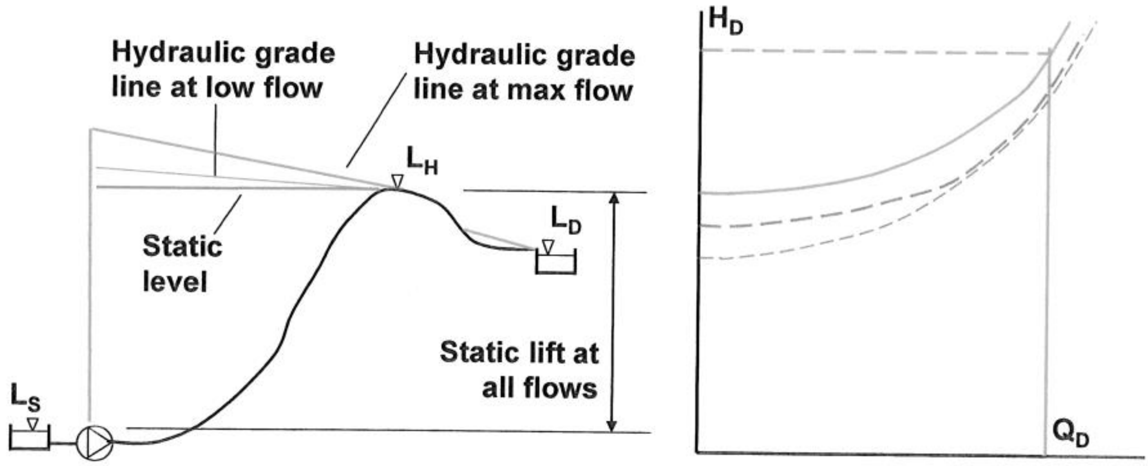


Figure 28 System with high point well above delivery level

So be aware that:-

- Pumping over a hill to a lower delivery level will change the system curves and may change the pump duty requirement.
- It will also potentially lead to conditions with the air valve at the high point open and free-surface flow in the downstream pipeline.
- In a potable system this is generally NOT ACCEPTABLE because the open air valve is a potential entry point for contamination. It is normally required to maintain a positive pressure of at least 0.5 barg (5 m head) in potable distribution systems even under transient, surge conditions.
- Open-channel flow may be acceptable in a raw-water or sewage pumping main.

So avoid pumping over hills if possible! But if no alternative what can we do?

Potable water systems

In a potable water system we need somehow to keep the downstream pressure up so that a positive pressure is maintained at the summit.

In a raw water or sewage pumping system if we allow open channel flow downstream of the summit the we have to be careful about the design details.

Figure 30 shows one possible option for a potable water system – a pressure-sustaining valve at the discharge point. The simplest solution would be to set the valve to maintain a pressure in the system just above the level of the high point, bBut this does mean throwing away a lot of energy through the valve.

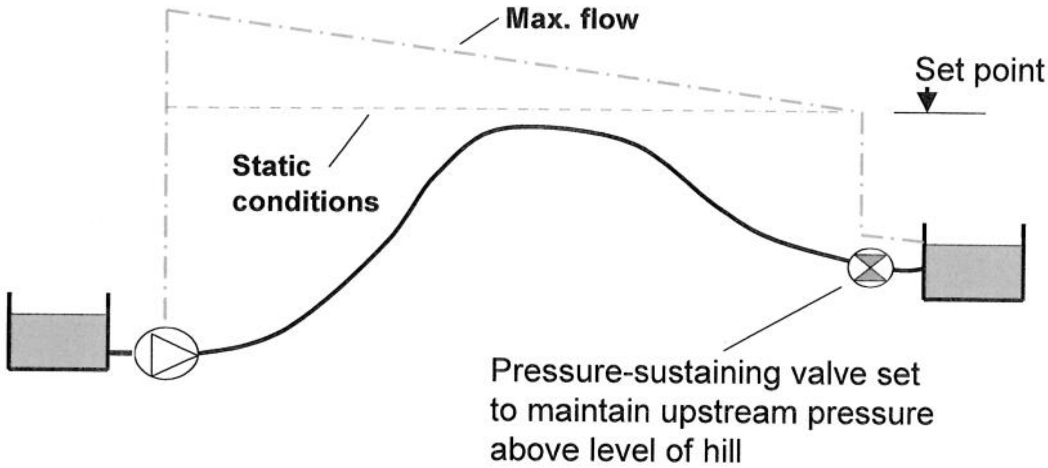


Figure 30 Potable water system – pressure sustaining valve (1)

An alternative is shown in Figure 31. Here the prv is set to maintain that same pressure but at the summit not at the position of the valve. Thus, as the flow increases and the hydraulic gradient gets steeper, so the valve opens and reduces the energy loss through the valve. The disadvantage of this system is that telemetry is now needed between the valve and a pressure sensor at the summit.

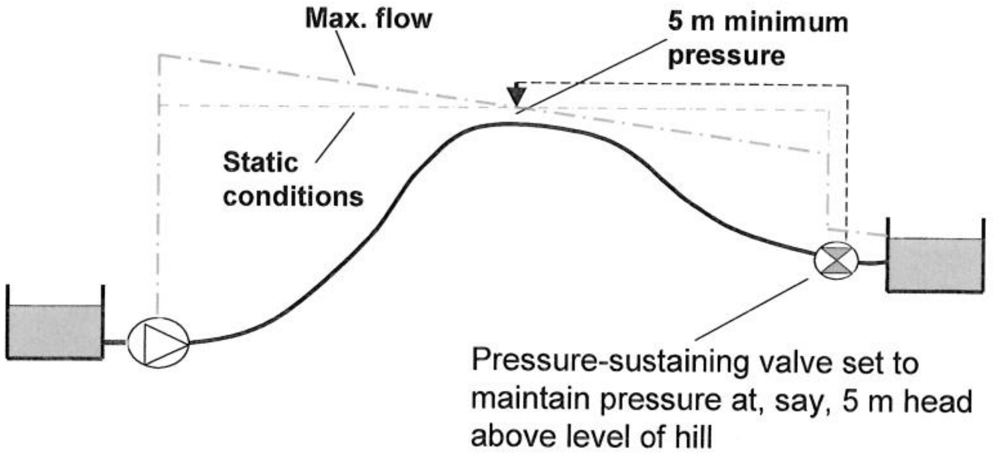


Figure 31 Potable water system – pressure sustaining valve (2)

Another possible option is to have a tank at the top of the hill acting to break the pressure so that the pumps always deliver to that tank and there is gravity flow in the downstream leg.

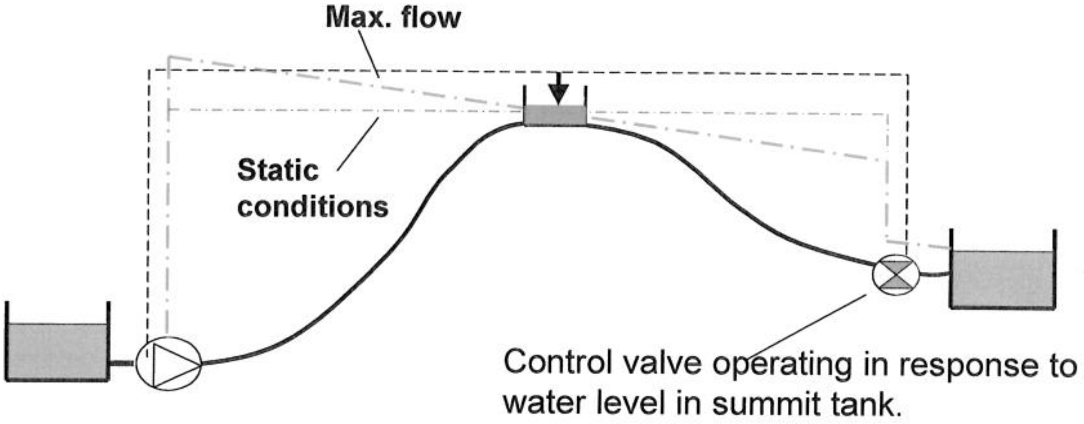


Figure 32 Break-pressure tank at top of hill

This reduces the pump lift but requires the addition of a tank at the top of the hill. It also requires telemetry to the pumps, so that they operate to maintain the tank full or at least to prevent it emptying, and to the downstream valve which would operate to keep the pipeline full. This type of system may enable the use of a smaller pipe in the downstream leg and it does have advantages from a surge point of view.

Raw water and wastewater systems

As noted above in a raw-water or wastewater system, contamination is rarely an issue and it is possible therefore to operate with open-channel flow in the downstream leg as illustrated in Figure 33.

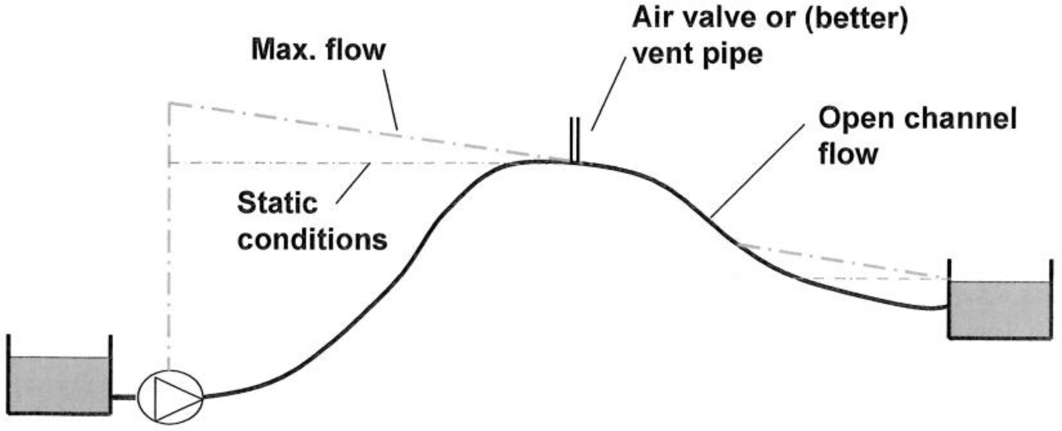


Figure 33 Pumping over a hill – wastewater system

It may also be possible to replace the air valve with a vent pipe – always worth doing as an air valve is a mechanical device and needs regular maintenance. If you do design for this mode of operation then the open channel section of pipe should run at no more than about 2/3 depth to allow the free movement of air in the space above the water.

Also there is a need to consider carefully the downstream leg if the profile is undulating. Air pockets can get trapped if the pipe profile rises up and down. It is also necessary to consider how long it takes for the system to reach equilibrium after the pumps start. If the sump is small then it may well be the case that the pumps turn off before the 'steady-state' conditions have been established. Velocities may be less than assumed in the design and, if self-cleansing velocities are not achieved then there may be a problem with sediment transport and deposition in the system as illustrated in Figure 34.

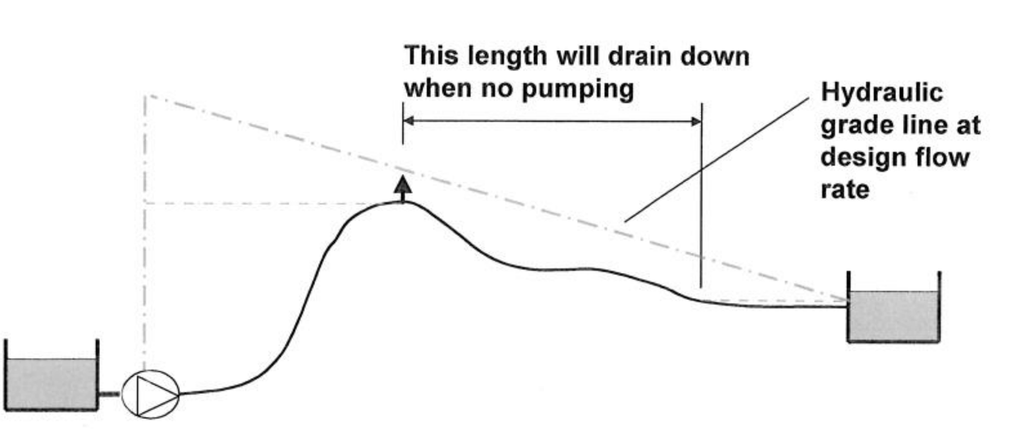


Figure 34 Potential problems of air trapping and sediment transport

Further issues of system design

Figure 35 shows a system with two reservoirs. Analysis requires balancing flows and pressures at the connection point. The actual distribution of flows depends on the relative levels of the two tanks and the size and length (hence head loss) in the two connecting pipes. Remember that although it may be possible to achieve a requires flow split by balancing head losses at the design flow by adjusting levels and pipe sizes, that flow distribution will not be maintained at other flow rates.

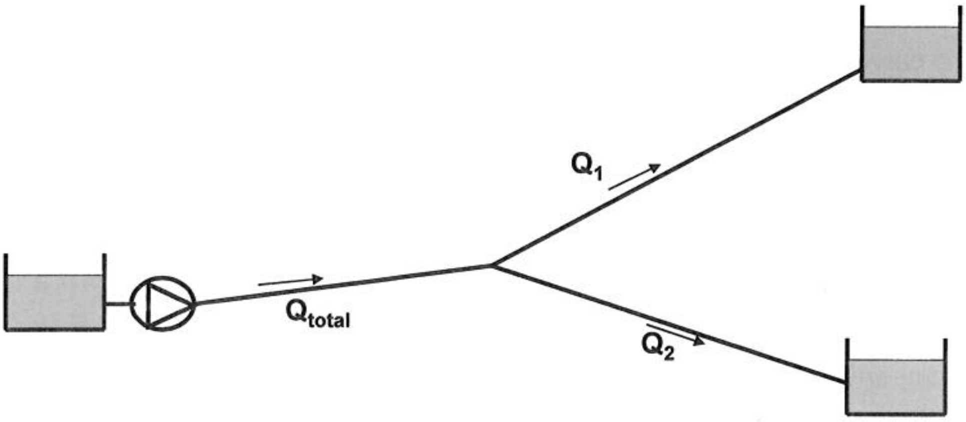


Figure 35 Pumping to two tanks

If it is necessary that a particular balance is required then a adjustable flow control valve may be necessary on one of the branches. Figure 36 shows this arrangement and it may be necessary to include a non-return valve on one or both legs to prevent back flow from one line to the other.

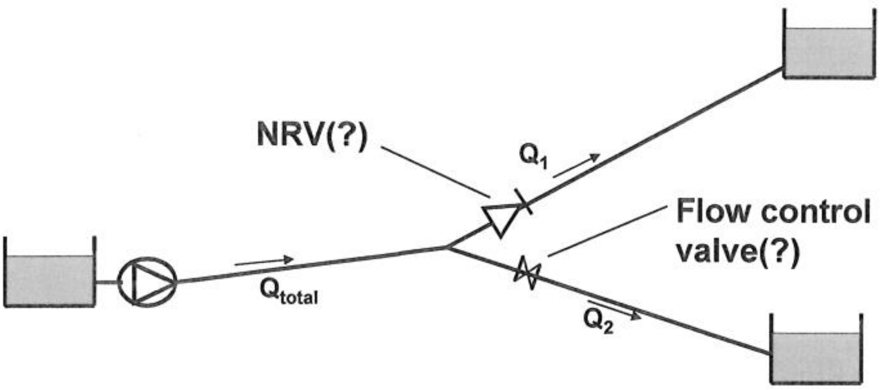


Figure 36 Flow control to achieve required distribution

With more branches and points of demand the system becomes a full distribution system as illustrated in Figure 37. Suc a system becomes very difficult to analyse by hand and almost invariably these days a suitable network program would be used. There are several, possibly many such programs available and the better ones can look at issues such as retention times in the network and water quality issues affected by the time to transfer water round the system.

Demand patterns vary, daily, weekly and seasonally and may be affected too by large industrial demands or other major point sources. It is very difficult to draw a system curve or even series of system curves for such a system and it may be necessary to run a large number of cases and plot the pump lifts and flows on a graph to find envelopes of all the conditions that might nee dto be catered for.

An example of a very large system with multiple internal pumping systems drawing from a main distribution ring is the Thames Water Ring Main which distributes water to most of London with water from several water treatment plants. The tunnel ring is the main distribution conduit and each zone within London is fed from three separate pumping stations drawing form the Ring Main.

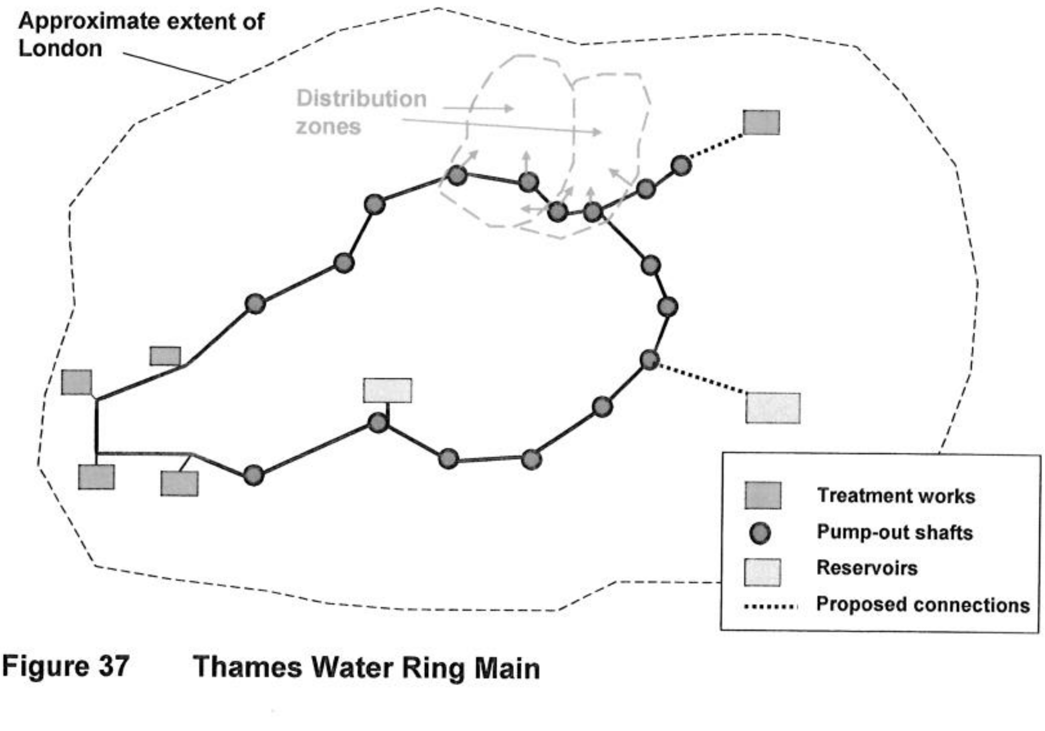


Figure 37 Thames Water Ring Main

5 SUMMARY OF PIPELINE SYSTEM DESIGN

1. Think in terms of the total energy line.

2. Unless energy is put into the flow (eg pumping) energy must be lost and the total energy line must reduce in the direction of flow.

3. The pipeline pressure head is the difference between the piezometric level and the pipeline invert. I.e. it is a function of the pipeline profile as well as the hydraulic gradient.

4. Generally the velocity head is small relative to the pipeline pressure. Thus the pressure in the pipe can usually be taken as the difference between the total energy line and the pipeline level.BUT this is not true if the pipeline velocities are high the pressures are low.

5. Draw the pipeline profile and total energy lines for the full range of flows and pipeline roughnesses.

6. Consider the cases of maximum and minimum pumping conditions and zero flow.

7. Draw the system curves and pump curves over the full range of possible operation. Consider appropriate duty requirements for pump(s).

8. Consider potential run-out, need for throttling, npsh requirements

9. At an early stage consider how the system is going to be controlled.

6 Example Design of Simple Pumping System

- Pump to deliver 230 l/s potable water from sump (max level +20maD, min level +16 maD) to tank (max level +46m, min level +41 m) over a distance of 4 km. Polyethylene, PE, delivery main has 4 no 90^0 bends and suction pipe has bellmouth intake and 2 no 90^0 bends.

- Requirement: to define pump duty

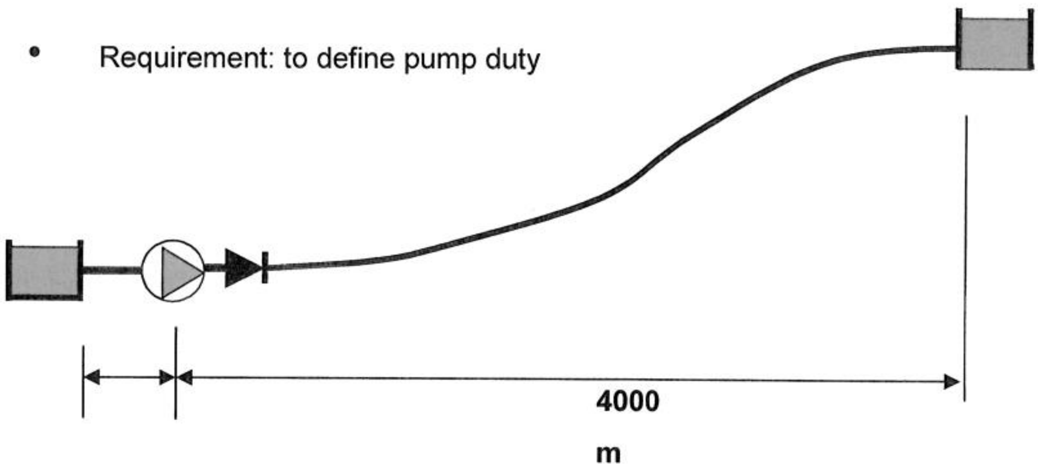


Figure 38 Definition of problem

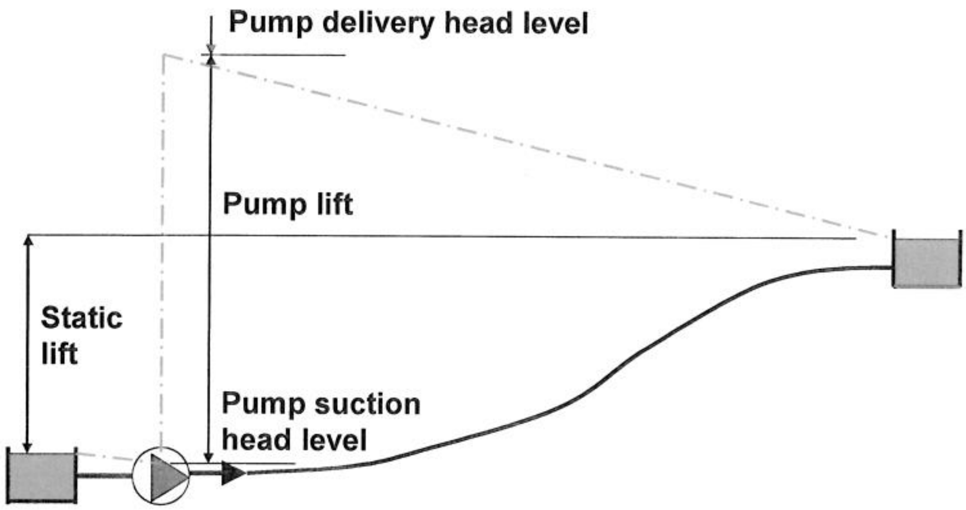


Figure 39 Reminder of what we are trying to calculate

Pipe sizing: **Could do full optimisation exercise but that requires costing data so assume for this exercise that optimum velocity is between 1.2 and 1.5 m/s:-**

Pipe diameter (nom)	**mm**	**400**	**450**	**500**	**560**	**630**	
Pipe int dia. (SDR 17)		**mm**	**350**	**394**	**437**	**490**	**552**
Pipe area	**m2**	**.096**	**.122**	**.173**	**.188**	**.240**	
Velocity at 230 l/s	**m/s**	**2.39**	**1.64**	**1.33**	**1.22**	**0.96**	

On the basis of the above figures chose pipe of 500 mm nom diameter.

STATIC LIFTS:

Maximum: **46 - 16 = 30 m**

Minimum: **41 - 20 = 21 m**

FRICTION LOSSES:

In suction main: **Small – therefore ignore**

In delivery main: **Use Colebrook –White formula (or other similar equation allowing direct solution):**

Maximum ks say 0.15 mm

Minimum ks when pipe new – say 0.03 mm

Flow (l/s)	50	100	200	300	400
ks = 0.03					
Hydr gradient	0.00023	0.00081	0.0029	0.0062	0.0106
Head loss (m)	0.92	3.24	11.6	24.8	42.4
ks = 0.15					
Hydr gradient	0.00030	0.0012	0.0045	0.0100	0.0178
Head loss (m)	1.2	4.8	18.0	40.0	71.2
Fittings losses (m):					
Velocity	0.29	0.58	1.16	1.73	2.31
Suction pipe: K ≈ 3.5	0.01	0.05	0.24	0.52	0.93
Delivery pipe: K ≈ 5.0	0.02	0.07	0.34	0.74	1.33
Total losses: Min (m)	0.95	3.36	12.2	26.1	44.7
Max (m)	1.23	4.92	18.6	41.3	73.5

Total pump lifts:

Flow (l/s)	0	50	100	200	300	400
Max static lift/ max roughness (m)	30	31.2	34.9	48.6	71.3	103.5
Min static lift/ min roughness (m)	21	22.0	24.4	33.2	47.1	65.7
Average static lift/ max roughness (m)	25.5	26.7	30.4	44.3	66.8	99.0

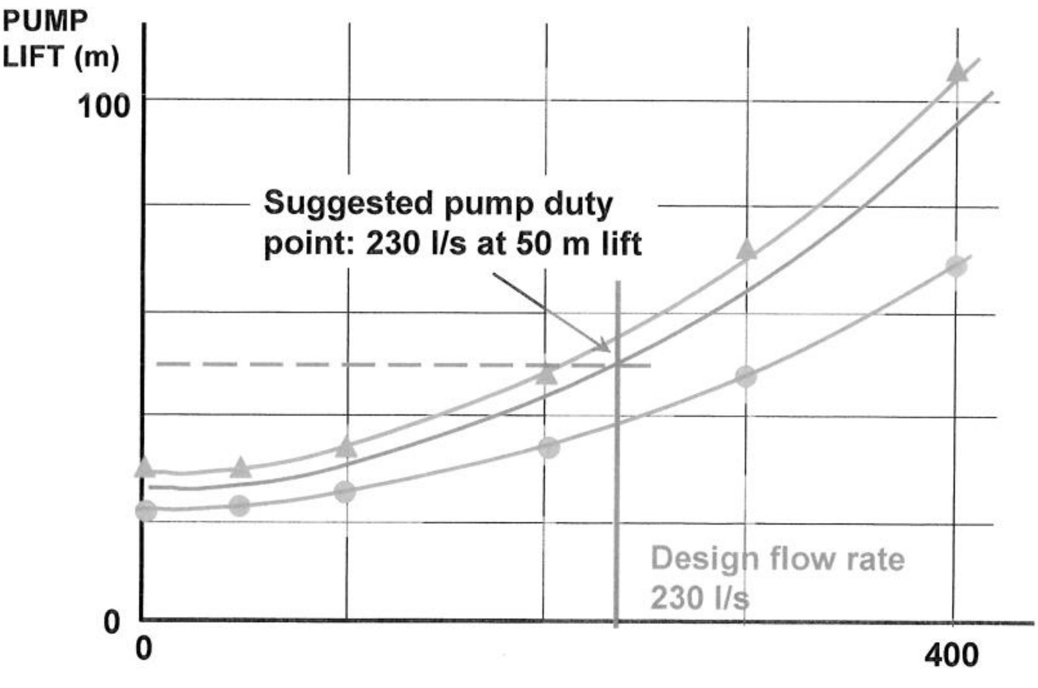


Figure 40 System curves

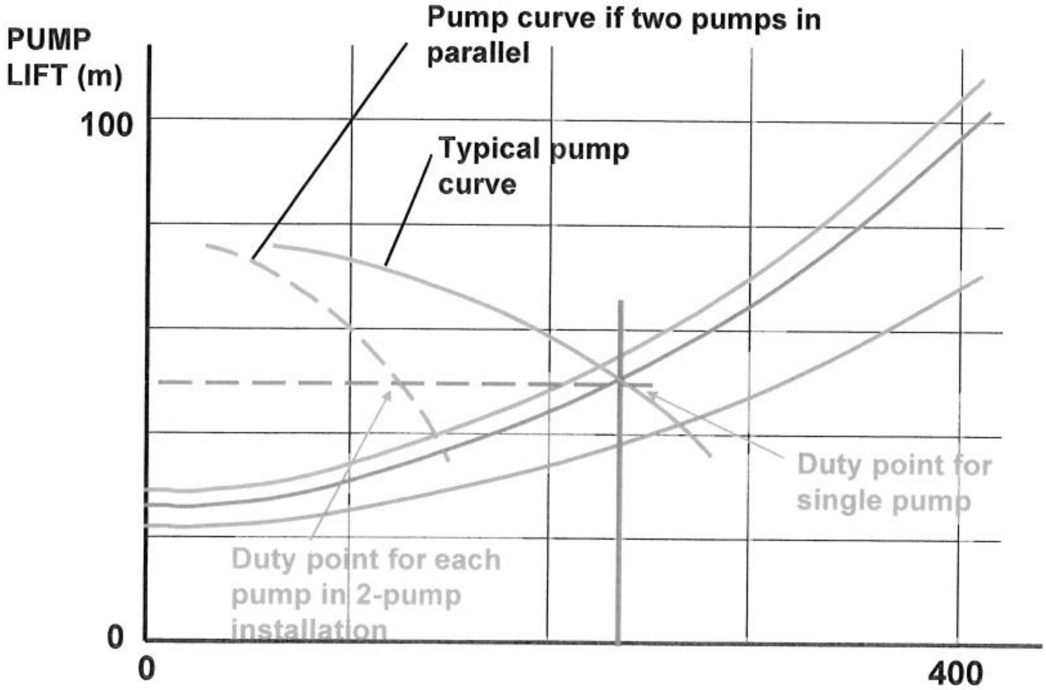


Figure 41 System Curves with single- and two-pump duties

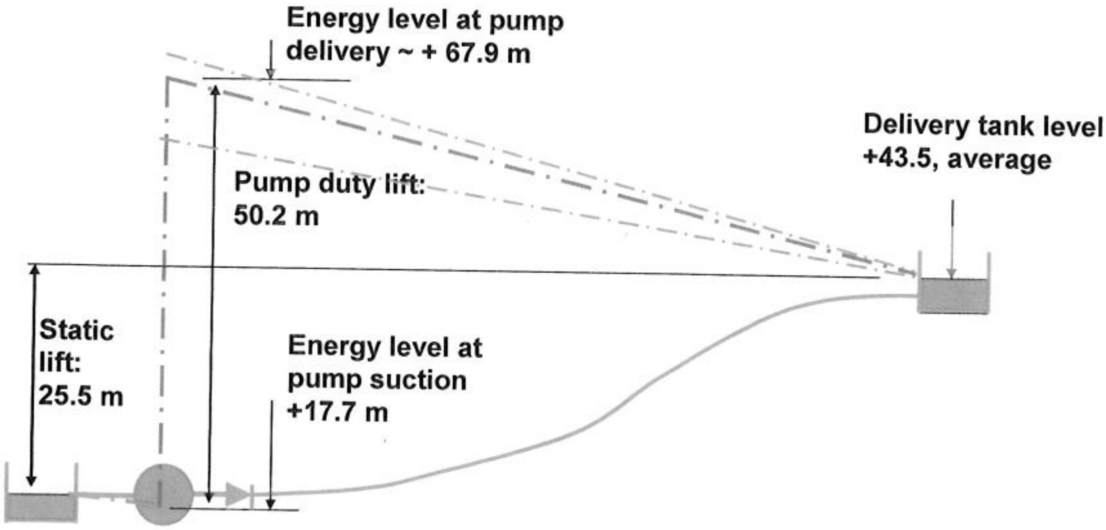


Figure 42 **Hydraulic Profiles at Duty Point**