

Lecture 17: Pump Cavitation and System Headloss

Pump Energy

Energy is needed to pump water. The power input required to lift water a given height at a given flowrate is:

$$\text{Power} = \rho g Q H \quad (Q \text{ in m}^3/\text{s}, H \text{ in metres, Power in W})$$

This formula is a basic expression of potential energy in a gravitational field: a mass m of water lifted to a height h acquires a potential energy of $E = mgh$. If the mass is lifted over some period of time (ie at a rate ρQ) the energy input likewise occurs at a rate, known (as we have seen) as power.

Strictly speaking the head H is not just the vertical lift but also includes the head losses that occur in the pipeline. This is known as the Pump Head, or the total pump energy per unit weight.

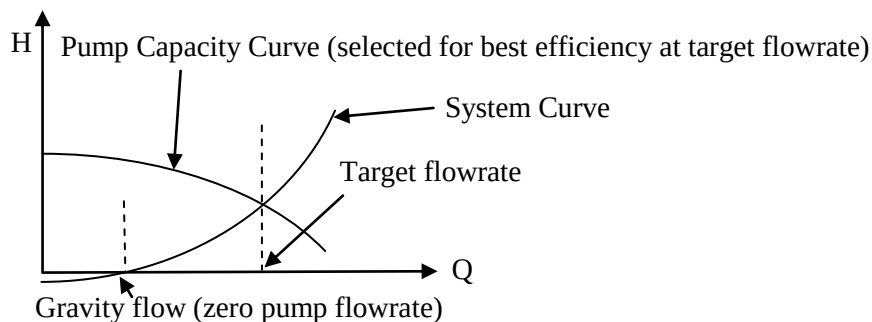
If a known total volume ∇ of water is to be lifted, the total energy (in kiloWatt-hours) is given by a simple formula that includes density, gravity and time conversions in one easy coefficient:

$$E \text{ (kWh)} = \nabla \text{ (m}^3\text{)} \times H \text{ (m)} / 367$$

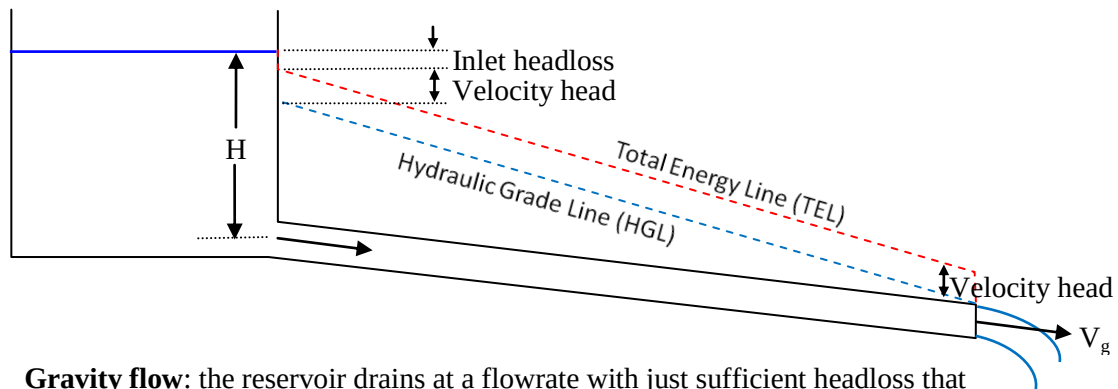
System Headloss and Pump Design

Pumps are designed to provide or restore head to systems that are otherwise undergoing losses. It is usually possible to derive a headloss curve as a function of flowrate. Headlosses are proportional to velocity head, so losses increase parabolically with discharge.

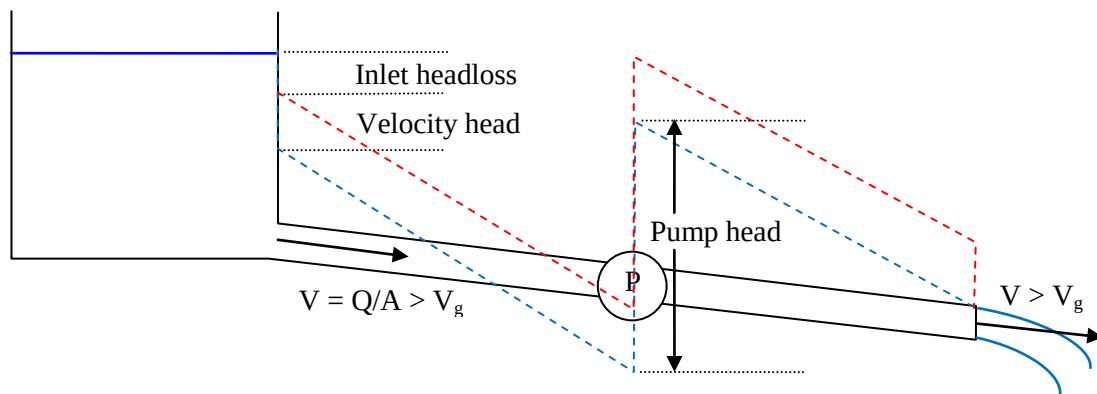
Since a pump capacity curve can be plotted on the same axes (head vs discharge), we can find a pump whose capacity curve crosses the headloss curve at the point of best efficiency:



Increasing the pumped flowrate has the effect of increasing losses throughout the system. Since headloss is proportional to velocity or discharge *squared*, the increase in losses can be significant:



Gravity flow: the reservoir drains at a flowrate with just sufficient headloss that the HGL meets the free surface water level at the outlet. Higher flowrates are not possible under gravity drainage as this would create negative pressure at the outlet.

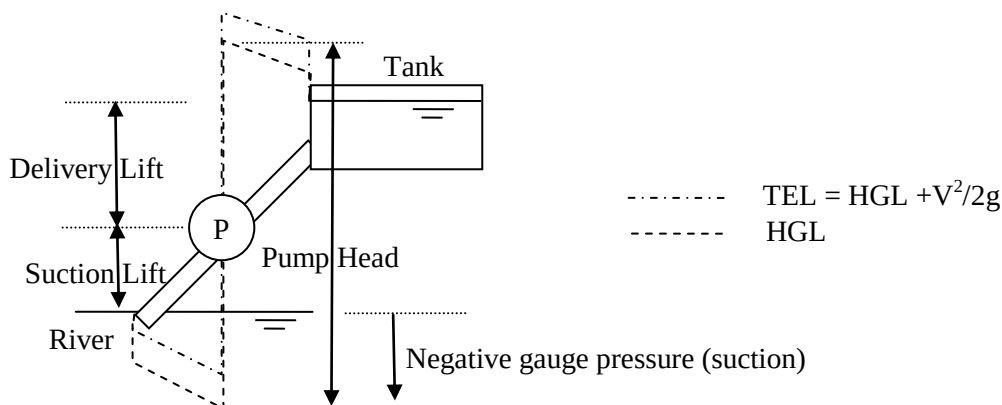


Pump flow: A pump increases the flowrate above the gravity flowrate, but the higher flowrate causes increased losses and steeper headloss gradients. The end condition is the same: the pipe HGL must meet the free surface level at the outlet.

Suction Lift, Delivery Lift and Headloss

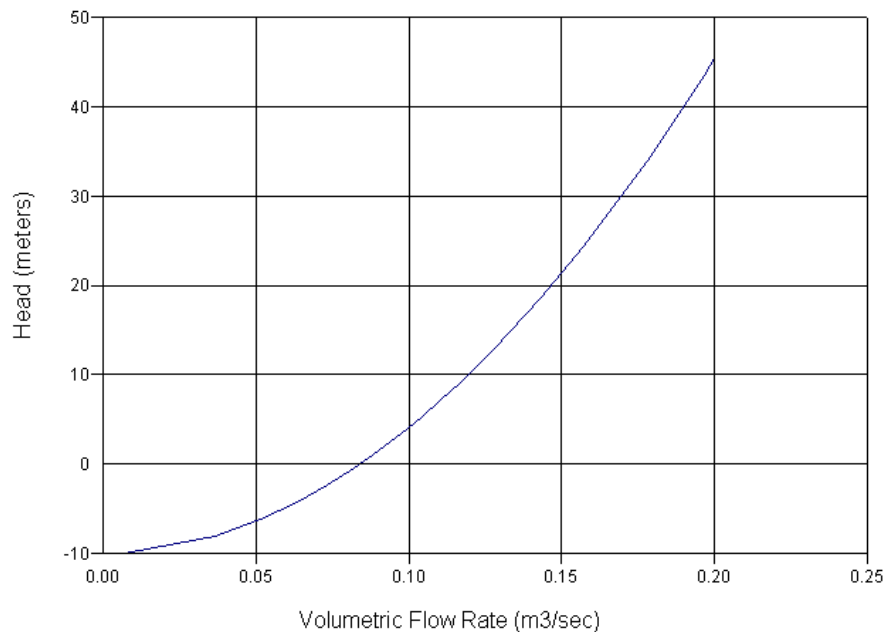
As might be expected, in most applications the gravity flowrate is zero or negative ie it would otherwise flow backwards or not at all. A *suction lift* must be applied to the fluid to raise it to the pump before delivery to a higher level. Friction and inlet/outlet headlosses occur at all points.

All three components, the suction lift, delivery lift and headloss, count towards the total pump head. A high suction lift (low water) results in a reduced maximum delivery lift. Headlosses increase with flowrate, and since this is limited by the head-discharge characteristic it is often necessary to increase the pump speed to increase flow. Gate valves provide limited controls on flowrate. A reflux valve prevents backflow through the pump.



The suction lift is in a region of negative gauge pressure ie absolute pressure less than atmospheric. If this absolute pressure falls as low as the vapour pressure of the fluid it will cavitate (see next section).

Example: The figure shows the system curve for a 200 mm horizontal steel pipe, 200 m long with multiple 30° bends, flowing from a 10 m deep water reservoir at 15°C to a free outlet. The graph shows that the gravity flowrate is about 0.085 m³/s (where the graph crosses the horizontal axis at H=0). It also shows that to increase the flowrate to 0.15 m³/s a pump head of about 22 m would be required. What size pump and speed is required?



Since this specification includes a target flowrate and pump head, the Head and Discharge relationships can be used to determine the size and speed of pump required to increase the flowrate by any given amount. The results depend on the model of pump and performance testing data. A solution based on the methods given in lecture 16 would be adequate.

The data for a 220 mm pump at 3000 RPM can be used:

$$Q_r = 0.845 \quad H_r = 2.64 \times 10^{-4}$$

Make any necessary conversions to ensure the same units are used: Convert m³/s to m³/hour

$$0.15 \text{ m}^3/\text{s} \Rightarrow 0.15 \times 3600 = 540 \text{ m}^3/\text{hour} = Q \text{ target flowrate}$$

Let's save ourselves some trouble and *guess* a standard size bigger than 220 mm that will do the job:

Say $D = 0.5 \text{ m}$

$$\text{Then } N = Q/(Q_r D^3) = 540/(0.845 \times 0.5^3) = 5112 \text{ RPM.}$$

This may be too fast: too much wear and tear on the pump.

Try $D = 0.7 \text{ m}$

$$\text{Then } N = 540/(0.845 \times 0.7^3) = 1863 \text{ RPM.} \quad \text{OK}$$

Cavitation

If a fluid's absolute pressure is reduced to the vapour pressure at that temperature it will boil. If the bubbles are carried to a region of higher pressure they will collapse with very high local shock pressures, of the order of gigaPascals, easily enough to damage moving parts and even corrode metal surfaces.

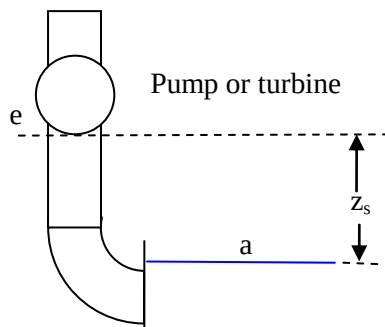
Since all turbomachinery and ship propellers as well as many hydraulic structures are subject to cavitation, special attention must be given to it in design.

A cavitation parameter σ can be calculated. When $\sigma = 0$ there is a high risk of cavitation:

$$\sigma = \frac{P - P_v}{\rho V^2 / 2} > 0 \quad \text{Cavitation parameter (condition for no cavitation)}$$

Where P is absolute pressure, P_v is the vapour pressure at that temperature (see fluid chemical data sheets) and V is an undisturbed velocity (ie ignoring the backflow disturbance of cavitation shock).

A cavitation index σ' is useful in the selection and location of turbomachinery with respect to the tailwater elevation or suction lift z_s , which largely determines the low pressure of a system:



e is the point of minimum pressure for both pump and turbine systems, and a is atmospheric pressure. Neglecting losses, Bernoulli's equation can be applied between the two sections:

$$\frac{P_e}{\rho g} + \frac{V_e^2}{2g} + z_s = \frac{P_a}{\rho g} + 0 + 0$$

For cavitation to occur, P_e must reduce to the vapour pressure P_v :

$$\sigma' = \frac{P_a - P_v - \rho g z_s}{\rho g H} = \frac{V_e^2}{2gH}$$

Where H is the total head gain or loss through the pump or turbine. For any homologous series there is a critical value of σ' , denoted σ'_c , measured by performance tests on a single unit, below which cavitation will occur:

$$\sigma' > \sigma'_c \quad \text{Cavitation Index (no cavitation at elevation } z_s \text{ above tailwater)}$$

Example: Test on a pump model indicate $\sigma'_c = 0.10$. A homologous unit is to be installed to pressurise water ($P_v = 3.5 \text{ kPa}$) to a head of 25 m. At what maximum elevation can the pump be installed above the tailwater ($P_a = 101.3 \text{ kPa}$)?

Solve for z_s : $z_s < \frac{P_a - P_v}{\rho g} - \sigma'_c H = (101300 - 3500) / 9800 - 0.1(25) = 7.5 \text{ m}$.

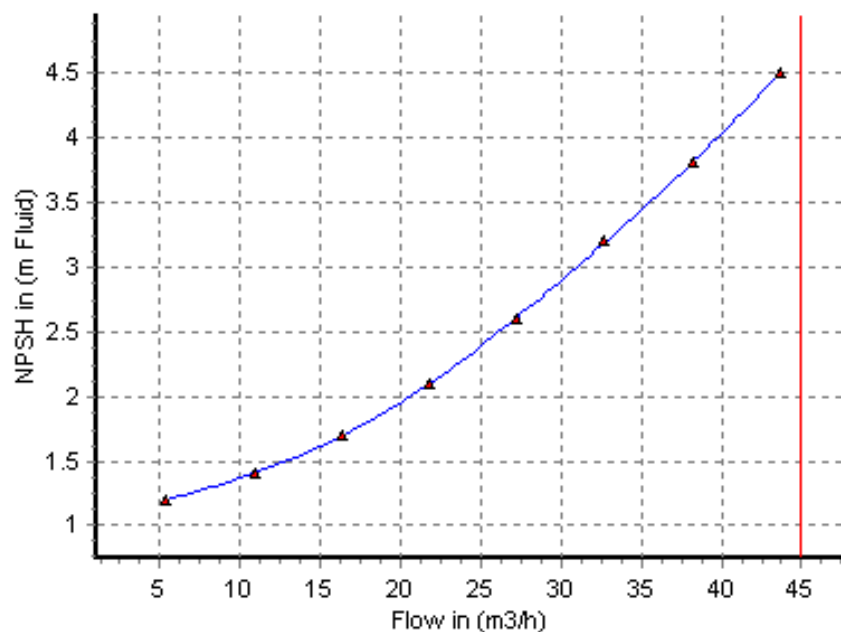
Net Positive Suction Head

NPSH is another commonly used control on cavitation, often reported with capacity and efficiency curves. A test is run on a machine to determine the maximum value of z_s for operation with no cavitation or significant impairment of efficiency, and NPSH is calculated:

$$NPSH = \sigma'_c H = \frac{P_a - P_v - \rho g z_s}{\rho g} = \frac{V_e^2}{2g}$$

Any homologous unit can now be operated at a setting where the suction lift is less than z_s as calculated:

$$z_s = \frac{P_a - P_v}{\rho g} - NPSH$$



NPSH Curve data for a 220mm centrifugal pump at 3000 RPM (image from FluidFlow © Flite Software Ltd)

Reading	Fox and McDonald 1992 (Reading 8) Section 11.5 Review Learning Outcomes for Section 5
Tutorial	Momentum Conservation and Turbomachinery
Software	See the online Tutorial “FluidFlow3 Software”