



Process engineering

Pumps, turbines and compressors

Centrifugal pumps (2)

This lesson will explain the relationship between capacity and head of a centrifugal pump. This knowledge is necessary so that, with the aid of this information, the installations can be operated in an optimal and responsible manner. Furthermore, the trend of the pipe characteristic will be dealt with. We will also take a detailed look at the changes in such a curve that might be caused by the operation. Combining the pump characteristic with the pipe characteristic produces the operating point of a pump. The lesson will be concluded with a discussion on the subject of cavitation and the closely associated concept of 'NPSH'. This is an abbreviation for 'Net Positive Suction Head'. It will be shown that a sufficiently large value of this NPSH is of crucial importance for the proper operation of a (centrifugal) pump.

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Lesson

1. Suitability and operating conditions

- head

A centrifugal pump is only suitable if it can supply the calculated manometric head and the volume flow that the operator requires. Whether the pump can satisfy these conditions depends on the design of the pump. The dimensions of the impeller, the shape of the vanes, the speed of the pump and the density of the liquid that must be displaced are some of the factors that determine the head that the centrifugal pump can supply.

- pump parameters

- series or parallel operation - efficiency

It is necessary for a pumping installation to have a number of pumps connected in series or in parallel, because operating conditions can change frequently. In this way the required head and capacity can always be met. The efficiency of the pump in particular is of major importance in this matter. Often, taking all these factors into consideration makes the choice for the right pump difficult, although a large number of pumps for various purposes are available nowadays.

2. Liquid flow in the impeller

When the pump is operating, the liquid flows from the suction pipe into the impeller with the absolute entry velocity c_1 (see figure 1).

The movement of the liquid particles, immediately after entering the impeller, can be considered to be a combination of the movement past a vane (relative movement) and the movement with the impeller. The first movement occurs with a relative entry velocity w_1 and the second with the tangential velocity u_1 of the impeller at the entry side. These two velocities can be considered to be the resolved components of the absolute velocity c_1 . The direction and magnitude of w_1 follow from the resolution of c_1 into u_1 and w_1 . The angles that are formed between c_1 or w_1 and the tangential velocity u_1 or its extension are α_1 and β_1 , respectively.

- impact free entry

For an impact-free entry of the liquid into the impeller, the curvature of the vane must be tangential to the calculated direction of w_1 on the entry side.

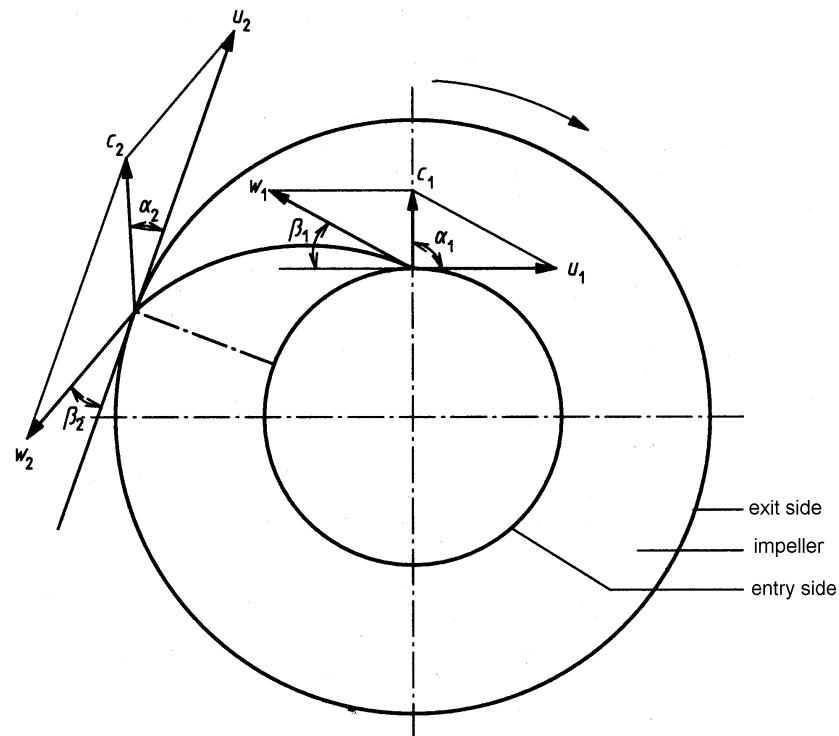
- ideal state of flow

After entering the pump, the liquid flows between the vanes of the impeller to the outer periphery of the impeller. Given a state of ideal flow, the liquid particles, when leaving the impeller, will have a relative exit velocity w_2 tangential to the vane at the exit side of the impeller. Their velocity is u_2 , which is equal to the tangential velocity at the exit side of the impeller.

- resultant

- volute casing

The result of the velocities u_2 and w_2 is the absolute exit velocity c_2 . The velocity c_2 is the velocity with which the liquid particles enter the volute casing of the pump. The directions of c_2 and w_2 with respect to the direction of u_2 (tangent of the impeller circumference) are determined by angles α_2 and β_2 .



c_1 = absolute entry velocity (m/s)

u_1 = tangential velocity entry side (m/s)

w_1 = relative entry velocity (m/s)

c_2 = absolute exit velocity (m/s)

u_2 = tangential velocity exit side (m/s)

w_2 = relative exit velocity (m/s)

a_1 = entry angle of c_1 with u_1 (absolute) (rad)

b_1 = entry angle of w_1 with u_1 (relative) (rad)

a_2 = exit angle of c_2 with u_2 (absolute) (rad)

b_2 = exit angle of w_2 with u_2 (relative) (rad)

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Figure 1

Centrifugal pump in which the absolute and relative flow velocities are indicated

- Euler's equation

The ideal theoretical head p_E is determined With Euler's equation from the tangential velocity u_2 , the absolute exit velocity c_2 and the angle a_2 between these two velocities. This equation is given by:

$$r_E = r * u_2 * c_2 * \cos \alpha_2 \quad (1)$$

In which:

r_E = the ideal theoretical head according to Euler (Pa)

r = the density of the liquid (kg/m³)

u_2 = the tangential velocity of the impeller (m/s)

c_2 = the absolute exit velocity (m/s)

α_2 = the angle between c_2 and u_2 (rad)

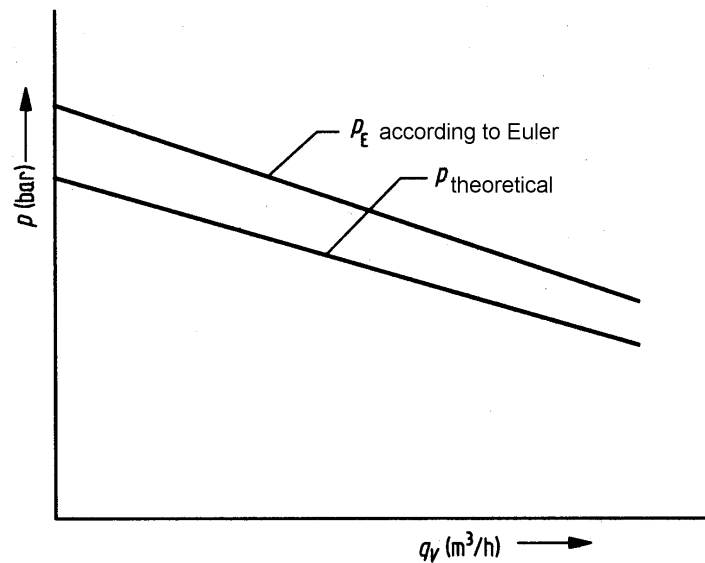
$$1 \text{ rad (radian)} = \frac{360}{2\pi}$$



3. The pump characteristic

The relationship between capacity and head is initially represented by a straight line in a graph. The theoretical relationship between the head and the capacity according to Euler will have a lower value, because the calculation of the ideal theoretical head did not take into account the actual number of vanes and their thickness. The head that can be achieved with the actual number of vanes is called the 'theoretical head' p_{th} . This is shown in figure 2.

- actual number of vanes



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Figure 2

The trend of Euler's head and the theoretical head as a function of the capacity of the pump

In this graph, the capacity is plotted in m^3/s (according to SI units), m^3/h or l/min on the horizontal axis. The head is plotted on the vertical axis in N/m^2 or bar. Often, the head h_E is plotted on the vertical axis because for a centrifugal pump it is independent of the density of the liquid pumped.

- head h_E

Question 1

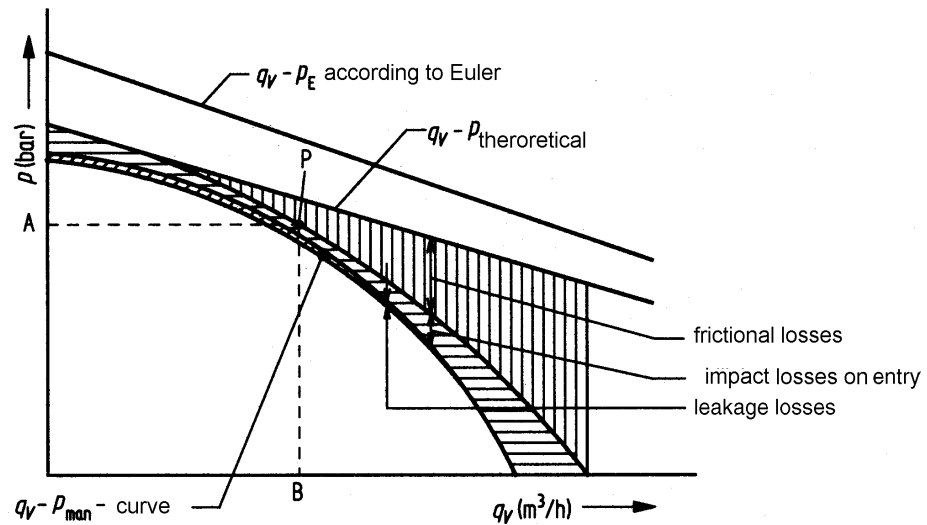
Show that the head of a centrifugal pump is independent of the density of the liquid to be pumped.

- decreasing head

The graph clearly illustrates that the head decreases with increasing capacity.

- hydraulic losses
- volumetric losses

When plotting this curve, the hydraulic and volumetric losses that occur in a pump were not taken into account. Hydraulic losses, for example, are losses due to liquid impact, friction and leakage (e.g. past the wear rings). When these losses are taken into account, the curve will have a shape as shown in figure 3.



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Figure 3

The head-capacity characteristic of a centrifugal pump. The losses due to friction, impact and leakage have been taken into account

Impact losses occur when the angle at which the liquid enters is not equal to the angle of the vane at the inlet side. The frictional losses are directly proportional to the square of the capacity (q_v^2).

Question 2

Why does the frictional loss increase with increasing capacity?

The angle of the vane at the inlet side of the impeller is only just right for the capacity for which the pump was designed. Only in that situation will the entry angle of the liquid be equal to the entry angle of the vane. In that case, there will be only slight losses due to impact. In the curve this has been designated by point P (design point). At this design point, with head A and capacity B, the curve for impact losses will nearly touch the curve for frictional losses. As the required capacity moves further away from this design point, the impact losses will increase further and, in doing so, the efficiency of the pump will decrease.

- design point

- frictional losses

Frictional losses will increase as the capacity increases. This is a quadratic relationship. At maximum head, the capacity will be zero ($q_v = 0$), in which case the frictional losses will also be zero.

The loss due to leakage depends on the head, i.e. a high head will give a high leakage loss and a low head will give a low leakage loss. Of course, the condition of the seals also plays a role. A leaking shaft seal and worn wear rings increase the losses due to leakage.

Question 3

Why do leakage losses increase with increasing head?

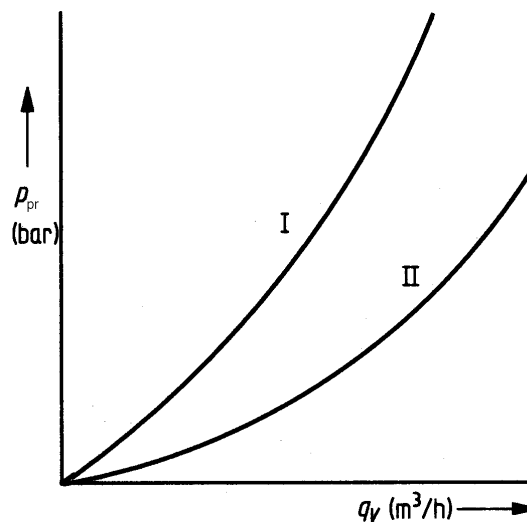


Now it can be seen that the actual pump characteristic is in fact a curved line that lies significantly lower than the theoretical line. This actual curve is called the q_v - p_{man} curve of the pump. This curve shows the relationship between the capacity and the manometric head (p_{man}). This curve is also often called the q_v - h_{man} curve.

4. Characteristics of pipe systems

- unvarying factors
- velocity of the liquid
- piping characteristic

We have already come across the concept of pipe resistance in straight pipes, fittings and accessories. The calculation of pipe resistances using tables and graphs was also discussed. It can be shown theoretically that the resistance of a pipe depends on a number of invariable factors and the velocity of the liquid in the pipe. The velocity of the liquid and the diameter of the pipe together determine the quantity of liquid flowing through the pipe per unit time. The graphical relationship between the resistance of the pipe (in bar) and the flow q_v (in m^3/h) gives a curve that is called the pipe resistance characteristic. Figure 4 shows such a curve.



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Figure 4

The curve of the resistance of a pipe for two different systems as a function of the capacity

Curve I is the characteristic of a pipe with a high resistance. The resistance increases drastically as the flow q_v increases.

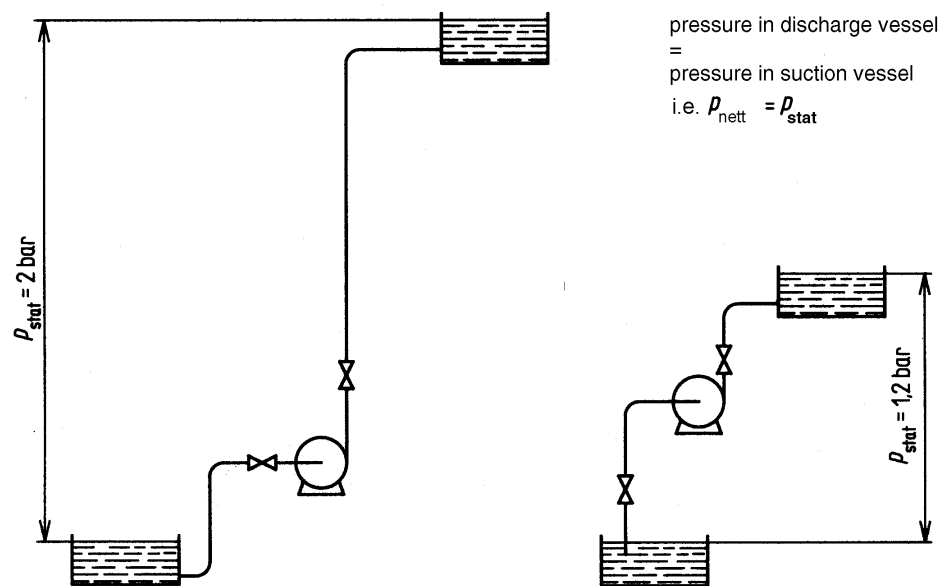
Curve II is the characteristic of a pipe with a low resistance. In comparison with characteristic I, the resistance increases far less as the flow q_v increases.

With the determination of the pipe characteristic in Figure 4, only the resistance caused by the flow of the liquid in the pipe has been taken into account. However, this occurs only in exceptional circumstances.



- height and pressure differences
- static pressure

In most situations, the pump must not only overcome the flow resistance, but also the differences in pressure and height between the suction vessel and the discharge vessel. This is called the static head. This pressure is always present; even when there is no flow. If we also want to incorporate this into the pipe characteristic, we will see that the starting point of the resistance curve no longer coincides with the origin, but that it is displaced upwards along the vertical axis. The amount of displacement is equivalent to the static head expressed in bar. These are curves I and II in figure 7 for the pipe systems shown in figures 5 and 6, respectively.



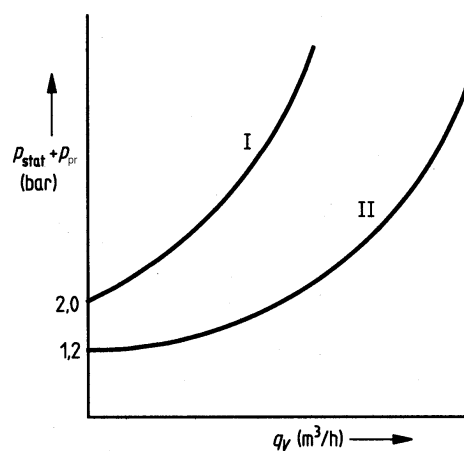
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Figure 5
Static head is 2 bar

Figure 6
Static head is 1.2 bar

Question 4

What is the effect of the increased static head on the resistance curves in figure 8?



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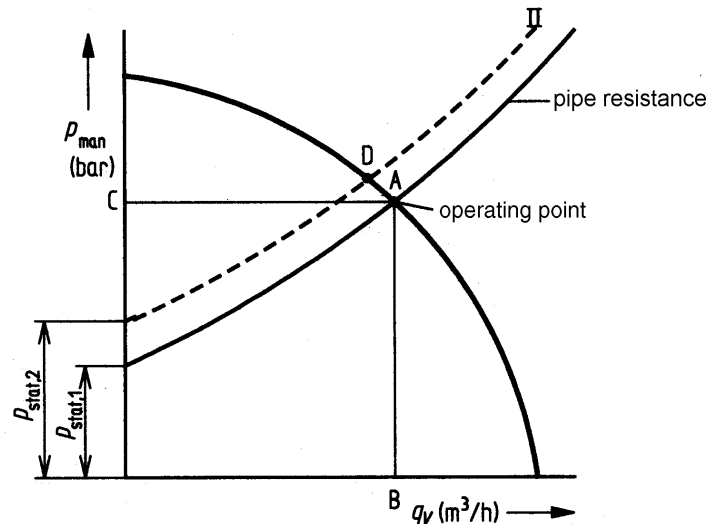
Figure 7
Two pipe characteristics. Curve I represents a static head of 2 bar and curve II a static head of 1.2 bar. The pipe resistance of system I is greater than the pipe resistance of system II



5. The operating point of a pump

- operating point

When we draw the characteristic of a given pump on the same graph as the characteristic of the pipe system, we see that these curves intersect each other at point A (figure 8). This intersection is called the 'operating point' of the pump.



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Figure 8

The pipe characteristic and the q_v - p_{man} characteristic as a function of the capacity of the of the pump

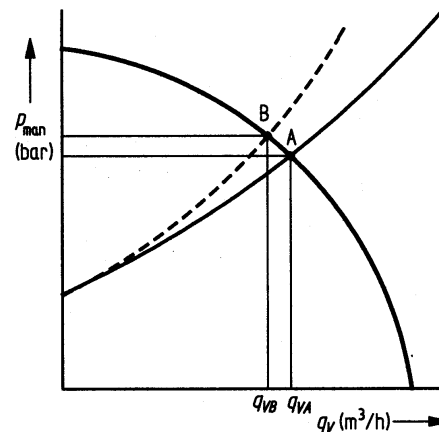
The pump will then deliver a volume B and this requires a manometric head C. If the static pressure increases, due to a lower level in the suction vessel or an increased level in the discharge vessel, then the entire resistance curve in figure 8 will be displaced upwards as indicated by the dashed line.

The operating point will be displaced to point D. This means that the pump must overcome a greater head, which results in a decreased capacity. If the pressure in the discharge vessel differs from the pressure in the suction vessel, there will be a pressure difference between the two vessels. With respect to the static head, this will have the same consequence as a change in the difference in levels between the suction vessel and the discharge vessel. In other words, the resistance curve will also be displaced. The resistance curve is displaced upwards when the pressure in the discharge vessel is higher than in the suction vessel and downwards when the pressure in the discharge vessel is lower than in the suction vessel.

- increased head
- decreased capacity

- throttling the discharge valve
- contamination of a filter

An entirely different situation results when the pipe resistance increases. This may be the result of throttling of the discharge valve or changing the opening in the control valve. Fouling of the pipes or filters may also result in an increased pipe resistance. This situation is shown in figure 9.



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Figure 9

A decreasing capacity as a result of an increasing pipe resistance (q_{vA} to q_{vB})

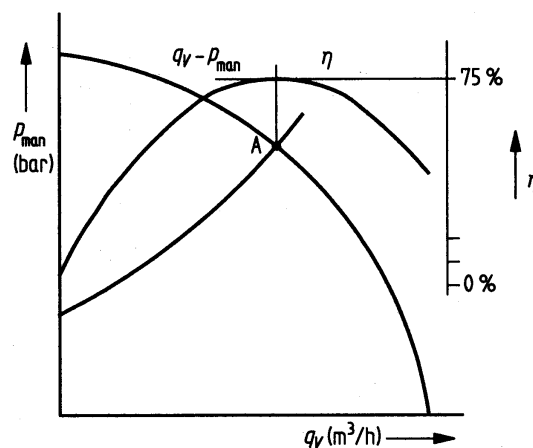
In this case, the static head remains the same and, hence, the starting point of the resistance curve remains the same. However, the slope of the curve is steeper and the operating point is displaced from point A to point B. Consequently, the pump once again needs the larger part of its head to overcome the resistance of the pipes and this results in a decreased capacity.

Question 5

How does throttling the discharge valve affect the power consumption?

- adapt to each other

When designing a pumping unit, care will always be taken to adapt the pump and the pipe circuit to each other in order to obtain maximum efficiency. This means that the operating point must be as close as possible to its maximum efficiency. This is shown in Figure 10. Clearly a change in the pipe resistance, resulting in either an increase or a decrease in the capacity, will cause a decrease in efficiency.



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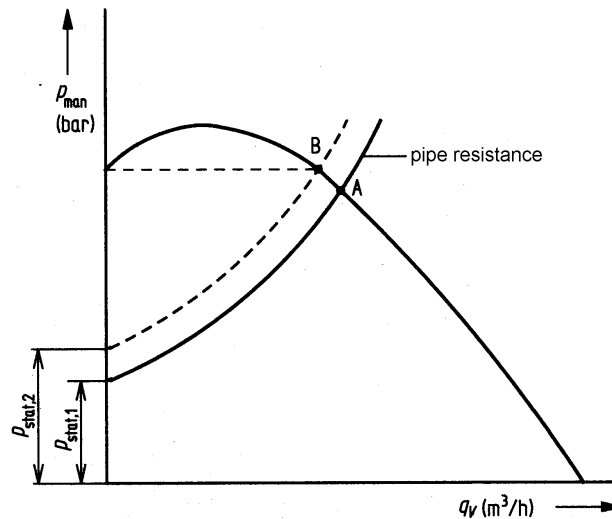
Figure 10

The curve of the pump efficiency η as a function of the capacity with respect to the operating point of the pump

**Question 6**

When a pump operates at optimal efficiency, what would in general be preferred: a high capacity or a high head?

If we bring about changes in the static head or the pipe resistance of a pump with an unstable characteristic, we obtain a situation as shown in figures 11 and 12.

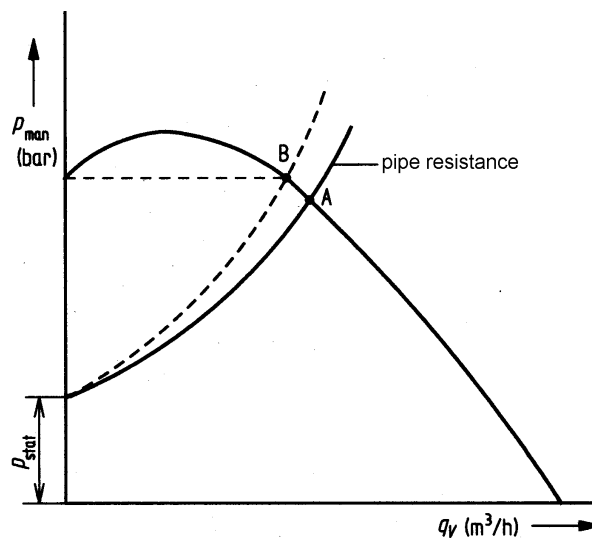


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Figure 11

Pipe and pump characteristic with increasing static head

Figure 11 shows the situation where the static head increases and figure 12 the situation where the pipe resistance increases. The operating point is displaced from A to B, with point A being situated in the stable part and point B at the beginning of the unstable part of the pump characteristic.



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Figure 12

Pipe and pump characteristic with constant static head and increasing pipe resistance



- stable and unstable part of the pump characteristic
- risk
- service life

The fact that the pump must overcome a greater head at point B, thereby having a lower capacity, also holds for this situation. Furthermore, there is the risk that point B will be displaced into the unstable part of the pump characteristic when the pressure in the suction pipe and the discharge pipe is not constant. This will result in a greatly varying capacity. This is not good for the service life of the pump.

6. The occurrence of cavitation

- cavitation

In pump engineering, the phenomenon of 'cavitation' is understood to be:

- the formation of vapour bubbles and afterwards, under the influence of increasing pressure further in the impeller;
- the 'collapse' of these vapour bubbles.

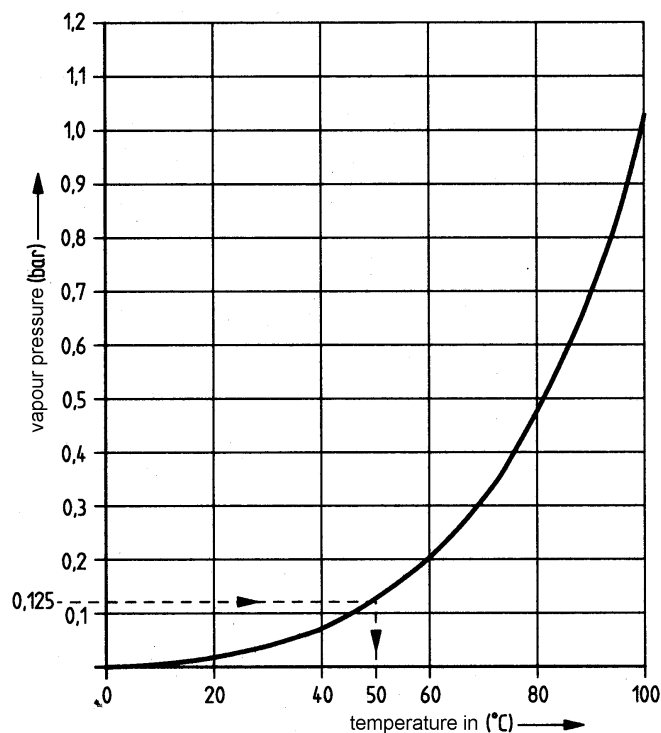
- cavitation erosion

Cavitation damages the pump. This damage is also called cavitation erosion.

- vapour pressure

Vapour bubbles are formed in a liquid when the pressure above the liquid is equal to the vapour pressure of the liquid. The temperature of the liquid also plays a major role. For instance, water boils at a temperature of 100 °C and at a pressure above the liquid of 1 bar. If the pressure is 0.125 bar, then water will boil at 50 °C (see figure 13). Other liquids will have other relationships between temperature and pressure (see figure 14). A liquid at a particular temperature will start to boil (formation of vapour bubbles) when the pressure above the liquid becomes equal to the vapour pressure at that particular temperature. For pumps this is very important.

- formation of vapour bubbles



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Figure 13

The relationship between temperature and vapour pressure for water



- large forces

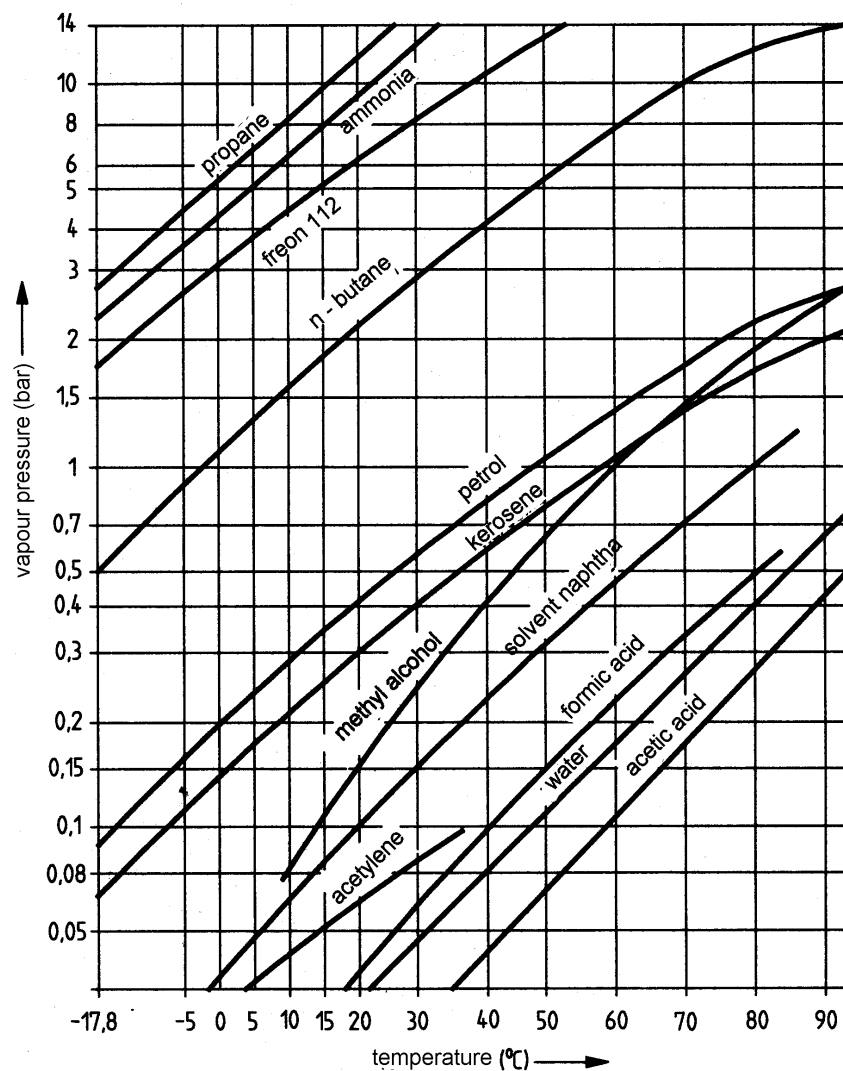
- rustling, crackling
sound

- avoid

Vapour bubbles will form throughout the liquid as soon as the pressure in the liquid becomes lower than the vapour pressure related to the temperature for that particular liquid. The liquid flow will carry the vapour bubbles via the impeller to a region of higher pressure. Under those new conditions it is impossible for the vapour phase to exist and, consequently, the vapour bubbles will suddenly collapse. During this collapse or implosion of the vapour bubbles, large forces will be exerted on the impeller and the walls of the impeller casing. These forces can cause damage, whereby pieces of metal can be removed from the metal surface (erosion).

Cavitation in a pump can be heard as a rustling or even a loud crackling sound, depending on the degree of vapour bubble formation. Cavitation will also cause a large decrease in the capacity of the pump. Obviously, cavitation must be avoided. Therefore, care must be taken to ensure that the pressure $p_{\text{suction flange}}$ remains above the vapour pressure. Therefore:

$$p_{\text{suction flange}} > p_{\text{vapour pressure}}$$



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Figure 14

The relationship between the temperature and the vapour pressure for various liquids



7. Net Positive Suction Head

- positive suction head available $NPSH_A$

The difference between $p_{\text{suction flange}}$ and $p_{\text{vapour pressure}}$ occurring in an installation is called the Net Positive Suction Head Available ($NPSH_A$).

$$NPSH_A \text{ (bar)} = p_{\text{suction flange}} - p_{\text{vapour pressure}} \text{ (bar)} \quad (2)$$

- positive suction head required ($NPSH_R$)

The manufacturer of the pump will indicate the difference between $p_{\text{suction flange}}$ and $p_{\text{vapour pressure}}$ that is required for proper operation of the pump. This is called the Net Positive Suction Head Required ($NPSH_R$).

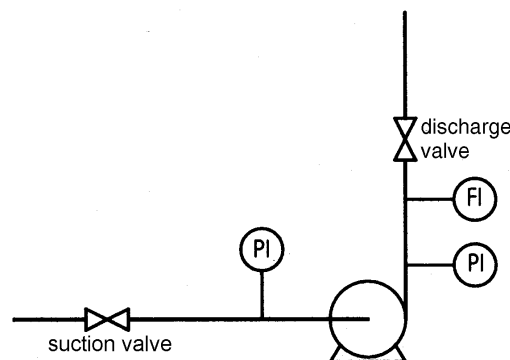
For a pump to operate properly, the $NPSH_A$ must never be smaller, but must always be larger than the $NPSH_R$. The usual value applied in practice is:

$$NPSH_A - NPSH_R = 0.1 \text{ bar}$$

- test bench

The $NPSH_R$ of a pump is tested by the manufacturer on a test bench according to an internationally established method.

Figure 15 shows a simplified version of the arrangement of a test bench for a pump.



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Figure 15

Arrangement of a test bench for a centrifugal pump for the determination of the Net Positive Suction Head Required

The $NPSH_R$ is determined for specific capacities of the pump that are kept constant during the test. By throttling the suction valve, p_{suction} is decreased. The capacity of the pump is kept constant by opening the discharge valve. At the moment that the value of the manometric head ($p_{\text{man}} = p_{\text{discharge}} - p_{\text{suction}}$) decreases by 3 % due to cavitation, the pressure is read from the suction pressure gauge. This pressure reduced by the vapour pressure is the $NPSH_R$. Figure 17 shows the curve of the $NPSH_R$.

Question 7

What is the vapour pressure of boiling water in an open vessel?

- volatile liquids

For liquids with a higher vapour pressure, i.e. volatile liquids like gasoline and benzene, p_{suction} must have a higher value in order for it to remain above the vapour pressure. A liquid is often pumped from a chamber that is at atmospheric pressure. In that case $p_{l,s} = 1 \text{ bar}$.



The equation for p_{suct} can then be written as:

$$p_{\text{suct}} = 1 - p_{\text{nett.s}} - p_{\text{pr.s}} - p_{\text{v.s}} \quad (3)$$

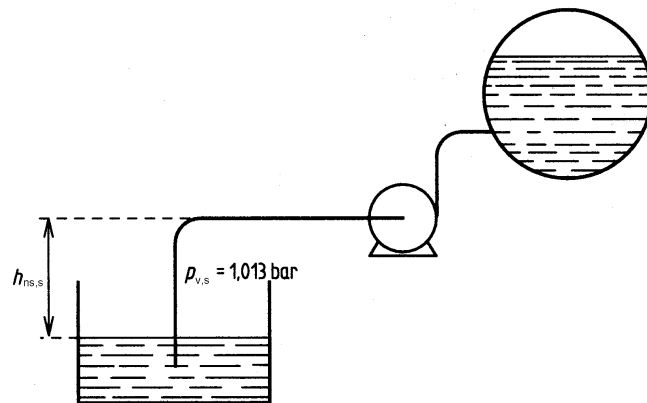
The best method to change p_{suct} is to change $p_{\text{nett.s}}$

By increasing the level of the liquid on the suction side or by placing the pump in a lower position, p_{suct} can reach a higher value and, hence, the NPSH_A will be greater. This reduces the risk of cavitation. A second method is to try to decrease the vapour pressure. The vapour pressure is dependent on the temperature of the liquid. Hence, the lower the temperature, the lower the vapour pressure.

However, in practice this solution is not attractive in view of the costs incurred.

- not advisable

7.1 An example of calculating NPSH



Data

atmospheric pressure	1.013 bar
vapour pressure	0.1 bar
velocity pressure	0.2 bar
resistance in suction pipe	0.2 bar
ρ liquid (= water)	1000 kg/m ³
g	10 N/kg
NPSH_R	0.2 bar

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Figure 16

Pump arrangement; open suction vessel, closed discharge vessel

$$\begin{aligned} 1. \quad h_{\text{ns.s}} &= 4 \text{ m (case 1)} \\ p_{\text{suct}} &= p_{\text{l.s}} - p_{\text{nett.s}} - p_{\text{pr.s}} - p_{\text{v.s}} \\ p_{\text{ns.s}} &= h_{\text{nett.s}} * \rho * g * 10^{-5} \text{ bar} \end{aligned}$$

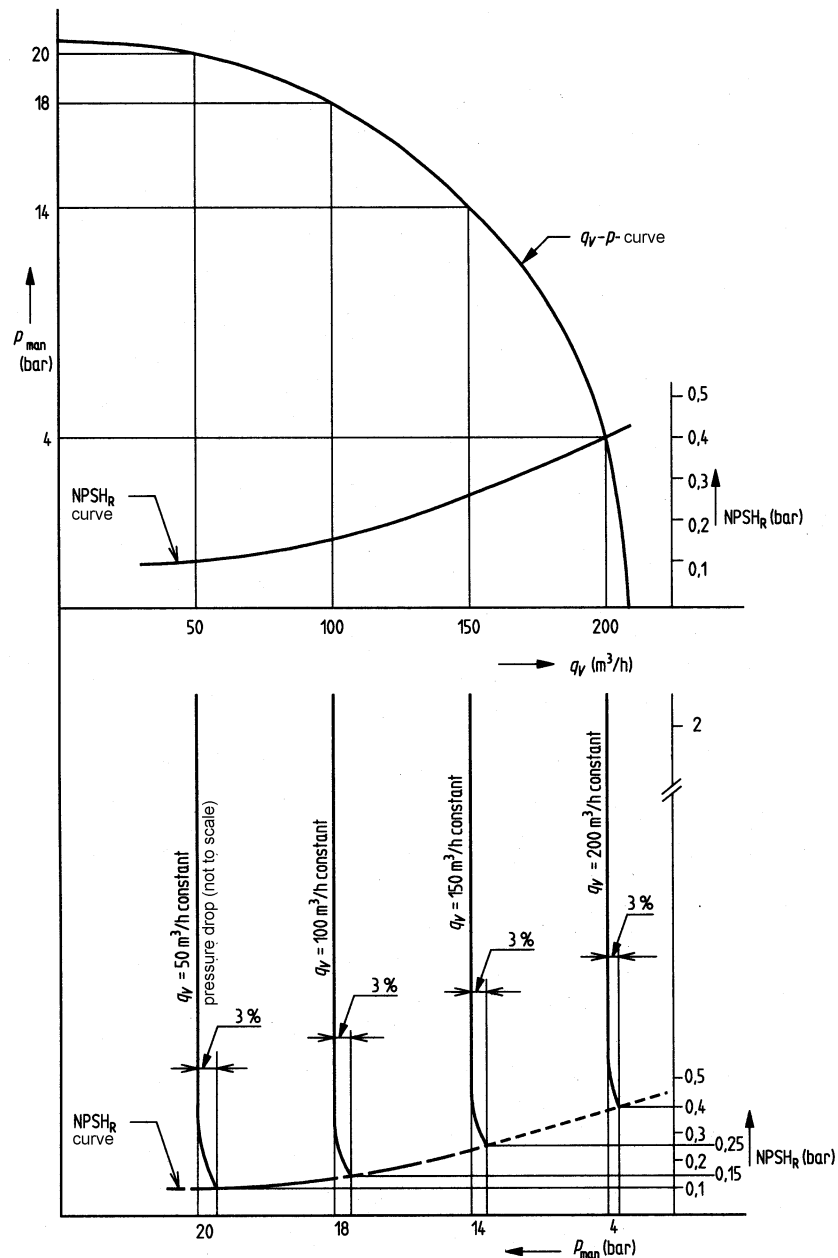
$$\begin{aligned} &\left[m * \frac{\text{kg}}{\text{m}^3} * \frac{\text{N}}{\text{kg}} * 10^{-5} \right] \\ &= 4 * 1000 * 10 * 10^{-5} \text{ bar} = 0.4 \text{ bar} \\ p_{\text{suction}} &= 1.013 - 0.4 - 0.2 - 0.2 \text{ bar} = 0.213 \text{ bar} \\ \text{NPSH}_A &= p_{\text{suct}} - p_{\text{vapour pressure}} = 0.213 - 0.1 = 0.113 \text{ bar} \end{aligned}$$

Hence the NPSH_A (0.113 bar) is smaller than the NPSH_R (0.2 bar). Therefore the pump will not operate properly because of cavitation.



$$\begin{aligned} 2. \\ h_{ns,s} &= 2 \text{ m (case 2)} \\ p_{suction} &= p_{l,s} - p_{nett,s} - p_{pr,s} - p_{v,s} \\ p_{ns,s} &= h_{nett,s} * \rho * g * 10^{-5} \text{ bar} \\ &= 2 * 1000 * 10 * 10^{-5} \text{ bar} = 0.2 \text{ bar} \\ p_{suction} &= 1.013 - 0.2 - 0.2 - 0.2 \text{ bar} = 0.413 \text{ bar} \\ NPSH_A &= p_{suction} - p_{vapour \text{ pressure}} = 0.413 - 0.1 = 0.313 \text{ bar} \end{aligned}$$

Hence, the $NPSH_A$ (0.313 bar) is greater than the $NPSH_R$ (0.2 bar). Therefore the pump will operate properly.



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Figure 17
The curve of the Net Positive Suction Head Required



Summary

It is possible to plot the relationship between the head and the capacity of a centrifugal pump in a graph. Theoretically, this would be a straight sloping line, but due to impact losses and frictional losses this line becomes a curve. This curve is usually called the $q_v - p$ curve or characteristic of the pump. On this curve there is a point at which there are hardly any impact losses. This is called the design point of the pump. Often, the efficiency of a pump and the required power are drawn in the same graph. The resistance curve of the pipe system to which the pump is connected can also be drawn in the same graph. The point of intersection of the $q_v - p$ curve and the resistance curve is the operating point of the pump. This point determines the capacity and the required head for that particular pipe system. In situations where the suction pressure at the inlet section of the pump becomes very low and approaches the vapour pressure of the pumped liquid at that particular temperature, vapour bubbles will form in the liquid. Further downstream in the impeller, the vapour bubbles will disappear under the influence of the increased pressure. This collapse of the bubbles is associated with large forces that can cause serious mechanical damage. It is for this reason that we discussed the concept of NPSH.



Test

Exercises

Do not send in your answers for correction

1. Which variables determine the manometric head that can be reached by a centrifugal pump?
2. In which way can a number of centrifugal pumps be combined?
3. Which conditions must be met for a impact-free entry into the impeller?
4. What is the difference between a stable and an unstable $q_v - p$ curve?
5. Which factors determine the head of a pump?
6. In which way can the impeller of a centrifugal pump be constructed?
7. On which principles is the operation of a centrifugal pump based?
8. Explain to what extent the head in a (centrifugal) pump will depend on the temperature of the liquid to be pumped.
9. In which way will the capacity of a pump decrease with time?
10. Which provisions with respect to seals can be made to prevent outside air from entering a pump when there is a subatmospheric pressure?
11. What does 'aligning' of a centrifugal pump mean?

Answers to questions in the lesson

1. The manometric head of a centrifugal pump according to Euler can be calculated from the following equation:

$$r_E = r * u_2 * c_2 * \cos \alpha_2 = [\text{kg/m}^3 * \text{m/s} * \text{m/s}]$$

The following is also true: $p = h * r * g = [\text{m} * \text{kg/m}^3 * \text{m/s}^2]$

And therefore $p_E = h_E * r * g$

$$\text{From which: } h_E = \frac{p_E}{r * g} = \frac{r * u_2 * c_2 * \cos \alpha_2}{r * g}$$

$$\text{It follows that } h_E = \frac{u_2 * c_2 * \cos \alpha_2}{g}$$



Do note that there is no term for density in this equation. Hence, the density of the liquid to be pumped does not determine the head of a centrifugal pump.

2. A greater capacity implies more liquid per unit time and therefore a greater velocity of the liquid. This will greatly increase the friction.
3. As the head of a centrifugal pump increases, an ever larger volume of liquid will, under the influence of the pressure difference between the discharge and suction sides, leak back from the discharge side via the wear rings to the suction side of the pump.
4. Due to the increase in static pressure, the origin of line II will be at a higher point. The result is that this line will be displaced upwards and parallel to itself.
5. The required power will not change substantially because, although the capacity of the pump will decrease, the manometric head will increase and the required power of a pump is determined by the product of these two variables.
6. The function of a pump is to transport liquids. The pressure that has to be overcome for this is a necessary evil. Hence, if a choice is at all possible, a larger capacity will be preferred.
7. This is equal to prevailing atmospheric pressure, i.e. an average of 1.013 bar.

Answers to the exercises

1. The variables that determine the head of a centrifugal pump are, amongst others:
 - the shape of the impeller (angle β_2) see figure 1;
 - the dimensions of the impeller;
 - the speed of rotation at which the pump operates.
2. Centrifugal pumps can be either connected in parallel or in series with each other.
3. An impact-free entry of the liquid into the impeller is possible when the curvature of the vane at the entry side is tangential to the vector w_1 . This vector is the resultant from the direction and magnitude of c_1 and u_1 (see figure 1).
4. A stable $q_v - p$ curve is understood to be one where the head p_{man} decreases as q_v increases from zero to its maximum value. This is not the situation for an unstable curve. In that situation, the value of p_{man} will at first increase as q_v increases and then it will decrease.



5. The head of a pump is determined by the static head and the pipe resistance belonging to that particular capacity. Hence, the design of the pipe system plays a major role in this situation. Also, remember changing the filters in the pipe system.
6. The simplest construction of an impeller for a centrifugal pump is a disc onto which a number of vanes have been fitted. In its simplest form it consists of a plate with a hub, which is connected to a ring plate by a number of curved vanes.
7. The impeller of a centrifugal pump must contain a liquid. The centrifugal force will spin the liquid to the periphery of the impeller. This creates a vacuum in the impeller, forcing more liquid to flow in. If for any reason gas were to flow in instead of liquid, the pump would cut out. This is one of the reasons why a centrifugal pump is not 'self-priming'.
8. The higher a pump is placed above the suction level, the lower the pressure on the suction side of the pump will be. As soon as this pressure becomes equal to the vapour pressure of the liquid to be pumped at that particular temperature, vapour bubbles will form in the liquid. This phenomenon will give rise to the much-feared cavitation and, in the most extreme situation, will cause the pump to cut out and, furthermore, cause mechanical damage to parts of the pump.
9. There is a gap between the (stationary) housing and the (rotating) impeller of a centrifugal pump. This gap is kept as small as possible. This is achieved by fitting wear rings. As these wear rings become more worn, a greater volume of liquid that is delivered by the impeller will flow back to the suction side of the impeller. The impeller will then have an (almost) identical capacity, while the amount of liquid flowing out of the pump will decrease. It is said that the volumetric efficiency decreases.
10. In that case use is made of a mechanical seal.
11. Aligning a centrifugal pump means to place the central axis of the pump and the central axis of the drive shaft of the drive (electric motor or steam turbine) accurately in line with each other.

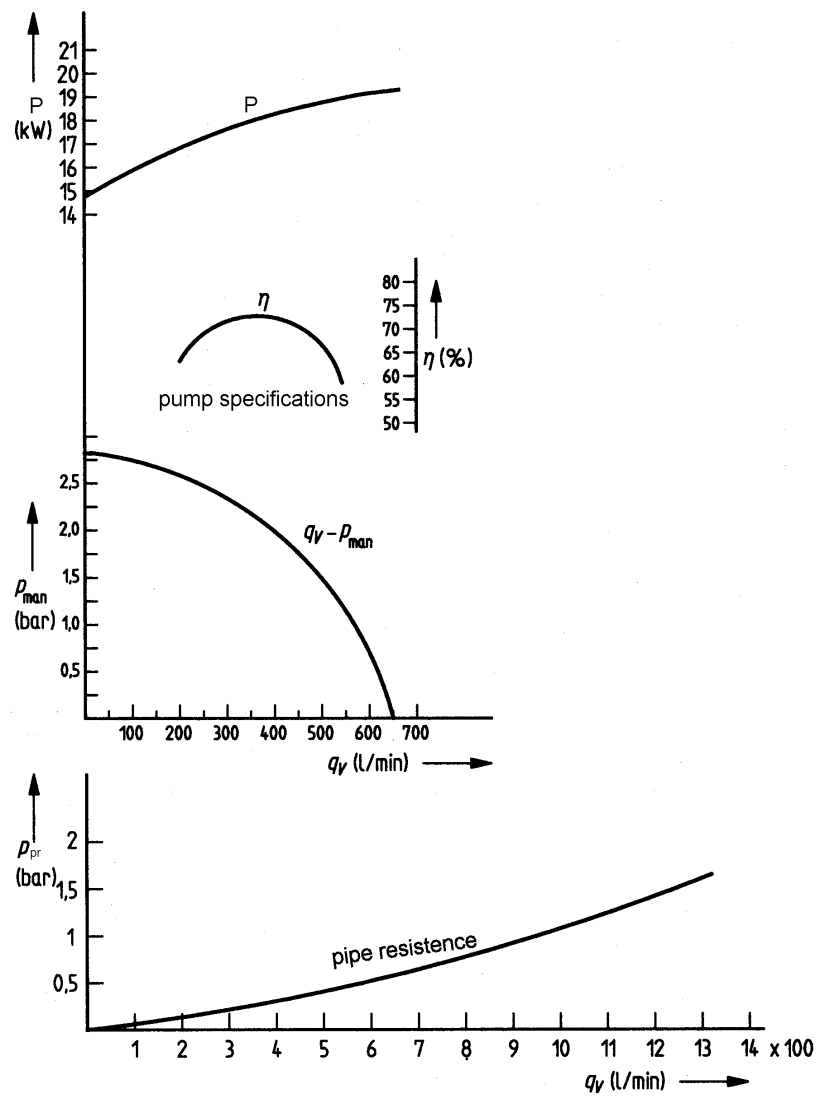
Problems and assignments

Answer and send in for correction

1. Which variables determine the pipe characteristic or resistance curve?
2. Draw an arbitrary $q_v - p_{\text{man}}$ characteristic for a centrifugal pump and indicate on this drawing how the capacity varies when the discharge valve is throttled.
3. Why do we sometimes also use the $q_v - h_{\text{man}}$ curve instead of the $q_v - p_{\text{man}}$ curve?



4. What are the disadvantages of a pump that has an unstable $q_v - p_{\text{man}}$ characteristic, and when would those pumps be quite suitable.
5. Why does a pump that transports warm liquids have a greater chance of cavitation than a pump that transports colder liquids, given that the product and all other conditions are identical.
6. Explain why a centrifugal pump is not self-priming.
7. What is the purpose of wear rings and where are they fitted? Clarify your answer with simple drawings.
8. Which factors need to be taken into account for pumps that have to operate in series.
9. How does the static head influence the capacity of a centrifugal pump?
10. Explain why the $q_v - p_{\text{man}}$ characteristic of a centrifugal pump deviates significantly from the ideal theoretical characteristic.
11. What is the capacity, the efficiency and the required power of a pump when the static manometric head is 1 and 2 bar respectively (see figure 19)? (Do note the scale indication)



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Figure 19