5 Cavitation

1. Causes, Indications, Effects

In case that the steam pressure (p_D) will be attained at entrance edge of the impeller vanes because of too high vacuum or too high temperature of the pumped liquid, cavities filled with vapor are being formed. This process is called cavitation. Due to the decrease in flow section area a loss of capacity and head will occur and consequently a decline in pump power output as well as efficiency. Perceptible indications of the cavitation during pump operation are more or less loud noises and an unsteadily working pump.

2. Cavitation Damages

The vapor bubbles are formed at the places of lowest pressure and then suddenly disappear if the pressure increases in the further flow course. In the case that this process occurs near the impeller blades or pump surfaces the materials of construction are being attacked by implosion of the vapor bubbles. Therefore this damage never appears at the place of origin of cavitation but at a point further away. The materials of construction are being pitted and can appear similar to the surface of sponge.

In case of extensive cavitation destructions will reach the guide vanes and the surroundings accordingly. In this condition vapor bubbles are formed at the end of the guide vanes if the back side (suction side) of the impeller blades slide past the guide vanes. These bubbles will collapse as soon as the front side (pressure side) of the following blade gets closer. Formation and collapsing of vapor bubbles will alternate periodically with the frequency resulting out of the product of speed and number of blades. Whereas the cavitation damages at the impeller are more or less spread out the guide vanes are damaged mainly at their ends and the surroundings accordingly. It is remarkable that the guide vanes - in case that the cavitation reaches them - are more affected than the impellers, due to the high frequency of the impact effects. Fig. 8 shows a diffuser damaged by heavy cavitation.

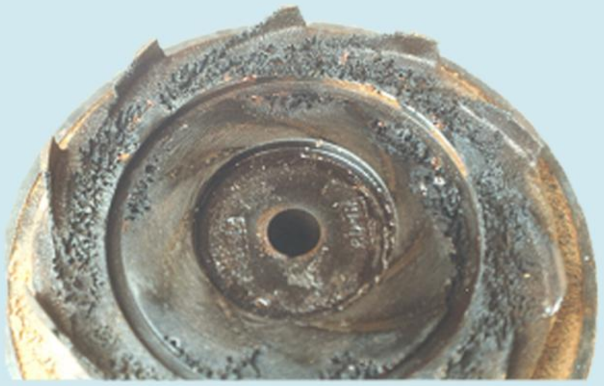

Fig. 8

The volume of vapor bubbles within the impeller is not equally distributed. Therefore unbalances occur on the one hand of mechanic type due to different liquid filling of the blade channels and on the other hand of dynamic type due to different pressure distribution over the circumference of the impeller. Shaft and bearing are subjected to violent pulsating loadings. The balancing of axial thrust for closed impellers may partly or completely become ineffective and the increased axial thrust resulting from that acting in combination with the afore mentioned forces will damage the bearing within short periods of time.

Frequently cavitation is combined with corrosion. This is also the case if the aggressiveness of the liquid is negligible. Caused by the collapse of the vapor bubbles the originally existing protective layers are being destroyed and renewal prevented. The metal surface therefore is permanently activated for the chemical attack. In this way even in case of slight cavitation it may lead to considerable damage to the materials.

3. To avoid Cavitation

Cavitation always occurs in case of vacuum at the inlet branch if a certain rate of flow is being exceeded. Fig. 9 shows the effect to the throttle curve of a diffuser pump for cold water with various vacuums in bar.

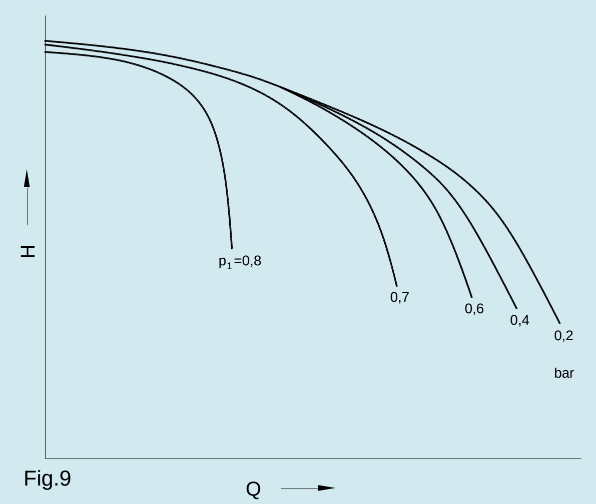

Fig.9

Only this range is free of cavitation being in congruence with that throttle curve being assigned to a smaller vacuum. It shows that the cavitation-free working range of a pump is getting smaller the higher the vacuum is. In case of very high vacuums no further contact exists to other curves, i.e. at each rate of flow the pump operates with cavitation.

The deviation of the pump curve as shown in Fig. 9 does not only occur at installations with geodetic suction head but e.g. also in case of transport of hot water with insufficient pressure above atmospheric at inlet of the pump. Only in case that the NPSHA (NPSH available) to the installation is higher than the required NPSH of the pump at point of operation cavitation-free transport is to be expected. In order to consider atmosphere pressure fluctuation, deviation from values being theoretically determined as well as test and manufacturing tolerances it is recommended to provide a safety margin of 0,5 m.

The required NPSH of a pump is being determined by the pump manufacturers without exception for a situation at which a certain cavitation has already occurred. It is common practice to permit a decrease of head of 3 % compared with the cavitation-free curve. This is for single-stage pumps. For multi-stage pumps the permissible decrease of head is meant for **one** stage.

Depending on the kind of liquid to be pumped, the application of the 3 % rule will result in deviation of NPSH with one and the same pump. Since a 3 % decrease in head at a chosen capacity will require a certain volume of vapor, a different NPSH will be ascertained because various liquids do not have equal vapor pressure. With this criterion liquid hydrocarbons for instance will require less NPSH. Since the characteristics of liquids are dependent on temperature, this will also influence NPSH. Therefore NPSH for water with rising temperatures decreases, when an equal deviation in head is permitted.

The NPSH determination for the characteristic curves of the EDUR technical documentation has been made with cold water. Therefore no further safety factor exceeding the usual 0,5 m safety factor is necessary. The required NPSH of centrifugal pumps however is not valid for liquids with an viscosity υ = 1 mm²/s.

1. Example of an Installation with Open Suction Tank

In a cooling circuit water with a maximum temperature of t = 25°C is taken from a tank. The water level inside this tank may go down to 5 m below center of the pump. The required rate of flow is Q = 80 m³/h. The suction pipe line consists of a foot valve with strainer, 12 m pipe length and 1 elbow; all dimensioned DN 125, i.e. the max. permissible flow velocity of 2 m/s will not be exceeded.

The NPSHA (NPSH available) has to be calculated

a) for an altitude of installation 30 m above sea level
b) for 800 m above sea level

To accomplish this we use the equation (14) from sheet 4 „Suction Capacity of Centrifugal Pump“

$$NPSHA = \frac{10{,}2 \cdot (p_I + p_b - p_D)}{\rho} - H_{VS} - H_{S\,geo}$$

p_I Inlet pressure
The water level of the tank is under atmospheric pressure, therefore neither a gauge pressure nor vacuum is available. $p_I = 0$.

p_b Atmospheric pressure at place of installation
a) up to an altitude of 100 m we can calculate with an atmospheric pressure of 1000 mbar. $p_b = 1{,}0$ bar.

b) from the table at the reverse of sheet 4 we take a value of $p_b = 0{,}9$ bar for an altitude of 1000 m.

p_D Steam pressure
From the table „Physical Parameter of Water“ we take the value for water at a temperature of t = 25°C: $p_D = 0{,}032$ bar.

ρ Density
Also from this table $\rho = 0{,}997$ kg/dm³, which may be rounded up to $\rho = 1{,}0$.

H_{VS} Friction losses in the suction pipe line

(see table „Friction Losses in Pipes“)

length of pipe line	12,0 m
equivalent pipe length for a footvalve with strainer at DN 125	26,0 m
elbow	2,7 m
total	40,7 m

Losses referred to 100 m pipe length DN 125 : 2,9 m. Losses referred to 40,7 m:

$$H_{VS} = \frac{2{,}9 \cdot 40{,}7}{100} = 1{,}18 \qquad \text{m}$$

$H_{s\,geo}$ Geodetic suction head

given with $H_{S\,geo} = 5$ m:

$$\text{a)}\quad NPSHA = \frac{10{,}2 \cdot (1{,}0 - 0{,}032)}{1{,}0} - 1{,}18 - 5 = 3{,}7 \qquad \text{m}$$

Considering a safety margin of 0,5 m a pump model has to be selected with a NPSH = 3,2 m or less. This demand can be met without any problem by a standard 80 m³/h pump and the standard speed of rotation n = 2900 rpm:

$$\text{b)}\quad NPSHA = \frac{10{,}2 \cdot (0{,}9 - 0{,}032)}{1{,}0} - 1{,}18 - 5 = 2{,}7 \qquad \text{m}$$

Due to the smaller atmospheric pressure in this case a pump with NPSH = 2,2 m at Q = 80 m³/h has to be selected. This inevitably leads to a bigger pump working at partial load or to an extremely more costly pump with a speed of rotation n = 1450 rpm.

2. Example of an Installation with Closed Suction Tank

10 m³ water have to be pumped from a condenser per hour. Temperature of the liquid is 50°C, vacuum in the tank is 0,8 bar, the water level inside the tank may go down to 1,5 m above center of the pump. Friction losses in the suction pipe line to be $H_{VS} = 0{,}2$ m. Atmospheric pressure at place of installation 1000 mbar in average.

Calculation: for water at t = 50°C a steam pressure $p_D = 0{,}1234$ bar has to be considered. In order to find out whether the water is under saturation condition first of all we have to calculate the **absolute** pressure in the tank:

$$p_{abs} = p_I + p_b = -0{,}8 + 1{,}0 = 0{,}2 \qquad \text{bar}$$

Therefore it is not under saturation condition and the equation (17) does not provide correct results. Therefore we are using the equation (15)

$$NPSHA = \frac{10{,}2 \cdot (p_I + p_b - p_D)}{\rho} - H_{VS} + H_{Z\,geo}$$

p_l Pressure in the tank

It is vacuum, p_l therefore has to be put in with minus sign. We can simplify this calculation since we already determined the absolute pressure to:

$p_l + p_b = 0{,}2$ bar.

The other values will not have to be explained:

$p_D = 0{,}1234$ bar
$\rho = 0{,}988$ kg/dm³
$H_{vs} = 0{,}2$ m
$H_{z\,geo} = 1{,}5$ m

$$\text{NPSHA} = \frac{10{,}2 \cdot (0{,}2 - 0{,}1234)}{0{,}988} - 0{,}2 + 1{,}5 = 2{,}09 \quad \text{m}$$

and with 0,5 m safety margin NPSHA = 1,6 m.

For a rate of flow of 10 m³/h there will be no problem to select a pump at n = 2900 rpm and a NPSH = 1,6 m. However in case that the pressure will be reduced to saturation condition guided by the equation (17) it will result - the safety margin of 0,5 m already considered - to:

$$\text{NPSHA} = H_{Z\,geo} - H_{VS} - 0{,}5 = 1{,}5 - 0{,}2 - 0{,}5 = 0{,}8 \quad \text{m}$$

Except for small capacities problems will arise to find a suitable standard pump and it will be necessary to abstain from a reasonable pump and look for special constructions or standard pumps with n = 1450 rpm.