CHAPTER TWO

PUMP

HYDRAULICS

Pump Hydraulics

In the previous chapters, we have seen the evolution of pumps, their construction, and their wide applicability. The many requirements of pressurized liquid lead to a large variety of pumps each designed and suited to a required application.

In spite of many different types of components, the basic mechanics and the principle of operation of the centrifugal pumps are similar.

Centrifugal Pumps are hydraulic machines that are used to energise and transfer a fluid within a system, at a flow that is dependent upon system needs. In order to understand how the pumps perform this function, it is essential to get familiar with some of the hydraulic terms associated with centrifugal pumps and of the liquids that they handle.

3.1 Specific gravity

The term 'specific gravity' refers to the ratio of the density of liquid to the density of water at 4°C (the density of water at this temperature is 1.000 kg/l). Specific gravity is a ratio and is hence a dimensionless quantity and is not expressed in any units.

$$\text{Specific gravity} = \frac{\rho_{\text{liquid}}}{\rho_{\text{water}}\, at\, 4^\circ C}$$

To find the specific gravity of a liquid, we must know its density in kilograms per meter cubed (kg/m□) or in grams per millimeter cubed (g/mm□). Then, divide this density by the density of pure water in the same units. If you use kg/m□, divide by 1000. If we use g/mm□, divide by 1 (that is, leave the number alone). It is important to use the same number of units in the numerator and denominator.

Materials with a specific gravity of less than 1 are less dense than water and therefore will float on it. Substances with a specific gravity of more than 1 are denser than water and will sink.

An object with a density of 100 kg/m□ has a specific gravity of 0.1, and can float on the surface of the body of water. An object with a density of 10 g/mm□ has a specific gravity of 10 and can sink rapidly.

3.2 Viscosity

Viscosity is best understood by imagining a styrofoam cup with a hole in the bottom. If honey is poured in this glass, it is noticed that the cup drains very slowly because the viscosity of honey is large compared to other liquids. If the same cup is filled with water, the cup will drain much more quickly. Viscosity is a measure of a fluid's resistance to flow.

It describes the internal friction of a moving fluid. A fluid with large viscosity resists motion because its molecular makeup gives it a lot of internal friction. A fluid with low viscosity flows easily because its molecular makeup results in very little friction when it is in motion.

In certain fluids called Newtonian fluids, the shear stress that causes the flow is directly proportional to the shear strain (rate of deformation).

The ratio of this shear stress to the shear strain is constant for a given fluid at a fixed temperature.

This constant is called the dynamic or absolute viscosity (µ) and often simply the viscosity. The viscosity of liquids decreases rapidly with an increase in temperature. Thus, upon heating, liquids flow more easily.

The dimensions of dynamic viscosity are force times time divided by area. The unit of viscosity, accordingly, is newton-second per meter square (N-s/m□).

For some applications, the kinematic viscosity is more useful than the absolute or dynamic viscosity.

Kinematic viscosity is obtained by dividing the absolute viscosity of a fluid by its mass density. (Mass density is the mass of a substance divided by its volume.)

$$\text{Kinematic viscosity} \quad \nu = \frac{\mu}{\rho}$$

The dimensions of kinematic viscosity are area divided by time. Its units are meter squared per second (m□/s).

Kinematic viscosity (ν) v is often expressed in stokes, St, where 10□St = 1 m□/s.

However, a more common unit of measure is centistokes (cSt).

Some other common viscosity units and conversion factor is listed below:

Kinematic Viscosity	**×**	**Specific Gravity**	**Absolute Viscosity**
Centistokes	×	S.G.	Centipoise
SSU × 0.2198 –*	×	S.G.	Centipoise

SSU –*	×	0.2198	=	Centistokes
Degree Engler –*	×	7.45	=	Centistokes
Seconds Redwood –*	×	0.2469	=	Centistokes

*** For centistokes greater than 50.**

3.3 Vapor Pressure

The vapor pressure of a liquid, pure or mixed, is defined as the pressure exerted by those molecules that escape from the liquid to form a separate vapor phase above the liquid.

If a quantity of liquid is placed in an evacuated, closed container, the volume of which is slightly larger than that of the liquid, most of the container is filled with the liquid. After a period, a vapor phase forms in the space above the liquid surface. This space consists of molecules that have passed through the liquid surface from liquid to gas. The pressure exerted by that vapor phase is called the vapor (or saturation) pressure. For a pure liquid, this pressure depends only on the temperature.

Following are some examples of vapor pressures for a few common liquids. The vapor pressure is 1 atm at 100°C for water, at 78.5°C for ethyl alcohol, and at 125.7°C for octane. Similarly, at 20°C, water has a vapor pressure of 0.023 atm (17.5 mm Hg). Isopropyl alcohol (rubbing alcohol) has a vapor pressure of 0.043 atm (33 mm Hg) at 20°C.

In a liquid solution, the component with the higher vapor pressure is called the light component (tendency to vaporize quicker), and that with the lower vapor pressure is called the heavy component.

3.4 Flow

The first and most important point to consider is that centrifugal pumps are volumetric machines. The liquid pumped is measured in terms of the volume flow rate. The units used are m□/h or gpm (US or Imperial).

It is worthy to note that any pump, for a single point of operation would always give the same volumetric flow rate for any liquid, be it hydrocarbon, water, or any other. Depending on the density of the liquid, the mass flow rate changes.

If we have a pump whose capacity is 20m□/h, then the mass flow rate would pump 20tph of water. However, the same pump when handling a hydrocarbon with a specific gravity of 0.8 would pump only 16tph.

3.5 Head

The pressure of the liquid can be stated in terms of meters (feet) of head of the liquid column (mlc). As in case of volumetric flow rate, the head generated by the pump in mlc for a single point of operation is the same for any liquid. Depending on the density of the liquid what changes is the reading on the pressure gage.

Any pump raises the liquid from one gradient (head) to another. Thus, the difference between the discharge head and the suction head is termed as 'differential head'.
The differential head developed by a pump is expressed in 'm' of liquid:

P_d = Discharge pressure (kg/cm□)

P_S = Suction pressure (kg/cm□)

ρ = Specific gravity of the liquid.

3.6 System Resistance

The flow rate delivered by the centrifugal pump is dependent on the total frictional and static head that it has to overcome.

The required head comprises of two components. These are:

1. A static component: h_S in meters 'm', which is independent of the flow through the pump. For example, if the liquid has to be raised from one height
2. A friction head loss component: h_f in meters 'm'. This head is proportional to the square of the flow rate – 'Q' in l/s.

The friction component is the summation of losses that occur as the liquid flows through the pipes and various equipment like heat exchangers.

To account for the losses, the entire flow path from the suction vessel to the discharge vessel is considered.

If this path has a large number of fittings, such as elbows (more bends), reducers, valves, and orifices, the losses are higher. To ease the calculations, nomogram of equivalent length of valves and fittings is used. An equivalent length is the length of pipe that would offer the same losses for a flow rate as offered by the fitting.

Thus, the suction and discharge paths, which may have a few fittings, are converted to an equivalent length. Using another nomogram, the friction loss due to the flow rate can be estimated.

Usually the total losses (static + frictional) on the suction side of the pump are calculated separately to establish the fact that there is adequate $NPSH-a$ available compared to $NPSH-r$ required, for the pump to operate satisfactorily. If $NPSH-a$ is less than $NPSH-r$, the pump will operate under conditions of cavitation which is undesirable. The next step is to evaluate the total system resistance H_t, which is the summation of losses on both the suction and discharge sides and includes the static lift H_{st}.

To compute the system resistance, consider the system as shown in Figure 3.1 where the pump flow rate is 100m□/h (27.8 l/s) of water, through steel pipes.

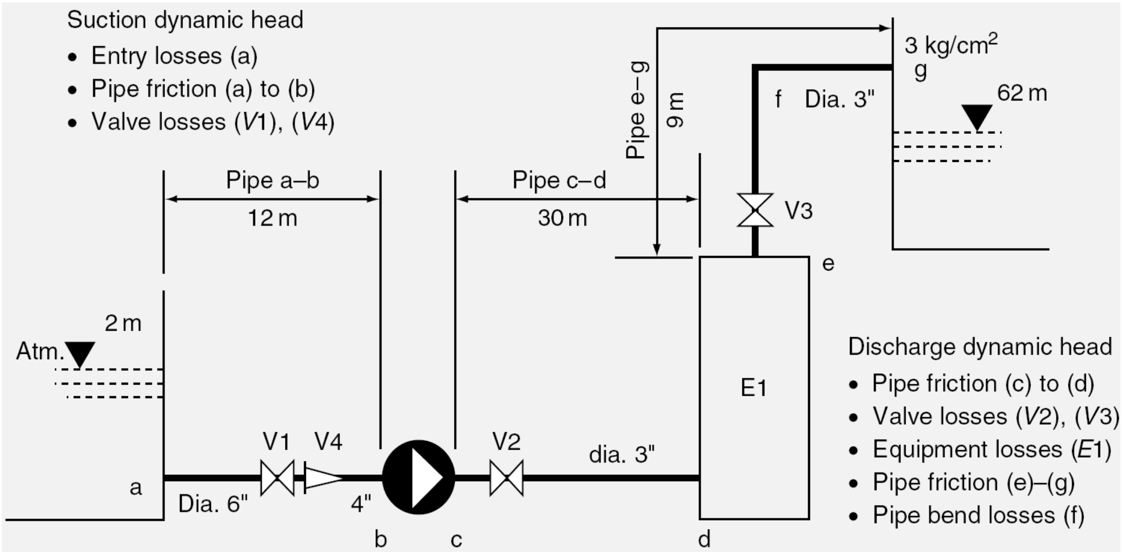


Figure 3.1 – A Typical Pumping System

3.6.1 Evaluate the Suction Side

Step 1: Calculate the Velocity in the Suction Pipe –6 in. (152.4mm)

$$\text{Velocity} = \frac{\text{Flow}}{\text{Area}}$$

$$= \frac{(100/3600)}{[(\pi/4)\times(0.1524)^2]}$$

$$= 1.52\,\text{m/s}$$

Step 2: Compute the Suction Head

The suction is from atmospheric vessel – h_a = 10.34 m

Suction height – H_s = 2 m

Step 3: Compute the Equivalent Pipe Length

Pipe length = 12m (assume almost all length is 6″ or 150mm nom. bore)

6″ Gate valve (fully open) V1 – equivalent length = 1m
6″ × 4″ eccentric reducer V4 – equivalent length = 1.4m
Entry losses – equivalent length = 6m
Total equivalent pipe length = 14.4m.

Step 4: Compute Friction Loss from Pipe Friction Tables

Friction loss for pipe length of 100m = 1.52 m/100m
Friction loss for pipe length of 20.4m = 0.31 m/100m

Step 5: Compute Total Suction Head

$$h_a + H_s - P_1 = 10.34 + 2 - 0.31 = 12.03m$$

Thus, the pump sees a differential head of:
Discharge head – suction head = 100.5 – 12.03 = 88.47 m.
The system resistance which the pump has to overcome is 88.47 m.

3.7 Pump Efficiency

The pump does not completely convert kinetic energy to pressure energy since some of the kinetic energy is lost in this process. Primarily, there are three areas where this energy is dissipated and not converted to useful work.

Pump efficiency is a factor that accounts for these losses. Pump efficiency is a product of the following three efficiencies:

1. Hydraulic efficiency (primarily, disk friction, which is the friction of the liquid with the impeller shrouds. This is a function of speed and impeller geometry. Other losses are shock losses during rapid changes in direction along the impeller and volute)
2. Volumetric efficiency (recirculation losses at wear rings, interstage bushes and other)
3. Mechanical efficiency (friction at seals or gland packing and bearings).

Some texts call the product of the first two efficiencies as internal efficiency of the pump.

Every pump is designed for a specific flow and a corresponding differential head, though it is possible to operate at certain percentage points away from the designed values.

However, the efficiency of the pump at the designed point is maximum and is called as the BEP. Efficiency at flows lower or higher than this design point is lower.

The efficiency of the pump has a close relationship to an important pump number called as the specific speed. This we shall cover in Section 3.11.

3.8 Hydraulic Power

If a pump were an ideal machine, the required input power to drive the pump would entirely lift the mass flow rate from one elevation to another. This power is called as the hydraulic power.

$$P_{H(kW)} = \frac{Q \times \rho \times g \times H}{3.6 \times 10^6}$$

Where:

Q	=	capacity in m□/h
ρ	=	liquid density in kg/□m at pumping temperature
H	=	differential head in m (meters of liquid column)
g	=	gravitational acceleration in m/s□.

When this hydraulic power is divided by pump efficiency, we get the shaft power.

$$P_{S(kW)} = \frac{P_H}{\eta_P}$$

3.9 Pump Characteristic Curve

With every pump, the manufacturer provides a curve depicting the performance or the behavior of the pump under various conditions. This is called as a characteristic curve of the pump.

Characteristic curve essentially comprises of four curves and these are:

1. Q vs H: Capacity vs differential head
2. Q vs efficiency: Capacity vs pump efficiency
3. Q vs power: Capacity vs shaft power
4. Q vs $\mathrm{NPSH-r}$: Capacity vs Net Positive Suction Head – required.

3.9.1 Flow Rate (Q) vs Differential Head (H) Curve

The Q vs H curve is a continuously drooping curve from shut-off (no flow) condition to BEP. API recommends that the curve from BEP to shut-off should rise by at least 10% for single-stage, single pump operation.

Any pump model can be assembled with trimmed impellers (usually not smaller than 20% of the maximum possible diameter).

The Q vs H curve of trimmed impellers in the operating range are parallel and below the Q vs H curve of the maximum diameter impeller.

The characteristic curves encompassing performance of all the possible impeller diameters for that model have efficiency depicted as iso-efficiency curves on the Q vs H curve.

On every Q–H curve, a small triangle is plotted to indicate the rated point of operation. The pump manufacturer guarantees this flow and the corresponding differential head.

Usually, the flow at rated point is in excess by 5 or 10% of the flow at which the pump will operate at most of the times or as specified by process demands. This operating point is called as the normal operating point.

Centrifugal pumps with radial impellers are started with discharge valves closed. At this point, there is no flow supplied by the pump and the entire liquid keeps churning in the casing. This point of operation is termed shut-off head and run time in this mode must be minimized.

3.9.2 Flow Rate (*Q*) vs Pump Efficiency (η_p)

The Q vs pump efficiency of the pump is an inverted 'U' shaped curve. At no flow, the efficiency is zero and then rises to a maximum value at a flow rate, which is termed as the BEP. Beyond this, the curve again drops.

The pumps operate in a range of flows but it has to be kept in mind that they are designed only for one flow rate point. Flow rates above and below this value result in higher hydraulic losses and hence lesser efficiency. The design point is the BEP.

3.9.3 Flow rate (*Q*) vs power (P_S)

The pump trial is carried using cold water as the liquid. As volumetric flow in m□/h, differential head in m, and pump efficiency are independent of the liquid pumped, the results obtained are valid for all service liquids.

Power obtained is for water and can be easily extrapolated for the liquid by multiplying it with the specific gravity of the service liquid.

3.9.4 Flow rate (Q) vs NPSH-r

$\mathrm{NPSH\text{-}r}$ is covered in detail in Section 3.12. However to initiate, it is the Net Positive Suction Head required by a pump to avoid a phenomenon called as cavitation.

$\mathrm{NPSH\text{-}r}$ on the characteristic curves is the measured suction head obtained while throttling the suction flow until a 3% drop in the differential head is observed at any particular flow rate (see Figures 3.2 and 3.3).

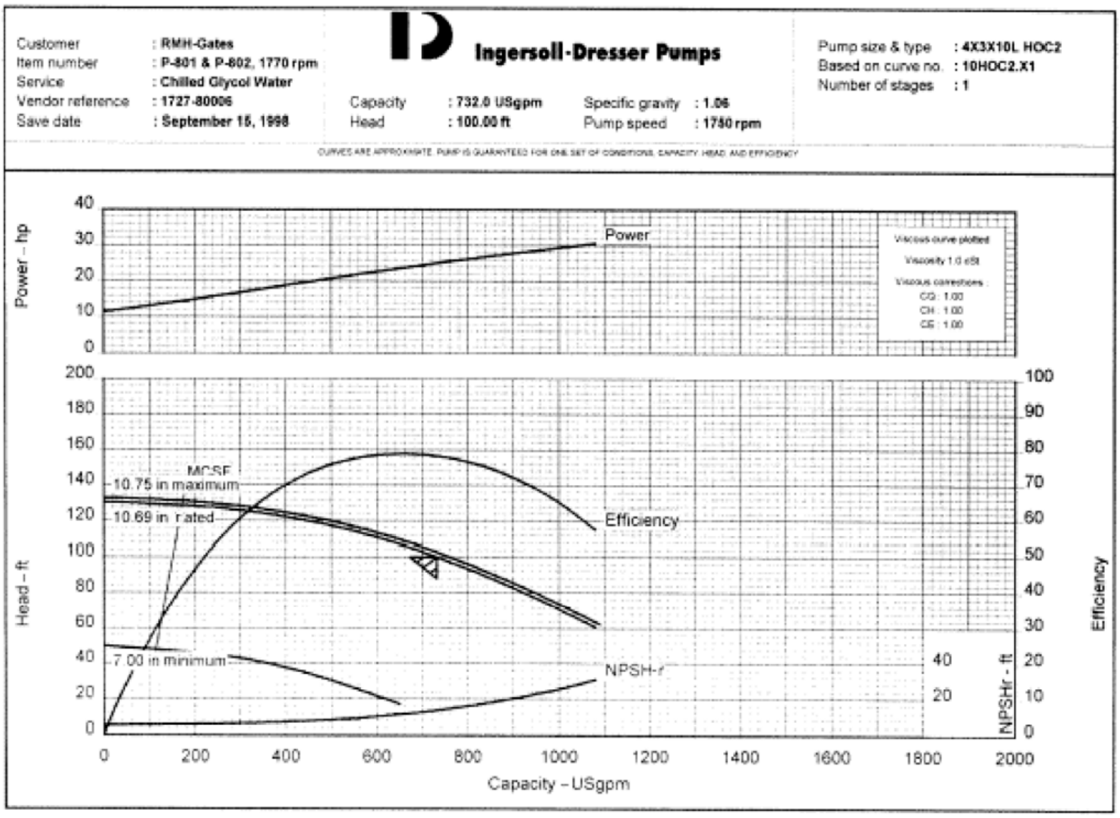


Figure 3.2 – A Typical Pump Characteristic Curve

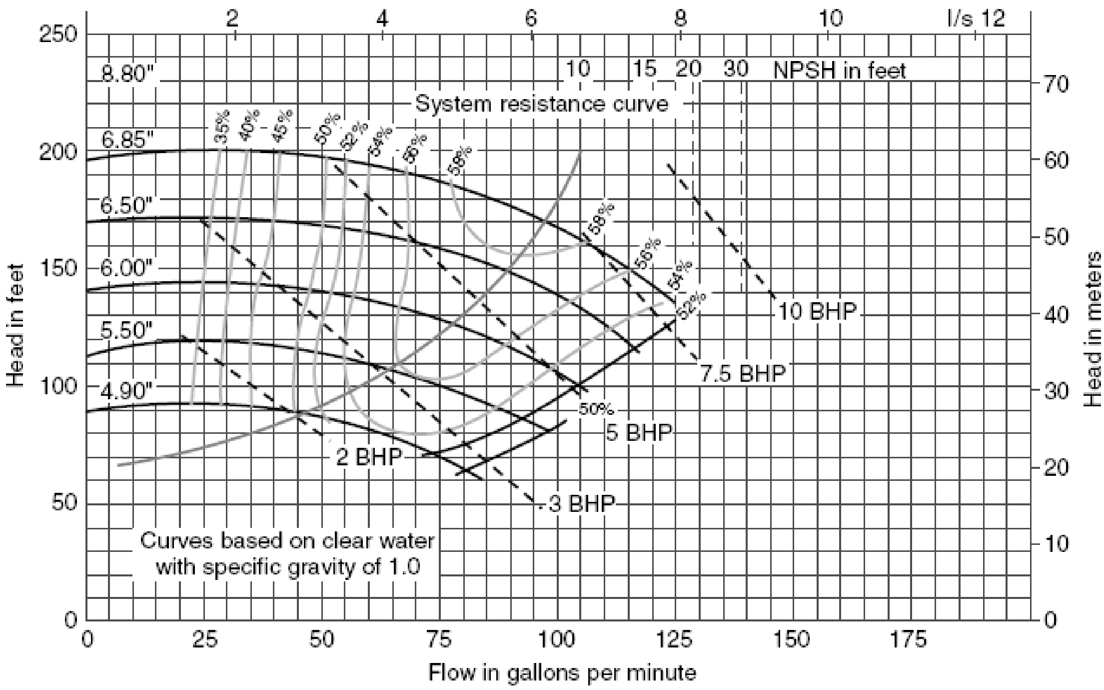


Figure 3.3 – Pump Curve at Various Diameters with System Resistance and ISO-Efficiency Curves

NPSH - r is dependant on the service liquid but it is known that cavitation resulting from cold water is most damaging as compared with most commonly pumped liquids (hydrocarbons, hot water) and so NPSH - r results obtained with cold water can be safely applied to other service liquids as well.

3.10 Curve corrections

The pump curves are generated while testing the pump using cold water as the liquid. The curve is fixed for a particular speed, impeller diameter, and water.

It is not necessary that the pumps' actual operation throughout its life will be for the same speed or impeller diameter and service. When any of these change, the pump flow and head generated will differ.

In certain cases, it is possible to predict the flow and head for alternate conditions using factors.

Thus, with the help of these factors, the curves can be corrected to obtain a performance map without retesting pump with modified conditions.

3.10.1 Affinity laws

The 'Affinity laws' are mathematical expressions that best define changes in pump capacity, head, and power absorbed by the pump when a change is made to pump speed, with all else remaining constant.

According to affinity laws

Capacity Q changes in direct proportion to the change in pump speed N ratio:

$$Q_2 = Q_1 \times \left(\frac{N_2}{N_1}\right)$$

Head H changes in direct proportion to the square of the speed N ratio:

$$H_2 = H_1 \times \left(\frac{N_2}{N_1}\right)^2$$

Power P changes in direct proportion to the cube of the speed N ratio:

$$P_2 = P_1 \times \left(\frac{N_2}{N_1}\right)^3$$

Where the subscript 1 refers to initial condition and 2 refers to new condition.

Important: The Affinity laws are valid only under conditions of constant efficiency.

The pump affinity laws mentioned above maybe utilized to determine the relationship between flow 'Q' and impeller diameter as well as to predict Head 'H' and Power 'P' values with change in impeller diameter, whilst speed is kept constant.

The results obtained however are approximate as these formulae are analogous to the centrifugal pump affinity laws. Hence we have;

$$Q_2 = Q_1 \times \left(\frac{D_2}{D_1}\right) \qquad H_2 = H_1 \times \left(\frac{D_2}{D_1}\right)^2 \qquad P_2 = P_1 \times \left(\frac{N_2}{N_1}\right)^3$$

The affinity laws described above require correction when performance is to be predicted following a change in impeller diameter. Due to the above mentioned constant efficiency factor, there is a discrepancy between the calculated impeller diameter and the achieved performance. This error becomes larger with the increase in cut of the impeller.

If, C is the calculated required percentage of impeller diameter and A is the actual required diameter percentage of the original diameter then:

$$\mathrm{A} = 16.2 + 0.838 \times \mathrm{C}$$

Thus, by calculation using affinity laws, it is computed that the impeller has to be trimmed to 84% of the original diameter then the actual trimming should be limited to 86.6% of the original diameter.

Note: There are a number of recommended empirical formulae to calculate impeller diameters to match reduced or increased pump flow rates. KSB – Centrifugal Pump Design recommends the following approximate formula for KSB pump impeller trim.

$$\left(\frac{D_2}{D_1}\right)^2 \text{approx.} = \frac{Q_2}{Q_1} \text{approx.} = \frac{H_2}{H_1}$$

In all cases, however, proceed with caution.

3.10.2 Viscosity Corrections

Under Section 3.2, we have discussed viscosity as a property of any fluid that is measure of its resistance to flow.

As the liquid flows through the pump, hydrodynamic losses are increased due to higher viscosity, as a result it is observed that when a viscous fluid is handled by a centrifugal pump:

- The brake horsepower requirement increases.
- There is a reduction in the head generated by the pump.
- Capacity reduction occurs with moderate and high viscosities.
- There is a decrease in the pump efficiency.

This is more evident in smaller pumps. For higher viscosities, larger pumps are used.

A viscosity correction chart from the Hydraulic Institute (as shown in Figure 3.4) provides coefficients for flow C_q, head C_h, and efficiency C_η.

These coefficients are used to modify the values of flow, head, and efficiency from the original curve. The new flow, head, and efficiency are obtained using the equations mentioned below.

$$Q_{vis} = C_q \times Q_w$$

$$H_{vis} = C_h \times H_w$$

$$\eta_{vis} = C_\eta \times \eta_w$$

Usually fluids more than 2 centipoise should be considered for viscosity correction.

e.g. – Q = 500 US-gpm, H = 80 ft, Viscosity = 1000 SSU.

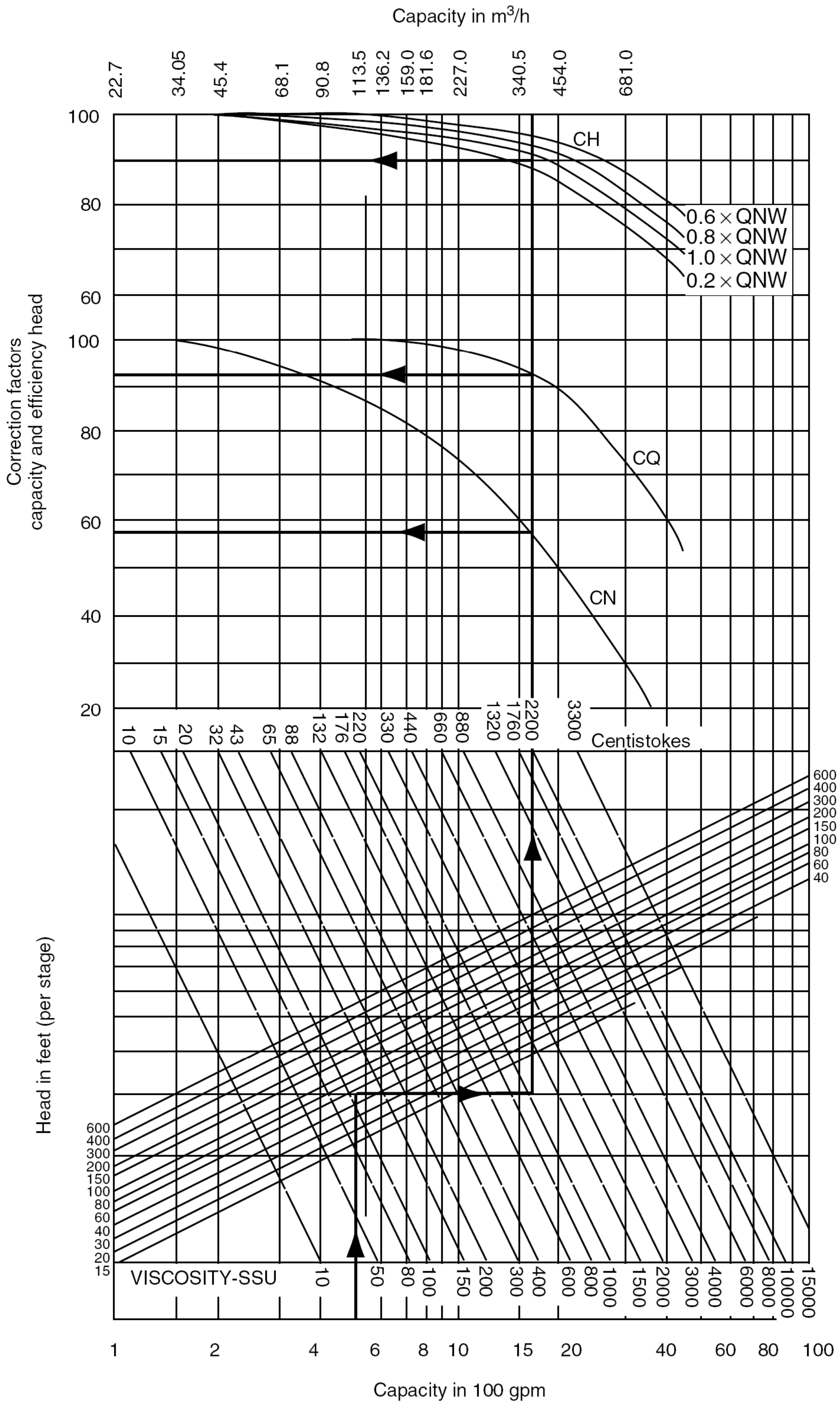


Figure 3.4 – Viscosity Correction Charts – Correction Factors, Hydraulic Institute

3.11 Specific speed

Specific speed is a number characterizing the type of impeller in a unique and coherent manner.

Specific speed is defined by the equation,

$$N_s = \frac{N\sqrt{Q}}{(H)^{3/4}}$$

Where:

N = pump speed

Q = flow at BEP at maximum impeller diameter (no corrections even if it is a double suction impeller)

H = head per stage at BEP at maximum impeller diameter.

It states that N_S is the speed in rpm at which a pump, if sufficiently reduced in size, would deliver (in US units) 1 gpm at a head of 1 ft. This definition is of little practical utility.

Specific speed is for an impeller, hence for multistage pumps only the first impeller is considered (in the equation, $H = H_{(total)}/\text{number of stages}$).

An index identifies the geometric similarity of pumps. Pumps of the same N_S but have different sizes are considered geometrically similar, one pump being a size-factor of the other.

However, many critical parameters used for impeller design and geometry, volute design, pumps efficiency, layout of pump model performance charts as required by pump manufacturers are based on the specific speed.

Though, in principle, this text uses SI units as a base, an exception has been made in the case of specific speed. There is so much empirical work done with specific speed in US gpm, ft, and rpm units that it is considered prudent to get acquainted with FPS units than to persist with SI units and create confusion and errors.

To assist in getting familiar, the conversion from FPS to metric units is given below:

1 US-gpm = 0.2271 m□/h → 1 British pgm = 0.2728 m□/h

1 ft = 0.3048 m

US N_S = 1.63 N_S (metric N_S)

Metric N_S = 0.614 US N_S → British N_S = 1.5 metric N_S

Since the 1900s, it was known that there existed a correlation between efficiency and the pump flow, head, and speed. In 1947, George Wislicenus generated curves (shown in Figure 3.5) of specific speed vs the pump efficiency. This was a statistical average of the data made available from the commercial pumps in those times.

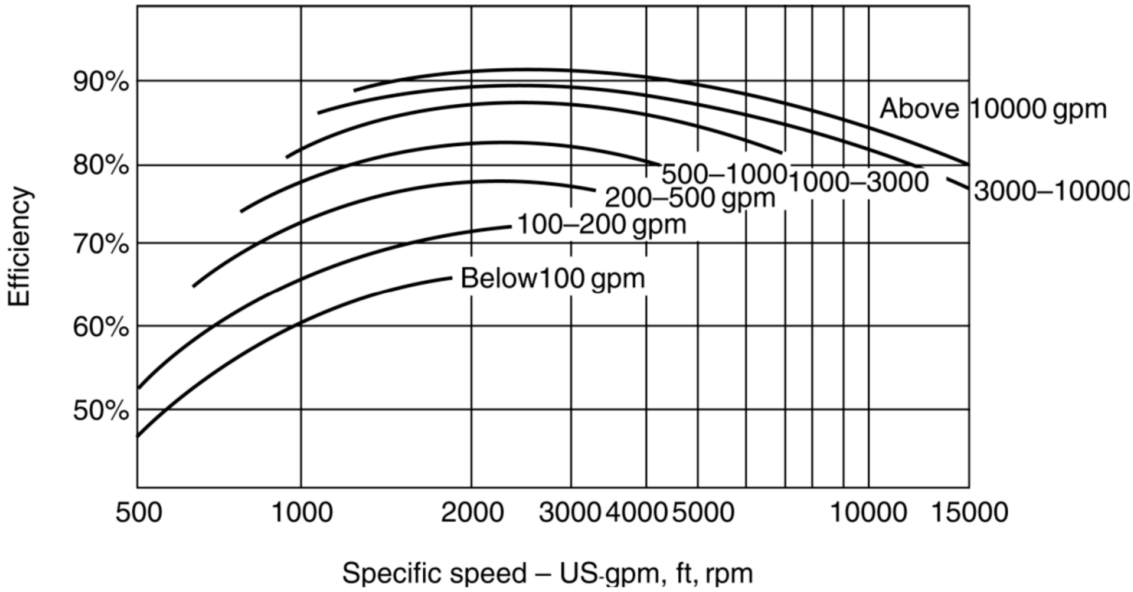


Figure 3.5 – An Early Chart – Relating Specific Speed with Single Stage Pump Efficiency (Original by George Wislicenus – 1947)

A more detailed study was generated taking into account various factors such as dimensional tolerance, surface roughness of wet parts, specified wearing ring clearances, and designs not exceeding specified suction-specific speed limits and similar curves.

The curves shown in Figure 3.6; is based on the work conducted by Eugene P. Sabini and Warren H. Fraser in their paper, 'The Effect of Specific Speed on the Efficiency of Single Stage Pumps' presented at the 1986 Pumps Users Symposium.

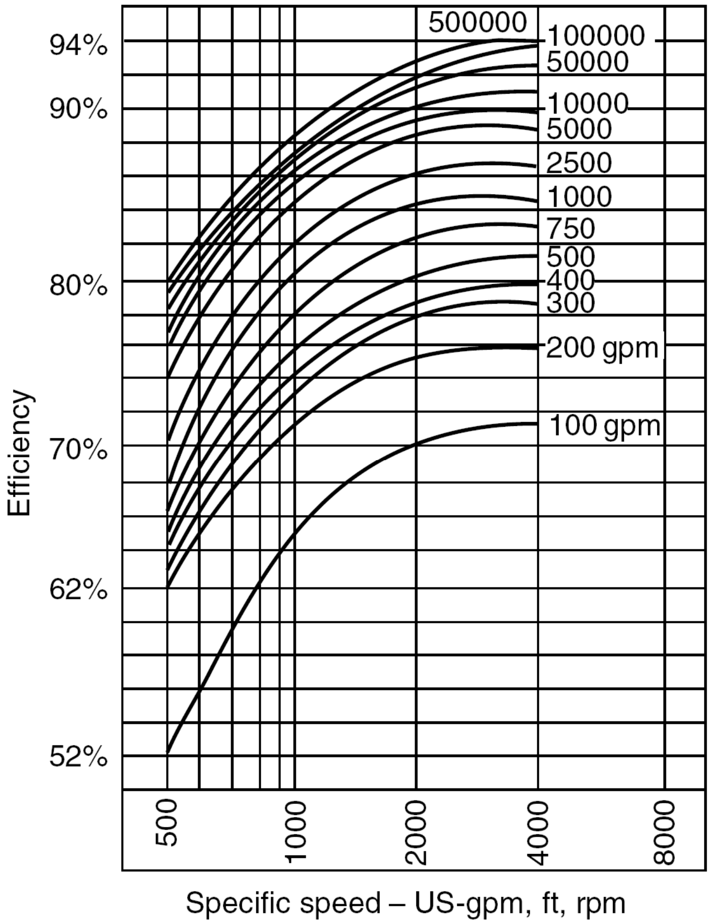
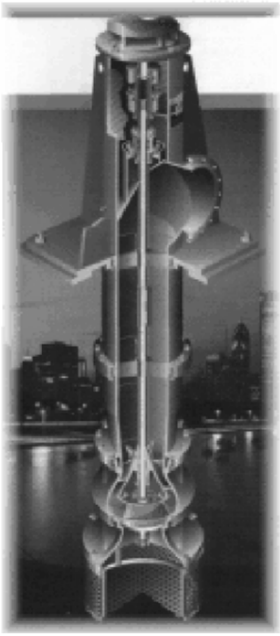


Figure 3.6 – Efficiencies of Single-Stage End-Suction and Double Suction Impeller Pumps

They considered the following in the preparation of the curves:

- Single-stage pumps only
- Finish and dimensional tolerance to within ±1% for vanes and hydraulic passages
- Surface finish of wet surfaces to be 2 × 10–6 per inch of impeller diameter or better
- Wearing ring clearances to be 0.0015 in. of ring diameter
- Suction-specific speed not exceeding 8500 (refer Section 3.13 – Units – USgpm, ft, rpm)
- Discharge recirculation within a range of 80–90%
- A uniform velocity profile at impeller inlet
- Fluid used was clean water at a temperature of 150°F or less
- Efficiencies were based on maximum impeller diameter
- Wet pit pump efficiencies were based on impellers with no back rings or balancing holes (Figure 3.7).



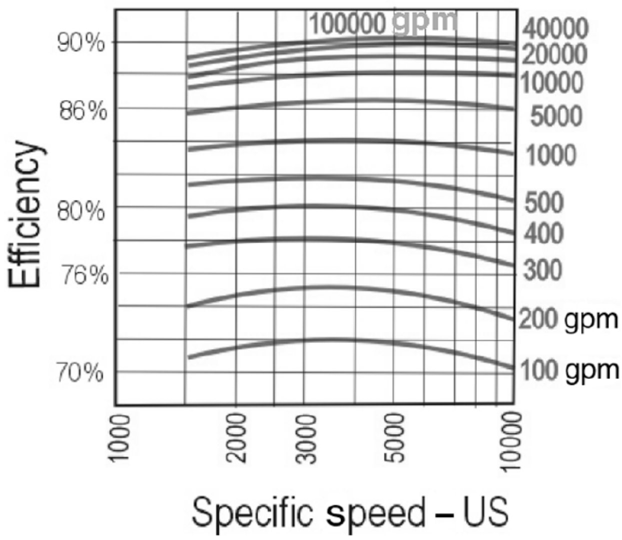


Figure 3.7 – Bowl Efficiencies of Wet Pit Centrifugal Pumps

A similar study was conducted by Lobanoff and Ross on pumps having six stages or less and operating at 3560 rpm. The study indicated that the efficiency for multistage pumps increases very rapidly to a specific speed of 2000 (US-gpm, ft, rpm) and stays constant until 3500 rpm. Then it begins to taper off a bit.

This is explained on the basis that hydraulic friction and shock losses for high specific speed pumps contribute greater percentage of total head than for low specific speed pumps.

The drop at low specific speeds is attributed to the fact that mechanical losses do not vary much over the range of specific speeds and are therefore a greater percentage of the total power consumption at the lower specific speeds.

Specific speed is a reference number that describes the hydraulic features of a pump, whether radial, semi-axial, or propeller type.

Another index related to the specific speed of the pump is the modeling law. It is usually applied to very large pumps in hydroelectric applications. It states that two geometrically similar pumps working against the same head will have similar flow conditions (same velocities at all sections) if they run at speed inversely proportional to their size. In this case, the capacity will vary the square of the size.

The optimum laying out of the performance chart or the family curves of a pump model is based on the specific speed (Figure 3.8). The BEPs of the family of pumps are usually lined up along the same specific speed. Their size is factored upward for higher flows and heads.

By far, it is the most important number of any pump model

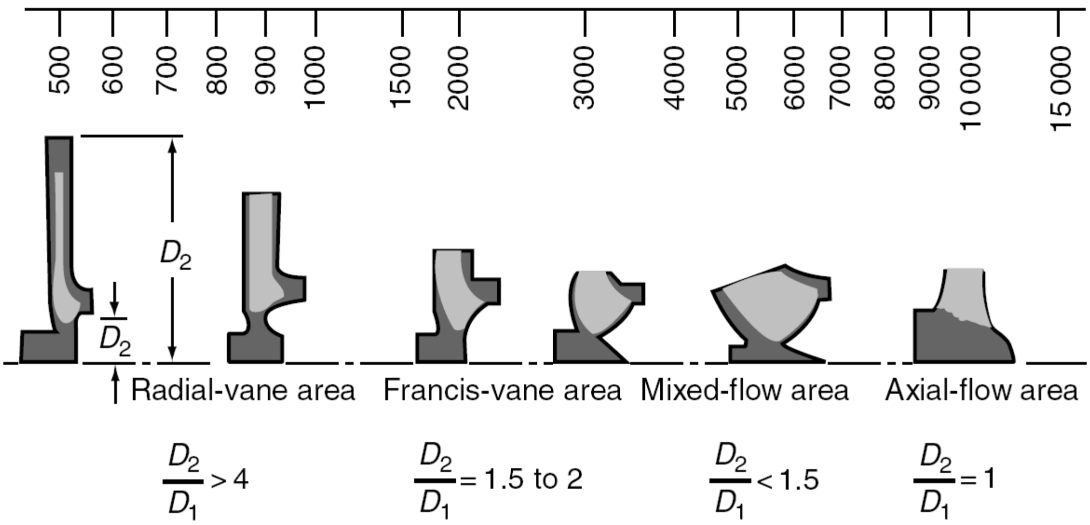


Figure 3.8 – Flow Area Chart

3.12 Cavitation, recirculation, and Net Positive Suction Head (NPSH)

3.12.1 Cavitation

Gases under pressure can dissolve in a liquid. When the pressure is reduced, they bubble out. Opening of a soda-water bottle is a good example.

In a somewhat similar way, when the liquid is sucked in the pump inlet, the pressure acting on the liquid surface drops. Under conditions, when the reduced pressure approaches the vapor pressure of the liquid (at that temperature), it causes the liquid to vaporize (see Figure 3.9). As these vapor bubbles travel further into the impeller, the pressure rises again causing the bubbles to collapse or implode.

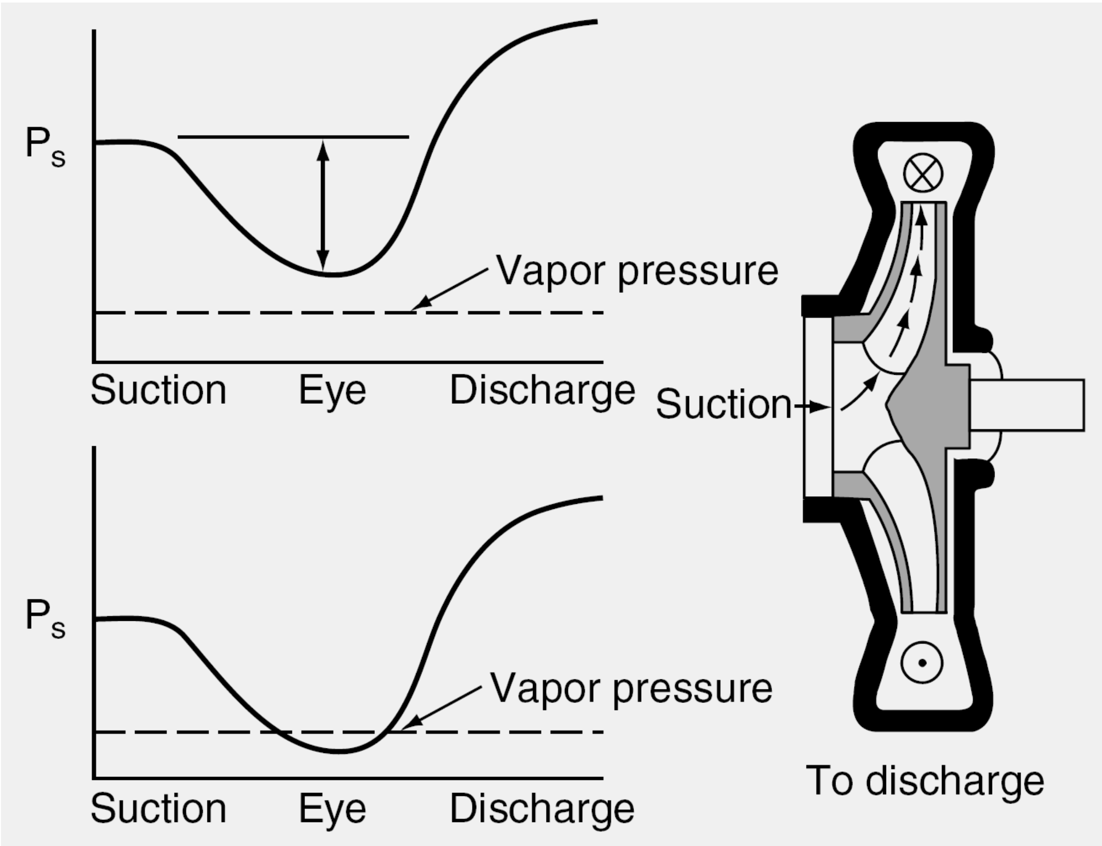


Figure 3.9 – Suction Pressure Falling below Vapor Pressure Causes Bubble Formation

This implosion adversely affects pump performance and could cause severe damage to pump internals. This phenomenon is called as 'cavitation'.

Cavitation damage to a centrifugal pump may range from minor pitting to catastrophic failure and depends on the pumped fluid characteristics, energy levels, and duration of cavitation.

Most of the damage usually occurs within the impeller; specifically, to the leading face of the non-pressure side of the vanes. This is the area where the bubbles normally begin to collapse and release energy on the vane. The net effect observed on the impeller vane will be a pockmarked, rough surface and severe thinning of the vanes from metal erosion.

3.12.2 Recirculation

Another type of cavitation seen in pumps is due to a phenomenon called as recirculation.

One of the complex problems associated with operation of pumps is that of recirculation. Recirculation is defined as flow reversal either at the inlet or at the outlet tips of the impeller vanes.

It is well established that cavitation type of damage seen on the inlet vanes and not associated with inadequate NPSH can be directly linked to the pump operation in the suction recirculation zone. Similar damage seen on the discharge vane tips too can be associated with pump operation in the discharge recirculation zone.

The suction and discharge recirculation may occur at different points as shown in Figure 3.10.

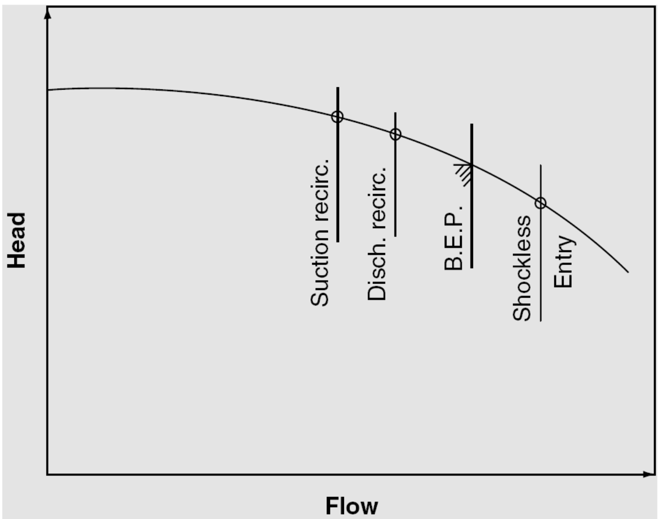


Figure 3.10 – Points on Curve where Recirculation can be Expected

The capacities at which the suction and discharge recirculation occurs are dependent on the design of the impeller at the inlet and outlet respectively. The casing has an influence on the intensity of the discharge recirculation but not on its inception.

Another observation made during extensive research indicated that when the inlet to outlet diameter ratio of the impeller equals or exceeds 0.5, the suction recirculation is in effect the capacity at which discharge recirculation occurs.

There are many explanations put forth to explain the phenomena of recirculation. Recirculation can occur at the suction as well as in the discharge as shown in Figure 3.11.

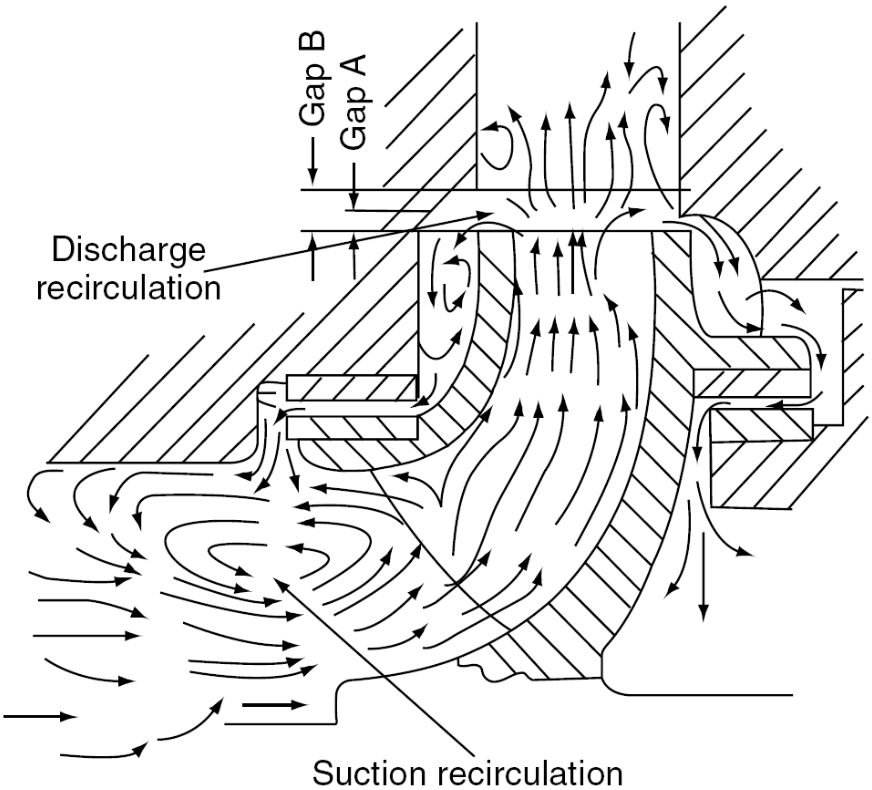


Figure 3.11 – Suction and Discharge Recirculation

One theory suggests that recirculation cavitation (rotating stall or separation) is the formation of vapor-fsilled pockets. This type of cavitation is different from the classical cavitation described earlier.

In suction recirculation, as the pump is operated to the left of the BEP, eddy currents begin to form at the eye of the impeller.

At this point of operation on the curve, there is no reduction in the flow rate through the pump. The eddy currents at the eye effectively reduce the flow channel size. As the flow rate is the same, the area is effectively reduced; it leads to an increase in the velocity of the liquid.

As the velocity increases, the pressure drops due to friction also increases. When there is a large drop in pressure below the liquid's vapor pressure, the pump experiences classical cavitation because of the initiating action of recirculation cavitation.

Another explanation provided for recirculation is that as the fluid flows over an impeller vane, the pressure near the surface is lowered and the flow tends to separate.

This separated region occurs when the incidence angle (see Figure 3.12), which is the difference between flow angle and pump impeller vane inlet angle, increases above a specific critical value.

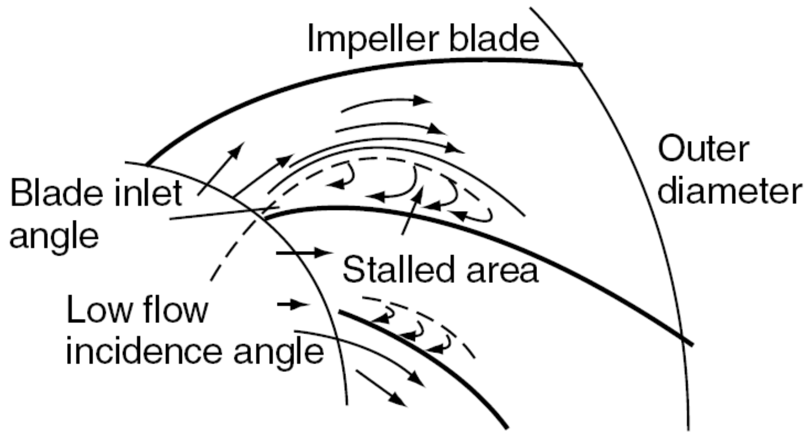


Figure 3.12 – Region of Stall in the Impeller

The stalled area eventually washes but as the rotation continues, it is reformed. The area contains a vapor surrounded by a turbulent flowing liquid at a higher pressure than the vapor pressure. This separated region will then fill with liquid from the downstream end.

The vapor pocket collapses, which causes damage to the surface of the impeller vane. This may occur up to 200–300 times per second.

The damage due to recirculation occurs on the opposite side of the vane where classical cavitation occurs.

This continuous recycling results in noise, vibration, and pressure pulsations. These results imitate classical cavitation, and thus recirculation is often incorrectly diagnosed as cavitation. Figure 3.13 shows regions within an impeller that are affected by cavitation and recirculation.

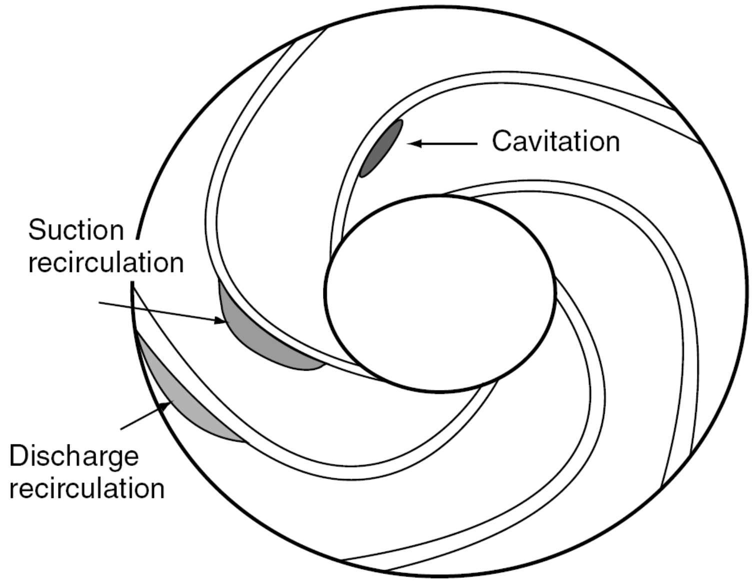


Figure 3.13 – Regions Within an Impeller that are Affected by Cavitation and Recirculation

It is often believed that only high-energy pumps (as per API 610 – sixth edition states: High-energy pumps are defined as pumping to a head greater than 650 ft (198 m) and more than 300 HP (224 kW) per stage) are affected by recirculation cavitation. However, an impeller constructed of cast iron or bronze can erode badly at much lower energy levels.

As the flow at the eye of the impeller recirculates, severe vortexing occurs. These vortices can pass through the impeller liquid channels and can initiate discharge recirculation, as shown in the figure on recirculation. The further the pump operates from its BEP, the greater the amount of vortexing.

Most pumps operate in either continuous or intermittent suction or discharge recirculation, especially those designed with high N_{ss} – suction-specific speeds (above about 11 000). (Suction-specific speed is covered in Section 3.13.) Impeller internal circulation usually shows up as cavitation noise and erosion damage, rotor oscillation, shaft breakage or surging in varying degrees depending on the pump design and application. Many of these problems can be avoided by designing the pump for lower suction-specific speed values and limiting the range of operation to capacities above the point of recirculation.

3.12.3 Net Positive Suction Head (NPSH)

The concept of NPSH involves two terms:

1. NPSH-r, called as the Net Positive Suction Head as required by the pump in order to prevent the inception of cavitation and for safe and reliable operation of pump.
2. 2. NPSH-a, called as the Net Positive Suction Head as made available by the suction system of the pump.

NPSH and its correlation to inception of cavitation has been a matter of great research and many theories.

This field is still misunderstood, misapplied, and misused that results in costly over design of new systems or unreliable operation of existing installations of pumps.

3.12.3.1 Net Positive Suction Head (NPSH) – required

The Hydraulic Institute defines NPSH-r of a pump as the NPSH that causes the total head (first stage head of multistage pumps) to be reduced by 3% due to flow blockage from cavitation vapor in the impeller vanes.

The above term is a practical method of exactly determining the point of minimum suction head for a pump.

The rated pump head is not achieved when the NPSH-a equals the NPSH-r of the pump. The head will be 3% less than the fully developed head value as shown in Figure 3.14.

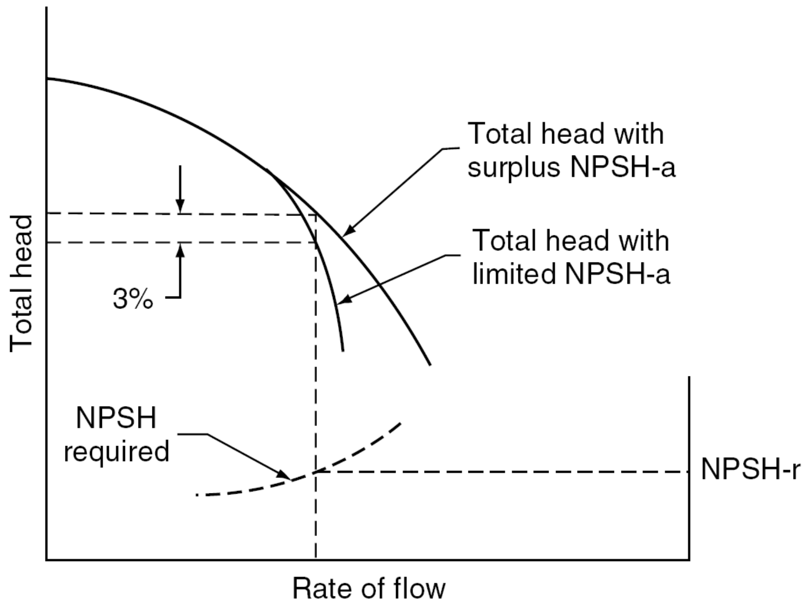


Figure 3.14 – When the Margin of NPSH-a and NPSH-r Lowers, the Differential Head Drops

Strange it may sound but NPSH-r by the above definition does not necessarily imply that this is the point at which cavitation starts; that level is referred to as incipient cavitation.

The NPSH at incipient cavitation can be from 2 to 20 times the 3% NPSH-r value, depending on pump design. The higher ratios are normally associated with high-suction energy pumps or pumps with large impeller inlet areas.

The suction energy level of a pump increases with:

- The casing suction nozzle size
- The pump speed
- The suction-specific speed
- Specific gravity of the pumped liquid.

Anything that increases the velocity in the pump impeller eye, the rate of flow of the pump, or the specific gravity, increases the suction energy of the pump (Figure 3.15).

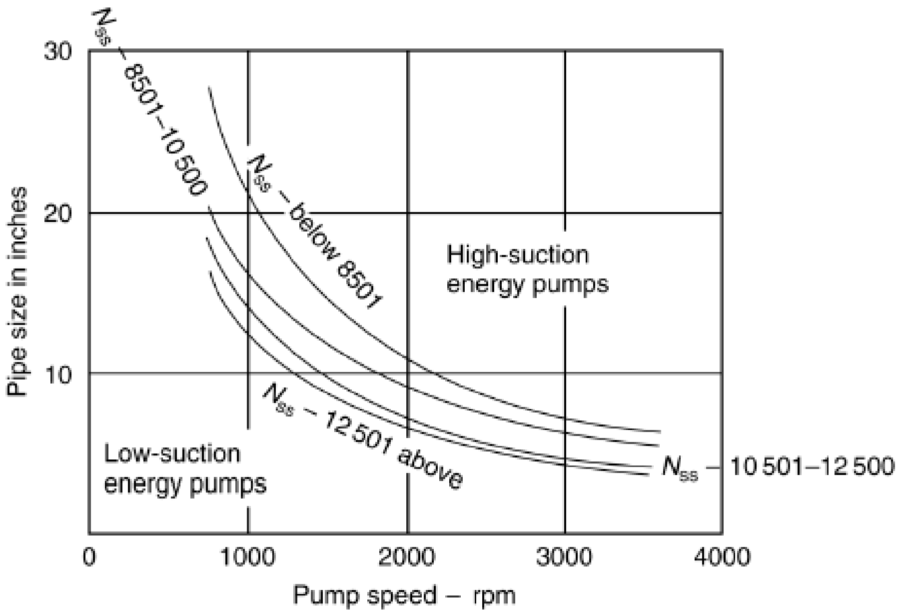


Figure 3.15 – Flow Rate Chart

Most standard low-suction energy pumps can operate with little or no margin above the NPSH-r value, without seriously affecting the service life of the pump.

Thus, we see that NPSH-r as per Hydraulic Institute's definition of 3% head drop is not an indicator of inception of cavitation and consequent pump damage.

It then becomes necessary to come up with a theoretical NPSH-r that can indicate the inception of cavitation that can then ensure a cavitation-free operation.

The theoretical derivation of NPSH-r or 'Cavitation-Free NPSH' is based on factors such as:

- Head loss due to friction
- Head drop due to fluid acceleration
- Head loss due to improper fluid entry into the impeller blade.

As the liquid in the suction pipe approaches the impeller eye, it has velocity and acceleration. In addition, it has to change its direction to enter the impeller. Losses in terms of liquid head occur due to each of the above and because of friction.

The pump inlet nozzle and impeller inlet vane geometry are designed to minimize the losses largely but cannot be eliminated.

Other factors like higher flow rates and recirculation due to higher clearance at wear rings and use of smaller diameter impellers in volutes can increase the losses.

The summation of the above losses is termed as net positive suction head as required by the pump or NPSH-r. In other words, NPSH-r is the summation of losses in the critical area between the suction nozzle and the leading edge of the first stage impeller blades.

Mathematically, the NPSH-r is expressed in the following equation:

$$NPSH - r = \frac{K_1 \times C_{M1}}{2g} + \frac{K_2 \times W^2}{2g}$$

The first term represents the friction and acceleration losses and the second term represents the blade entry losses.

To understand this equation, we need to learn the inlet flow velocity triangle of the pump impeller.

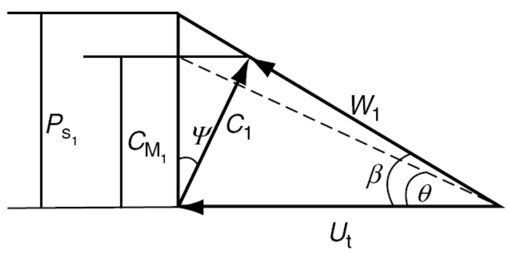


Figure 3.16 – Impeller Inlet Velocity Triangle

C_{M1} = Average meridional (plane passing through shaft axis) velocity at blade inlet

= $\dfrac{Q\,m3/gec}{A_{m^2}}(m/s)$

U_t = Peripheral blade velocity (m/s)

= $(\pi \times D \times N)/60$

D = Impeller eye diameter in meters

N = Speed in rpm

W_1 = Relative velocity (m/s)

C_1 = Absolute velocity of flow (m/s)

P_{S_1} = C_{M_1}/R (R_1 is a factor used to determine the vane outlet angle)

θ = Angle of flow approaching the impeller

= Blade angle at outer radius of impeller eye

ψ = Pre-rotation angle (usually not more than 30°)

α = Incidence angle = (β – θ)

The angle of the inlet edge of the impeller vane, at the point where the vane joins the front shroud, measured in a plane tangent to the shroud surface, is β.

U_t is the peripheral velocity of that same point.

W_1 is the relative velocity (relative to the impeller) of the liquid just before entering the vanes.

C_{M1} is the meridional velocity of the liquid just before entering the vanes. The meridional velocity C_{M1} is the velocity relative to the casing. It lies in the meridional plane (the plane that passes through the shaft centerline).

(If you were standing in the suction nozzle of the casing, facing the impeller, C_{M1} velocity would hit you squarely in the back.) The capacity is such that these three vectors create a right triangle.

If there is 'no pre-rotation' or 'no pre-swirl' or 'shock less-entry', then the included angle θ = β.

Going back to the NPSH-required equation, the first term of '$K_1 C_{M1}^2/2g$' represents the friction and acceleration losses and the second term '$K_2 C_{M1}^2/2g$' represents blade entry losses.

In low-suction energy pumps, the first term is the prime factor and in the high-suction energy pumps the latter term gains prominence.

The constant K_1 is influenced largely by the pump suction nozzle approaching the impeller eye.

The angle of incidence α that influences K_2 is the difference between the inlet angle β and the flow angle θ. The angle β is determined from C_{M1} multiplied by the factor R_1 that allows for the effects of recirculated flow Q_L and non-uniform velocity distribution.

The flow Q_L may vary depending on the wear that has occurred causing additional leakages from wearing ring clearances and balance lines of a multistage pump. In lower specific speed pumps, the percentage of leakage flow is a higher percentage of the total flow and hence there is a greater impact on the NPSH-r of such pumps.

All the discussions on cavitation and recirculation were based on phenomena that lead to vapor formation and implosion.

The boiling of the liquid or vapor formation during cavitation or recirculation is a thermal process and is dependent on the properties of the liquid. These properties include pressure, temperature, specific and latent heats of vaporization. If liquid has to boil, the latent heat of vaporization has to be derived from the liquid flow. The extent of cavitation depends on the proportion of vapor released, the rapidity of liberation, and the vapor specific volume.

This is accounted by a gas to liquid ratio factor C_b. In a paper by D.J. Vlaming, 'A Method of Estimating the Net Positive Suction Head Required by Centrifugal Pumps', ASME 81-WA/FE-32 has provided values of C_b that indicate that cold water has the potential for causing most damage by way of cavitation.

Taking all these into consideration, it is found that cold water is the most damaging of the commonly pumped liquids. Similarly, this difference applies to water at different temperatures.

A review of the properties of water and its vapor at several temperatures shows that the specific volume of vapor decreases rapidly as pressure and temperature increase. Hence, due to this, cold water is again more damaging than hot water. Even field experience tends to corroborate the above study. Pumps handling certain hydrocarbons or hot water operate satisfactorily with lower NPSH-a than does cold water.

Thus, pumps tested with cold water to detect NPSH-r are found to be good enough for the above-mentioned services.

A considerable research material is available on this topic and Terry L. Henshaw in his paper; 'Predicting NPSH for Centrifugal Pumps' has compiled the works of ma y people on this subject.

Their research findings are listed below. It makes an interesting study into this complex number called as the NPSH-r:

- The current industry standard to define the NPSH requirement of a centrifugal pump as NPSH available with cool water, which creates cavitation in the eye of the impeller sufficient to cause the (one stage) head of the pump to drop 3%.
- Because of the 3% head-drop definition, a pump with a larger-diameter impeller will require less NPSH than the same pump with a smaller-diameter impeller.
- The amount of NPSH required to achieve 100% head is typically 1.05–2.5 times the NPSH-r for the 3% head drop.
- The amount of NPSH required to suppress all cavitation is typically 4–5 times the NPSH-r for the 3% head drop, although this ratio can vary from 2 to 20.
- 'Incipient' cavitation causes minimal damage to the impeller.
- The peak cavitation erosion rate occurs at an NPSH-a value above that of the 3% NPSH-r and below that coincident with incipient cavitation.
- Cool water among the many liquids is the most damaging to a cavitating pump.
- • For cool-water services, the 3% head-drop NPSH is not sufficient to prevent cavitation erosion to the impeller.
- For most pumps, at best efficiency flow rate (BEP), the NPSH-r, based on the 3% head drop, does not vary with speed to the exponent of 2. The exponent is more typically 1.5. Therefore, suction-specific speed at $\mathrm{BEP}, N_{SS\text{-}BEP}$ increases as speed increases.
- The shape of the NPSH-r curve varies with the percentage head drop. The $\mathrm{NPSH}-r_3$ decreases as flow rate decreases, reaching a minimum value, normally at or below 40% of the BEP. The NPSH-r curves for 1 and 0% head drop increase as the flow rate decreases below BEP.
- Field experience has revealed that an increase in pump failure rates occur when suction-specific speeds (calculated at BEP and in US units) exceed around 10 000, with a pronounced increase at 11 000.
- The Hydraulic Institute Standards uses N_{ss} value of 8500 (US) as the basis for their maximum speed recommendations.
- At no-prerotation flow rate, the NPSH-a for incipient cavitation is equal to the NPSH-a for the 3% head drop plus 'peripheral velocity head' ($U_1{}^2/2g$).
- To reduce the $\mathrm{NPSH}-r_3$ at BEP, impellers are typically designed, at the BEP flow rate, such that $P_{s_1} > C_{M_1}$ (refer Figure 3.16). The $P_{s_1} > C_{M_1}$ ratio is typically about 1.25. Therefore, the flow rate coincident with no prerotation is typically about 25% larger than the BEP flow rate.

3.12.3.2 Net Positive Suction Head (NPSH) – available

Every pump has an associated inlet system comprising of vessel, pipes, valves, strainers, and other fittings. The liquid, which has a certain suction pressure, experiences losses as it travels through the inlet system.

Thus, the net inlet pressure (in absolute terms) of the pipe and fitting losses is what is available at pump inlet and this is called as the Net Positive Suction Head – available or NPSH-a.

Now knowing what is NPSH-a and NPSH-r, it becomes clear that their difference has to be greater than the vapor pressure of liquid at that temperature to avoid vaporization of liquid.

As a convention, a mathematical simplification is done. The vapor pressure of the liquid is subtracted from the NPSH-a. In pump terminology, NPSH-a includes vapor pressure correction.

Thus, all we need to insure is that the NPSH-a is greater than NPSH-r. Their difference in such a case would then be in the true sense of the word Net Positive Suction Head. Some common examples of NPSH calculations are provided below.

Calculating NPSH-Available: Pressurized Flooded Suction

$$Vapor\ pressure - P_{vap} = 0.45\,kg/cm^2$$

$$Pipe\,losses - H_f = 1.5\,m$$

$$Specific\,gravity = 0.8$$

$$P_g - gage\ pr. = 0.5\,kg/cm^2$$

$$H_g\ in\,meters\ `m' = 0.5 \times 10.2\ /\ 0.8 = 6.4\,m$$

$$H_{st}\ in\,meters\ `m' = +\ 0.2\,m$$

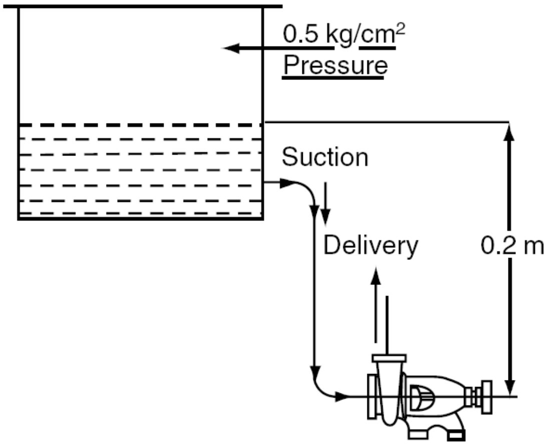
$$H_f = 1.5\,m$$

$$H_a - atmospheric\ pressure = 10.325\ /\ 0.8 = 12.9\,m$$

$$H_{vap}\ in\,meters\ `m' = 0.45 \times 10\ /\ 0.8 = 5.7\,m$$

$$H_a + H_s + H_{st} - H_f - H_{vap}$$

$$NPSH - a\ in\,meters\ `m' = 12.9\ + 6.4\ + 0.2\ - 1.5\ - 5.7\ = 12.3\ m$$



Calculating NPSH-Available: Atmospheric Flooded Suction

$$\text{Vapor pressure} - P_{vap} = 0.45\, kg/cm^2$$

$$\text{Pipe losses} - H_f = 1.5\, m;$$

$$\text{Specific gravity} = 0.8$$

$$P_g - \text{gage pr.} = 0\, kg/cm^2$$

$$H_g \text{ in meters ‘}m\text{’} = 0 \times 10.2 / 0.8 = 0\, m$$

$$H_{st} \text{ in meters ‘}m\text{’} = +4\, m$$

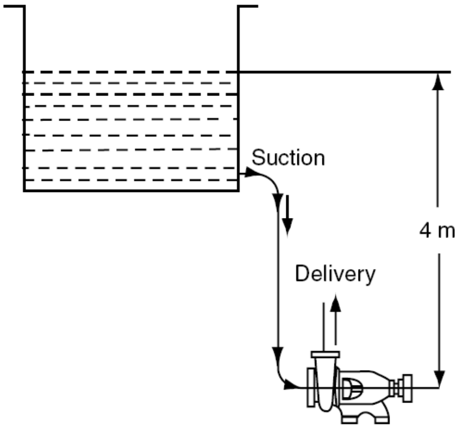
$$H_a - \text{atmospheric pressure} = 10.325 / 0.8 = 12.9\, m$$

$$H_f \text{ in meters ‘}m\text{’} = 1.5\, m$$

$$H_{vap} \text{ in meters ‘}m\text{’} = 0.45 \times 10 / 0.8 = 5.7\, m$$

$$H_a + H_g + H_{st} - H_f - H_{vap}$$

$$NPSH - a \text{ in meters ‘}m\text{’} = 12.9 + 0 + 4 - 1.5 - 5.7 = 9.7\, m$$


Suction
4 m
Delivery

3.12.3.3 Calculating NPSH-Available: Vacuum Flooded Suction

$Vapor\ pressure - P_{vap} \quad = 0.45\, kg / cm^2$

$Pipe\,losses - H_f \quad = 1.5\, m;$

$Specific\,gravity \quad = 0.9$

$P_g - gage\ pressure \quad = 600 mm - Hg$

$H_g\ in\ meters\ 'm' \quad = (600/1000) \times 13.6\ /\ 0.9 = 9.1 m$

$H_{st}\ in\ meters\ 'm' \quad = +10.2 m$

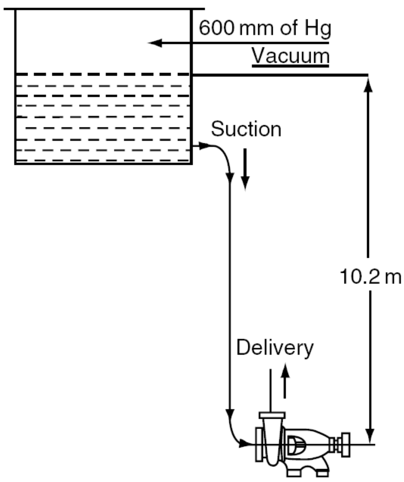
$H_a - atmospheric\ pressure \quad = 10.33 \times 10 / 0.9 = 11.5 m$

$H_f\ in\ meters\ 'm' \quad = 1.5 m$

$H_{vap} \quad = 0.45 \times 10\ /\ 0.9 = 5.1\ m$

$$H_a + H_g + H_{st} - H_f - H_{vap}$$

$NPSH - a\ in\ meters\ 'm' \quad = 11.5 + 9.1 + 10.2 - 1.5 - 5.1 = 6.0 m$



3.12.3.4 Calculating NPSH-Available: Negative Suction

$Vapor\ pressure - P_{vap} \quad = 0.45\,kg/cm^2$

$Pipe\,losses - H_f \quad = 1.5\,m;$

$Specific\,gravity \quad = 0.8$

$P_g - gage\ pressure \quad = 0\,kg/cm^2$

$H_g\ in\,meters\ 'm' \quad = 0 \times 10.2\ /0.8 = 0\,m$

$H_a - atmospheric\ pressure \quad = 10.325\ /0.8 = 12.9\,m$

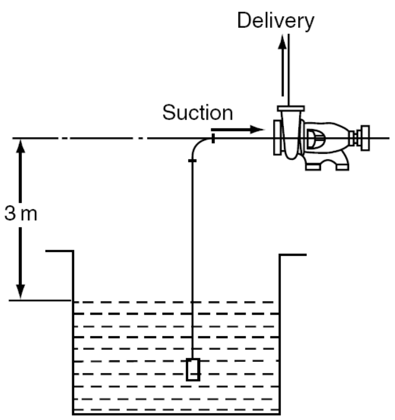
$H_{st}\ in\,meters\ 'm' \quad = -3\,m$

$H_f\ in\,meters\ 'm' \quad = 1.5\,m$

$H_{vap}\ in\,meters\ 'm' \quad = 0.45 \times 10.2\ /\ 0.8 = 5.7\ m$

$$H_a + H_g + H_{st} - H_f - H_{vap}$$

$NPSH - a\,in\,meters\ 'm' \quad = 12.9 + 0 + (-3) - 1.5 - 5.1 = 2.7\,m\,(Satiafactory)$



3.12.3.5 Net Positive Suction Head (NPSH) – Margin

The simple approach considered in regard to NPSH margin is the net between the available and required NPSH.

It is a requirement that the NPSH-a available must be equal to or greater than the NPSH-r stipulated by the pump manufacturer. Most pump specifications quote a margin of not less than 1 to 1.5 m over the entire range of pump operation.

When the difference between NPSH-a and NPSH-r is less than the stated margin, it calls for an exact determination of the NPSH-r by carrying out the NPSH-r test.

Another approach adopted to define the margin is by taking the ratio of NPSH-a and NPSH-r.

The table given below offers suggested minimum NPSH margin ratio guidelines (NPSH-a/NPSH-r), within the allowable operating region of the pump (with standard materials of construction). It is based on the experience of the many pump manufacturers with many different pump applications.

Minimum NPSH Margin Ratio Guidelines (NPSH-a/NPSH-r)			
	Suction Energy Levels		
Application	Low	Medium	High
Petroleum	1.1-a	1.3-c	
Chemical	1.1-a	1.3-c	
Electrical Power	1.1-a	1.5-c	2.0-c
Nuclear Power	1.5-b	2.0-c	2.5-c
Cooling Towers	1.3-b	1.5-c	2.0-c
Water/Waste water	1.1-a	1.3-c	2.0-c
General Industry	1.1-a	1.2-b	
Pulp and Paper	1.1-a	1.3-c	
Building Services	1.1-a	1.3-c	
Slurry	1.1-a		
Pipeline	1.3-b	1.7-c	2.0-c
Water Flood	1.2-b	1.5-c	2.0-c
'a' – Or 0.6 m (2 ft) whichever is greater 'b' – Or 0.9 m (3 ft) whichever is greater 'c' – Or 1.5 m (5 ft) whichever is greater.			

Table 3.1 –

Vertical turbine (misnomer) pumps often operate without NPSH margin without damage, but with slightly reduced discharge head. Such pumps generally have lowsuction energy, and cavitation noise is normally not an issue. NPSH-a has to be equal to or larger than the NPSH-r over the allowable operating region of the pump, including a low water level.

High and very high suction energy pumps operating with the margins specified in the table will have acceptable bearing and seal life. However, they may still be susceptible to impeller erosion and higher noise levels.

In addition to the margins specified in the table additional requirements of suction head arise due to:

- Increase in wearing ring clearances due to wear. This increases the leakage flow to the impeller eye and disturbs the inlet flow pattern
- Gas content in the liquid
- Improperly designed inlet piping and pump casing that cause non-uniform suction flow or turbulence
- Operation of the pump on the farther right hand side of BEP. In this region, the NPSH-a reduces and NPSH-r increases.

3.13 Suction-Specific Speed

Suction-specific speed is defined by the equation:

$$N_{ss} = \frac{N\sqrt{Q}}{(\text{NPSH - r})^{3/4}}$$

Where:

N	=	pump speed
Q	=	capacity at BEP at maximum impeller diameter (it gets halved for a double suction impeller)
NPSH-r	=	Net Positive Suction Head (required) at BEP at maximum impeller diameter

Studies carried out have empirically established that pump models with Nss less than 11 000 (US units: Q – US gpm, N – rpm, NPSH-r – feet) have a more stable operation and are more reliable.

Therefore, it is commonly used as a basis for estimating the safe operating range of capacity for a pump. The higher the Nss is, the narrower is its safe operating range from its BEP. Most users prefer that their pumps have Nss in the range of 8000–11 000 for optimum and trouble-free operation.

It is usually recommended that such pumps (Nss > 11 000) should not be operated at flow rates below 60–70% of the BEP.

When the pump is operated below this range, it may experience:

- Impeller and casing erosion
- Shaft deflection and stress
- Radial and thrust bearing failures
- Seal problems.

The above are attributed to the recirculation of liquid at the impeller inlet, which has been covered in the earlier section.

For a smooth operation, the liquid enters the impeller at a particular designed angle. This inlet angle is meant for flows at the BEP, however, at lower flows, the liquid enters the impeller at a much different angle and is unable to make an entry into the impeller.

As a result, it is forced back into the pump suction pipe. The liquid keeps recirculating in front of the impeller.

Evidence of recirculation at impeller inlet is:

- Suction pressure gage fluctuations
- Noisy operation
- High vibrations at low flow rates.

If a higher Nss pump model is thus encountered, users prefer to buy a lower speed pump even though it may cost more.

3.14 3.14 Performance Calculation Procedure

For a centrifugal pump, the performance calculation's aim is to determine the pump efficiency. This value can be read on the characteristic curves provided by the manufacturer. The deviation of the calculated efficiency from the rated efficiency indicates the performance degradation of the pump.

3.14.1 Flow Measurement

Flow measurement can be taken from a flow measuring device, if fitted. In cases where flow measurement devices are not installed, non-invasive ultrasonic flow meters can be used to measure the flow from the pump.

The flows are usually indicated as mass flow in kg/h. It is recommended to convert this to volume flow in m□/h.

M = Mass flow (kg/h)

Q = Volumetric flow (m□/h)

δ = Density at pumping temperature (kg/m□)

$$Q = \frac{M}{\delta}$$

3.14.2 Differential head

h_s = Suction head (m)

P_s = Suction pressure (kg/cm□)

ρ = Specific gravity at pumping temperature

$$h_s = \frac{10 \times P_s}{\rho}$$

Discharge head h_d in m is calculated in a similar manner.

Differential head – h in m is further calculated as:

$$H = h_d - h_S$$

3.14.3 Hydraulic power

The next step in this process is to calculate the hydraulic power. This is calculated in the following manner.

g = gravitational acceleration – 9.81 m/s□.

$$P_{H(kW)} = \frac{Q \times \delta \times g \times g}{3.6 \times 10^6}$$

The hydraulic power is minimum power required to pump the fluid. This will be the power if the pump had an efficiency of 100%. However, this is not possible in practice. To obtain the actual pump efficiency we need to go to the next step of calculating the energy being provided to the pump by the prime mover. Let us assume that in this case the prime mover is an electrical motor.

3.14.4 Motor power

The electrical power is fed to the motor to its terminals. However, we are interested in the power that is delivered by the motor at its coupling with the pump. Thus, we need to consider the efficiency of the motor too.

Efficiency of the motor is not a fixed number but changes with the load on the motor. A part load efficiency of the motor is much lower than its efficiency at full load. This table of motor efficiency with respect to its load is provided by the motor manufacturer.

V = Measured voltage in volts
I = Measured current in ampere
$\cos\varphi$ = Measured power factor
η_e = Motor efficiency

Motor Power, PM in kW at its coupling is:

$$P_M = \frac{\sqrt{3} \times V \times I \times \cos\phi \times \eta_e}{1000}$$

3.14.5 Pump efficiency

Having performed the above calculations, we are now in a position to derive the pump efficiency. The ratio of pump hydraulic power to the motor power gives the pump efficiency.

η_p = Pumping efficiency (hydraulic efficiency)

$$\eta_P + \frac{P_H}{P_M}$$