

# **CHAPTER FOUR**

## **CENTRIFUGAL**

### **PUMP OPERATION**

#### **AND**

### **CHARACTERISTICS**

## **Centrifugal Pump Operation and Characteristics**

One of the most attractive features of a centrifugal pump is its ability to perform in a system under a wide range of operating conditions.

Pumps are able to operate with satisfactory under varying operating conditions and with a range of fluid types. These could be variations in:

- Flow rate
- Suction and discharge heads
- Speed
- Type of fluids handled and its many properties
- Fluids with varying properties.

The pump is designed for a discrete value of flow rate, differential head, and speed. This is the best efficiency point of operation or 'BEP'. However, in practical applications, the pumps are rarely operated at the operating parameters for which the pump has been designed.

Among the parameters stated, the flow rate  $Q$  and differential head  $H$  of the pump vary a great deal during normal operation.

Consider this case where a pump discharges into a delivery pipe that is connected to the bottom of a vessel situated at a certain height. As liquid is discharged into the vessel, the height of the liquid in the vessel increases and this increases the differential head that the pump has to generate. As the differential head  $H$  increases, the flow rate of the pump  $Q$  decreases.

If in the above case, the power requirement increases then the load on the motor could result in a possible drop in speed of the prime mover.

Other cases include where process and utility requirements to cover a greater range of flow rate and differential head may demand a simultaneous operation of multiple pumps within a single system. The performance of one pump in a common system is certainly affected by the behavior of the other pumps.

Even though a pump may perform with wide variations in the parameters and conditions, the pump performance is not unaffected by these changes.

It is therefore important to determine the behavior of the pump as it responds to the variation in the parameters under operating conditions that are different from parameters that were considered during the design of the pump.

The basics of the pump hydraulics have been covered in chapter 3 and this chapter would cover the behavior of the pump in response to the changes in its environment.

We may revisit some of the fundamentals covered in the earlier chapters to understand the variation in the behavior of the pump.

## **PRACTICAL CENTRIFUGAL PUMPS**

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### **5.1 Behavior of Hydraulic Properties of Pumps**

The hydraulic properties of any centrifugal pump are studied taking the shaft speed of the pump  $N$  as a constant.

After the speed is considered constant, the behavior of the differential head  $H$  with respect to the flow rate  $Q$  is obtained by throttling the discharge valve of the pump. The various openings of the discharge valve result in different flow rates and corresponding heads.

This experiment provides the relationship of  $Q$  with  $H$  that can be represented as Differential Head:

$$H = f(Q)$$

This is the fundamental characteristic of any centrifugal pump.

During this experiment, simultaneous readings of Power are noted and the efficiency values are also computed.

Even these can be represented as Hydraulic power:

$$P = f(Q)$$

Pump efficiency:

$$\eta = f(Q)$$

The curves generated from the above functions are called 'performance curves' or the centrifugal pump characteristic curves.

The fourth characteristic representing the function of NPSH with respect to the flow rate  $Q$  is a supplementary characteristic.

In the first case, the values of the flow rate  $Q$  is plotted along the X-axis (abscissae). The units that maybe used are  $m^3/h$ ,  $m^3/s$ ,  $l/s$ , and  $l/min$ , US-gpm, Imp-gpm.

The values of differential head  $H$  (meters, feet), Power  $P$  (kW, HP), and Efficiency (as percentage, decimal fractions) are plotted on the Y-axis (coordinates)

The performance of most centrifugal pumps is given in terms of its capacity, discharge head, efficiency, and input power. Because these quantities are directly interdependent, a series of curves are used to express pump performance. Pump manufacturers provide the design characteristic curves for their individual pumps. These curves are then used as a basis for pump acceptance tests.

## PRACTICAL CENTRIFUGAL PUMPS

In Figure 5.1, the above-mentioned curves for some representative pumps are shown.

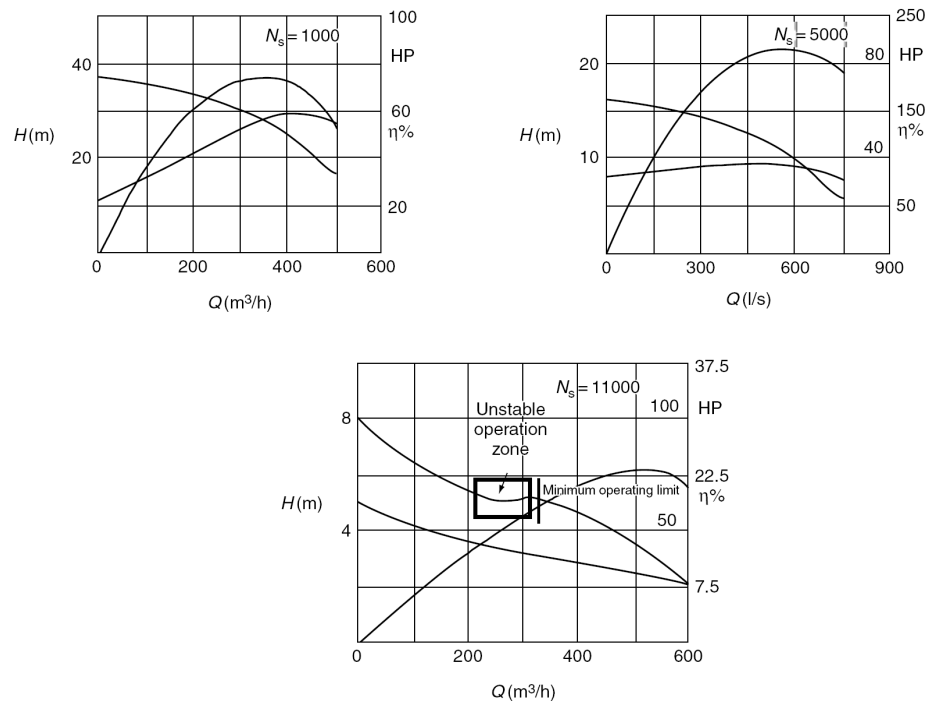


Figure 5.1 – Pump Design Characteristic Curves

As the three graphs indicate, the specific speeds of the pump have quite an impact on the nature of these characteristic curves. The shape of the curves for each characteristic may appear different but the trends are similar. We shall discuss these after having a look at the curves.

## PRACTICAL CENTRIFUGAL PUMPS

### 5.1.1 Head–Flow Characteristics

One of the most important characteristics of a pump is its capacity; that is, the amount of fluid it moves per unit time. The capacity of a centrifugal pump decreases as the pressure at the pump discharge increases.

When the discharge valve is completely closed, the head developed by the pump is called the shut-off head. At this point, the pump will obviously not deliver any liquid. This is also the maximum head any pump can develop. The shut-off head is represented as  $H_{so}$ .

Now, if the discharge valve is opened completely and the flow is directed into the atmosphere, the pump discharges at maximum capacity.

The shape of the H–Q curve as seen in the graphs in Figure 5.1 is dependent on the specific speed or the shape of the blades of the impeller.

The other factors that have a bearing on the shape of this curve are:

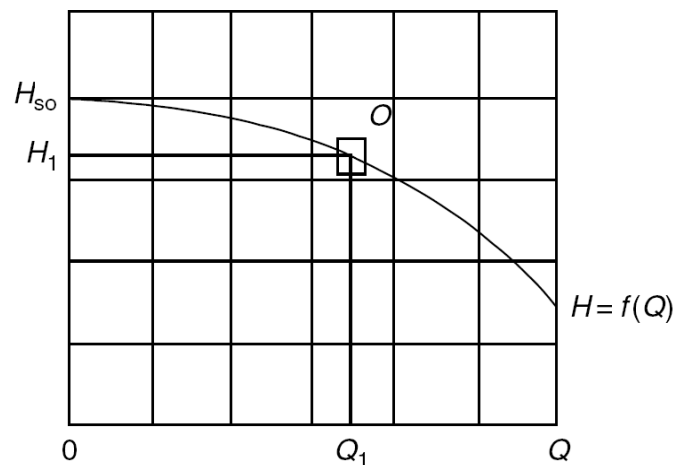
- Number of blades or impeller vanes
- The type of the pump casing (volute casing, diffuser ring, vane less guide ring, or concentric casing).

The H–Q curves of any pump are of two types:

1. Stable H–Q curve
2. 2. Unstable H–Q curve.

### 5.1.2 Stable Head–Flow Characteristics

Stable H–Q curves is one in which the differential head  $H$  progressively falls with the increase in flow rate  $Q$ . This is shown in Figure 5.2.



## **PRACTICAL CENTRIFUGAL PUMPS**

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Figure 5.2 – Stable H-Q Curves

## PRACTICAL CENTRIFUGAL PUMPS

### 5.1.3 Unstable Head–Flow Characteristics

In this type of curve, the  $H$  generated by the curve starting from  $H_{so}$  rises to a maximum value at point A and then falls progressively.

The drop in the curve indicates that for a single system resistance requirement of  $H_1$ , there can be two possible flow rates  $Q_1$  and  $Q_2$ . This is not a desirable characteristic of a pump.

In actual operation, the pump with unstable characteristics and head requirement, that is greater than  $H_{so}$ , has a pump flow that oscillates between  $Q_1$  and  $Q_2$ , leading to a modulating flow rate and possible vibrations in the pipeline (Figure 5.3).

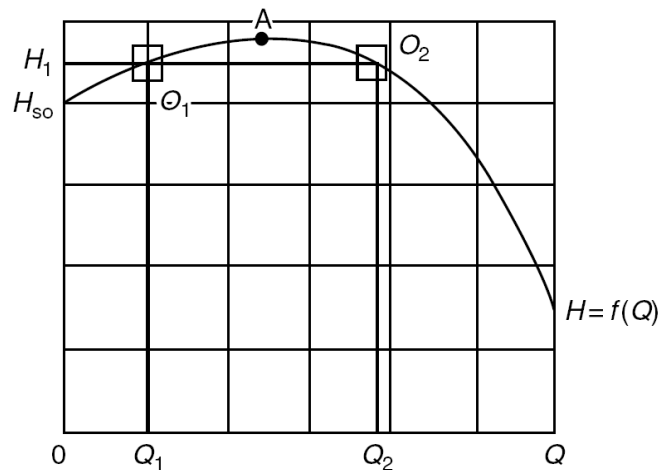


Figure 5.3 – Unstable H–Q Curve

In single-stage centrifugal pumps, the stability of the curve can be obtained by:

- Reducing the number of vanes/blades
- Modification of the vane geometry
- Reducing the vane outlet angle
- Modifications of the diffuser shape and return vanes in a multistage pump.

API 610, the design standard for centrifugal pumps recommends that all pumps shall have stable H–Q curves, which implies that the head, at rated capacity should continuously rise until it reaches the shut-off head.

In case of parallel operation of pumps, the head rise from the rated capacity to shut-off as in the case of single-/double-stage pumps shall be 10–20%. In multistage pumps, a lower percentage is recommended.

## PRACTICAL CENTRIFUGAL PUMPS

### 5.1.4 Steepness of the Head–Flow Characteristics

The Steepness of the H–Q curve is usually associated with a stable H – Q curve. It is defined as percentage fall in head from shut-off conditions to BEP of the pump (Figure 5.4).

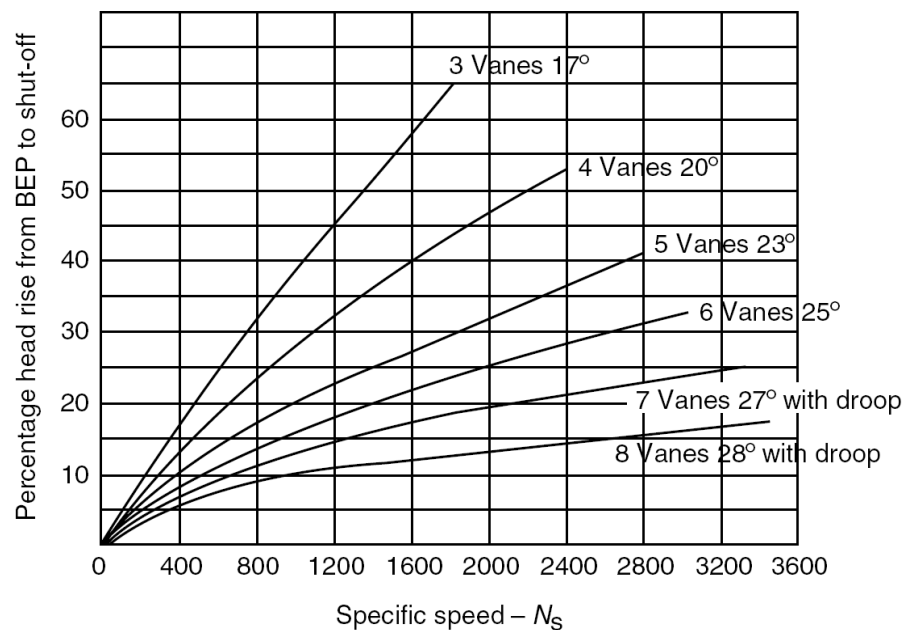


Figure 5.4 – Steepness of the H–Q Curve

This is represented by the formula given below:

$$\text{Steepness Percentage} = \frac{H_{SO} - H_{BEP}}{H_{BEP}} \times 100$$

The number of vanes and blade outlet angle determine the steepness of the H–Q curve. The relationship of the above factors with the specific speed of the pump is shown in Figure 5.4. The figure represents some of the practically used combinations of vane number and outlet blade angle in the industry.

Another way of expressing steepness of the H–Q curve is by using the ratio of the change in differential head H to the change in flow rate Q,  $\Delta H/\Delta Q$ .

## **PRACTICAL CENTRIFUGAL PUMPS**

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In H-Q curves, where  $\Delta H/\Delta Q$  is smaller, the curves are said to have flat characteristics and while those with a higher value of  $\Delta H/\Delta Q$  are called as steep characteristics.

In most cases, the application of the pump determines whether the curve should be of the flat or steep type.

For example: pumps used in the firewater service need to have very flat characteristic curves.

Consider this scenario, where a pump is feeding water to a header system that is being used to extinguish fires. One or many fire hydrants maybe opened simultaneously to fight the fire and the required flow rate may increase sharply. As the flow from the pump increases it is important that the pressure from the pump should not drop substantially. A drop in head delivered may affect the height at which the water has to be sprayed.

It is observed that as the specific speed of the pump goes on increasing the H-Q curves keep getting steeper.

In addition, in high specific speed pumps, there is a relatively steep slope toward shutoff and beyond BEP.

The procedure for selecting an axial flow pump is much the same as for any other process pump; however, the duty point is limited to a range within 75–115% of the pump's BEP.

This is more critical with the axial flow designs than with radial vane designs because the hydraulics become unstable at 70–75% of BEP (shown in the Figure 5.4).

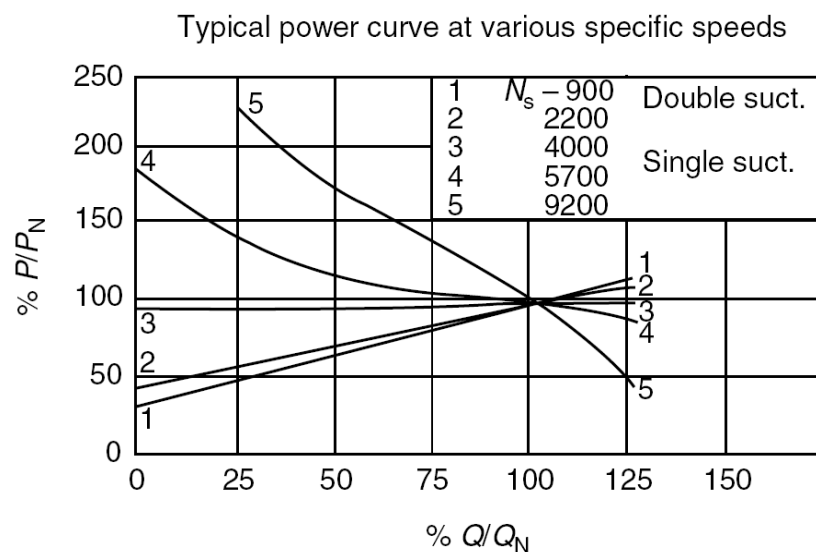
This phenomenon may also be seen in mixed-flow impeller pumps. As a result, this flow from such pumps is rarely throttled. They are also not started with a closed discharge valve. It will be seen in the subsequent section that the power curves too favor the starting of such pumps with an open discharge valve.

## PRACTICAL CENTRIFUGAL PUMPS

### 5.1.5 Power–Flow Characteristics

The nature/shape of power–flow rate characteristic curves are also dependent on the specific speed of the pump.

Centrifugal pumps having low to medium specific speeds have P–Q curves that rise upward. For higher specific speeds the P–Q curves maybe approximately flat and horizontal (Figure 5.5). In case of propeller (axial flow) pumps, which have very high specific speeds, the power for the pump falls as the flow rate increases.



$Q_N$  and  $P_N$  are flow rate and power at normal operating point

Figure 5.5 – The P–Q curves as a function of specific speed of pumps

The pumps that are driven by AC induction motors should be started with minimum load on them so as to limit the starting current. The above characteristics indicate as to how the pumps should be started so the above condition is fulfilled.

Therefore, it is recommended that radial impeller pumps be started with discharge valves closed whereas propeller pumps be started with discharge valves fully open.

The pump P–Q curves are of two types:

1. Non-overloading
2. Overloading

These terms are derived from the fact that the demand of power by the pump increases or decreases with the increase in flow rate.

## **PRACTICAL CENTRIFUGAL PUMPS**

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In mixed flow and axial flow pumps, it is observed that the power curve remains flat or tapers downward. Thus, it is rare that overloading of the motor occurs due to increase in flow rate by the pump. These curves are called 'Non-overloading' P-Q curves.

In centrifugal pumps with radial impellers, the power curves rises with an increase in flow rate. Thus, when the system resistance on the pump drops, there is a tendency for the motor to trip on account of motor overload.

The P-Q curves of such pumps are called 'Overloading' P-Q curves.

### **5.1.6 Efficiency–Flow Characteristics**

The efficiency vs power curve  $\eta - Q$  curve initially rises to a peak, which as mentioned earlier is called as the BEP. Subsequently, as the flow rate is increased the efficiency drops.

It is observed that this drop in efficiency occurs after BEP gets sharper with the increase in specific speed.

Thus, a radial impeller curve will show gradual drop in efficiency post BEP whereas it would be quite steep in the case of an axial flow pump.

## PRACTICAL CENTRIFUGAL PUMPS

### 5.2 The Cause of the H-Q Curve

To understand the nature of the H-Q curve of a radial impeller pump, we have to learn the velocity triangles of the liquid flow at the outlet of the impeller.

The three cases of flow rate  $Q$  less than, equal to and greater than  $Q$  at BEP are shown in Figure 5.6. The changes in the corresponding vectors are also indicated.

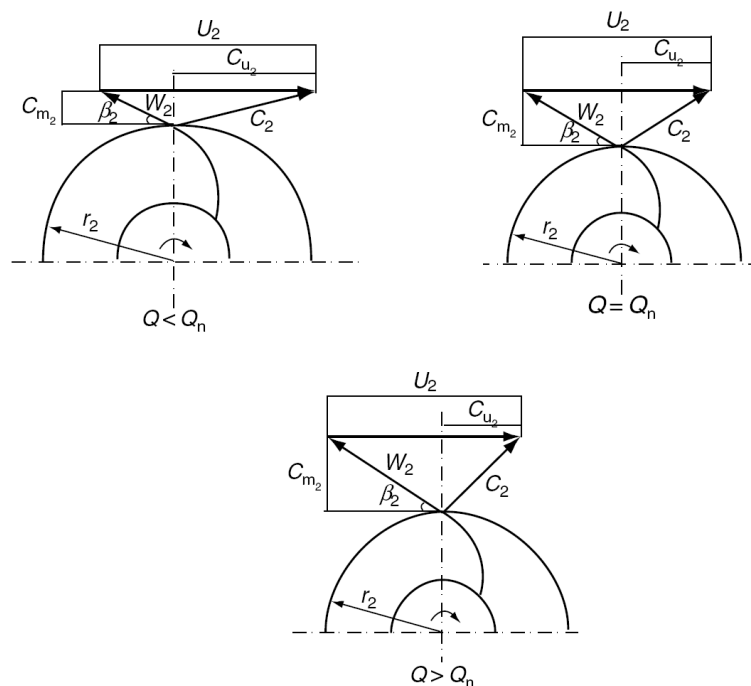


Figure 5.6 – Velocity Triangles at Outlet of Impellers

The main velocities are:

- Peripheral velocity:  $U_2 = \pi \times N \times d_2 / 60$  – As speed is constant,  $U_2$  is constant
- Absolute velocity:  $C_2$
- Relative velocity:  $W_2$  – Direction is along the vane outlet angle
- Radial velocity at impeller discharge:  $C_{m2} = Q \times \text{Area}$

When the flow rate  $Q$  changes, the following do not change:

- Peripheral velocity at the outlet  $U_2$ , as the speed  $N$  is constant
- Blade outlet angle  $\beta_2$ , contained between the direction of the relative velocity  $W_2$  and the vector  $U_2$

## **PRACTICAL CENTRIFUGAL PUMPS**

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At the same time, the relative velocities of flow  $W$  through the impeller passages shall increase or decrease based on variations in the flow rate. As a result, the magnitudes of the vectors forming the velocity triangle change. The shape of the parallelogram of velocities changes and this changes the peripheral component of the absolute velocity  $C_{u2}$ . This produces a change in the theoretical differential head  $H$  given by

$$H = U_2 \times \frac{C_{u2}}{g}$$

From the comparison of the outlet velocity triangles, it follows directly that when the flow rate is reduced the circumferential component of the absolute velocity  $C_{u2}$  increases, and the meridional component  $C_{m2}$  decreases. Increasing the flow rate  $Q$  reduces the component  $C_{u2}$  and thus this reduces the differential head  $H$  as developed by the pump.

In this manner, it is explained that when the flow rate increases the head developed by the pump decreases, hence the reason for the  $H$ - $Q$  curve.

The power absorbed to pump the liquid is directly proportional to the product of flow rate  $Q$  and the head developed  $H$ . The changes in  $Q$  and  $H$  described determine the power of self-regulation of impeller pumps.

This is an essential and valuable feature in the exploitation of impeller pumps, since if the total head of the pump increases during operation; the pump automatically reacts by reducing the discharge so that the impeller can overcome the increased resistance.

Conversely, a reduction in the resistance in the delivery pipe stimulates the pump to increase the discharge.

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### 5.3 The Inlet Velocity Triangle

When the flow rate of a pump changes there are changes in the liquid flow direction even at the inlet of the impeller (Figure 5.7)

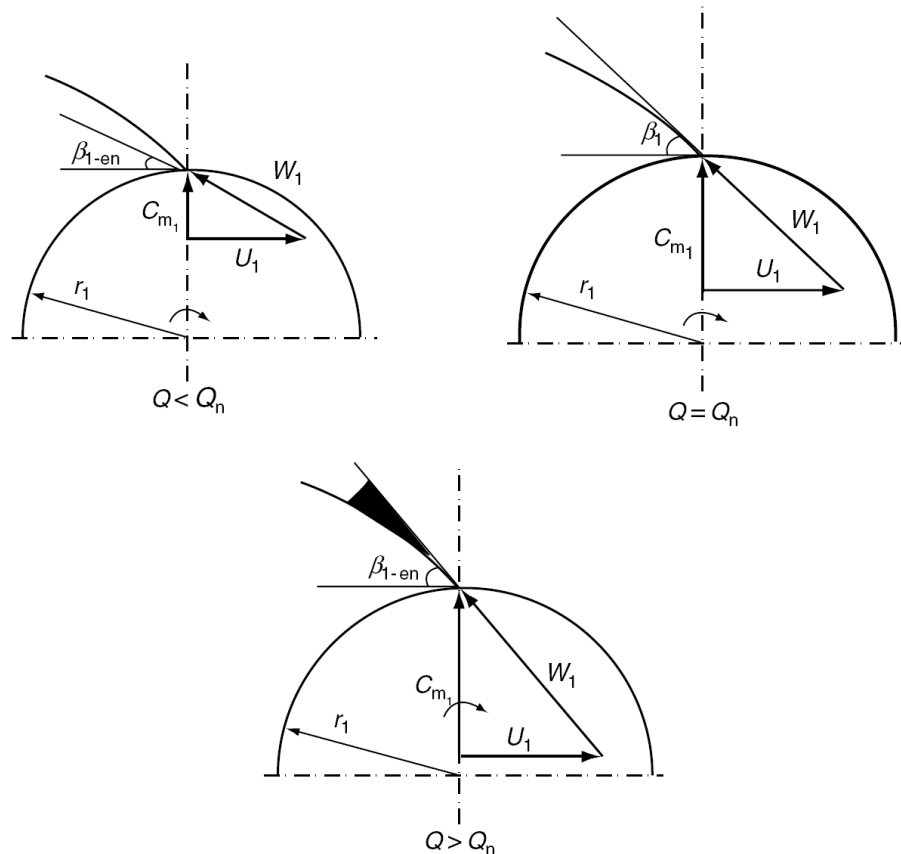


Figure 5.7 – Inlet Velocity Triangle

The three main velocities are:

1. Peripheral velocity:  $U_1 = \pi \times N \times d_1 / 60$  – As speed is constant,  $U_2$
2. Absolute (Meridional) velocity:  $C_{m1} = Q \times \text{Area}$
3. Relative velocity:  $W_1$

When the pump flow rate changes, there is a change in the angle of entry of the liquid,  $\beta_{1-en}$ . It is smaller than the blade inlet angle  $\beta_1$ , when the flow rate is less than the flow at BEP.

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At BEP, the flow is such that the liquid inlet angle is the same as the blade inlet angle  $\beta_1$ .

Similarly, when the flow rate  $Q$  is more than the specified flow rate at BEP, the liquid inlet angle  $\beta_{1-en}$  becomes greater than the blade inlet angle  $\beta_1$ .

During conditions of flow rate less than or greater than BEP flow rate, flow separation occurs and leads to the formation of eddies.

At  $Q < Q_{BEP}$ , eddies are formed on the inner surface of the vane; the converse happens in case of  $Q > Q_{BEP}$ , when eddies are formed on the outer surface of the vane.

In practical cases, a certain pre-whirl is present and this tends to reduce the difference between  $\beta_{1-en}$  and  $\beta_1$  at flow rates less than or greater than flow at BEP. Thus, the losses are less than that assumed by a straight flow into the impeller.

The velocity triangles at inlet provide a good insight to the cause of the losses that occur at flow rates other than that at BEP.

These flow rates other than BEP also cause disturbances and hydraulic losses in the diffuser ring and the volute casing.

It is described that such hydraulic losses are proportional to the cube of the flow rate.

### **5.4 Multiple pump operation**

A process application may require operation of more than one in a single system.

There could be a case where several pumps would discharge their flows in a common delivery pipe. The configuration of such pumps is called as Parallel Operation.

There could also be a case when a single pump cannot meet the total head requirement of the system and it has to be shared by various pumps. The configuration of such pumps is called as Series Operation.

When pumps are operated in parallel, they work against a common pressure. In this case, the flow rates are added. For a pump in series, the heads are added for a common total head.

Multiple pump operations are sensitive to the individual characteristics of the pump.

## PRACTICAL CENTRIFUGAL PUMPS

The total flow rate or the total head developed may not be a simple addition of the individual flow rates and heads developed. This is especially in cases where the pumps do not have similar characteristic curves.

Before we take up the various cases of operation, it is recommended to the reader to revisit the topic on System Resistance covered under Section 3.6.

### 5.4.1 Parallel Operation with Similar Q–H Curves

When several pumps with similar Q–H characteristic curves feed into a common discharge, the combined Q–H characteristic curve is obtained by simply adding up the abscissae corresponding to all pumps.

If two pumps with the same Q–H curve are operated in parallel, the combined curve is obtained by doubling the abscissa for each point on the individual pump characteristic, while keeping the total value of the head as constant.

When a single pump operates, the flow rate is say  $Q_1$ . The combined Q–H curve is obtained by doubling the abscissa for the same head, as shown in Figure 5.8.

#### Figure 5.8 – Parallel operation of two pumps with similar characteristics

It would be normal to assume that this would be the operating point of the pump when these are operated in parallel. However, this is not the case.

The point of operation is dependent on the system resistance.

In a parallel operation with a system resistance curve as shown in the figure, a single pump would deliver a flow rate of  $1 \times Q$ . When the next pump is also started, the system

resistance takes a turn upwards and intersects the Q–H curve of the combined flows at

a point,

$2 \times Q$ . This is due to the square relationship of the system resistance with the flow rate.

## 5.5 The Cause of the P–Q Curve

As described earlier, hydraulic losses occur for every flow rate other than the flow rate at the BEP. As the flow rate moves further from the BEP in either direction the losses increase. These losses as mentioned are proportional to the square of the flow rate.

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### **PRACTICAL CENTRIFUGAL PUMPS**

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The other two losses that contribute to the pump inefficiency are recirculation losses and mechanical losses. None of these losses is grossly affected with the change in flow rate.

Thus, there is a certain value of the flow rate of the pump when sum of the losses are minimum. Thus, efficiency is at a maximum when the losses are minimal. At every other flow rate the efficiency is lower.

This explains the inverted U shape of the P-Q curve of a centrifugal pump.

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### 5.6 The Effect of Speed Changes on Characteristic Curves

All the discussions in the preceding chapters have been based on the prime assumption that the speed of the pump is constant. In many practical applications, pumps operate with variable speed drives and in this chapter, we will see the effect of change in speed on the characteristic curves on pumps.

In Figure 5.8 the Q-H curve-b at the normal speed  $N$  is a typical Q-H curve of a centrifugal pump.

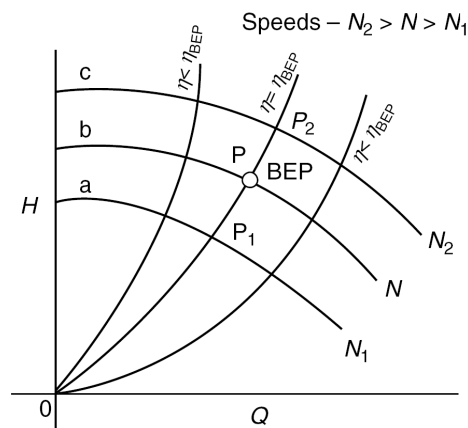


Figure 5.9 – Speed Changes Effects on Characteristic Curves

When the speed of the pump is changed, the flow rate  $Q$  and the head developed  $H$  also changes.

If we consider the velocity triangles at the inlet and outlet, they remain similar and all the velocities vary in proportion to the speed of the pump.

The flow rate is direction proportional to the speed  $N$ :

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

The total heads is proportional to  $u_2^2$ , and hence it is proportional to  $N^2$ :

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

## PRACTICAL CENTRIFUGAL PUMPS

The power  $P$  of a pump is proportional to the product of flow rate  $Q$  and head developed  $H$ . Therefore,  $P$  is proportional to  $N^3$ .

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

The relationships derived above help to establish the characteristics of the pump at different speeds, once the  $Q$ - $H$  curve at the normal speed  $N$  is known.

The relationships derived are based on a major assumption that the efficiency remains constant when transferring a point on one characteristic to the homologous point on another characteristic.

The results obtained by these equations often differ from the values obtained in practical application. The difference becomes larger when the speed change is more than  $\pm 25\%$ .

Let us consider a point  $P_2$  on the  $Q$ - $H$  curve-c in Figure 5.8. The homologous points  $P$  and  $P_1$  on the  $Q$ - $H$  curve-b and  $Q$ - $H$  curve-c are given by

$$Q_1 = \frac{N_1}{N} \times Q \qquad Q_2 = \frac{N_2}{N} \times Q$$

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

Resolving the above equations, we get:

$$H_x = \left( \frac{Q_x}{Q} \right)^2 \times H$$

The above equation is of the form of a parabola passing through the point  $P$  and the origin  $O$ . These are shown in Figure 5.8.

The operating  $P$  corresponds to a certain specific speed given by:

$$N_s = \frac{N \times \sqrt{Q}}{H^{\frac{3}{4}}}$$

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If we substitute the values of Q and H by their proportional speeds we get a constant as shown below.

$$N_s = \frac{N \times c_1 \sqrt{N}}{c_2 \times N^{\frac{3}{4}}}$$

$$N_s = \frac{c_1}{c_2}$$

We thus obtain the relationship that:

$$N_s = \text{constant}$$

Thus, the parabola passing through homologous points  $P_1$ ,  $P$ , and  $P_2$ , bound by the affinity relationships have the same specific speed.

These are called as Constant Specific Speed Curves.

As efficiency of the pump is a direct function of specific speed, the parabola containing homologous points have the same specific speed and same efficiency.

### 5.7 The Complete Characteristic Curve

Thus, if we are to integrate all the characteristics in a single three-dimensional figure, it would look in the manner shown in Figure 5.9. The coordinates are H, Q, and  $\eta$ .

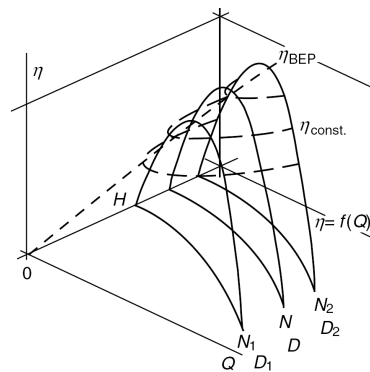


Figure 5.10 – H-Q, and Efficiency Curves

The curves represent the function  $H = f(Q)$  and  $\eta = f(Q)$  are plotted along their respective axes. The Q-H curve varying as a function of impeller diameter or pump speed is also represented in this plot.

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The new curves as seen in Figure 5.9 are the  $\eta_{\text{const}}$  curves called as the iso-efficiency curves. However, we could even flatten these into a two-dimension figure as shown in Figure 5.10.

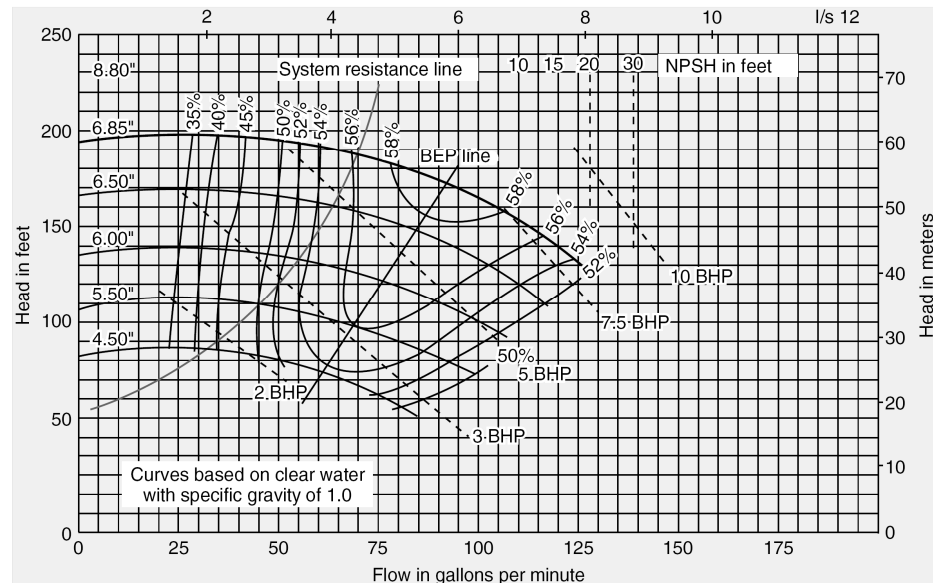


Figure 5.11 – Iso-Efficiency Curves

## 5.8 Multiple pump operation

A process application may require operation of more than one in a single system.

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When pumps are operated in parallel, they work against a common pressure. In this case, the flow rates are added. For a pump in series, the heads are added for a common total head.

Multiple pump operations are sensitive to the individual characteristics of the pump.

## PRACTICAL CENTRIFUGAL PUMPS

The total flow rate or the total head developed may not be a simple addition of the individual flow rates and heads developed. This is especially in cases where the pumps do not have similar characteristic curves.

Before we take up the various cases of operation, it is recommended to the reader to revisit the topic on System Resistance covered under Section 3.6.

### 5.8.1 Parallel operation with similar Q-H curves

When several pumps with similar Q-H characteristic curves feed into a common discharge, the combined Q-H characteristic curve is obtained by simply adding up the abscissae corresponding to all pumps.

If two pumps with the same Q-H curve are operated in parallel, the combined curve is obtained by doubling the abscissa for each point on the individual pump characteristic, while keeping the total value of the head as constant.

When a single pump operates, the flow rate is say  $Q_1$ . The combined Q-H curve is obtained by doubling the abscissa for the same head, as shown in Figure 5.11.

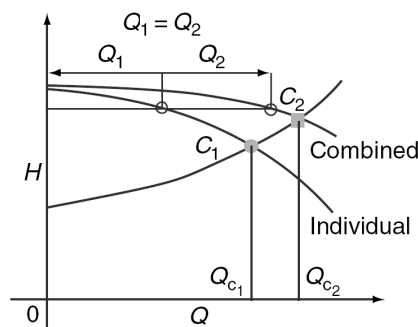


Figure 5.12 – Parallel Operation of Two Pumps with Similar Characteristics

It would be normal to assume that this would be the operating point of the pump when these are operated in parallel. However, this is not the case.

The point of operation is dependent on the system resistance.

In a parallel operation with a system resistance curve as shown in the figure, a single pump would deliver a flow rate of  $Q_{c1}$ . When the next pump is also started, the system resistance takes a turn upwards and intersects the Q-H curve of the combined flows at a point,  $Q_{c2}$ . This is due to the square relationship of the system resistance with the flow rate.

As this can be seen from the figure,  $Q_{c1}$  is not exactly half the flow rate of  $Q_{c2}$ .

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In case there is a need to regulate the flow of the system, it is more efficient to throttle only one pump.

Pumps in parallel are used whenever a flat type of characteristic curve (small  $dH/dQ$ ) is required. That is, the pump discharge head decreases gently with an increase in flow rate.

The combined efficiency for two pumps in parallel is given:

$$\eta = \frac{H \times (Q_1 + Q_2)}{K \times (P_1 + P_2)} \times 100$$

Where

- $\eta$  = combined efficiency [%]
- $H$  = total head
- $Q_1$  = flow rate for pump 1
- $Q_2$  = flow rate for pump 2
- $K$  = unit conversion constant
- $P_1$  = brake power input for pump 1
- $P_2$  = brake power input for pump 2

The combined NPSH-r for pumps in parallel is equal to that for the most limiting pump, the largest NPSH-r in this case.

### 5.8.2 Parallel Operation with Different Q–H Curves

Figure 5.12 shows how the two pumps with dissimilar Q–H characteristic curves. These are represented by curves 1 and 2.

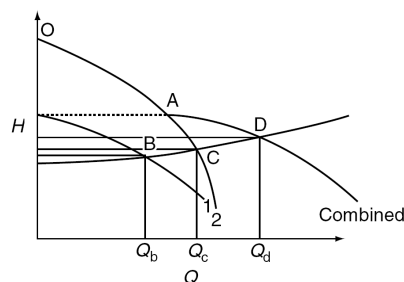


Figure 5.13 – Parallel operation with different Q–H curves

The combined curve of such a combination is represented by OAD. However, until the operating point reaches A, the first pump does not deliver any flow as it sees for itself shut-off conditions.

It is only after point A that pump1 can begin to contribute to the total flow.

It is for this reason that in such a case, the pump2 should be started first and then pump1.

## PRACTICAL CENTRIFUGAL PUMPS

### 5.8.3 Parallel Operation with Flat and Steep Q-H Curves

There are three possible combinations of pump operations with the flat and steep type of characteristics.

These are:

1. Both pumps have flat characteristics
2. Both pumps have steep characteristics
3. One flat and steep curves

The Figures 5.13 and 5.14 are plotted for pumps operating in parallel and having similar flat and steep characteristics respectively.

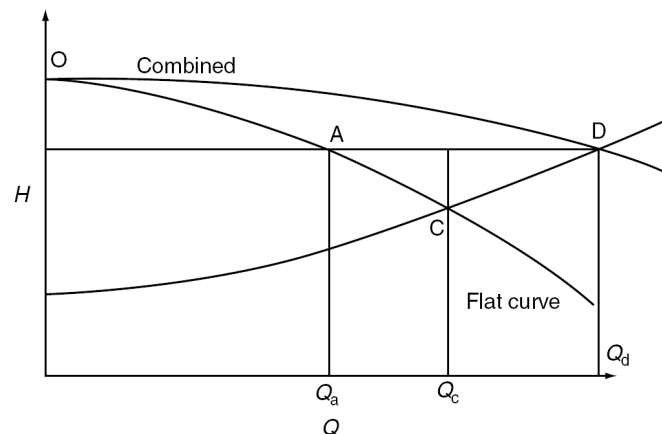


Figure 5.14 – Pumps with Flat Curves

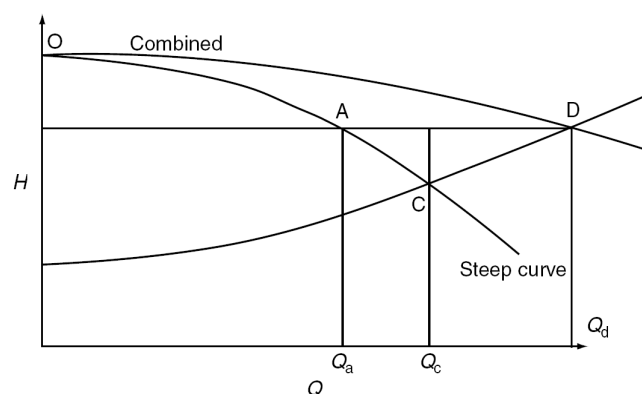


Figure 5.15 – Pumps with Steep Curves

## PRACTICAL CENTRIFUGAL PUMPS

As shown in the figures above the combined Q-H characteristics intersect the system resistance curve at point D. For the same differential head, the individual pump characteristic would intersect at point A.

It can be observed that in case of a flat curve, the difference ( $Q_c - Q_a$ ) would be greater than ( $Q_c - Q_a$ ) of a pump with steep characteristics.

The last combination is already covered in the previous section.

### 5.8.4 Series Operation of Pumps

When two pumps with similar characteristics are operated in series, the combined curve is formed by doubling the head ordinate for each point on the individual pump characteristic, while keeping the same value of flow rate  $Q$  (Figure 5.15).

For pumps with different characteristics, the method is similar.

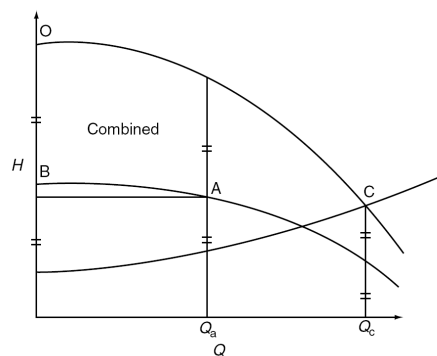


Figure 5.16 – Series Operation of Pumps

When two pumps are connected in series, it follows that at any given instant the rates of flow through the two pumps must be same. The total head of the two pumps on the other hand, are only equal if the individual head discharge characteristics of the pumps are identical.

Series operation of single stage pumps is seldom encountered; more often, multistage centrifugal pumps are used.

## PRACTICAL CENTRIFUGAL PUMPS

### 5.9 Pump Characteristics – Viscous Liquids, Liquids with Considerable Solids

In Section 3.10, 'corrections', we have discussed as to how the behavior of the pump is affected while handling liquids with higher viscosity.

We had also seen the Viscosity Correction Chart provided by the Hydraulic Institute to look up the values of coefficients of flow, head, and efficiency. The coefficients corrected the pump characteristics obtained with clear liquid to the derated performance as expected while pumping viscous fluids.

An example of performance deterioration with increase in liquid viscosity is shown in Figure 5.16.

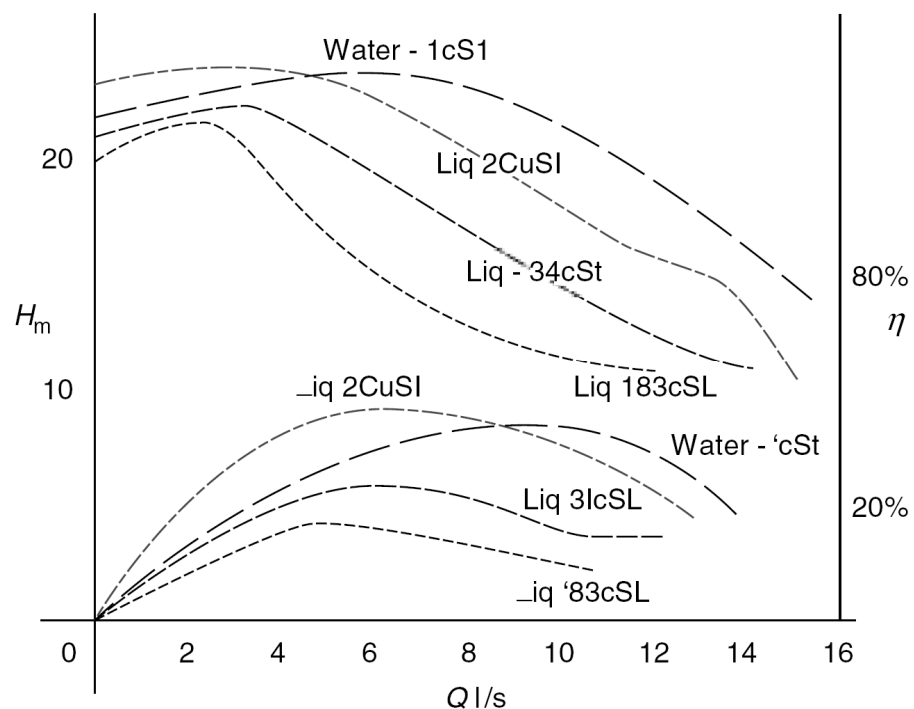


Figure 5.17 – Q–H and Q– $\eta$  Characteristic Comparison of Liquid with Increasing Viscosities

When solids are suspended in liquids, the characteristics of a pump begin to differ from those obtained for pumping clear liquids.

## **PRACTICAL CENTRIFUGAL PUMPS**

Solids in liquids tend to increase the hydraulic resistance in proportion to the concentration of the solid in the liquid.

As a result, both the pump flow rate and the head developed are lower. The power required is higher and consequently, the efficiency of the pump is lower.

If the solid particles are fine in nature, they tend to form a homogenous mixture with the liquid, the shape of the pump characteristics begin to seem like a pump handling a liquid with higher viscosity.

The Q-H curves for pumps transporting mixtures of water and solids such as sand, slag, sugar beet, potatoes, fishes etc. can only be determined by experiments.

The Q-H curves while pumping various mixtures formed by various percentages of solids in the liquids can be compared with that of clear water and nomogram can be created. This nomogram can then be used to predict the pump characteristics for a particular mixture.

### **5.10 Pump Characteristics – Abnormal Operation**

The normal operation of a pump is considered when direction of rotation of the pump is in accordance with the backward vanes of the impeller. The flow of the pump is from the suction (lower head) to the discharge (higher head).

The pump characteristics considered in the preceding sections are based on the normal operation of the pump as described above and this shown in the right corner of Figure 5.17.

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