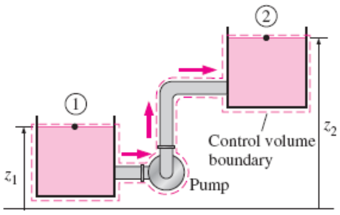
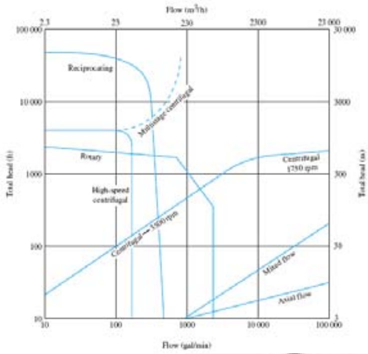



$$NPSH = \frac{(P_s - P^v)}{\rho g}$$

# Fluid Mechanics (0905241)

## Hydraulic Analysis of Pumps and Piping Systems




Prof. Zayed Al-Hamamre

$$h_p = (Z_2 - Z_1) + \frac{P_2 - P_1}{\rho g} + h_f + h_m$$

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# Content

- **Conservation Principle of Energy (Review)**
- **Pump System Heads**
- **System Performance Curves**
- **Net Positive Suction Head (NPSH)**
- **Pump Characteristic Curves**
- **Pump Selection**



# Introduction

- **Pumps:** Devices for supplying energy or head to overcome energy losses.

- If the pump is placed between two points 1 and 2 in a pipeline, the mechanical energy balance becomes:

$$g(z_2 - z_1) + \frac{1}{2}\left(\bar{u}_2^2 - \bar{u}_1^2\right) + \frac{P_2 - P_1}{\rho} = w_p - w_f$$

$w_p$ : specific work done by pump on the liquid

• No pump is not 100% efficient, the efficiency of the pump is defined as:

$$\eta = \frac{w_p}{W_{total}} = \frac{\dot{m} w_p}{\dot{W}_{total}} = \frac{\dot{w}_p}{\dot{W}_{total}}$$

Where $W_{total}$ is the total specific work supplied to the motor of the pump.



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# Introduction

$\dot{m} = \rho Q$ : Mass flow rate of the pumped liquid

$\dot{w}_p$ : Power added to the liquid by the pump

$\dot{W}_{total}$ : Power supplied to the motor of the pump



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# Conservation Principle of Energy

- Dividing MEB by gravitational acceleration, g, gives head form of energy balance:

$$(z_2 - z_1) + \frac{1}{2g}\left(u_2^2 - u_1^2\right) + \frac{P_2 - P_1}{\rho g} = \frac{w_p}{g} - \frac{w_f}{g}$$

Let:

$\frac{w_p}{g} = h_p$ :Total energy head done on the system by pump for example; in SI units m.

$\frac{w_f}{g} = h_f$ : Energy head losses due to friction; in SI units m

**Head form of (MEB)**

$$(z_2 - z_1) + \frac{1}{2g}\left(u_2^2 - u_1^2\right) + \frac{P_2 - P_1}{\rho g} = h_p - (h_f + h_m)$$

| Potential head | Velocity head | Pressure head | Pump head | Head losses |
|---|---|---|---|---|



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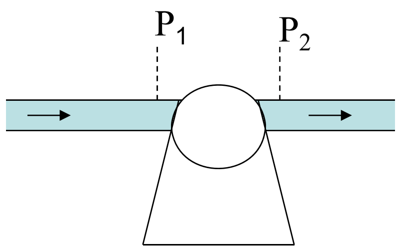
# Example

The motor which drives a pump is supplied by 2.8 horsepower. The pressure at the suction side of the pump is 30 psia and the pressure at the discharge side is 100 psia. The pump is pumping 50 gpm. What is the efficiency of the pump?



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# Example Contd.



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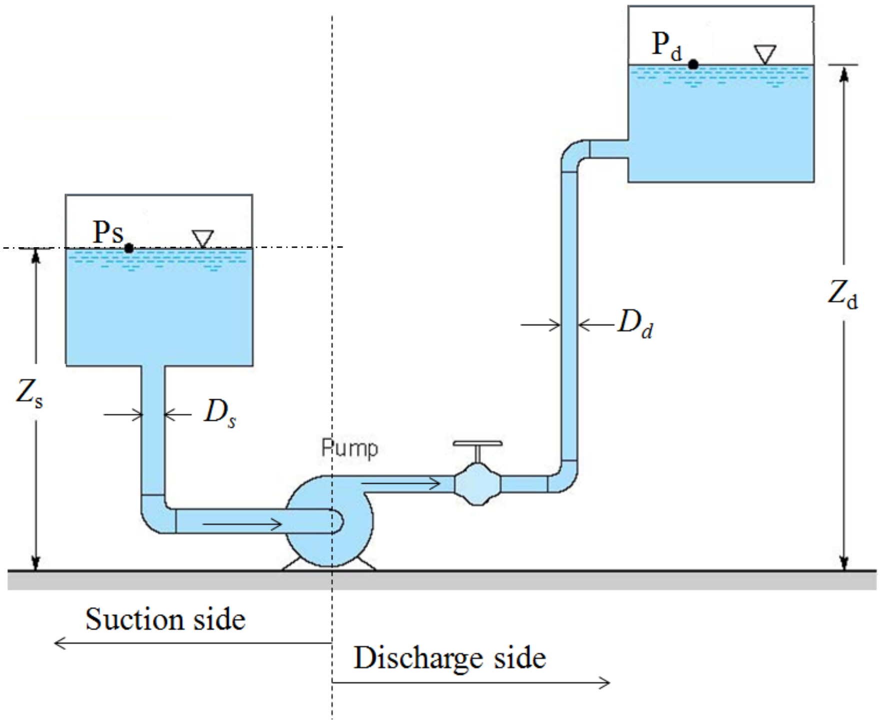
# Analysis of Pumps and Piping Systems



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# Pump System Heads

**A. Suction head ($h_s$): it works in favor of the pump**

$$h_s = Z_s + \frac{P_s}{\rho g} - (h_{fs} + h_{ms})$$

$Z_s$ : Suction static head, if it has negative value it will be Suction static lift.

$P_s$ : gas absolute pressure above the liquid in the tank of the suction side.

$h_{fs}$ : head losses due to friction in the suction side.

$h_{ms}$ : head losses due to fittings in the suction side.



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# Pump System Heads

**B. Discharge head ($h_d$): it works against the pump**

$$h_d = Z_d + \frac{P_d}{\rho g} + (h_{fd} + h_{md})$$

$Z_d$ : discharge static head, if it has negative value it will be Suction static lift.

$P_d$ : gas absolute pressure above the liquid in the tank of the discharge side.

$h_{fd}$ : head losses due to friction and fittings in the discharge side.

$h_{md}$ : head losses due to fittings in the discharge side.



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# Pump System Heads

**C. Total head ($h_p$):**

$$h_p = h_d - h_s = \frac{w_p}{g}$$

This definition can be obtained by applying the head form of mechanical energy balance between one point (1) at the free surface of the suction tank and another point (2) at the free surface of the discharge tank:

$$(Z_d - Z_s) + \frac{1}{2g}\left(\bar{u}_2^2 - \bar{u}_1^2\right) + \frac{P_d - P_s}{\rho g} = h_p - h_L$$

$$\bar{u}_2 \approx 0 \quad ; \quad \bar{u}_f \approx 0$$

$h_f$ is the total head losses in the suction and discharge sides:

$$h_L = h_{fs} + h_{fd} + h_{ms} + h_{md}$$



# Pump System Heads

**C. Total head ($h_p$):**

$$(Z_d - Z_s) + \frac{P_d - P_s}{\rho g} = h_p - h_L$$

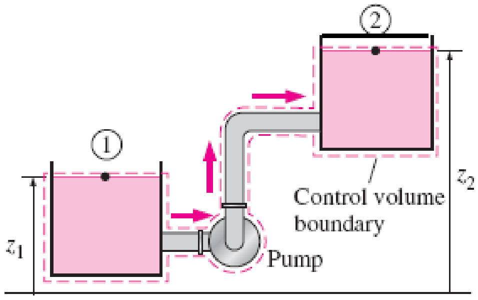
Rearrange the equation as:

$$h_p = Z_d - Z_s + \frac{P_d}{\rho g} - \frac{P_s}{\rho g} + h_L$$

$$= \left(Z_d + \frac{P_d}{\rho g} + h_{fd} + h_{md}\right) - \left(Z_s + \frac{P_s}{\rho g} - h_{fs} - h_{ms}\right)$$

$$= h_d - h_s$$

# Total System Heads




A volumetric flow rate Q of water is required to be moved from tank 1 to tank 2

Applying the Mech E. Equation between point 1 and point 2

$$Z_1 + \frac{P_1}{\rho g} + \frac{V_1^2}{2g} + h_p = Z_2 + \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + h_f + h_m$$

$$h_f = \sum_i f_i \frac{L_i}{D_i}\frac{V_i^2}{2g}$$ : head losses due to friction (pipe section)

$$h_m = \sum_i K_{L,i} \frac{V_i^2}{2g}$$ : head losses due to minor losses (fitting and valves)

($h_f$ and $h_m$ together: $h_L$)

Depend on he velocity in the pipes



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# Pump System Heads

For the current situation, the velocity terms at 1 and 2 are zero, hence, the required pump head for the system to operate:

$$h_p = (Z_2 - Z_1) + \frac{P_2 - P_1}{\rho g} + h_L$$

Useful work against elevation and pressure difference — Constant for a given system

Friction and minor losses dissipate energy — Depend on the flow rate (Q)

- At any different flow rate, elevation difference and pressure difference are constant.
- When there is no flow, Q = 0, then there will be no losses, but the system must have a head ($h_p$) to overcome the elevation difference and pressure difference before any flow can occur.
- The higher the flow rate goes, the higher are the friction and minor losses and the higher the system head required.



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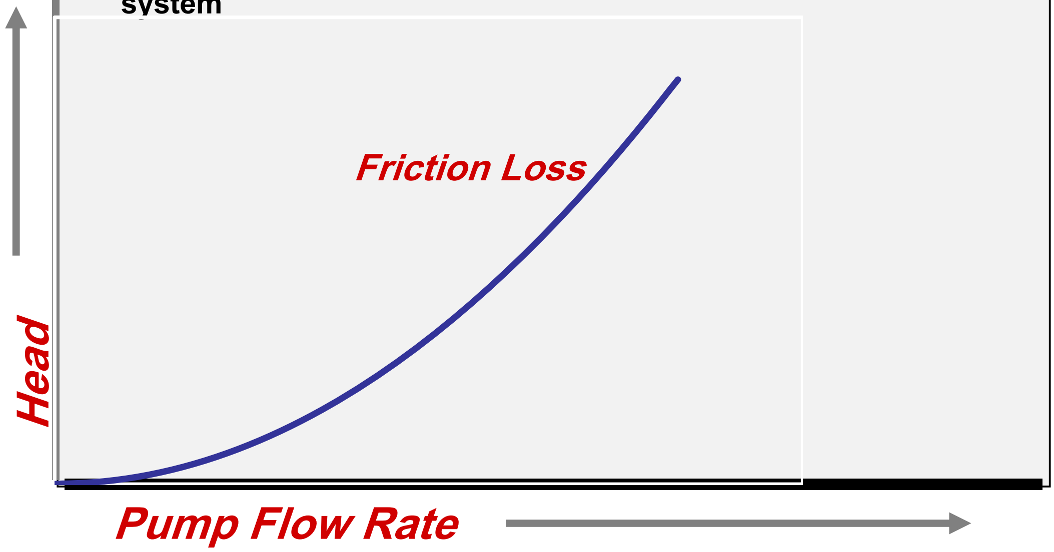
# Total System Heads

- The friction loss is mapped onto the graph
- The amount of friction loss varies with flow through the system



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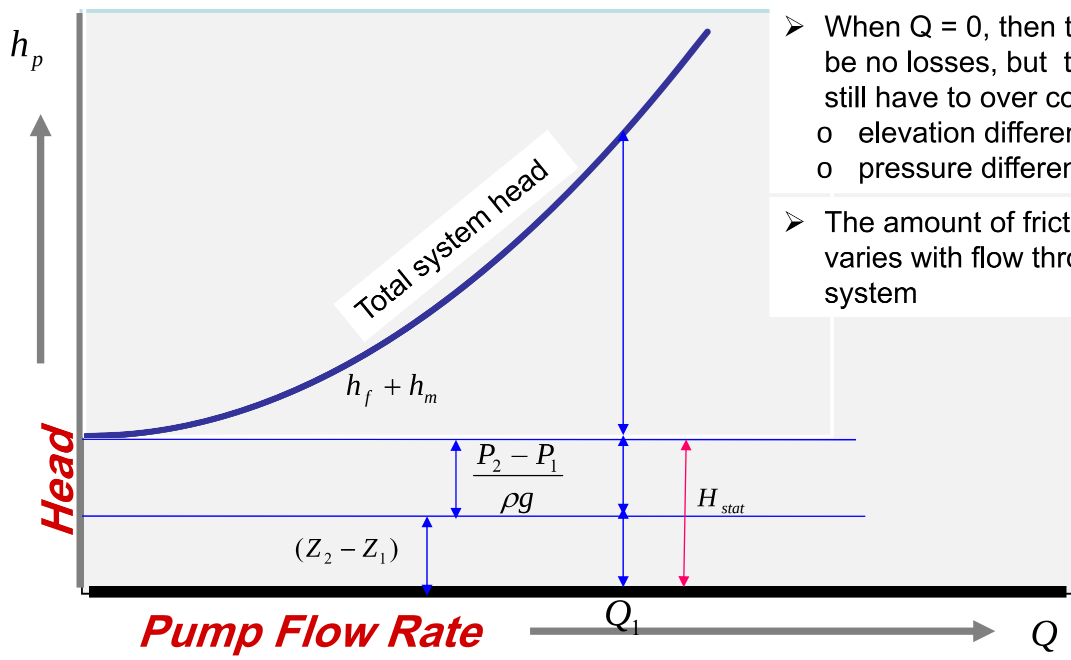
# Total System Heads




- When Q = 0, then there will be no losses, but the system still have to over come the
  - elevation difference and
  - pressure difference
- The amount of friction loss varies with flow through the system



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## System Performance Curves

- The total head, $H_t$ , that the pump delivers includes the elevation head and the head losses incurred in the system.
- The friction loss and other minor losses in the pipeline depend on the velocity of the water in the pipe, and hence the total head loss can be related to the discharge rate
- For a given pipeline system (including a pump or a group of pumps), a unique system head-capacity *(H-Q)* curve can be plotted.
  - This curve is usually referred to as a system characteristic curve or simply system curve.
  - System Performance Curve is a mapping of the head required to produce flow in a given system, i.e. it is a graphic representation of the system head (pressure drop) over a range of flow rates starting from zero to the maximum expected value of Q.



## System Performance Curves

Step #1, Horizontal Axis

- **The System's flow rate in plotted on the horizontal axis ( X axis)**
- **Usually expressed in Gallons per Minute**

***System Flow Rate*** →

# System Performance Curves

Step #2, Vertical Axis

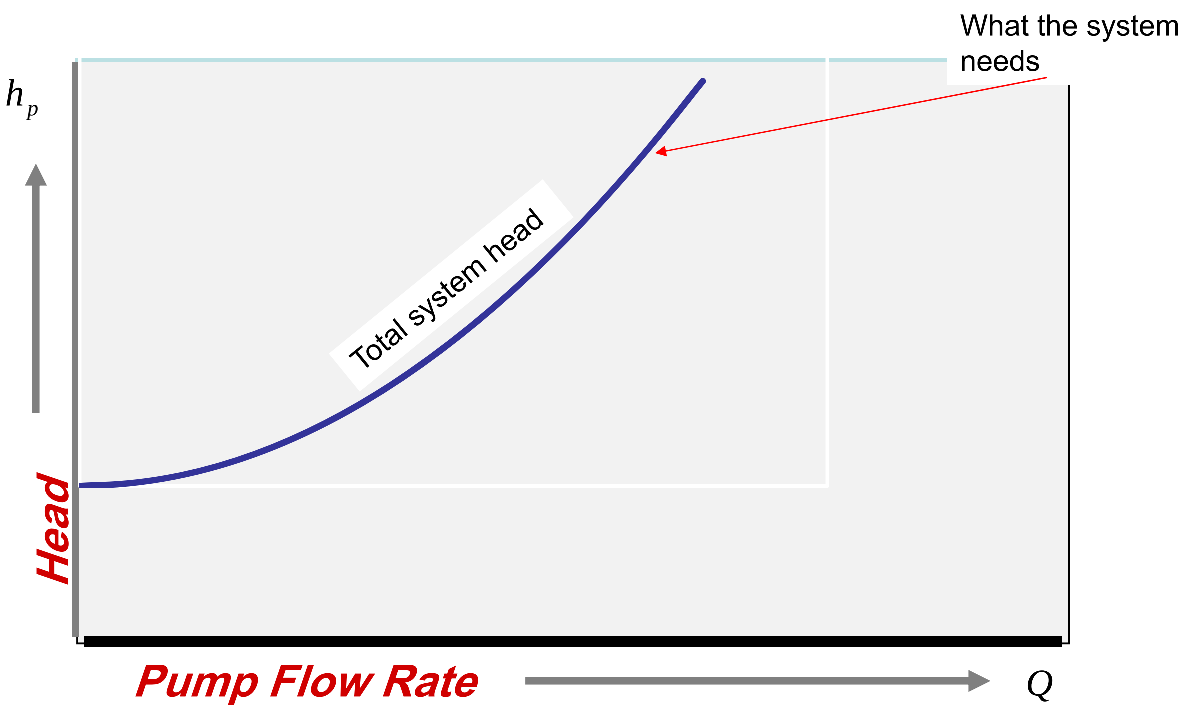
- The head the system requires is plotted on the vertical axis (Y axis)
- Usually express in Feet of Water

Head

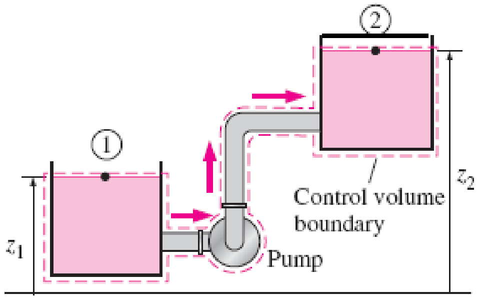
Pump Flow Rate



# System Performance Curves



# Example




A volumetric flow rate Q of water is required to move from tank 1 to tank 2

Applying the Mech E. Equation between point 1 and point 2

$$h_p = \Delta Z + \frac{\Delta P}{\rho g} + \frac{\Delta V^2}{2g} + h_f + h_m$$

$$h_L = h_f + h_m = \sum_i f_i \frac{L_i}{D_i} \frac{V_i^2}{2g} + \sum_i K_{L,i} \frac{V_i^2}{2g}$$

$$= \left(\sum_i f_i \frac{L_i}{D_i} + \sum_i K_{L,i}\right) \frac{V_i^2}{2g}$$



# Example Contd.

But $Q = VA$

$$h_L = \left(\sum_i f_i \frac{L_i}{D_i} + \sum_i K_{L,i}\right) \frac{8}{g \pi^2 D^4} Q^2$$

Constant

$f_i = F(Re, \varepsilon/D)$

$K_{L,i} = F(Re)$ or constant

Almost Constant for a given system and relatively in dependent of the flow rate

System head difference that the we need the pump to provide:

$$h_p = \Delta Z + \frac{\Delta P}{\rho g} + CQ^2$$

$C$ : Constant with all information about system geometry, video

# Cavitation (Review)

- In general, **cavitation** occurs when the liquid pressure at a given location is reduced to the vapor pressure of the liquid.
- For a piping system that includes a pump, **cavitation** occurs when the absolute pressure at the inlet falls below the vapor pressure of the water.
- This phenomenon may occur at the inlet to a pump and on the impeller blades, particularly if the pump is mounted above the level in the suction reservoir.
- Under this condition, vapor bubbles form (water starts to boil) at the impeller inlet and when these bubbles are carried into a zone of higher pressure, they collapse abruptly and hit the vanes of the impeller (near the tips of the impeller vanes). **causing**:
  - o Damage to the pump (pump impeller)
  - o Violet vibrations (and noise).
  - o Reduce pump capacity.
  - o Reduce pump efficiency



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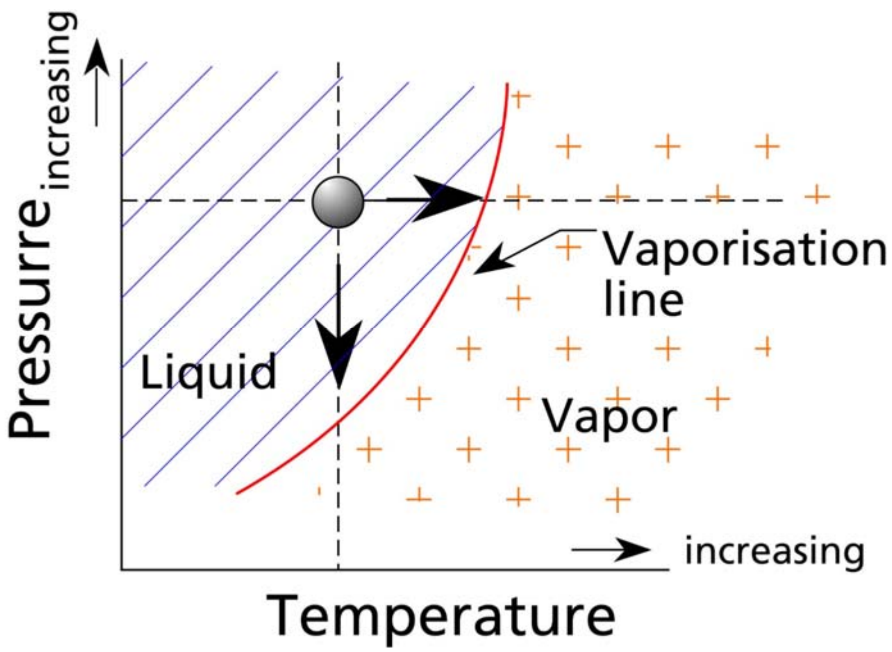
# Cavitation (Review)



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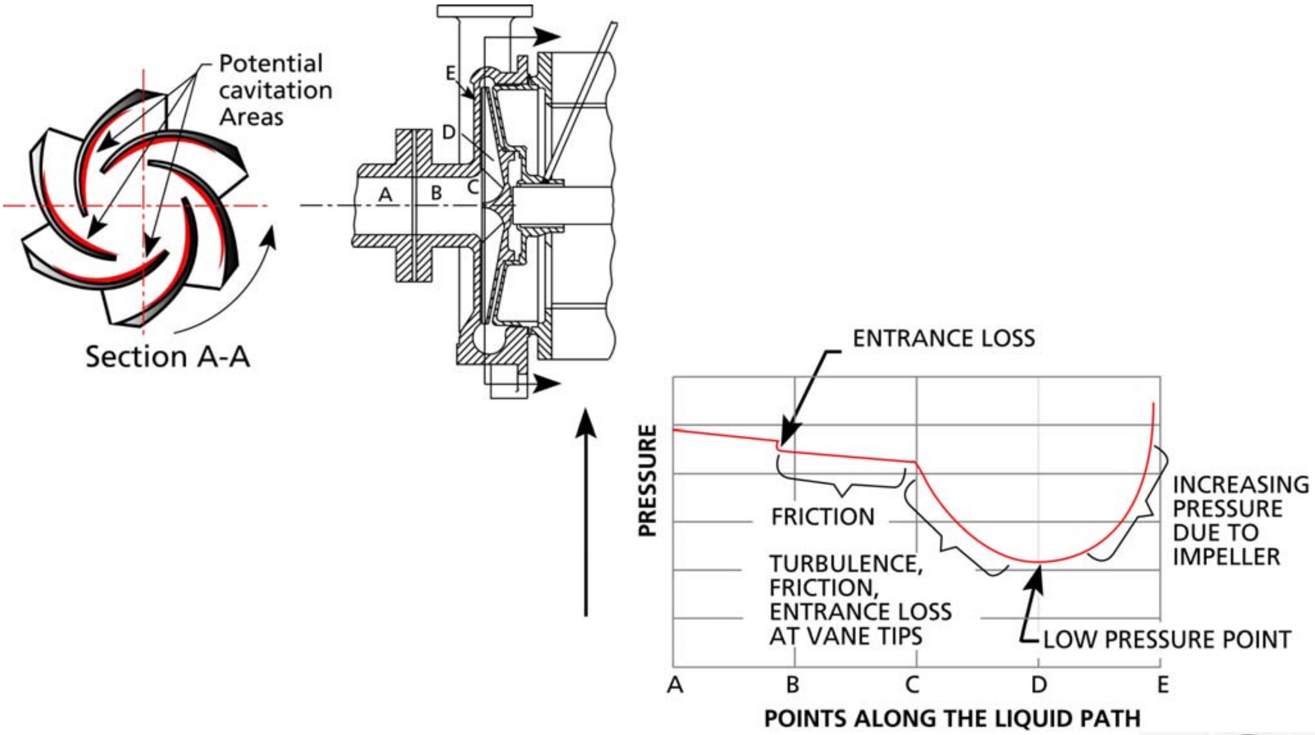
## Cavitation (Review)






## NPSH

Pump design

Flow speed of the liquid through the pump

→ Minimum inlet pressure which can be tolerated to prevent cavitation, such that

$$P_s > P^v$$

where $P_s$ : Pressure at the pump suction

$P^v$ : Liquid vapor pressure

- Then, the **Net Positive Suction Head (NPSH)** is the head (pressure) at the pump suction minus the vapor pressure head

$$NPSH = \frac{(P_s - P^v)}{\rho g}$$

# NPSH

- There are two values of NPSH of interest.

i. The first value for NPSH of concern is the available NPSH, denoted ***(NPSH)$_A$***, which represents the head that actually occurs for the particular piping system. This value can be determined experimentally, or calculated if the system parameters are known.

Define the suction head as $h_s = \dfrac{P_s}{\rho g}$

Then $$NPSH_A = h_s - \frac{P^v}{\rho g}$$



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# NPSH

ii. The second is the required NPSH, denoted ***(NPSH)$_R$***, that must be maintained or exceeded so that cavitation will not occur and usually determined experimentally and provided by the manufacturer.

- For proper pump operation (no cavitation) :

$$(NPSH)_A > (NPSH)_R$$



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## Thoma's cavitation constant

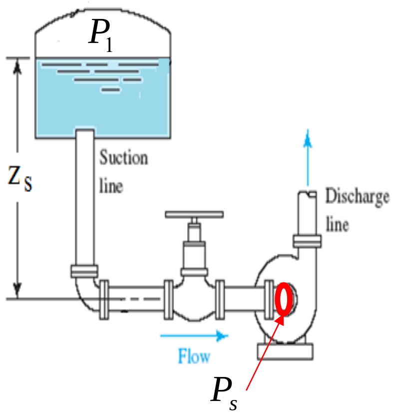
**The cavitation constant:** is the ratio of *(NPSH)*$_R$ to the total dynamic head ($H_t$) is known as the Thoma's cavitation constant ($\sigma$)

$$\sigma = \frac{(NPSH)_R}{h_p}$$

***Note***: If the cavitation constant is given, we can find the maximum allowable elevation of the pump inlet (eye) above the surface of the supply (suction) reservoir.



## Example




$$NPSH_A = h_s - \frac{P^v}{\rho g}$$

$$h_s = \frac{P_1}{\rho g} + Z_s - \frac{V_1^2}{2g} - h_f - h_m$$

Dynamic head (due to fluid acceleration)

Losses due to friction

Minor losses

➢ All terms should be figured for designing the piping system

➢ To avoid cavitation, make sure that

$$(NPSH)_A > (NPSH)_R$$

Calculated

provided by the manufacturer

## Effect of flow rate on the system heads

As flow rate increases $h_{fs}$ and $h_{fd}$ increases then:

- The suction head ($h_s$) will decrease.

- The discharge head ($h_d$) will increase.

- The total head ($h_p$) which the pump is required to impart the flowing liquid increases.

- The available net positive suction head (*NPSH*) will decrease.



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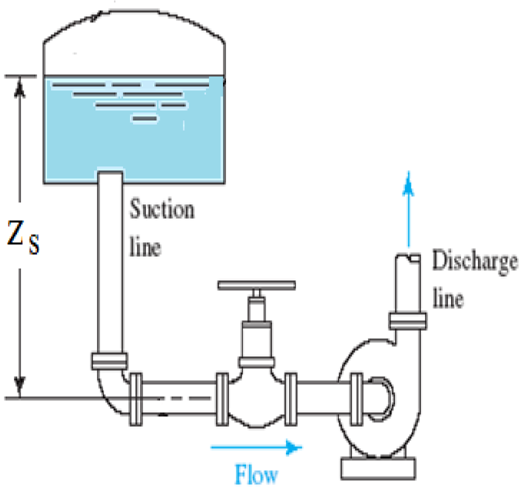
## Example

Determine suction head and the available *NPSH* for the system shown in the figure . The liquid reservoir is a closed tank with a pressure of -20 kPa above water at 70°C (the vapor pressure is 31.2 kPa, density is 978 kg/m3, and kinematic viscosity is 4.13 ×$10^{-7}$ $m^2/s$). The atmospheric pressure is 100.5 kPa. The water level in the tank is 2.5 m above the pump inlet. The pipes in the suction side is a 1.5-in Schedule 40 steel pipe with a total length of 12.0 m. The elbow is standard 90° and the valve is a fully open globe valve. The flow rate is 95 L/min.



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# Example Contd.



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# Example Contd.



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# Example Contd.



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## Pump Characteristic Curves

- Pump manufacturers provide information on the performance of their pumps in the form of curves, commonly called pump characteristic curves (or simply pump curves).

- In pump curves the following information may be given:
  - the discharge on the x-axis,
  - the head on the left y-axis,
  - the pump power input on the right y-axis,
  - the pump efficiency as a percentage,
  - the speed of the pump (rpm = revolutions/min).
  - the NPSH of the pump.


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## Pump Characteristic Curves

Step #1, Horizontal Axis

- The pump's flow rate is plotted on the horizontal axis ( X axis)
- Usually expressed in Gallons per Minute

***Pump Flow Rate*** →


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# Pump Characteristic Curves

Step #2, Vertical Axis

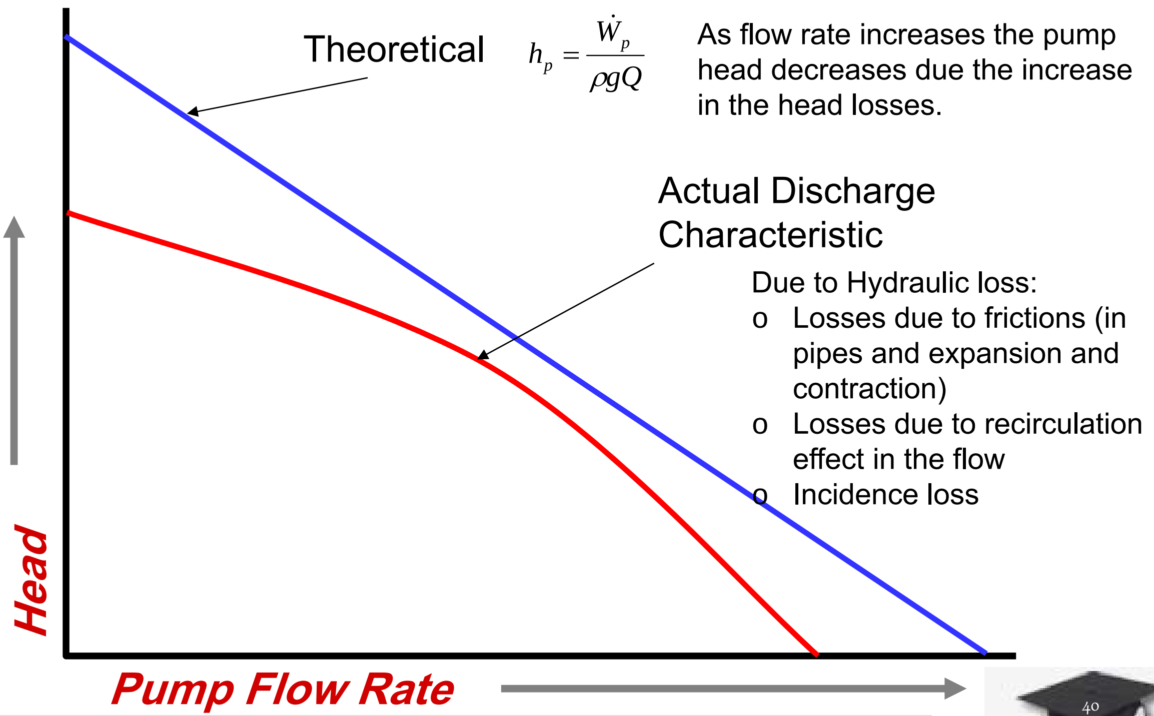
- The head the pump produces is plotted on the vertical axis (Y axis)
- Usually express in Feet of Water

***Head***

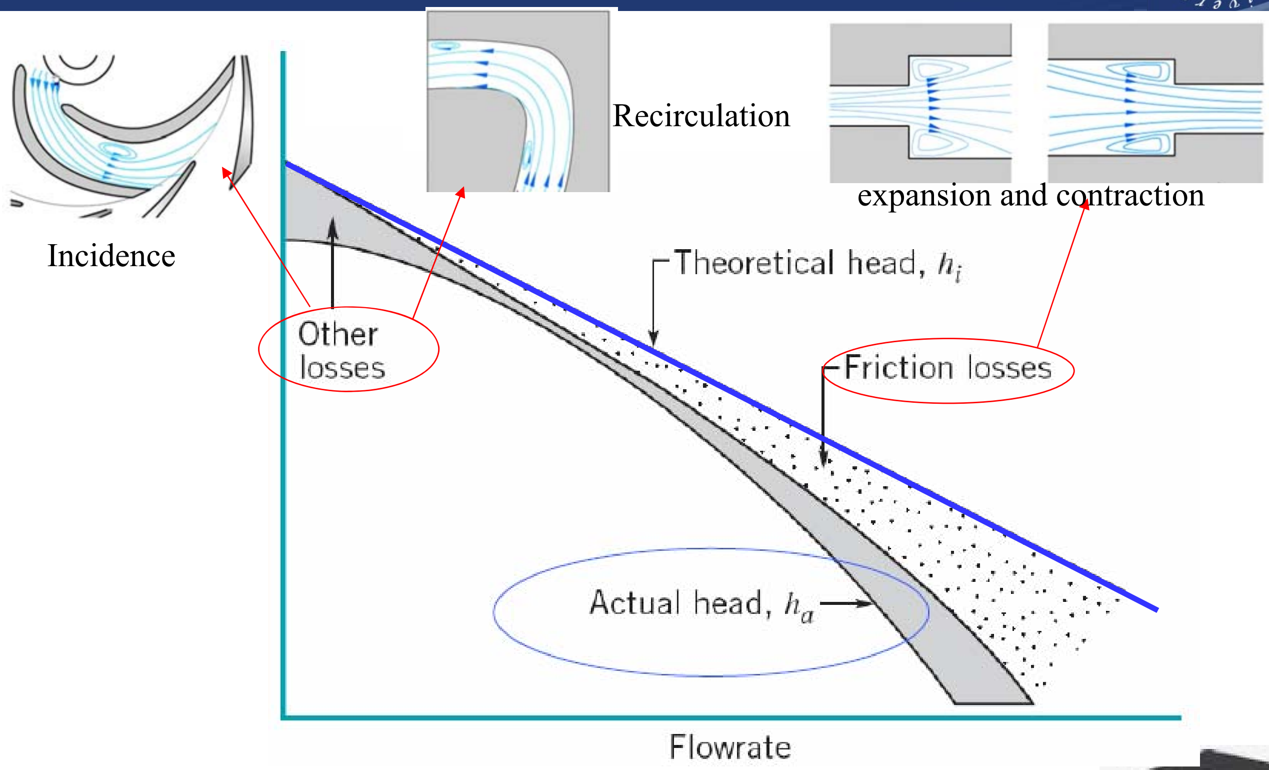
***Pump Flow Rate***



# Pump Characteristic Curves



# Pump Characteristic Curves

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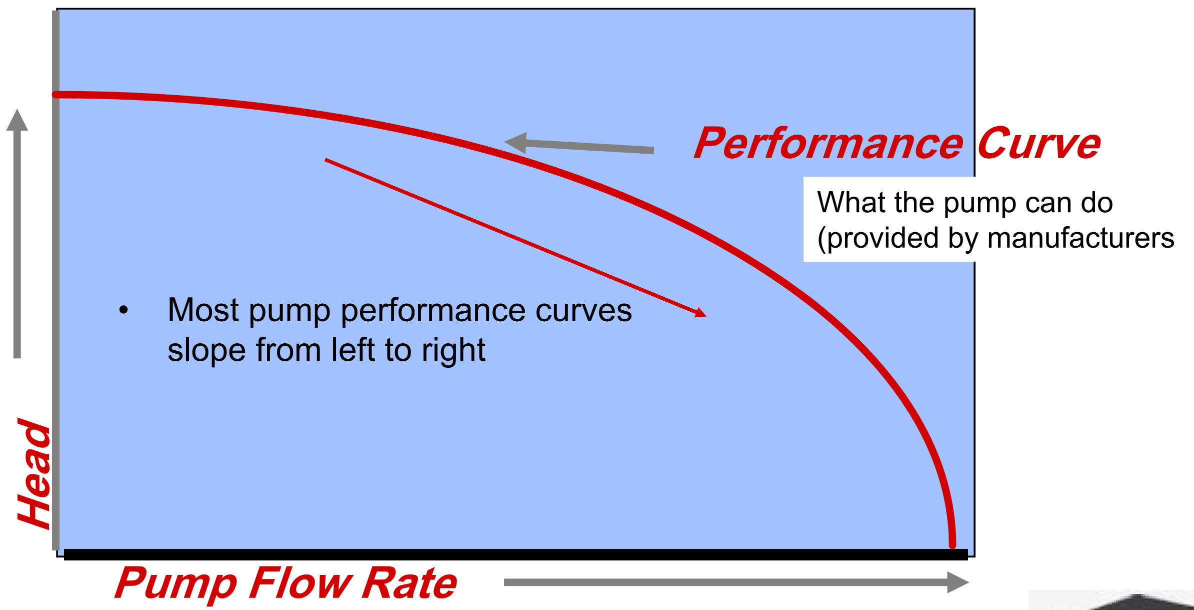
# Pump Characteristic Curves

Step #3, Mapping the Flow and the Head




- Most pump performance curves slope from left to right

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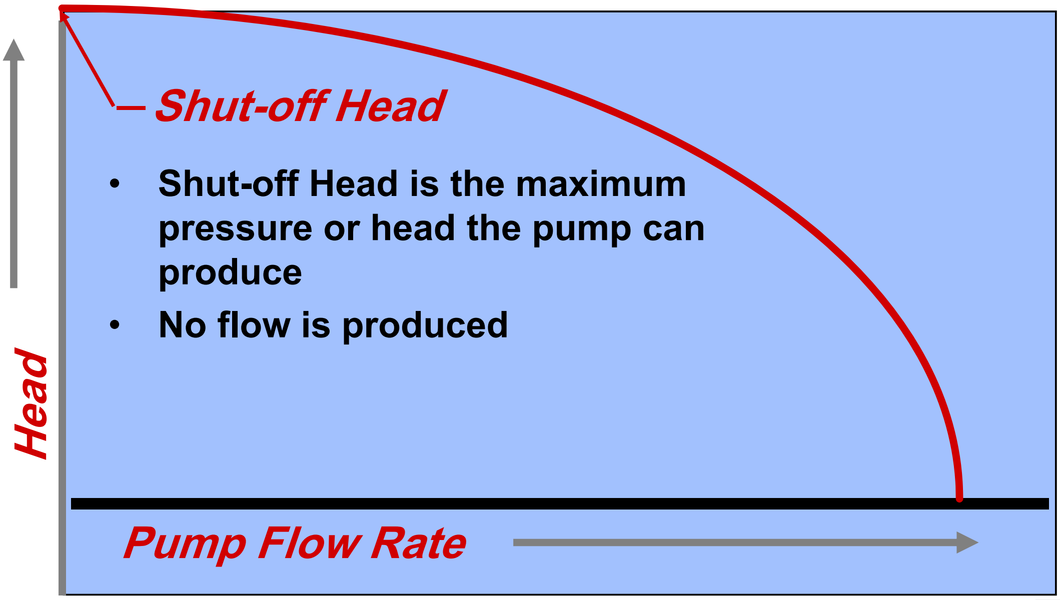
# Pump Characteristic Curves

Important Points




- Shut-off Head is the maximum pressure or head the pump can produce
- No flow is produced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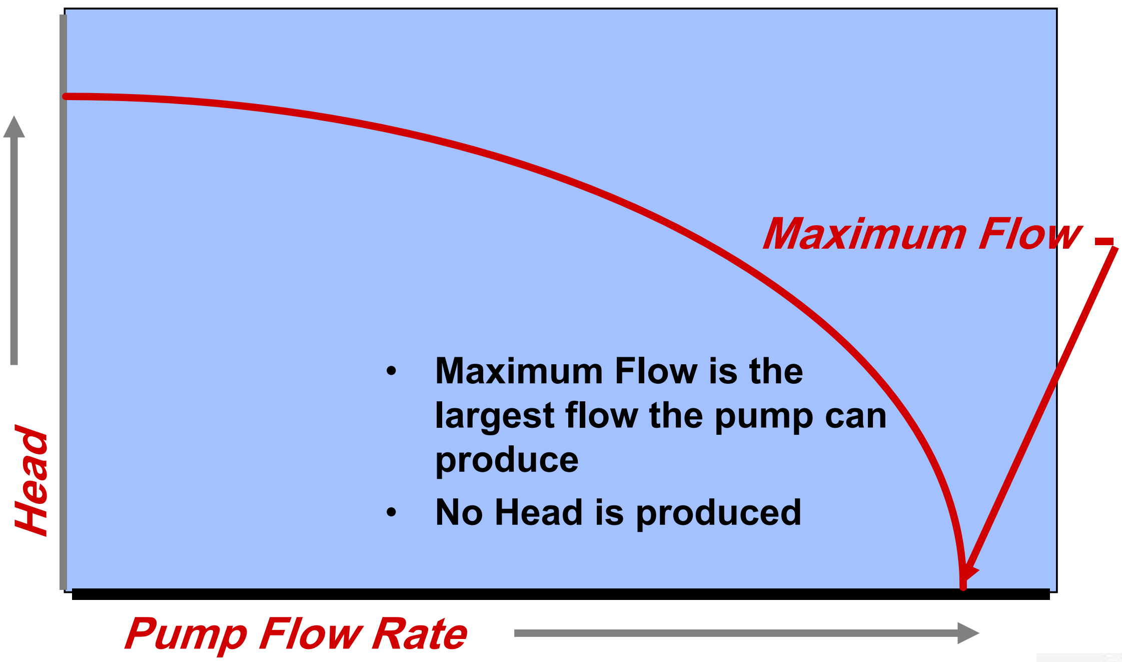
# Pump Characteristic Curves

Important Points




- Maximum Flow is the largest flow the pump can produce
- No Head is produced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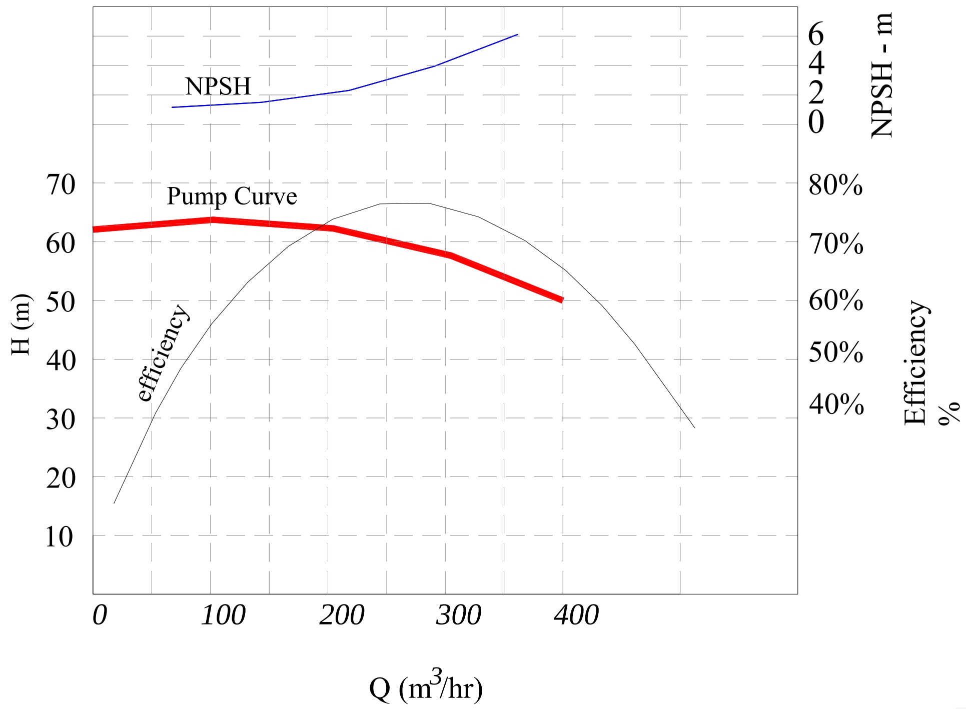
# Pump Characteristic Curves



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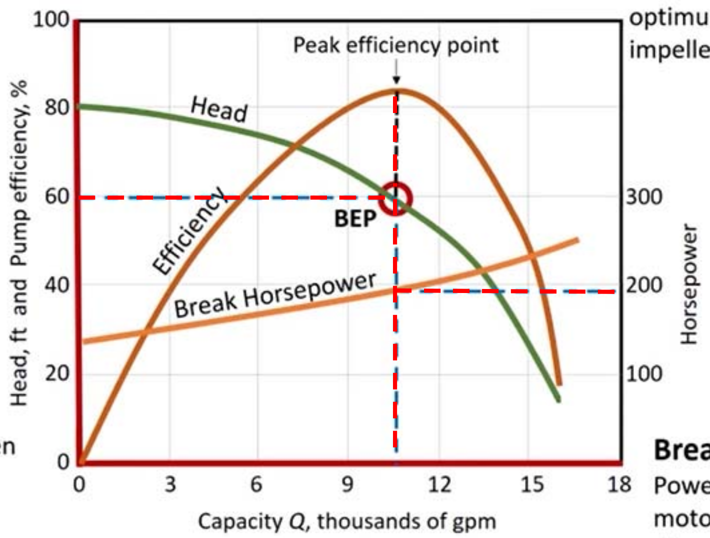
# Pump Characteristic Curves

**Head Curve**
Feet of head for a given flow rate

**BEP - Best Efficiency Point**
The flow at which the pump operates at the highest or optimum efficiency for a given impeller diameter




**Efficiency**
Pump efficiency for a given flow rate

**Break Horsepower**
Power required from the motor to drive the pump at a given head to deliver a given volumetric flow rate



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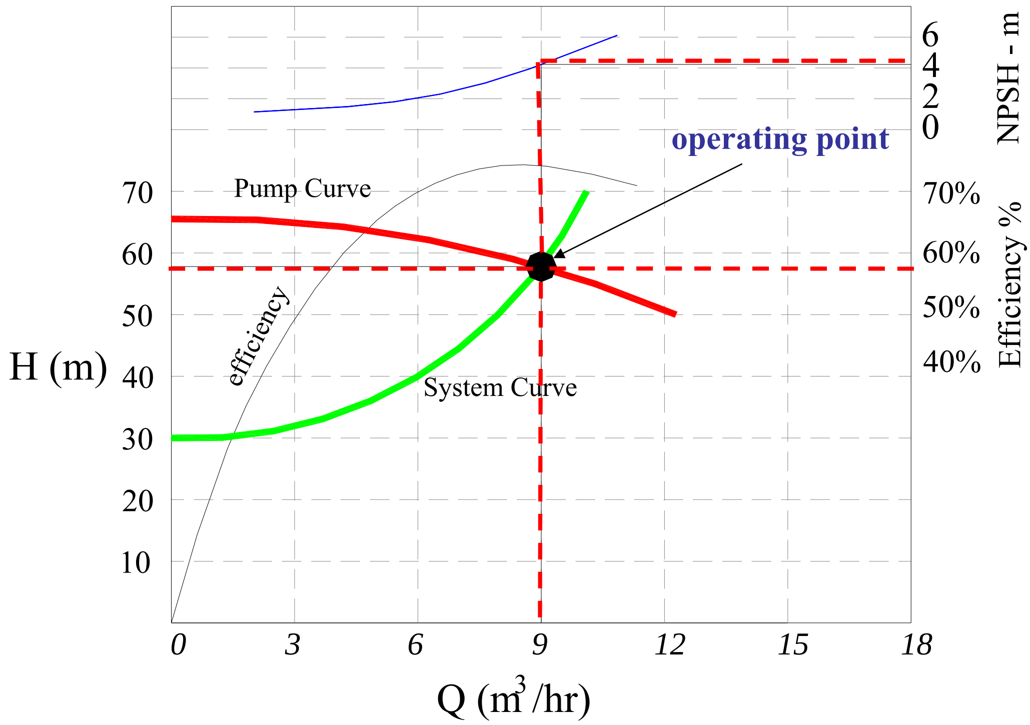
# Performance Curves



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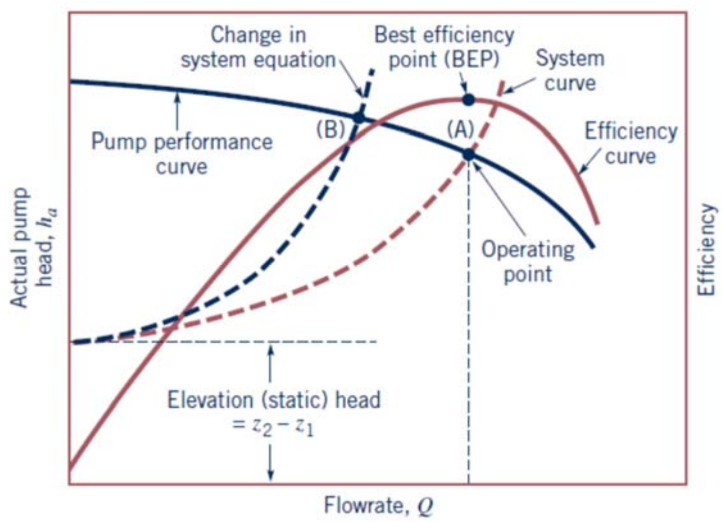
# Performance Curves

- There is also a unique relationship between the actual pump head gained by the fluid and flow rate, which is governed by the pump design.
- Pipe friction increase due to wall fouling.

(A)→ (B) flowrate ↓
efficiency ↓



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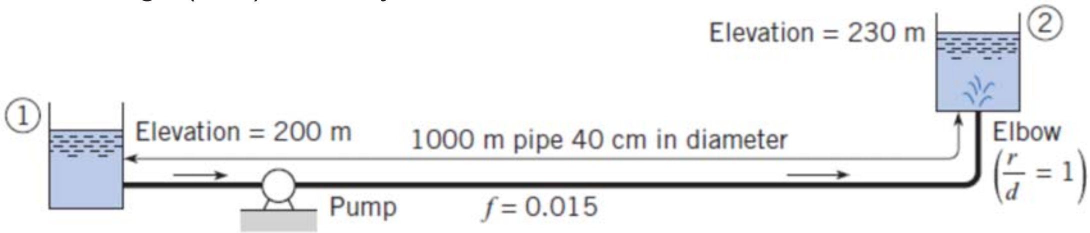
# Example

1. A pump has the head-versus-discharge curve shown.
2. The friction factor is *f* 0.015.

**Find:** Discharge ($m^3 s$) in the system.




***Plan***

1. Develop an equation for the system curve by applying the energy equation.
2. Plot the given pump curve and the system curve on the same graph.
3. Find discharge *Q* by finding the intersection of the system and pump curve.



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# Example Contd.



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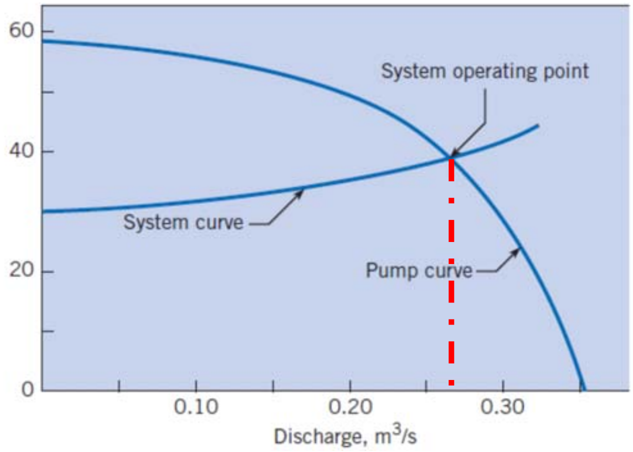
# Example Contd.

| $Q(m^3/s)$ | $h_p = 30\ m + 127Q^2\ m$ |
|---|---|
| 0 | 30 |
| 0.1 | 31.3 |
| 0.2 | 35.1 |
| 0.3 | 41.4 |



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# Example

For the following pump, determine the required pipes diameter to pump 60 L/s and also calculate the needed power.

*Minor losses 10 $v^2/2g$*

*Pipe length 10 km*

*roughness = 0.15 mm*

*$\Delta Z$ = 20 m*

| Q L/s | 70 | 60 | 50 | 40 | 30 | 20 | 10 | 0 |
|---|---|---|---|---|---|---|---|---|
| $h_p$ | 31 | 35 | 38 | 40.6 | 42.5 | 43.7 | 44.7 | 45 |
| $\eta_P$ | 40 | 53 | 60 | 60 | 57 | 50 | 35 | – |



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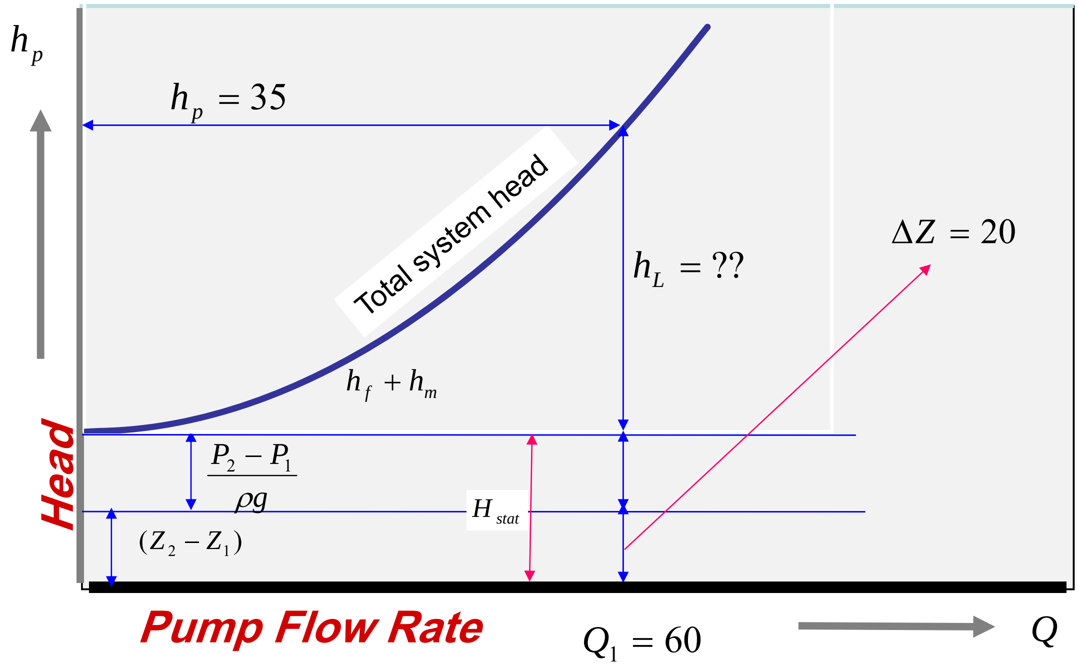
# Total System Heads






# Example Contd.

# Example Contd.





# Example Contd.



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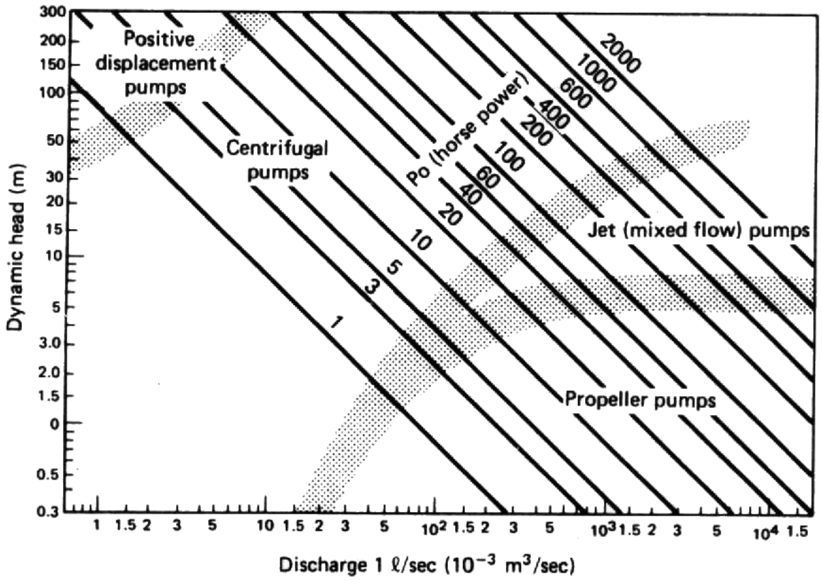
## Selection of A Pump

- In selecting a particular pump for a given system:
- The design conditions are specified and a pump is selected for the range of applications.
- A system characteristic curve (H-Q) is then prepared.
- The H-Q curve is then matched to the pump characteristics chart which is provided by the manufacturer.
- The matching point (operating point) indicates the actual working conditions.

# Selection of A Pump

- It has been seen that the efficiency of a pump depends on the discharge, head, and power requirement of the pump.
- The approximate ranges of application of each type of pump are indicated in the following Figure.

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# Characteristic Curves of Centrifugal Pumps

- Because centrifugal pumps are not positive-displacement types, there is a strong dependency between capacity and the pressure that must be developed by the pump.
- This makes their characteristic curves somewhat more complex.
- The typical performance characteristic curves for centrifugal pump are its total head, efficiency, and power plots versus the capacity or discharge *Q.*
- Pump performance is affected strongly by size of impeller and its rotational speed (rpm). Thus, they must be include in the composite performance charts.

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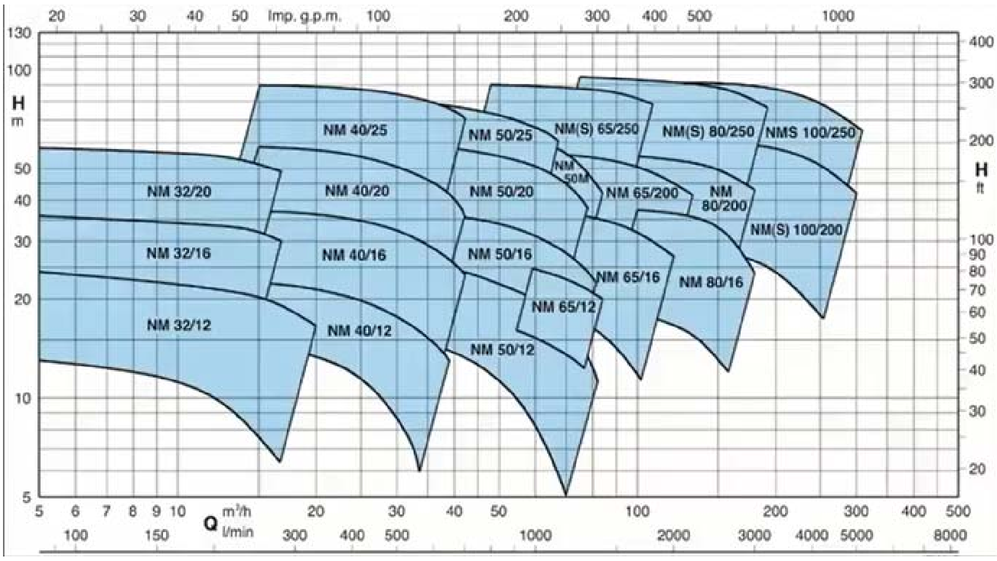
# Manufacturers’ Data for Centrifugal Pumps

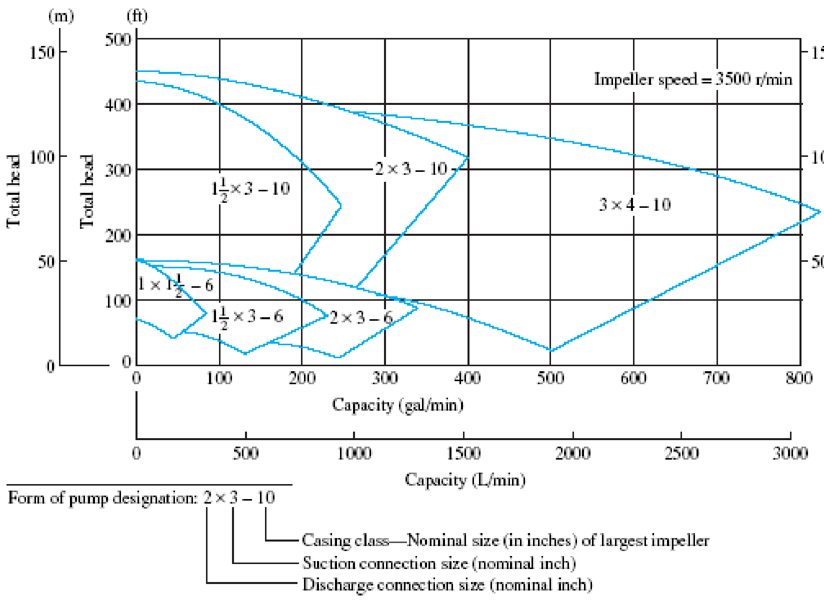
- Manufacturers publish families of pump curves to identify model flow head ranges.
- Each od these model model with different geometry, speed and specific flow head

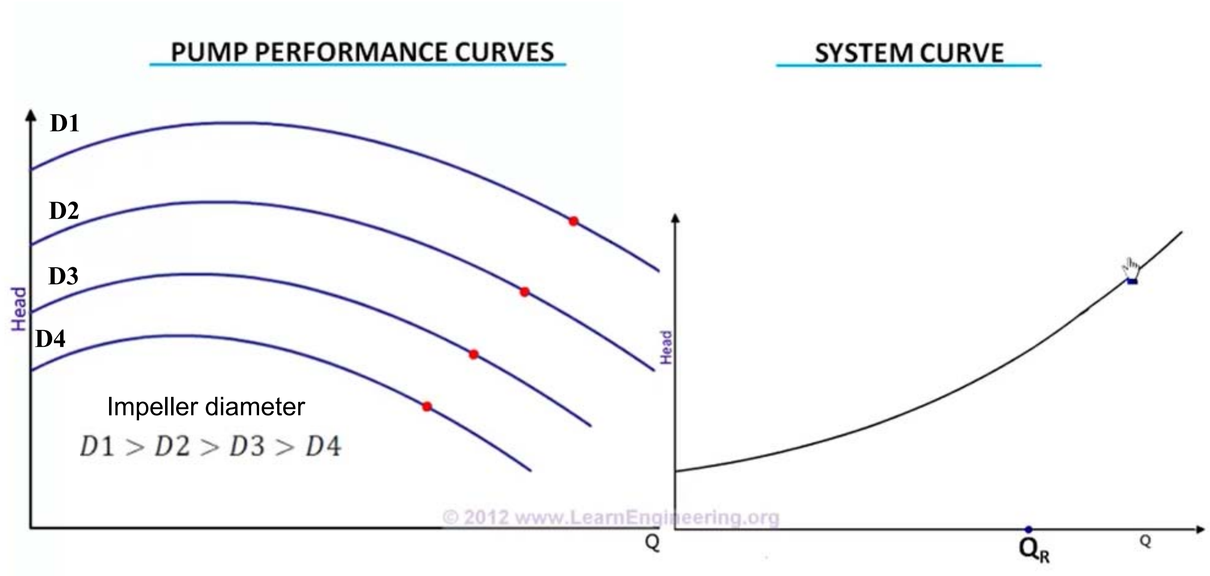




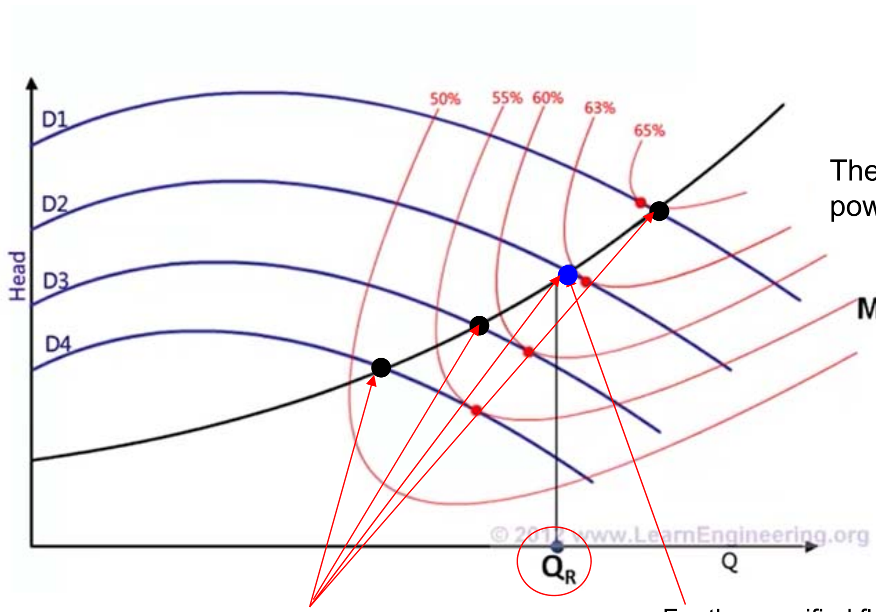

# Manufacturers’ Data for Centrifugal Pumps

- A composite rating chart which allows a quick determination of the centrifugal pump size:







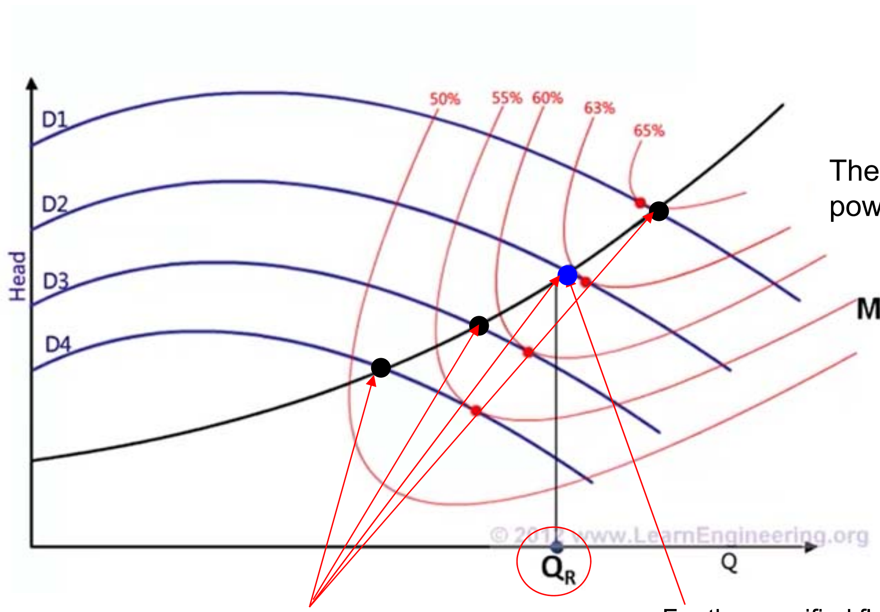




The required shaft power input to the pump $= \dfrac{\rho g Q h}{\eta}$

**MOTOR SELECTION**

Possible operating points,

For the specified flow rate $Q_R$, D2 is selected, η = 63 %

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**MOTOR SELECTION**

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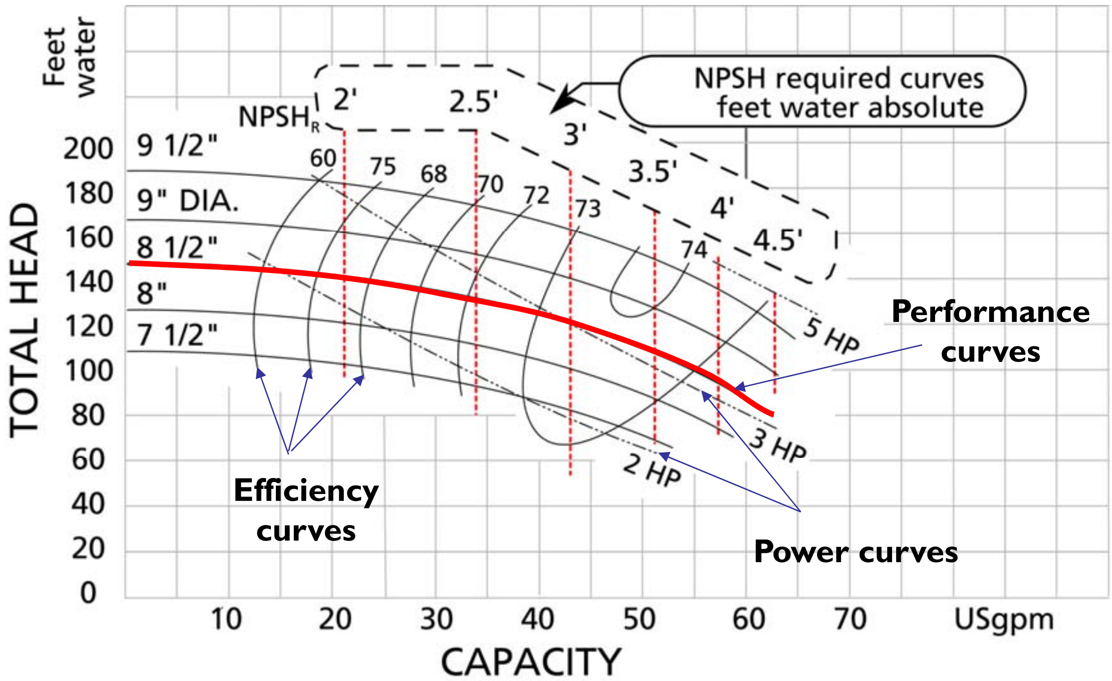

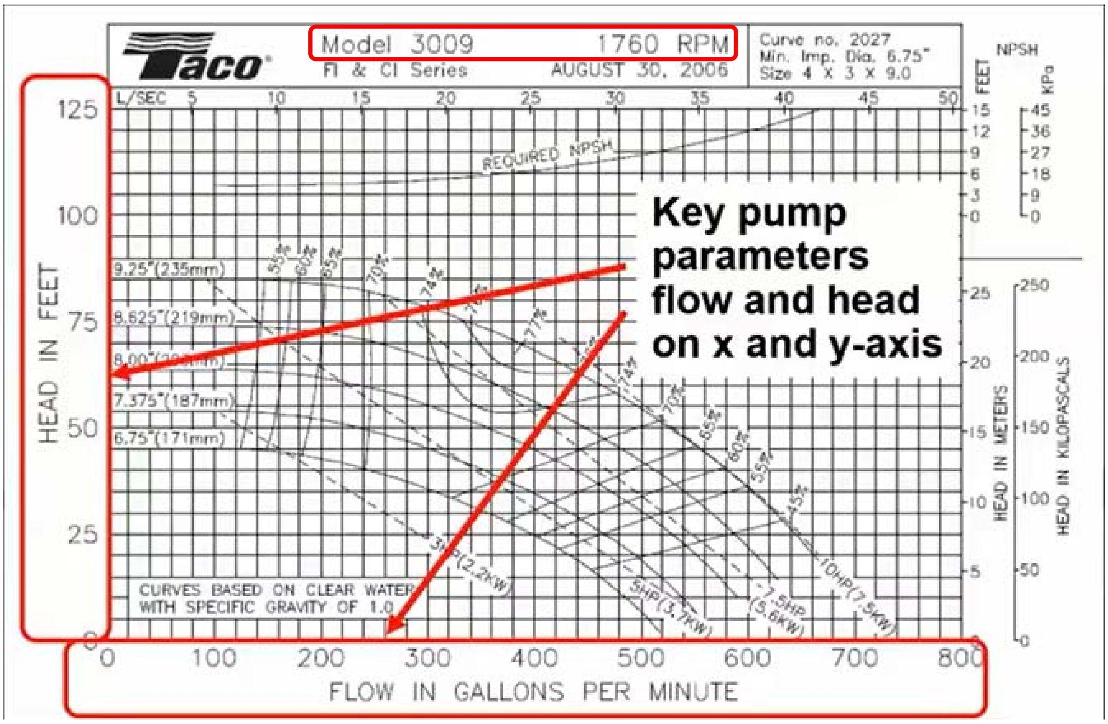




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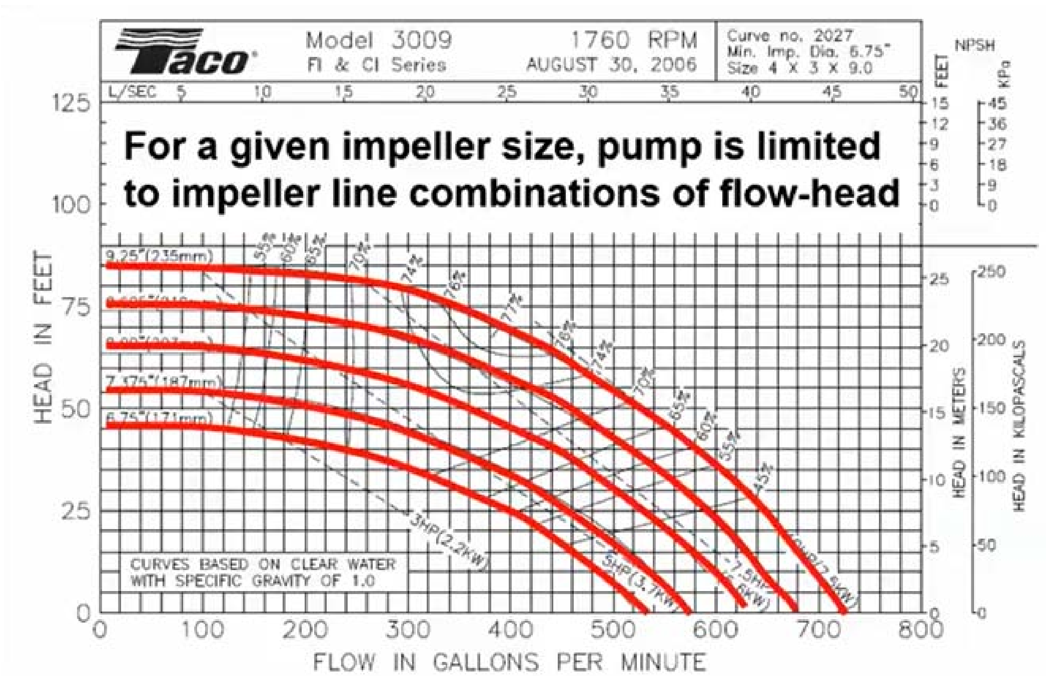
- A single pump model is available with several impeller sizes



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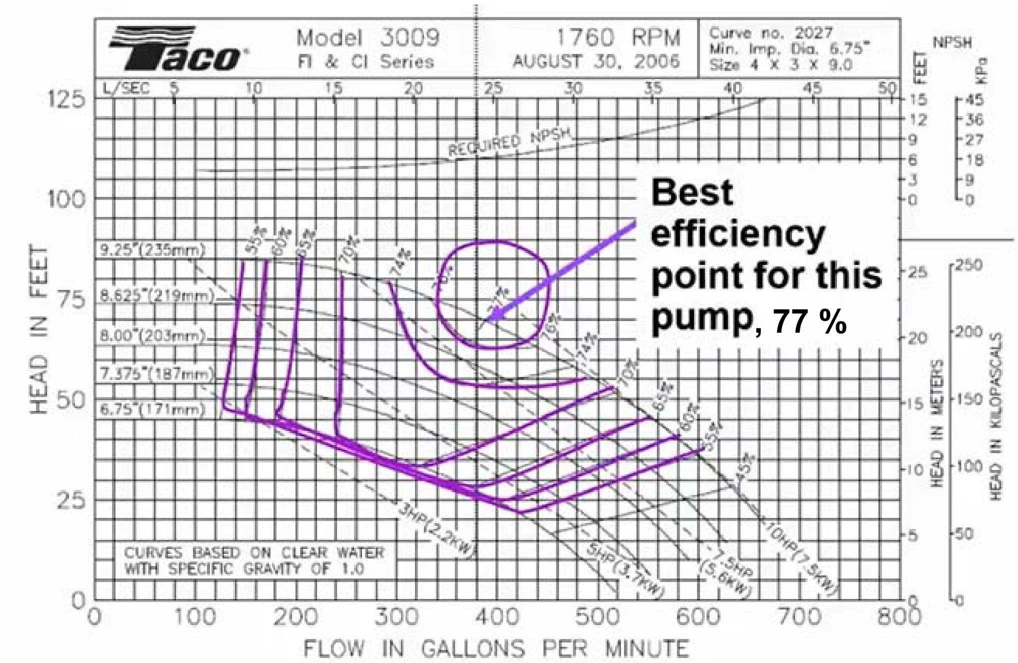
- Pump efficiency depends on where on the flow-head location



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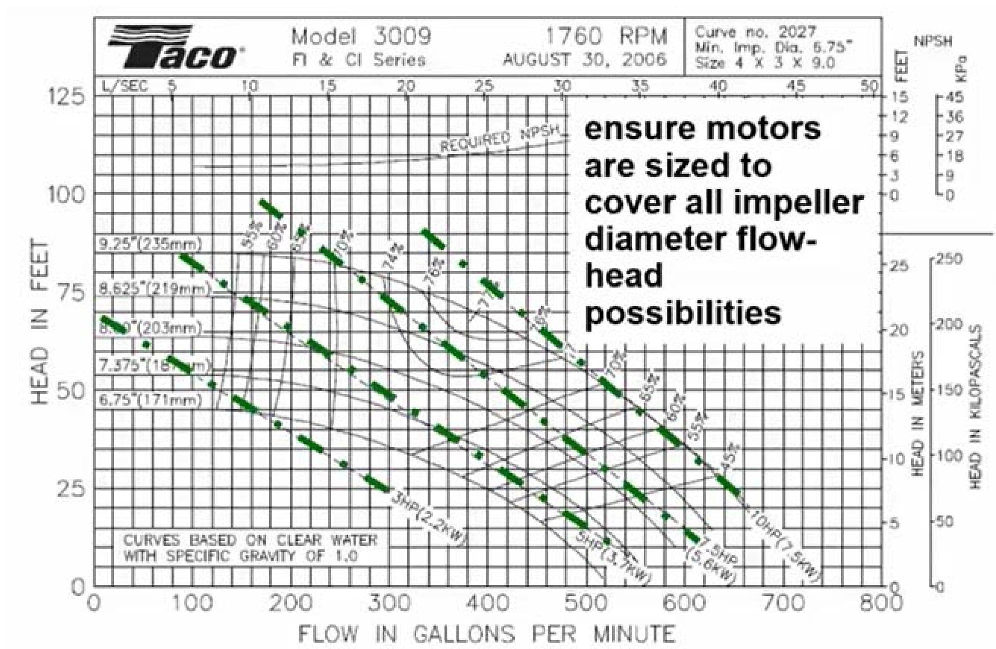
- Lines of HP indicate the Brake horsepower (provided by motor) and required to operate the pump at a specific flow-head location



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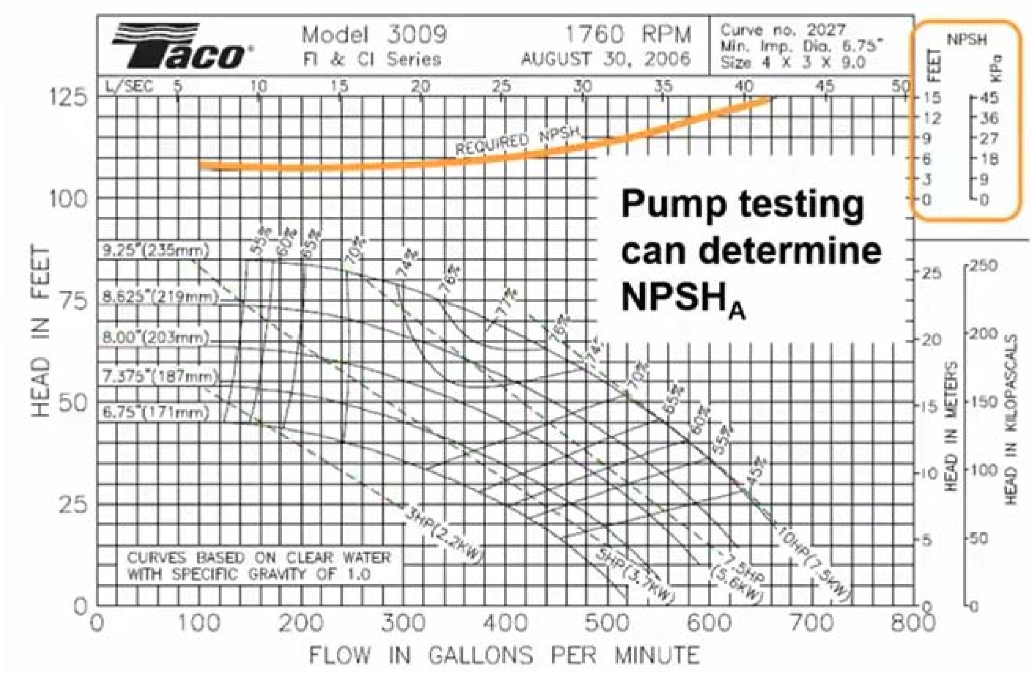
- The $NPSH_R$, indicates the minimum inlet pressure needed to prevent cavitation at a given flow rate (the higher the flow rate the higher the $NPSH_R$)



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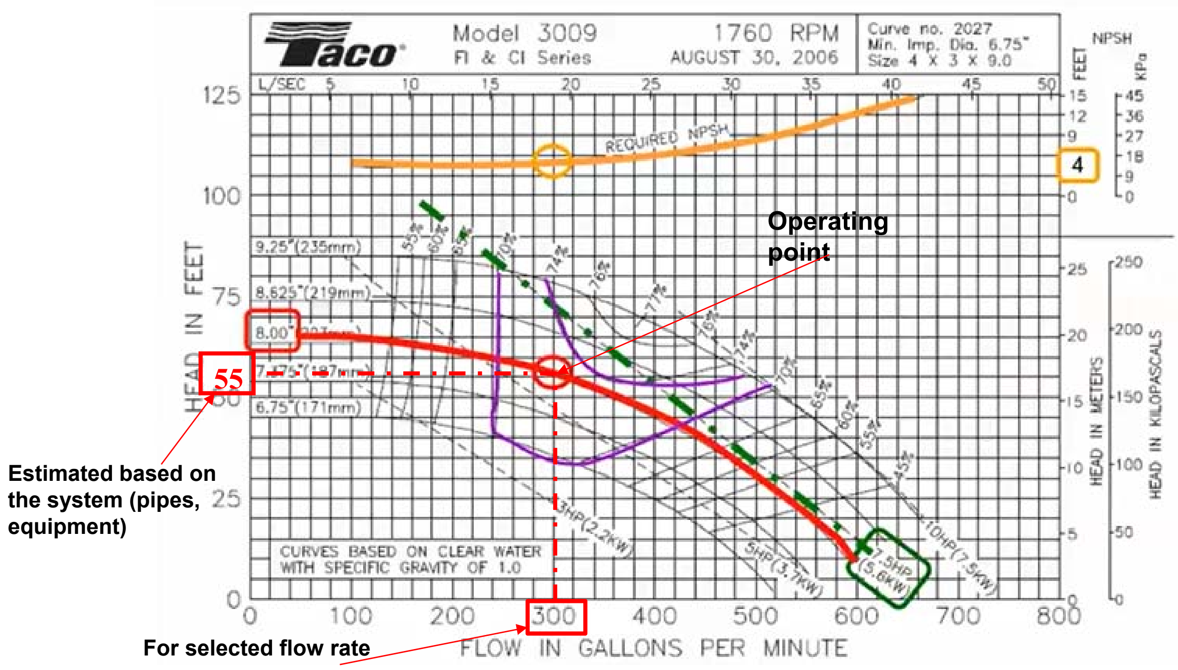


- In this case, an 8.0” impeller would be chosen, to operate at 73 % efficiency with a 7.5 HP motor and 4 ft $NPSH_R$



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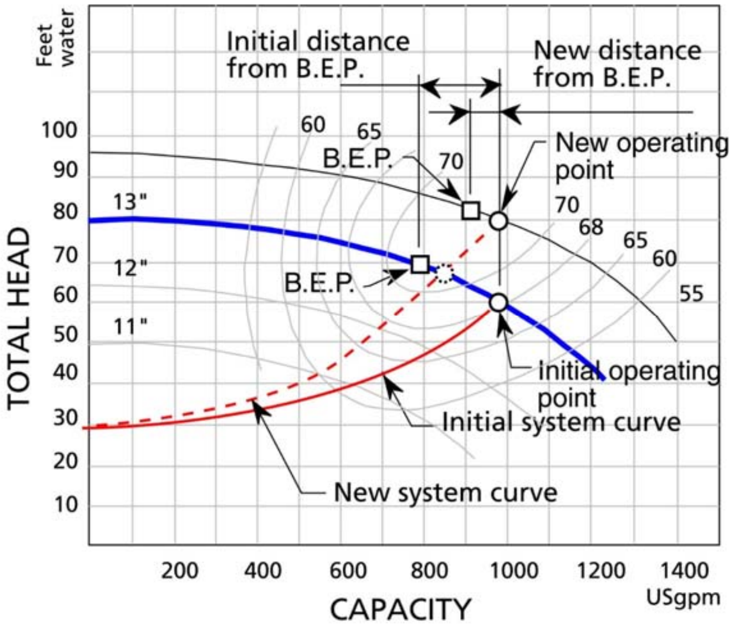


Why increase the impeller diameter when the flow must be increased?



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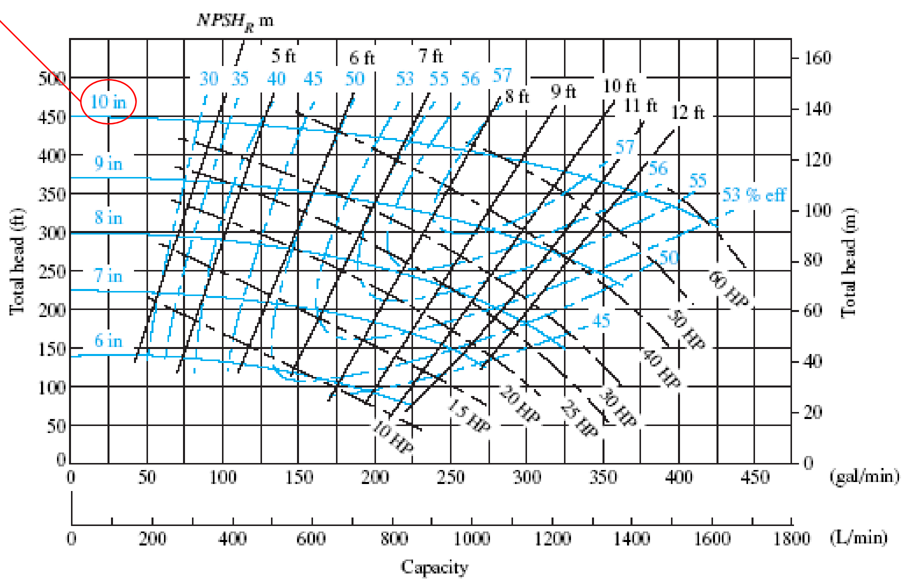
## Some characteristic Curves of Centrifugal Pumps

Impeller diameter




**Performance chart of 2×3-10 centrifugal Pump at 3500 rpm**



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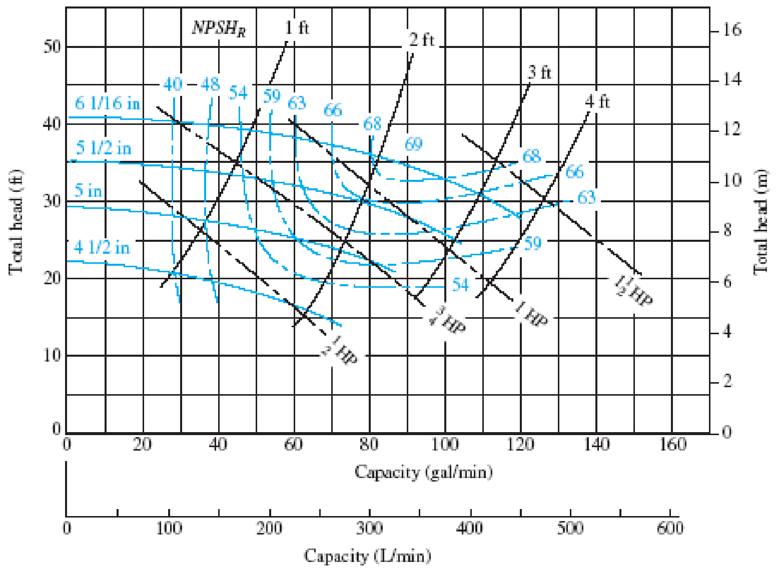
## Some characteristic Curves of Centrifugal Pumps




**Performance chart of 1.5×3-6 centrifugal Pump at 1750 rpm**



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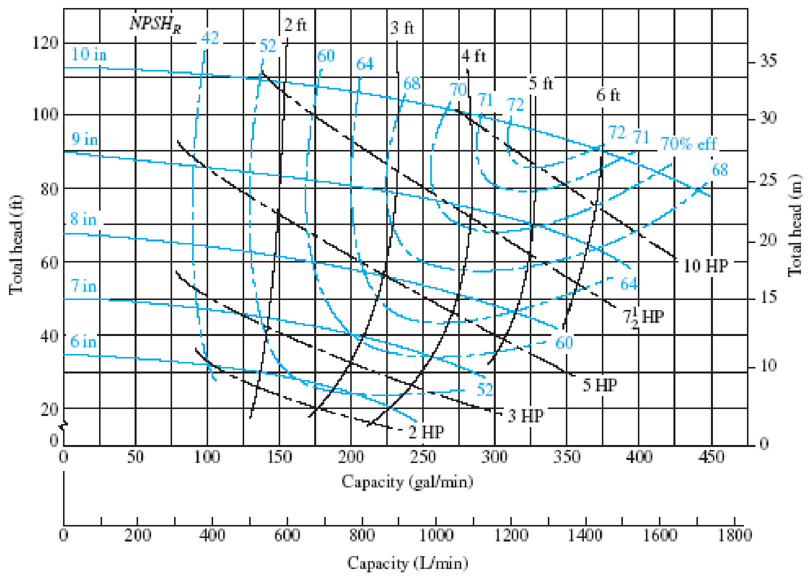
## Some characteristic Curves of Centrifugal Pumps




**Performance chart of 3×4-10 centrifugal Pump at 1750 rpm**



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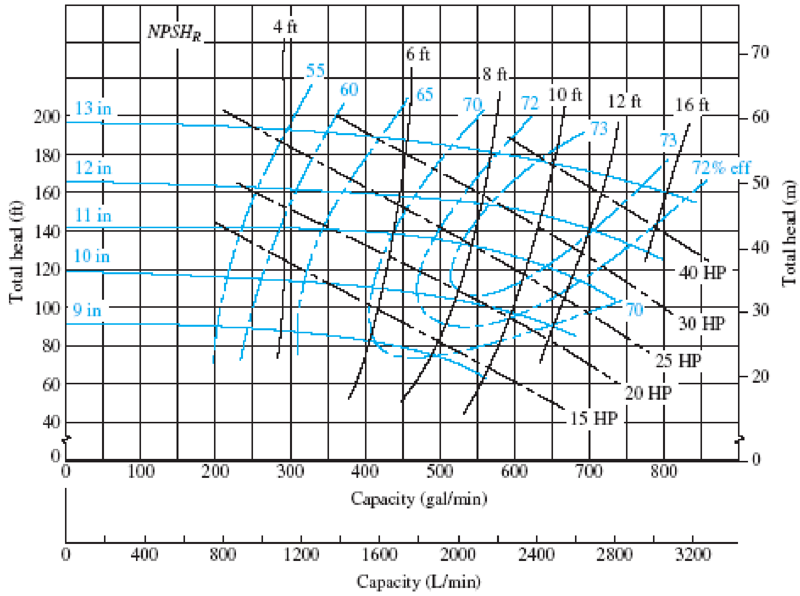
## Some characteristic Curves of Centrifugal Pumps




**Performance chart of 3×4-13 centrifugal Pump at 1750 rpm**



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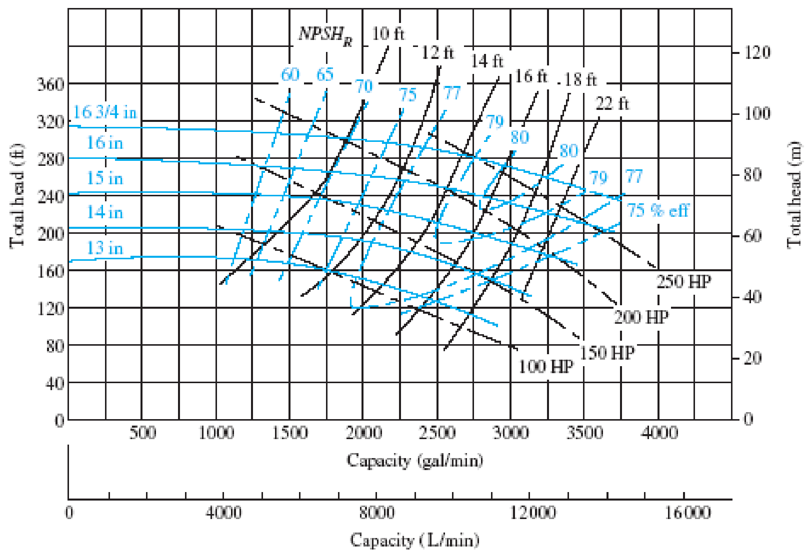
## Some characteristic Curves of Centrifugal Pumps




**Performance chart of 6×8-17 centrifugal Pump at 1780 rpm**



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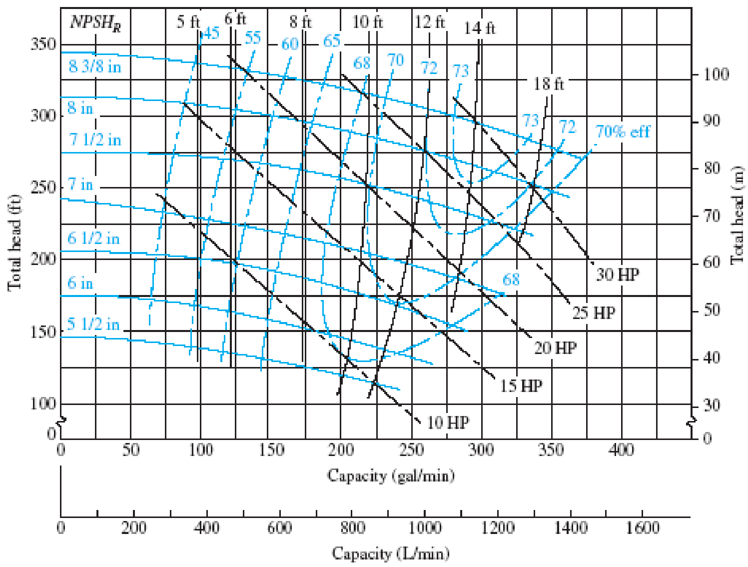
## Some characteristic Curves of Centrifugal Pumps




**Performance chart of 2×3-8 centrifugal Pump at 3560 rpm**



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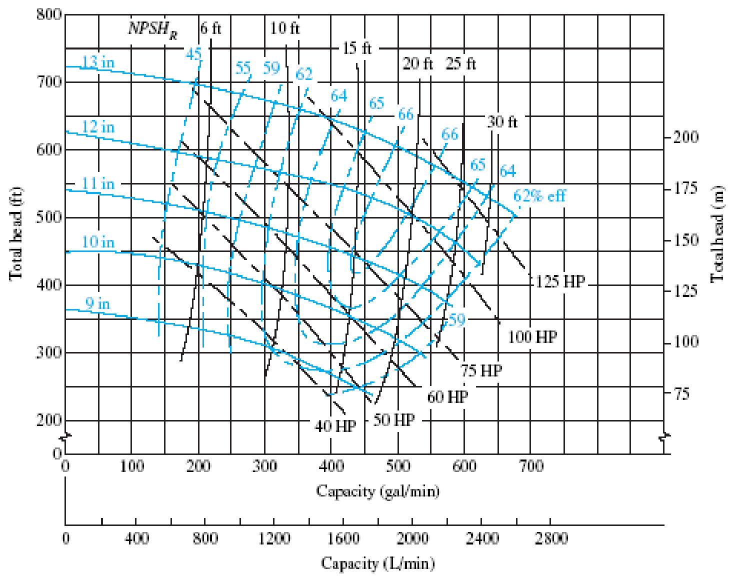
## Some characteristic Curves of Centrifugal Pumps




**Performance chart of 1.5×3-13 centrifugal Pump at 3560 rpm**



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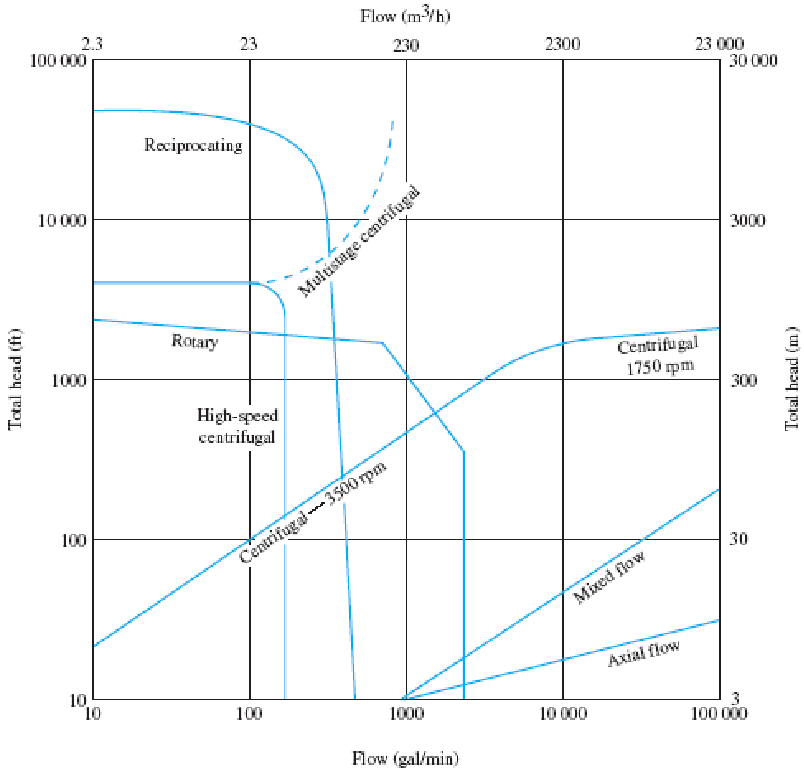
# Useful charts for pump selection:



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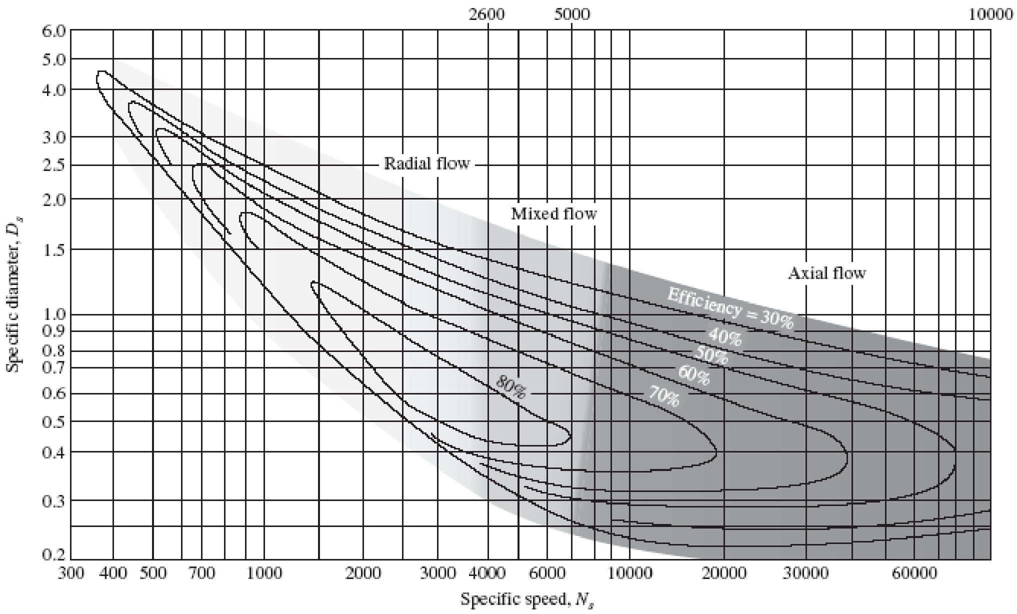
# Useful chart for kinetic pump selection:

Specific speed, $N_s$ $\left\{ N_s = \frac{N\sqrt{Q}}{H^{3/4}} \quad N = \text{Rev/min} \quad H = \text{Head, m} \quad Q = \text{Flow, Lit/s} \right\}$




$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$ $\quad D_s = \frac{D H^{1/4}}{\sqrt{Q}}$ $\quad N$ = Rev/min $\quad Q$ = Flow, U.S. gpm $\quad H$ = Head, ft $\quad D$ = Diameter, in



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