

Chapter Four

Hydraulic Machines

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- 7- Problems; sheet No. 4 (Turbines)

1- Introduction:

There are two basic types of hydraulic machines:

- Pumps: these machines add energy to fluids.
- Turbines: these machines convert the fluid energy into useful mechanical work.

2- Pumps:

Pumps are used to convert mechanical energy to fluid energy. In most of the cases pumps are used for raising fluids from a lower to a higher level. This is achieved by creating a low pressure at the inlet or suction end and high pressure at the outlet or delivery end of the pump.

Classification of pumps:

- a- Positive displacement pumps. (figures (1) through (3))
- b- Rotodynamic or (dynamic pressure pumps). (figures (4) and (5))
 - Centrifugal pump.
 - Axial pump.
 - mixed flow pump.

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cutaway of a lobe pump

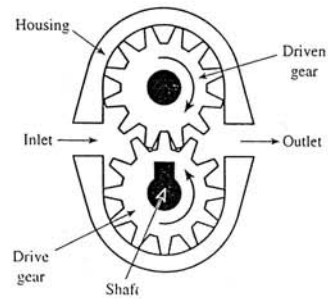
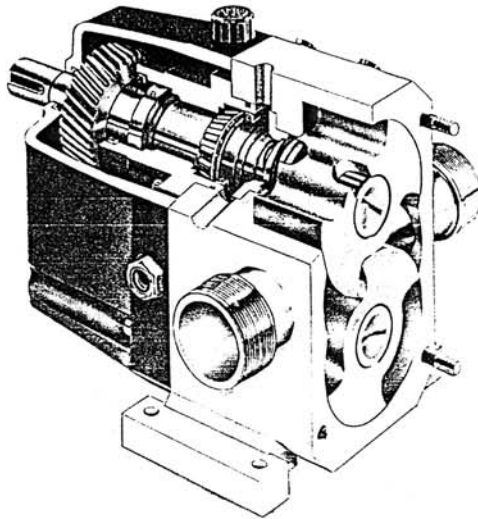
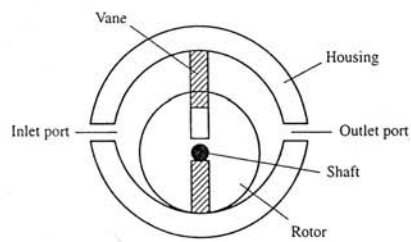


Fig.2 gear pump operation



Vane pump operation

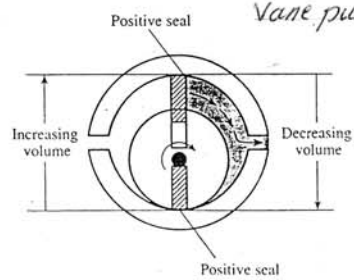


Fig.1 lobe and vane pumps

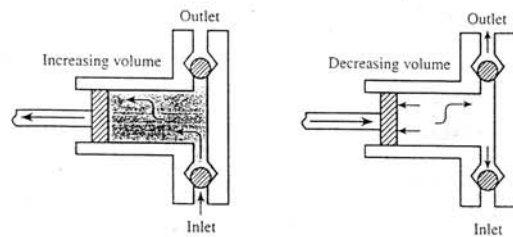
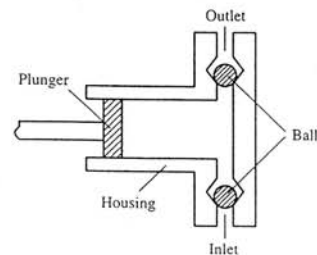


Fig.3 operation of a piston pump.

Positive displacement pumps

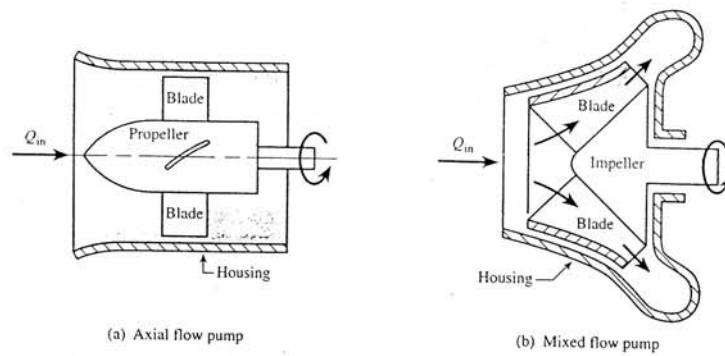


Figure 4 Axial flow and mixed flow dynamic pumps.

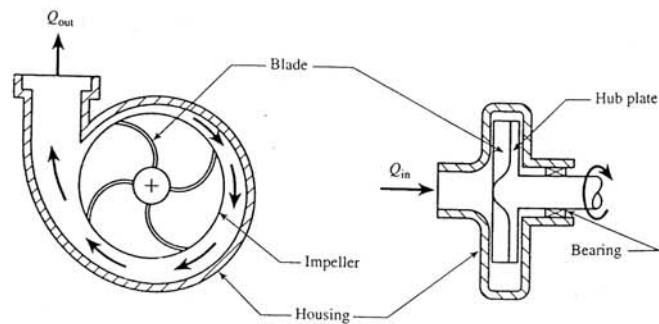


Figure 5 Construction features of a centrifugal pump.

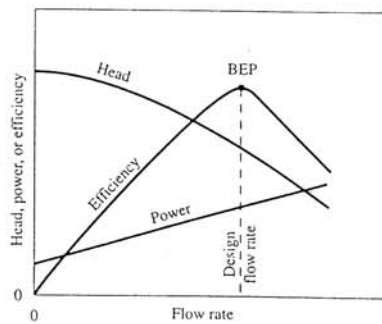


Fig. 6 Typical performance characteristics for centrifugal pumps.

Centrifugal pump:

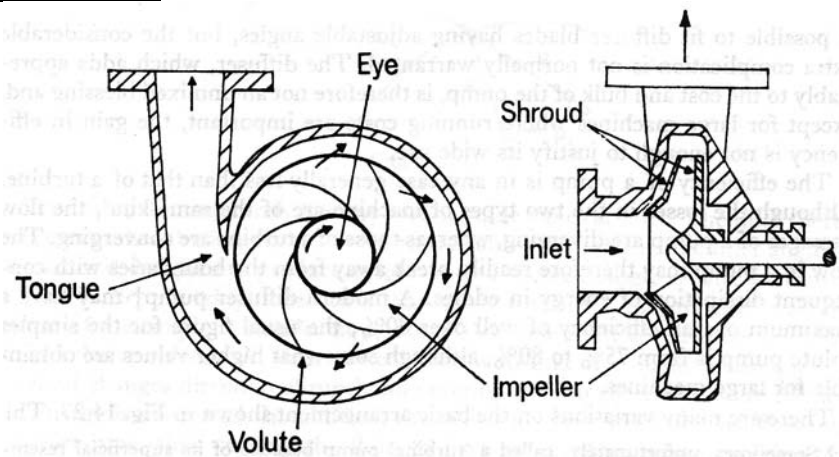
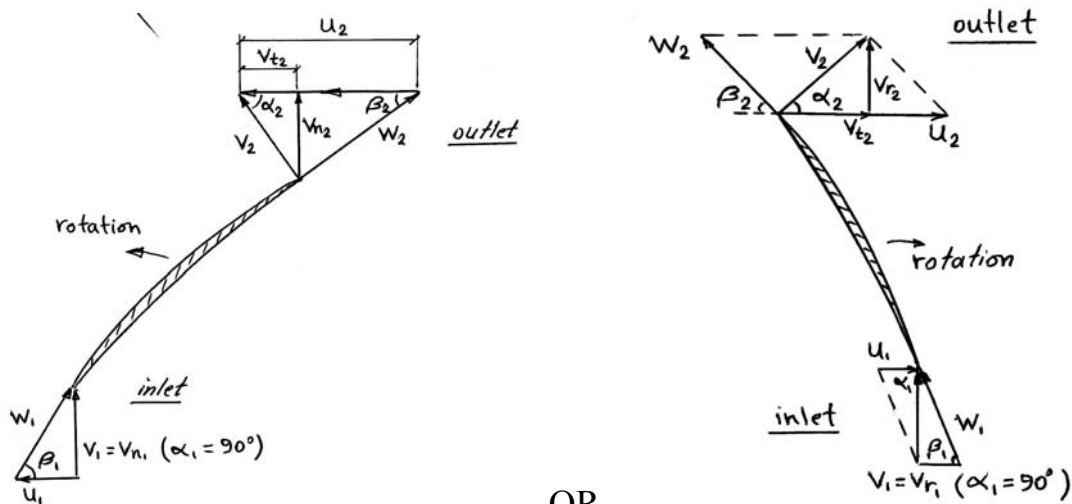


FIG. 14.27 Volute-type centrifugal pump

Consider the impeller of a centrifugal pump shown in figure, friction is neglected and the relative velocity of the fluid is always tangent to the blade.



OR

Where V = absolute fluid velocity.
 W = relative fluid velocity.
 u = peripheral velocity of the impeller.
 α = angle between u and V .
 β = blade angle.

For pump, the moment-of-momentum equation is:

$$T = \rho Q [(rV_t)_{out} - (rV_t)_{in}]$$

where T is the torque acting on the fluid.

$$T = \rho Q (r_2 V_2 \cos \alpha_2 - r_1 V_1 \cos \alpha_1)$$

The power is:

$$Power = \rho Q(u_2 V_2 \cos \alpha_2 - u_1 V_1 \cos \alpha_1)$$

The theoretical pump head is:

$$H_{th} = \frac{u_2 V_2 \cos \alpha_2 - u_1 V_1 \cos \alpha_1}{g}$$

The hydraulic (or manometric) efficiency of the pump:

$$\eta_h = \frac{H_{act}}{H_{th}}$$

$$\text{where } H_{act} = H_{th} - H_{losses}$$

The hydraulic losses (H_{losses}) include shock loss, fluid friction loss, and loss due to circulatory flow.

Overall efficiency:

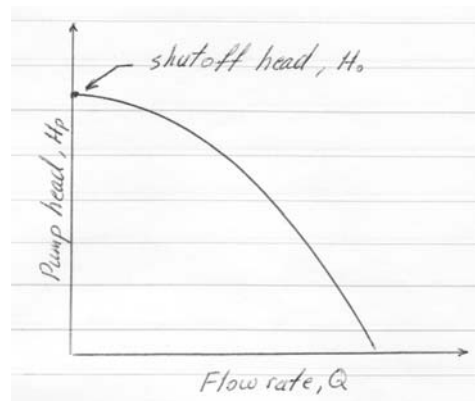
$$\eta = \frac{\rho Q H_{act}}{T \omega}$$

In addition to hydraulic losses, the efficiency of a pump is reduced by bearing and packing friction and by disk friction as well as by the leakage effect.

Head-Flow rate curves for centrifugal pumps:

The maximum head produced by a centrifugal pump is called:

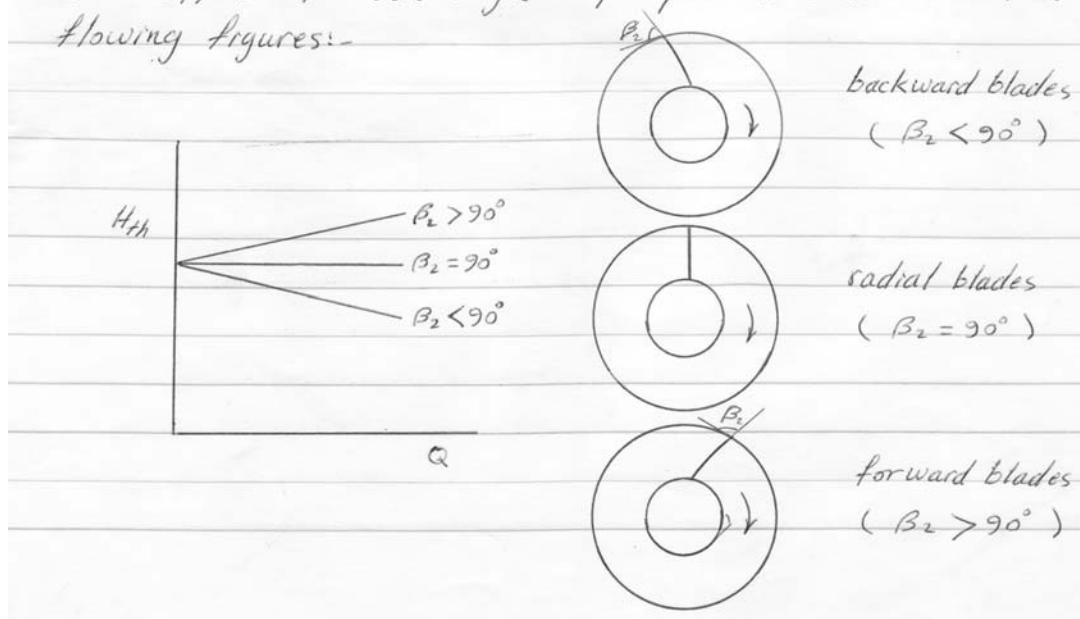
Pump shutoff head (H_0).



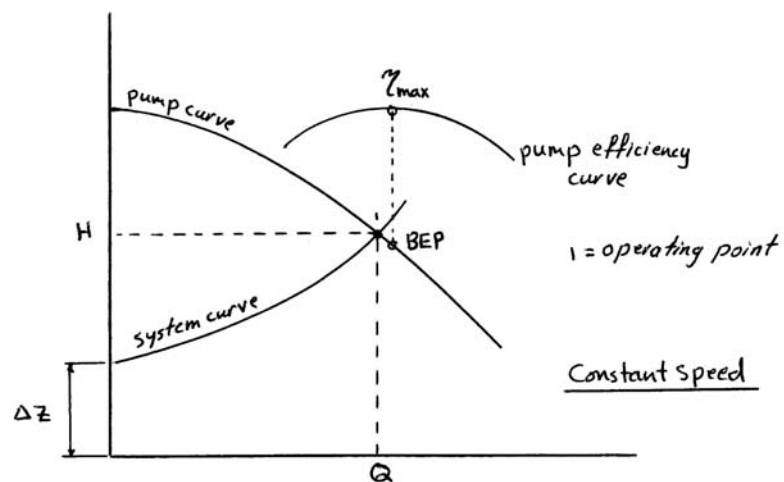
The equation for the (Q-H) curve can be written as:

$$H = H_0 - A Q^2 \quad \text{where } A = \text{costant}$$

* The effect of blade angle on pump head is shown in the following figures:-



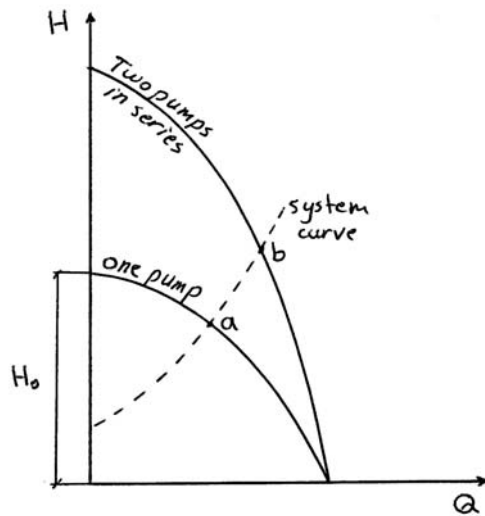
Operating point of a pump:




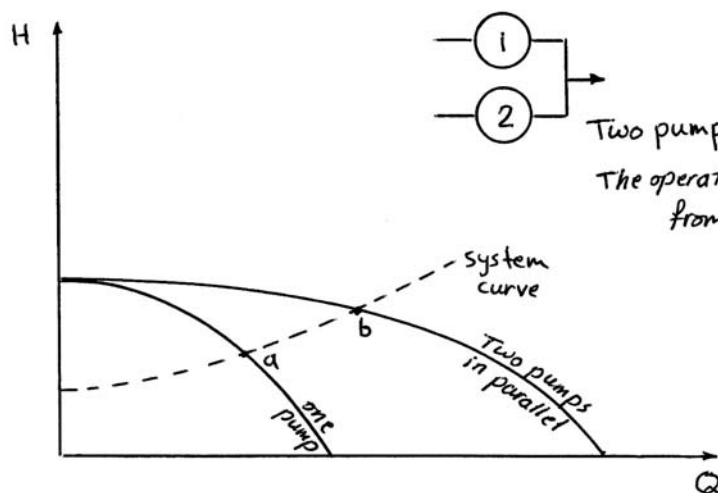
The actual pump operating head and flow rate are determined by the intersection of the pump and system curves.

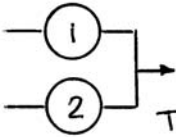
It is important to select a pump such that the intersection of the pump and system curves is near the BEP.

Pumps operating in series and in parallel:




Two pumps in series
The operating point shifts
from a to b




Two pumps in parallel
The operating point shifts
from a to b

The figures show the performance curves for two identical pumps in series and in parallel.

- Pumps in series tend to increase the head (H).
- Pumps in parallel tend to increase the flow rate (Q).

Similarity laws for centrifugal pumps:

Flow rate similarity law:
$$\left(\frac{Q}{ND^3} \right)_m = \left(\frac{Q}{ND^3} \right)_p$$

Head similarity law:
$$\left(\frac{gH}{N^2 D^2} \right)_m = \left(\frac{gH}{N^2 D^2} \right)_p$$

Input power similarity law:
$$\left(\frac{power}{\rho N^3 D^5} \right)_m = \left(\frac{power}{\rho N^3 D^5} \right)_p$$

Overall efficiency similarity law:
$$\eta_m = \eta_p$$

where: p= prototype ; m= model ; N= pump speed ; D= impeller dia.

Specific speed (N_s):

In SI system:
$$N_s (SI) = \frac{N\sqrt{Q}}{(gH)^{\frac{3}{4}}}$$

where N in (rad/s)
Q in (m³/s)
g in (m/s²)
H in (m)

In the US customary system:
$$N_s (US) = \frac{N\sqrt{Q}}{H^{\frac{3}{4}}}$$

where N in (rpm)
Q in (gpm)
H in (ft)

The conversion factor between them is:
$$N_s (US) = 2730 N_s (SI)$$

Also:
$$N_s (Eur) = \frac{N_s (US)}{17100}$$

The specific speed of a series is usually defined for the point of best efficiency (BEP).

Note: when determining specific speed N_s for a multistage pump, use H_1 , the first stage head rather than the total head of the pump.

Multistage pumps:

General view:

In multistage pumps the liquid being transferred is passed successively through several stages.

Total head H developed by n stages is: $H = \sum_1^n H_i$

where H_i = head developed by a stage.

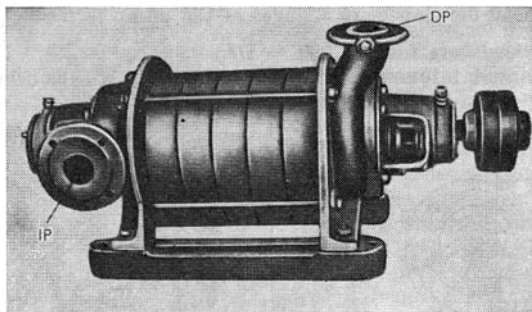
If the stages are identical then: $H = nH_i$

All the impellers of the multistage pumps are similar in design. However, in some cases the first stage impeller is made with double entry to improve cavitation characteristics and increase permissible suction lift of the pump.

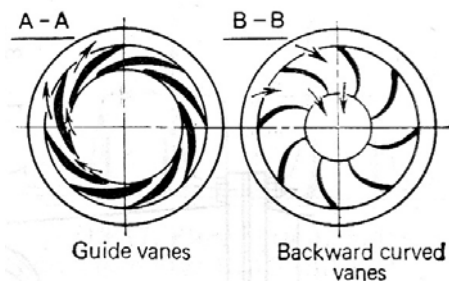
Types of multistage pumps:

1- Multistage chamber pump.

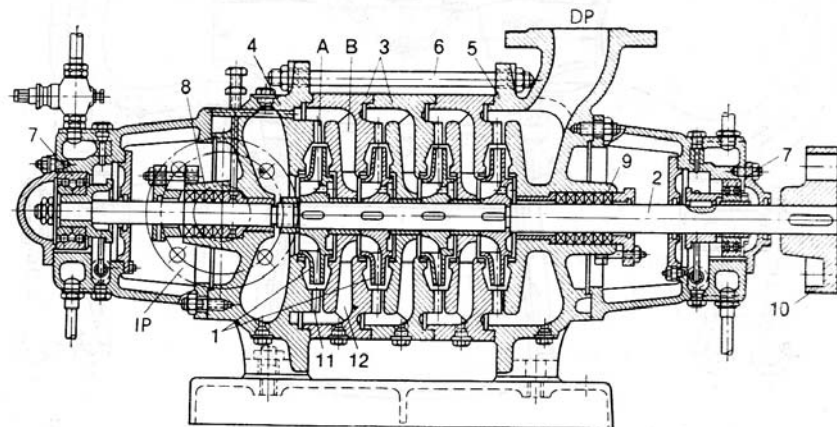
The external and axial cross sectional views of this type are shown in the following figures. An important element of this type is a guide vane assembly which passes water to the inlet of the next stage (sections A-A and B-B shown in figures). The assembly is intended to minimize hydraulic losses and ensure axial entry of the flow to the impeller (without swirling).



Multistage chamber pump, external view

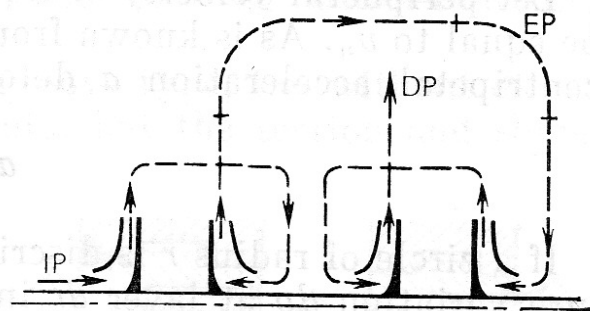
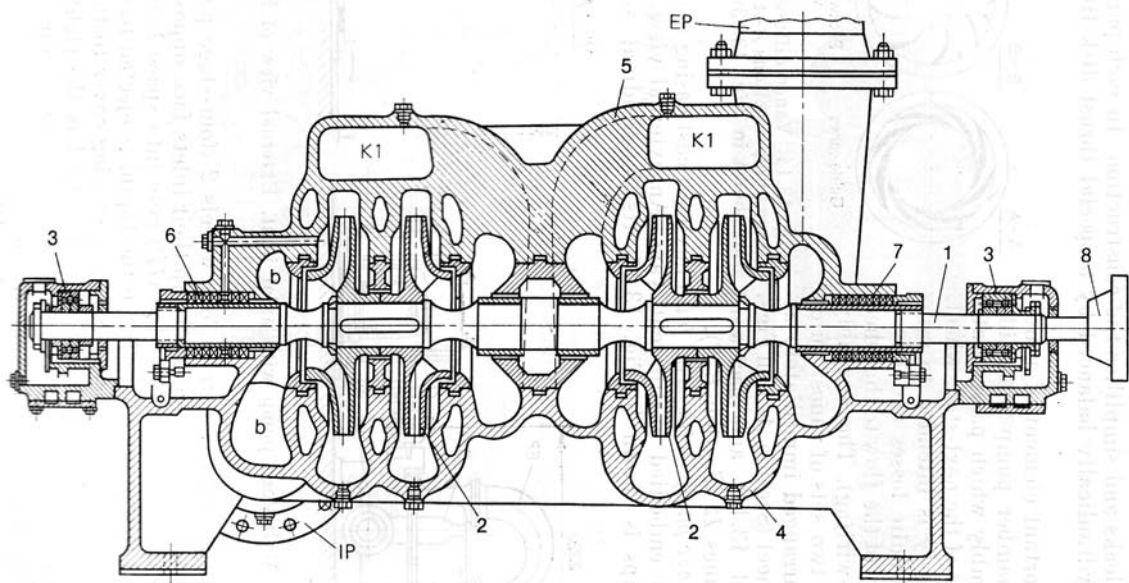
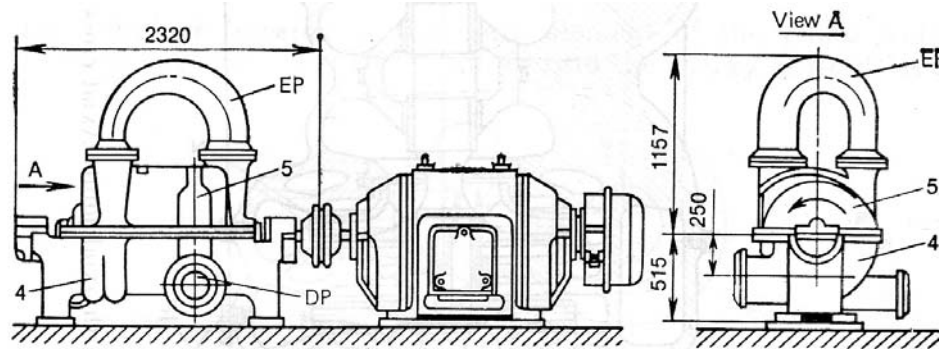


Guide vane assembly



Multistage chamber pump, axial cross sectional view

The external and axial cross sectional views of this type are shown in the following figures. This type have a cast casing made of two halves are embodied in different ways. The liquid flow path is shown in figure.



Pump selection:

Each pumps manufacturer uses its own system for designating pumps. Choice of a manufacturer may be based on established practice, location, or cost. Once a manufacturer is chosen, pump selection is a four-step process as will be shown in the following example:

Example: Select a pump to deliver $0.11 \text{ m}^3/\text{s}$ of water at 36.6 m total head. Choose the appropriate model and driver speed. Specify the pump efficiency, driver power, and NPSH requirement.

Solution:

Assume the selection is to be made from the full-line catalog of the Peerless Pump Company (USA).

The required delivery and head in U.S customary system are:

$Q = 1750 \text{ gpm}$, $H = 120 \text{ ft}$

Step 1: *select a pump type suited to the application.* The Peerless product line catalog specifies a maximum delivery and head of 2500 gpm and 660 ft for series AE pumps. Assume the selection is to be made from this series.

Step 2: *use selector charts.* The desired operating point is not within any pump contour on the 3500 rpm selector chart (Fig D.1). From the 1750 rpm chart (Fig D.2) select a model 6AE14 pump. From the performance curve for 6AE14 pump (Fig D.6) choose a 13 in. impeller.

Step 3: *use performance charts.* On the performance chart for the 6AE14 pump, project up from $Q = 1750 \text{ gpm}$. Project across from $H = 120 \text{ ft}$ on the ordinate. The intersection is the pump performance at the desired operating point. ($\eta \approx 85.8\%$, power $\approx 64 \text{ hp}$)

From the operating point, project down to the NPSH curve. At the intersection, read $(\text{NPSH})_R \approx 17 \text{ ft}$.

Step 4: *verify* that the system operating condition has been predicted accurately and the pump has been selected correctly.

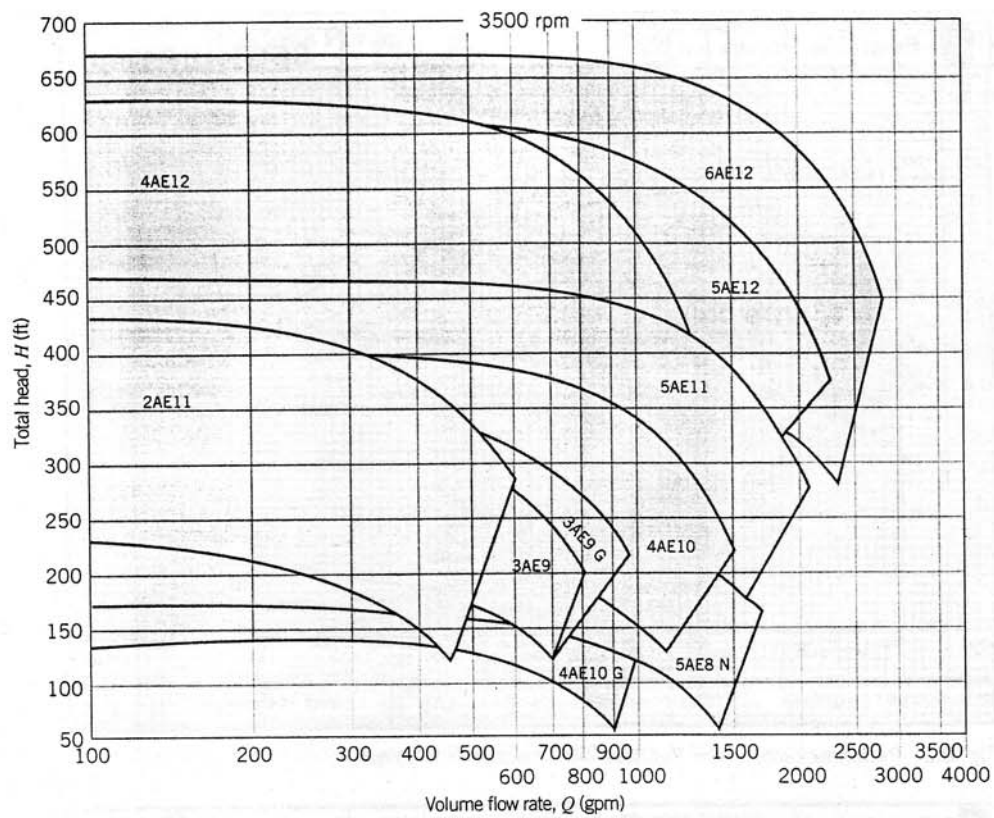


Fig. D.1 Selector chart for Peerless horizontal split case (series AE) pumps at 3500 nominal rpm.

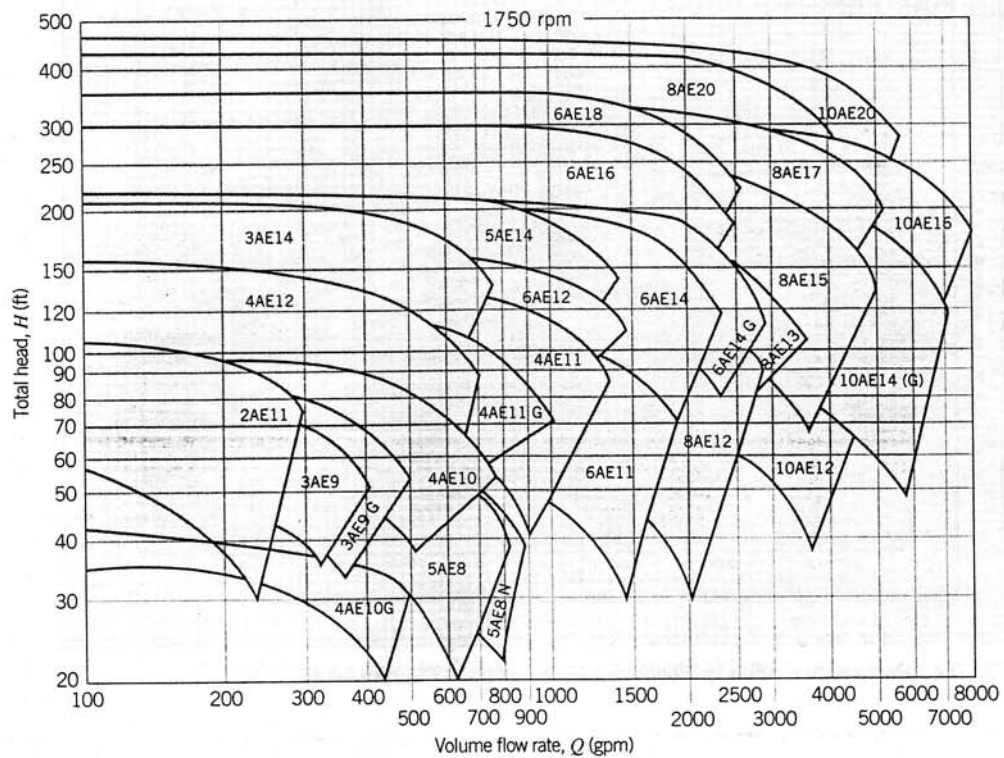


Fig. D.2 Selector chart for Peerless horizontal split case (series AE) pumps at 1750 nominal rpm.

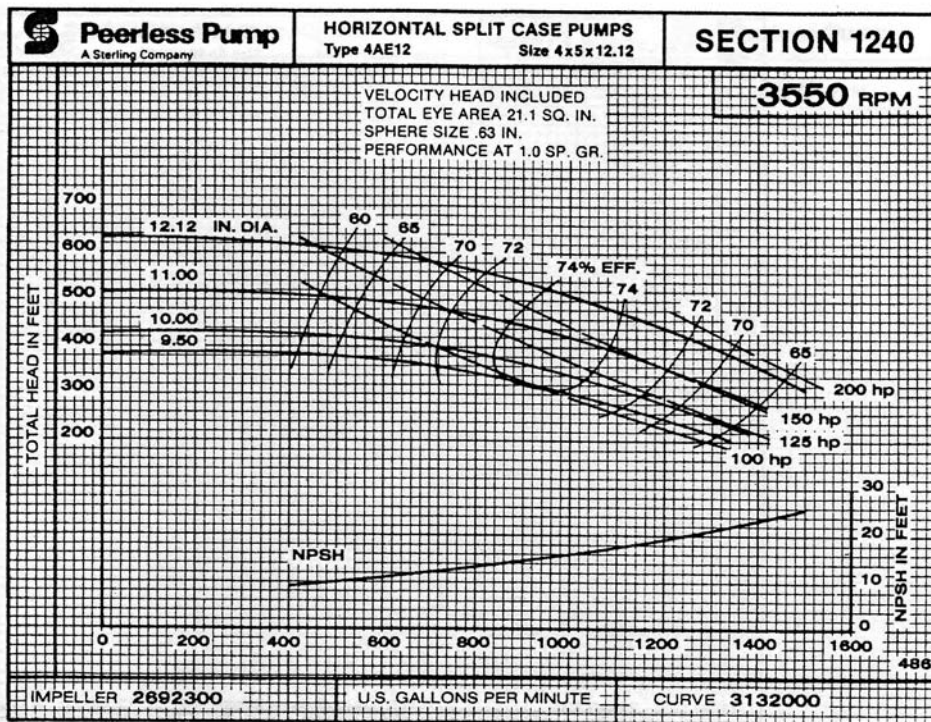


Fig. D.5 Performance curve for Peerless 4AE12 pump at 3550 rpm.

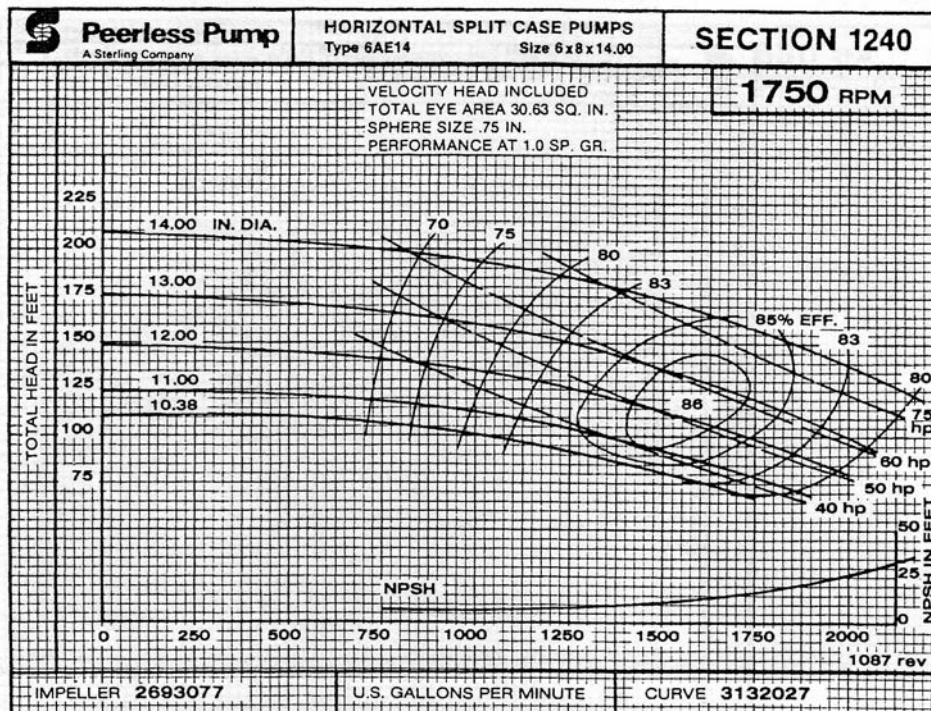


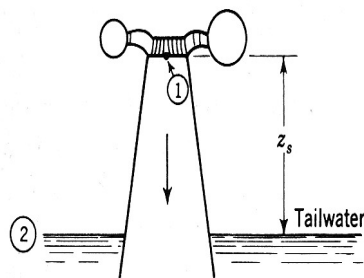
Fig. D.6 Performance curve for Peerless 6AE14 pump at 1750 rpm.

3- Turbines:

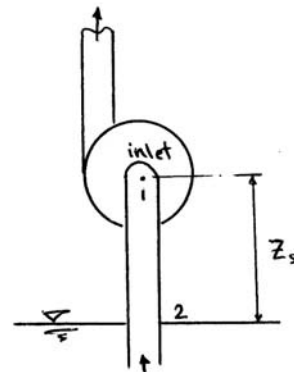
4- Cavitation in hydraulic machines:

When a liquid flows into a region where its pressure is reduced to vapor pressure, it boils and vapor pockets develop in it. The vapor bubbles are carried along with the liquid until a region of higher pressure is reached, where they suddenly collapse. This process is called cavitation.

Cavitation parameter (σ):



Turbine



Pump

$$\sigma = \frac{p_{atm} - p_v - \gamma Z_s - h_{losses}}{\gamma H} \quad \text{cavitation parameter}$$

where:

p_{atm} = atmospheric pressure.

p_v = absolute vapor pressure.

Z_s = as shown in above figures.

H = net head across the machine.

For cavitation not to occur at (1) the pressure must be greater than the vapor pressure of the liquid.

or $\sigma > \sigma_c$ where σ_c is the critical cavitation parameter.

Net Positive Suction Head (NPSH):

$$(NPSH)_A = \frac{p_{atm} - p_v - \gamma Z_s - h_{losses}}{\gamma}$$

where $(NPSH)_A$ = available net positive suction head.

Pump manufacturers specify a required value of net positive suction head $(NPSH)_R$.

To avoid cavitation: $(NPSH)_A \geq (NPSH)_R$

5- Examples:

1- A centrifugal pump impeller has an outside diameter of 200 mm and rotates at 2900 rpm. Determine the actual head generated if the vanes angle at outlet is 25° and the radial component of flow velocity is constant through out the impeller is 3 m/s. Assume hydraulic efficiency as 75%.

Solution:

$$u_2 = \frac{\pi DN}{60} = 30.37 \text{ m/s}$$

$$H = \frac{u_2 V_2 \cos \alpha_2}{g} = \frac{u_2 V_{t2}}{g} \quad (*)$$

$$V_{t2} = u_2 - \frac{V_{r2}}{\tan \beta_2} = 23.94 \text{ m/s}$$

from equation (*)

$$H = \frac{30.37 \times 23.94}{9.81} = 74.11 \text{ m}$$

$$\eta_h = \frac{H_{act}}{H_{th}}$$

$$H_{act} = 0.75 \times 74.11 = 55.5 \text{ m} \quad (\text{Ans})$$

2-

Solution:

1- A centrifugal water pump has an impeller with $d_1=0.2$ m, $d_2=0.6$ m, $\beta_1=20^\circ$, $\beta_2=10^\circ$, $b_1=50$ mm, $b_2=19$ mm. For 1800 rpm, neglecting losses and blade thickness, determine:
a- the discharge for shock less entrance when $\alpha_1=90^\circ$. b- α_2 and H_{th} . c- the power required.
d- the pressure rise through the impeller.

[0.216 m³/s ; 15.11° ; 129 m ; 273.4 kW ; 1030 kN/m²]

2- A centrifugal water pump with impeller diameters $d_1=102$ mm, $d_2=254$ mm, and $b_1=76.2$ mm, $b_2=38.1$ mm, $\beta_2=60^\circ$ is to pump 0.142 m³/s at 19.5 m head. Determine: β_1 , the speed, and the power. Neglect losses and assume no shock at the entrance ($\alpha_1=90^\circ$).

[43.56° ; 1146 rpm ; 27.2 kW]

3- The impeller of centrifugal pump has an outer diameter of 25 cm and an effective outlet area of 170 cm². The blades are bent backward so that the direction of the outlet relative velocity makes an angle of 148° with the tangent drawn in the direction of impeller rotation. The diameters of the suction and delivery pipes are 15 cm and 10 cm respectively. The pump delivers 1860 l/min at 1450 rpm. The gauge points on the suction and delivery pipes close to the pump show heads of 4.6 m below and 18 m above atmosphere respectively. The head losses in the suction and delivery pipes are 2 m and 2.9 m respectively. Find the manometric efficiency assuming that water enters the pump without shock and whirl.

[75%]

4- A three-stages centrifugal pump has impeller 40 cm in diameter and 2 cm wide. If the blade angle at outlet is 45° and the area occupied by the thickness of the vanes may be assumed 8% of the outlet area. If the pump delivers 3.6 m³/min when running at 920 rpm, Calculate: a- power of the pump. b- the head. c- specific speed. Assume mechanical efficiency as 88% and manometric efficiency as 77%.

[65.75 kW ; 75.69 m ; $N_s(\text{U.S.})=1033$]

5- Tests of a 0.57 m diameter centrifugal water pump at 2134 rpm yield the following data:

Q (m ³ /s)	0	0.054	0.108	0.162	0.216	0.27
H (m)	102	102	102	99	90	66
Power (kW)	100.6	119.3	153	190	246	246

What is the BEP? What is the specific speed? Estimate the maximum discharge possible.

[0.162 m³/s ; $N_s(\text{U.S.})=1412$]

6- A pump with a critical value of σ of 0.2 is to pump against a head of 20 m. The barometric pressure is 98.5 kPa abs, and the vapor pressure is 5.2 kPa abs. Friction losses from the reservoir to the pump are 0.5 m. Find the maximum allowable height of the pump relative to the water surface in the reservoir.

[no more than 5 m]

7- A centrifugal pump is installed above an open water tank. The pump is to provide a flow rate of $0.015 \text{ m}^3/\text{s}$. Under this flow condition, the pump manufacturer specifies $(\text{NPSH})_R$ of 5 m. Determine the maximum elevation that the pump can be installed above the water free surface without pump cavitation. The head loss h_l in the 100 mm dia. suction pipeline is due to a pump inlet strainer having a loss coefficient of 15. Other head losses are negligibly small, $p_{\text{atm}} = 101 \text{ kPa}$, $p_v = 2.3 \text{ kPa}$. [no more than 2.28 m]

8- A centrifugal pump with a 91.5 cm impeller dia. operates at 800 rpm. If the speed is to be increased to 1200 rpm, determine the impeller diameter that should be used so that the same shaft input power would be required. [71.7 cm]

9- A 30 cm diameter centrifugal water pump is geometrically similar to a 38 cm diameter pump. The larger pump has the following test performance characteristics at the best efficiency point: speed=800 rpm, pump head=15 m, flow rate= $0.126 \text{ m}^3/\text{s}$, eff.=80%, input shaft power=23.6 kW. Determine the following for the smaller pump operating with water at 1000 rpm: a- flow rate, b- pump head, c- input power, d- maximum efficiency. [$0.0775 \text{ m}^3/\text{s}$; 14.6 m ; 14.14 kW ; 80%]

10- The 330 mm dia. centrifugal pump whose performance curves for water are given in figure (1) has been selected to operate in the fluid system of figure (2). The following data are given for this system:

Pipe dia.= 152.5 mm

Total pipe length= 91.5 m

Suction pipe length= 7.6 m

Pipe friction factor= 0.03

Loss coefficients $(K_{\text{strainer}} + K_{1 \text{ elbow}}) = 3$
and $(K_{\text{valve}} + K_{2 \text{ elbow}}) = 1.5$

Elevation difference $(Z_2 - Z_1) = 4.6 \text{ m}$

Elevation difference $(Z_3 - Z_1) = 1.55 \text{ m}$

Determine: _

a- the pump flow rate.

b- the pump efficiency.

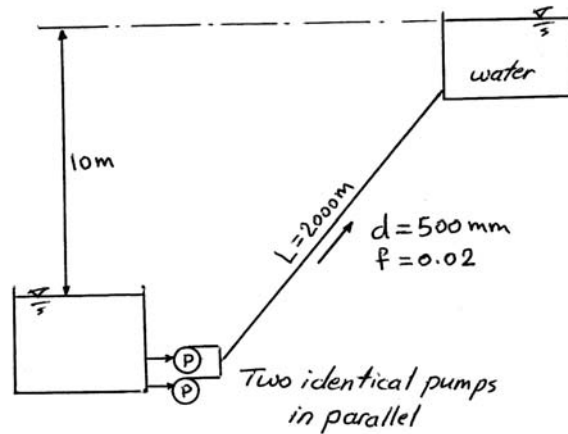
c- whether or not the selected pump is suitable for this application.

(take: $p_{\text{atm}} = 101 \text{ kPa}$, $p_v = 2.3 \text{ kPa}$)

d- the input power required to drive the pump.

[$54.25 \times 10^{-3} \text{ m}^3/\text{s}$; 74% ; suitable ; 10.5 kW]

11- A centrifugal pump has the following approximate expression for the characteristic curve: $H=24.4-470.66Q^2$ where H in (m) and Q in (m^3/s). Find the discharge through the system of figure for two pumps in parallel. (Note: minor losses in the system can be neglected). [0.254 m^3/s]



12- Find the discharge through the system shown in figure of problem (11) for pumps A and B in series; ($L=1600$ m, $d= 300$ mm, $H= 30$ m, $f= 0.02$). Neglect minor losses in the system. [79.5 l/s]

Pump A:

Q (l/s)	0	56.6	72.5	85.8	108	116	127	134
H (m)	21.3	18.3	16.8	15.2	12.2	10.7	9.1	6.1

Pump B:

Q (l/s)	0	74	112	140	161	174	177
H (m)	24.4	21.3	18.3	15.2	12.2	9.1	6.1

* Approximate expressions for the characteristic curves:

Pump A: $H=21.3-787.75Q^2$

Pump B: $H=24.4-470.66Q^2$

where H in (m) and Q in (m^3/s).

Problem No. 10

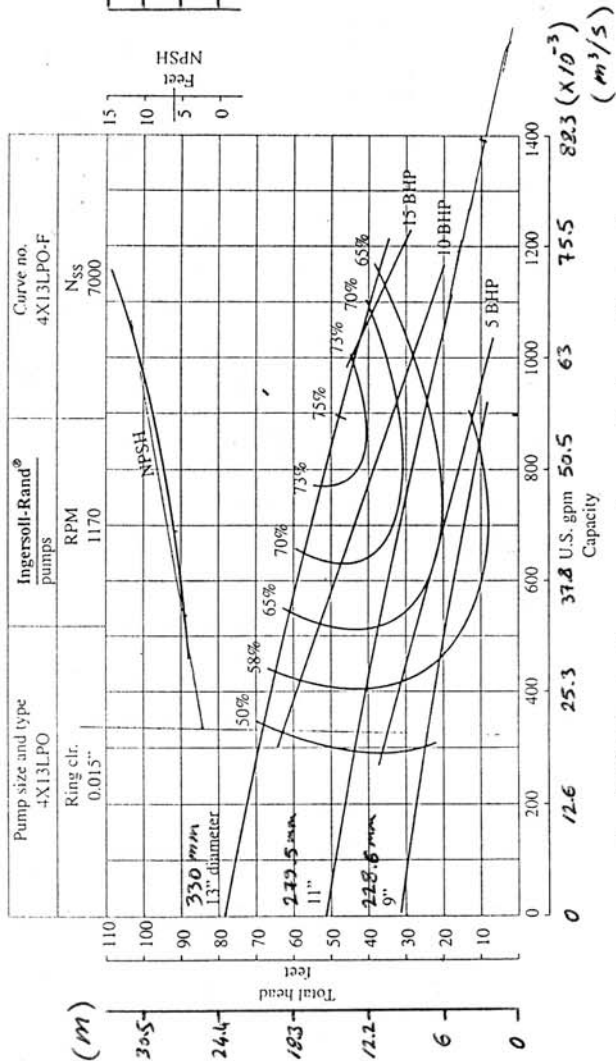


Figure (1) Actual performance curves for a centrifugal pump. (Courtesy of Ingersoll-Dresser Pump Co., Phillipsburg, New Jersey.)

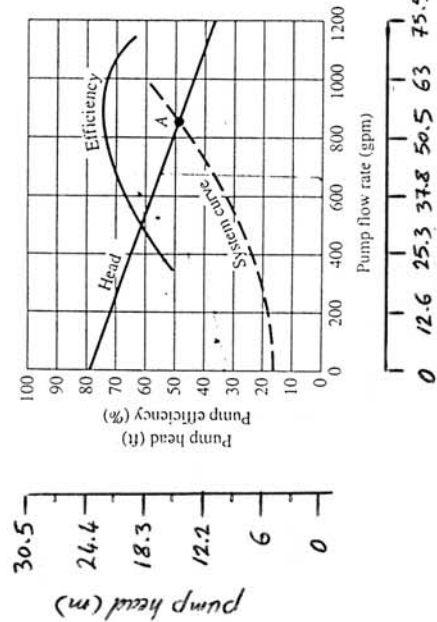


Figure (3)

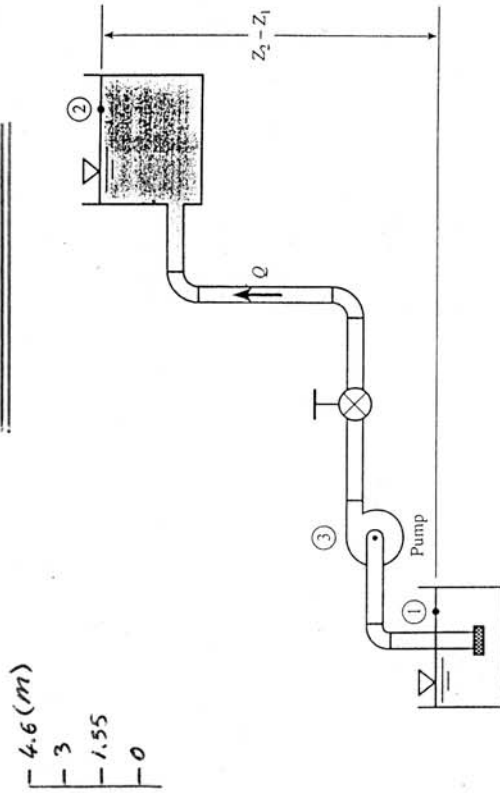


Figure (2) Typical fluid system with a centrifugal pump.

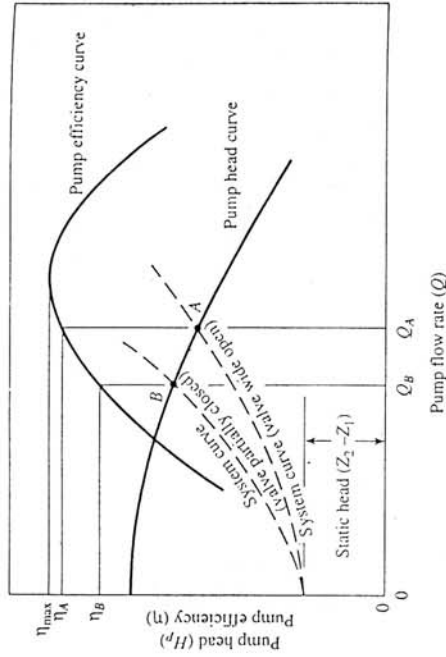


Figure (4) Intersection of the system curve with the pump head curve gives the operating point of the system.