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HYDRAULICS

HYDRAULIC MACHINERY

TOPIC 5.1- 5.4

by

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HYDRAULIC MACHINERY

5.1
- Introduction to hydraulic machinery

5.2
- Classifications of Pumps:
  - Positive Displacement and Rotodynamic

5.3
- Pump: Description of a Centrifugal Pump

5.4
- Pump Operating In Combination:
  - Single, In Series and In Parallel

5.1 : INTRODUCTION TO HYDRAULICS MACHINES

5.1.1 : Pump

- ❖ Pump are used to transfer fluid in a system, either at the same level or to a new height.
- ❖ The flow rate depends on the height to which the fluid is pumped and the relationship between "head" and flow rate is called the " pump characteristic".

5.1.2 : Characteristics that should be considered when selecting a pump

- ➢ The nature of liquid to be pumped
- ➢ The required capacity (volume flow rate)
- ➢ The conditions on the suction (inlet) side of pump
- ➢ The conditions on the discharge (outlet) side of the pump
- ➢ The total head on the pump
- ➢ The type of system to which the pump is delivering the fluid
- ➢ The type of power source (electric motor, diesel engine, steam turbine)
- ➢ Space, weight and position limitation
- ➢ Environment condition
- ➢ Cost of pump purchase and installation
- ➢ Cost of pump operation
- ➢ Governing codes and standards

After pump selection, the following items must be specified;

- ➢Type of pump and manufacturer
- ➢Size of pump
- ➢Size of suction connection and type (flanged, screwed etc)
- ➢Size and type of discharge connection
- ➢Speed of operation
- ➢Specifications for driver (for eg: for an electric motor-power required,
  speed, voltage, phase, frequency, frame size, enclosure type)
- ➢Coupling type, manufacturer and model number
- ➢Mounting details
- ➢Special materials and accessories required, if any
- ➢Shaft seal design and seal materials

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Modern hydraulics turbines and centrifugal pumps are highly efficient machines with few differences in their characteristics.

For each design there is a definite relationship between;

- •The speed of rotation, *N*
- •Discharge of flow ,*Q*
- •Head, *H*
- •Diameter, *D* of the rotating element
- •Power, *P*

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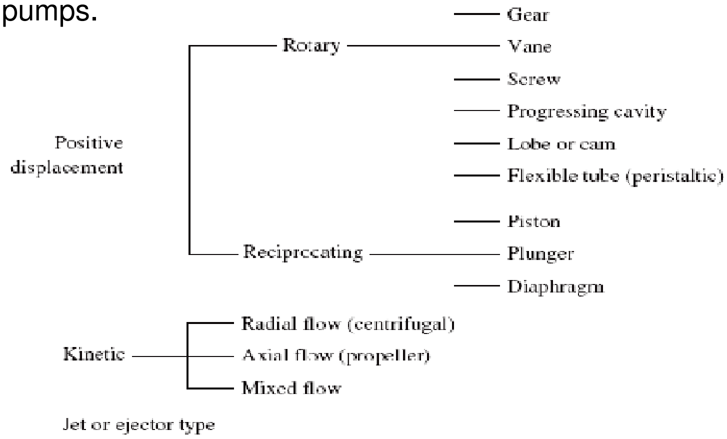
Chapter 5: Hydraulic Machinery by N Adilah A A Ghani

5.2 : CLASSIFICATIONS OF PUMPS

- Pumps are typically classified as either positive-displacement or kinetic pumps.



5.2.1 Positive-Displacement Pumps

- Positive-displacement pumps ideally deliver a fixed quantity of fluid with each revolution of the pump rotor or drive shaft.
- Most positive-displacement pumps can handle liquids over a wide range of viscosities.
- Some of the examples are gear, piston, vane, screw, progressive cavity, lobe, piston, diaphragm and peristaltic pumps.

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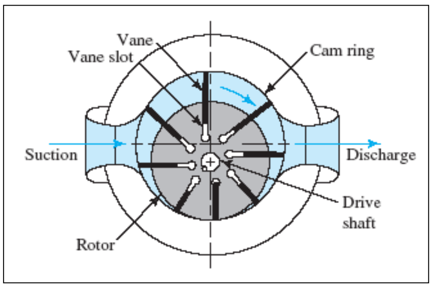
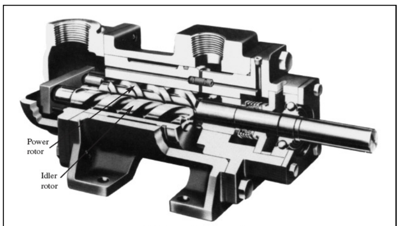
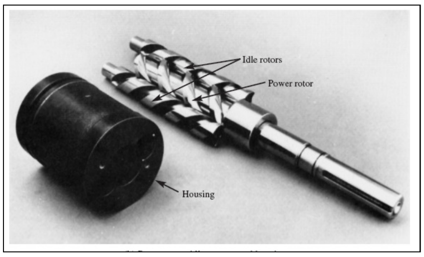
Vane pump



Screw pump



(a) Cutaway of pump assembly



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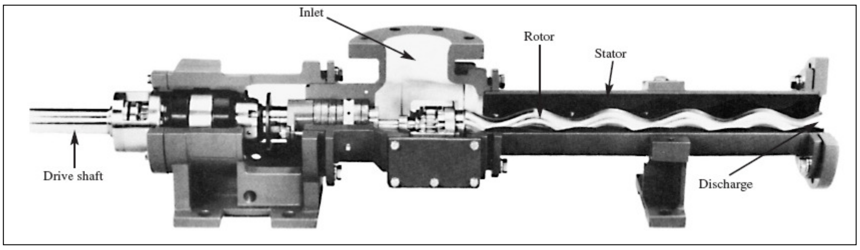
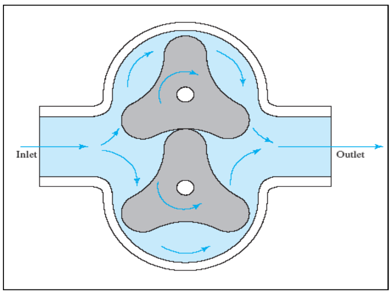
Diaphragm pump



Lobe Pump



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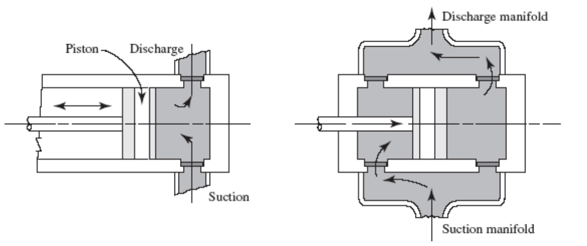
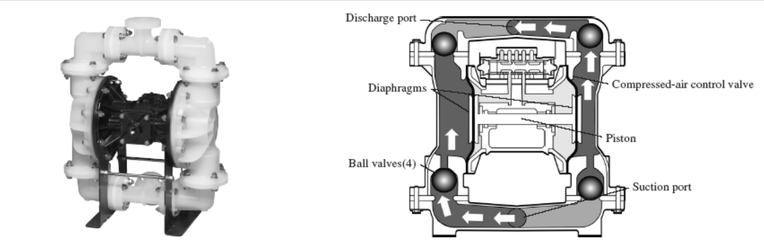
Piston pump

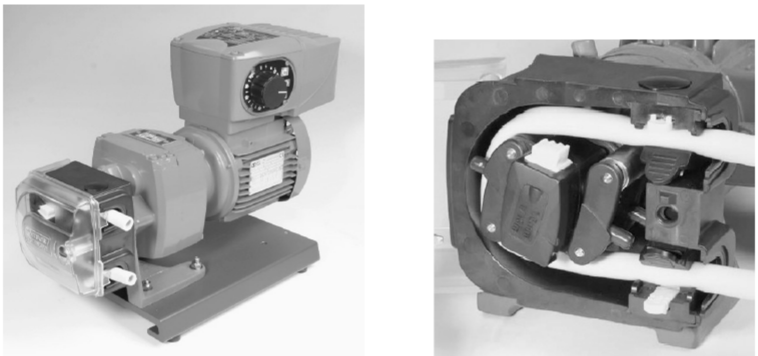


(a) Single acting—simplex (b) Double acting—duplex



(a) Diaphragm pump with nonmetallic housing (b) Diagram of the flow through a double-piston diaphragm pump

Peristaltic pump



(a) Peristaltic pump with variable-speed drive system (b) Peristaltic pump with case open to show tubing and rotating drive rollers

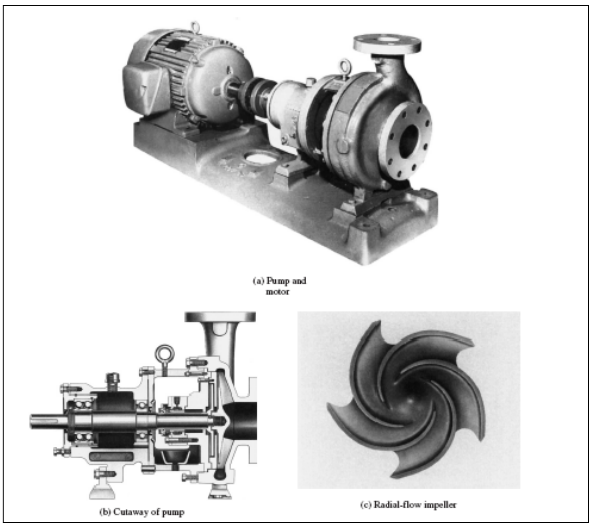
5.2.1.1 Performance Data for Positive-Displacement Pumps

- The operating characteristics of positive-displacement pumps make them useful for handling such fluids as water, hydraulic oils in fluid power systems.
- Some disadvantages of some designs include pulsating output, susceptibility to damage by solids and abrasives, and need for a relief valve.

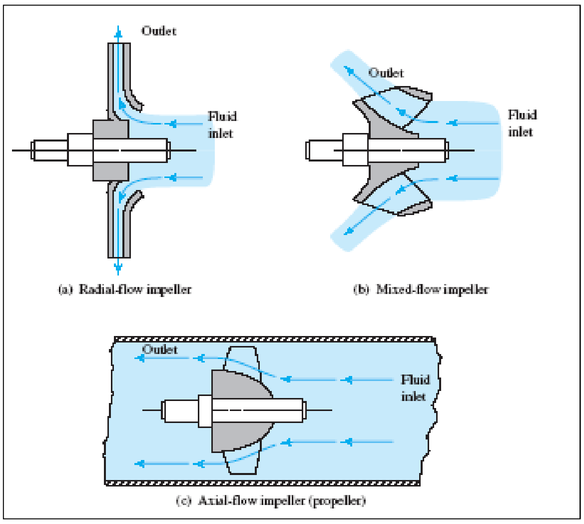
5.2.2 Kinetic Pump (Rotodynamic)

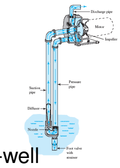
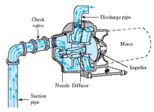
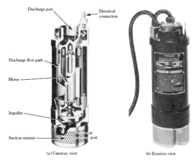
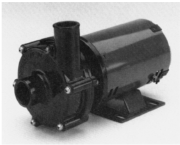
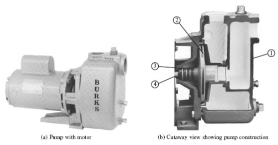
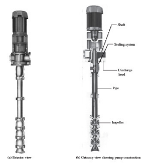
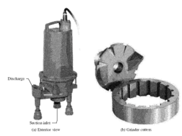
- *Kinetic pumps* add energy to the fluid by accelerating it through the action of a rotating *impeller*.
- The propeller type of pump (axial flow) depends on the hydrodynamic action of the propeller blades to lift and accelerate the fluid axially, along a path parallel to the axis of the propeller.

• Figure shows the basic configuration of a radial flow centrifugal pump, the most common type of kinetic pump.



• Figure shows the basic design of radial, axial, and mixed-flow impellers.



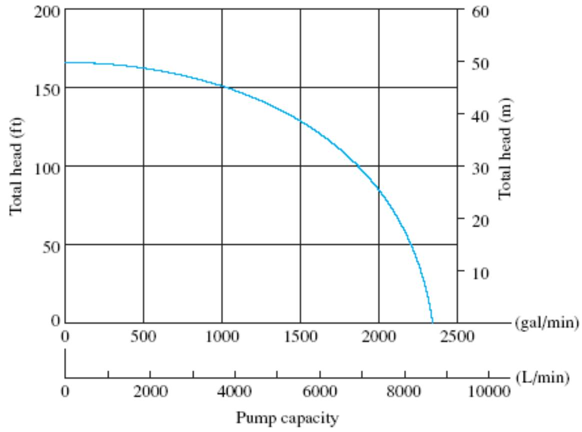
Example of Kinetic pump	
Jet  deep-well  shallow-well	Portable Submersible 
Small Centrifugal 	Self-priming 
Vertical Turbine 	Centrifugal Grinder 

5.3 DESCRIPTION OF CENTRIFUGAL PUMPS

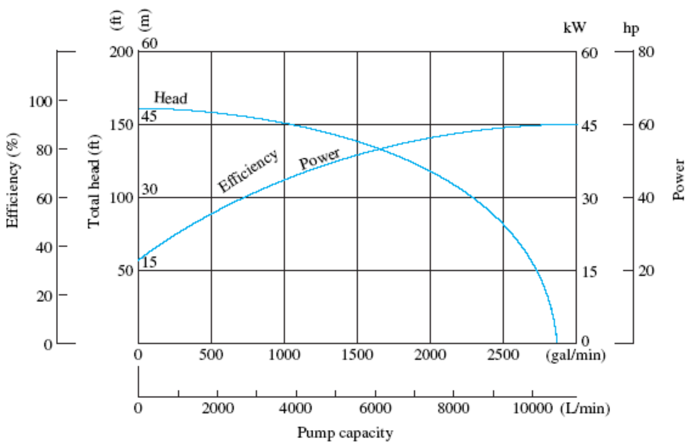
- Because centrifugal pumps are not positive-displacement types, there is a strong dependency between capacity and the pressure that must be developed by the pump.
- This makes their performance ratings somewhat more complex.
- Efficiency and power required are also important to the successful operation of a pump.

5.3.1 Performance Data for Centrifugal Pump

- The typical rating curve plots the total head on the pump versus the capacity or discharge *Q*, as shown in below.



- Figure shows a more complete performance rating of a pump, superimposing head, efficiency, and power curves and plotting all three versus capacity.



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5.3.2 Affinity Law for Centrifugal Pumps

- It is important to understand the manner in which capacity, head, and power vary when either speed or impeller diameter is varied.
- These relationships, called *affinity laws.*
- The symbol *N* refers to the rotational speed of the impeller, usually in revolutions per minute (r/min, or rpm).

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- When *speed varies*:

1. Capacity varies directly with speed:

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

2. The total head capability varies with the square of the speed:

$$\frac{h_{a_1}}{h_{a_2}} = \left(\frac{N_1}{N_2}\right)^2$$

3. The power required by the pump varies with the cube of the speed:

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

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- When *impeller varies*:

1. Capacity varies directly with impeller diameter:

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

2. The total head varies with the square of the impeller diameter:

$$\frac{h_{a_1}}{h_{a_2}} = \left(\frac{D_1}{D_2}\right)^2$$

3. The power required by the pump varies with the cube of the impeller diameter:

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3$$

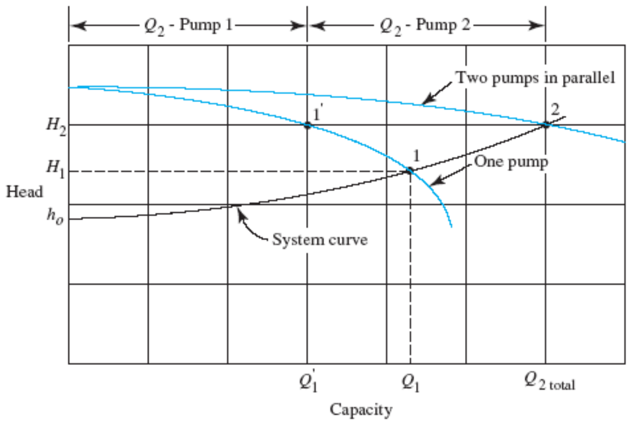
- Efficiency remains nearly constant for speed changes and for small changes in impeller diameter.

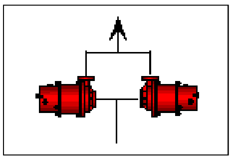
5.4 PUMP OPERATING IN COMBINATION

- It may be more efficient to have multiple pumps operating in some combination in a line than it would be to have one very large pump.
- This situation often arises in cases were the conditions (required discharge or pumping head) may vary dramatically with time.
- In these cases, only those pumps that are required to meet the particular discharge and head requirements of the moment can be employed and the others can be shut off to save the power.

5.4.1 Operating Pump In Parallel

- Many fluid flow systems require largely varying flow rates that are difficult to provide with one pump without calling for the pump to operate far off its best efficiency point.



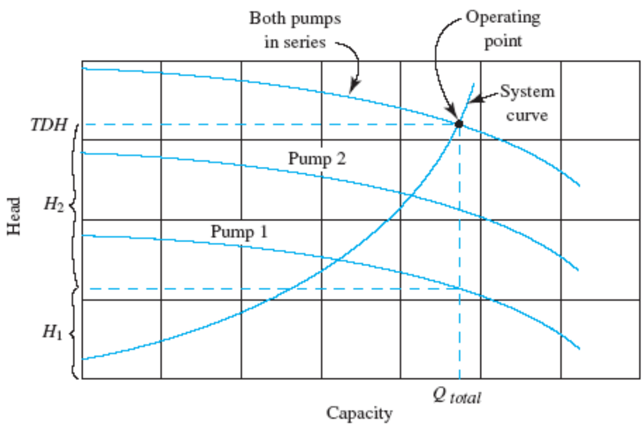


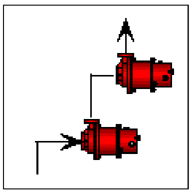
- When two or more pumps are arranged in parallel, their resulting performance curve is obtained by adding their flowrates at the same head.

5.4.2 Operating Pump In Series

- Directing the output of one pump to the inlet of a second pump allows the same capacity to be obtained at a total head equal to the sum of the ratings of the two pumps.

- This method permits operation against unusually high heads.





- When two (or more) pumps are arranged in serial, their resulting pump performance curve is obtained by adding heads at the same flowrate.

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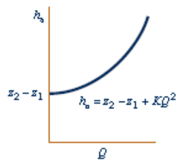
12.4.4 System Characteristics and Pump Selection

The system equation relates the actual head gained by the fluid to the flowrate.

A typical flow system in which a pump is used is shown in Fig. 12.14. The energy equation applied between points (1) and (2) indicates that

$$h_a = z_2 - z_1 + \sum h_L \quad (12.26)$$

where h_a is the actual head gained by the fluid from the pump, and Σh_L represents all friction losses in the pipe and minor losses for pipe fittings and valves. From our study of pipe flow, we know that typically h_L varies approximately as the flowrate squared; that is, $h_L \propto Q^2$ (see Section 8.4). Thus, Eq. 12.26 can be written in the form



$$h_a = z_2 - z_1 + KQ^2 \quad (12.27)$$

where K depends on the pipe sizes and lengths, friction factors, and minor loss coefficients. Equation 12.27, which is shown in the figure in the margin, is the ***system equation*** and shows how the actual head gained by the fluid from the pump is related to the system parameters. In this case the parameters include the change in elevation head, $z_2 - z_1$, and the losses due to friction as expressed by KQ^2. Each flow system has its own specific system equation. If the flow is laminar, the frictional losses will be proportional to Q rather than Q^2 (see Section 8.2).

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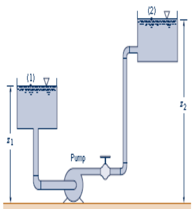
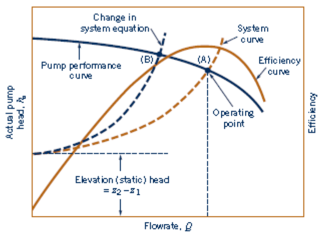


■ Figure 12.14 Typical flow system.



■ Figure 12.15 **Utilization of the system curve and the pump performance curve to obtain the operating point for the system.**

The intersection of the pump performance curve and the system curve is the operating point.

There is also a unique relationship between the actual pump head gained by the fluid and the flowrate, which is governed by the pump design (as indicated by the pump performance curve). To select a pump for a particular application, it is necessary to utilize both the *system curve*, as determined by the system equation, and the pump performance curve. If both curves are plotted on the same graph, as illustrated in Fig. 12.15, their intersection (point A) represents the operating point for the system. That is, this point gives the head and flowrate that satisfy both the system equation and the pump equation. On the same graph the pump efficiency is shown. Ideally, we want the operating point to be near the best efficiency point (BEP) for the pump. For a given pump, it is clear that as the system equation changes, the operating point will shift. For example, if the pipe friction increases due to pipe wall fouling, the system curve changes, resulting in the operating point A shifting to point B in Fig. 12.15 with a reduction in flowrate and efficiency. The following example shows how the system and pump characteristics can be used to decide if a particular pump is suitable for a given application.

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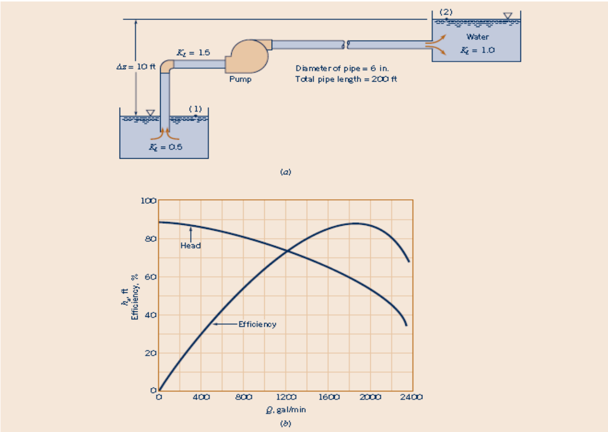
EXAMPLE 12.4 Use of Pump Performance Curves

GIVEN Water is to be pumped from one large, open tank to a second large, open tank as shown in Fig. E12.4*a*. The pipe diameter throughout is 6 in., and the total length of the pipe between the pipe entrance and exit is 200 ft. Minor loss coefficients for the entrance, exit, and the elbow are shown, and the friction factor for the pipe can be assumed constant and equal to 0.02. A certain centrifugal pump having the performance characteristics shown in Fig. E12.4*b* is suggested as a good pump for this flow system.

FIND With this pump, what would be the flowrate between the tanks? Do you think this pump would be a good choice?



■ Figure E12.4*a*, *b*

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SOLUTION

Application of the energy equation between the two free surfaces, points (1) and (2) as indicated, gives

$$\frac{p_1}{\gamma} + \frac{V_1^2}{2g} + z_1 + h_a = \frac{p_2}{\gamma} + \frac{V_2^2}{2g} + z_2 + f\frac{\ell}{D}\frac{V^2}{2g} + \sum K_L \frac{V^2}{2g} \quad (1)$$

Thus, with $p_1 = p_2 = 0$, $V_1 = V_2 = 0$, $\Delta z = z_2 - z_1 = 10$ ft, $f = 0.02$, $D = 6/12$ ft, and $\ell = 200$ ft, Eq. 1 becomes

$$h_a = 10 + \left[0.02\frac{(200\text{ ft})}{(6/12\text{ ft})} + (0.5 + 1.5 + 1.0)\right]\frac{V^2}{2(32.2\text{ ft/s}^2)} \quad (2)$$

where the given minor loss coefficients have been used. Since

$$V = \frac{Q}{A} = \frac{Q(\text{ft}^3/\text{s})}{(\pi/4)(6/12\text{ ft})^2}$$

Eq. 2 can be expressed as

$$h_a = 10 + 4.43\, Q^2 \quad (3)$$

where Q is in ft^3/s, or with Q in gallons per minute

$$h_a = 10 + 2.20 \times 10^{-5} Q^2 \quad (4)$$

Equation 3 or 4 represents the system equation for this particular flow system and reveals how much actual head the fluid will need to gain from the pump to maintain a certain flowrate. Performance data shown in Fig. E12.4*b* indicate the actual head the fluid will gain from this particular pump when it operates at a certain flowrate. Thus, when Eq. 4 is plotted on the same graph with performance data, the intersection of the two curves represents the operating point for the pump and the system. This combination is shown in Fig. E12.4*c* with the intersection (as obtained graphically) occurring at

$$Q = 1600 \text{ gal/min} \quad \textbf{(Ans)}$$

with the corresponding actual head gained equal to 66.5 ft.

Another concern is whether the pump is operating efficiently at the operating point. As can be seen from Fig. E12.4*c*, although this is not peak efficiency, which is about 86%, it is close (about 84%). Thus, this pump would be a satisfactory choice, assuming the 1600 gal/min flowrate is at or near the desired flowrate.

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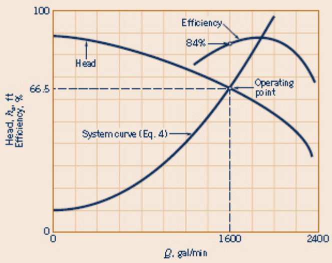
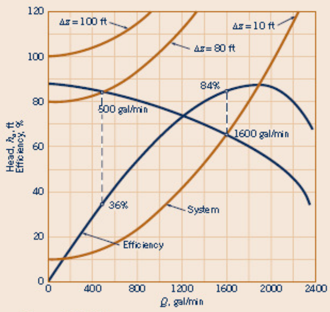


■ Figure E12.4*c* (*Continued*)



■ Figure E12.4*d*

The amount of pump head needed at the pump shaft is 66.5 ft/0.84 = 79.2 ft. The power needed to drive the pump is

$$\dot{W}_{shaft} = \frac{\gamma Q h_a}{\eta}$$
$$= \frac{(62.4 \text{ lb/ft}^3)[(1600 \text{ gal/min})/(7.48 \text{ gal/ft}^3)(60 \text{ s/min})](66.5 \text{ ft})}{0.84}$$
$$= 17{,}600 \text{ ft}\cdot\text{lb/s} = 32.0 \text{ hp}$$

COMMENT By repeating the calculations for $\Delta z = z_2 - z_1 = 80$ ft and 100 ft (rather than the given 10 ft), the results shown in Fig. E12.4*d* are obtained. Although the given pump could be used with $\Delta z = 80$ ft (provided that the 500 gal/min flowrate produced is acceptable), it would not be an ideal pump for this application since its efficiency would be only 36%. Energy could be saved by using a different pump with a performance curve that more nearly matches the new system requirements (i.e., higher efficiency at the operating condition). On the other hand, the given pump would not work at all for $\Delta z = 100$ ft since its maximum head ($h_a = 88$ ft when $Q = 0$) is not enough to lift the water 100 ft, let alone overcome head losses. This is shown in Fig. E12.4*d* by the fact that for $\Delta z = 100$ ft the system curve and the pump performance curve do not intersect.

Note that head loss within the pump itself was accounted for with the pump efficiency, η. Thus, $h_s = h_a/\eta$, where h_s is the pump shaft work head and h_a is the actual head rise experienced by the flowing fluid.

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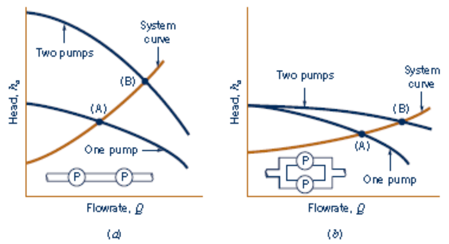
For two pumps in series, add heads; for two in parallel, add flowrates.

Pumps can be arranged in series or in parallel to provide for additional head or flow capacity. When two pumps are placed in *series*, the resulting pump performance curve is obtained by adding heads at the same flowrate. As illustrated in Fig. 12.16*a*, for two identical pumps in series, both the actual head gained by the fluid and the flowrate are increased, but neither will be doubled if the system curve remains the same. The operating point is at (A) for one pump and moves to (B) for two pumps in series. For two identical pumps in *parallel*, the combined performance curve is obtained by adding flowrates at the same head, as shown in Fig. 12.16*b*. As illustrated, the flowrate for the system will not be doubled with the addition of two pumps in parallel (if the same system



■ Figure 12.16 **Effect of operating pumps in (*a*) series and (*b*) in parallel.**

curve applies). However, for a relatively flat system curve, as shown in Fig. 12.16*b*, a significant increase in flowrate can be obtained as the operating point moves from point (A) to point (B).

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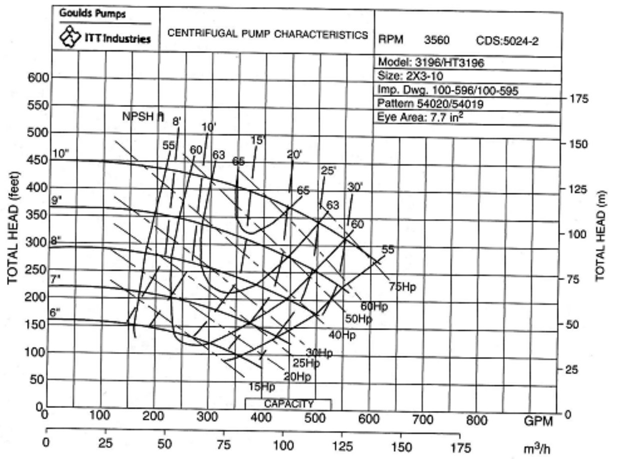
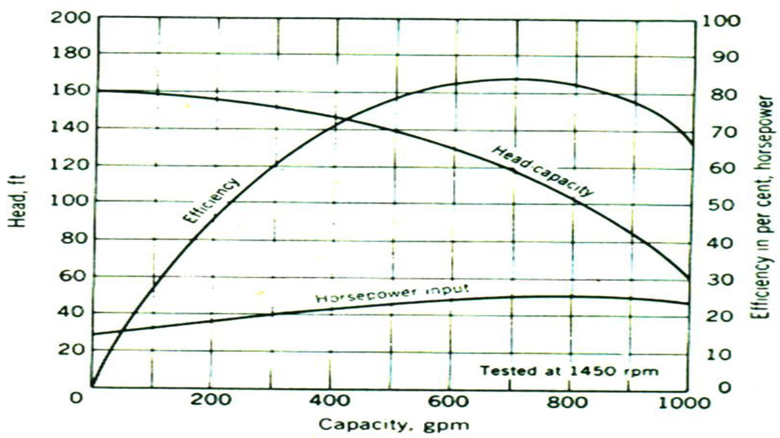
EXAMPLE 5.1

A centrifugal pump must deliver at least 1500L/min of water at a total head of 80.0m. List its performance characteristic (pump efficiency, power requires, size of impeller and NPSH).



EXAMPLE 5.2

Determine the specific speed of the pump (rated capacity 700gpm) whose operating characteristics are shown below. If this pump were operated at 1200rpm, what head and discharge would be developed at rated capacity, and what power would be required?



EXAMPLE 5.3

Water is pumped from a river through a 150mm diameter pipeline 950m to an open storage tank with a water level 45m above the river. A pump is available and has the discharge head performance characteristics shown below.

a. Calculate the duty point for the pump when the Darcy friction factor for the pipeline f= 0.04.

b. Determine the revised duty point when a second similar pump is connected in parallel into the system.

Total head (m)	30	50	65	80	87	94
Discharge (l/min)	2000	1750	1410	800	500	0

Solution 5.3 (a)

1.Plot graph of pump discharge-head characteristics.
2.Calculate head loss in the pipeline for a range of discharge values using the Darcy-Weisbach formula.
3. Add value of static head.
4. Construct a graph of the pump curve and the pipe curve.
5. From the graph, determine the duty point (location where the two curves cross)

Solution 5.3 (b)

1. Find the new characteristic for pump in parallel.
2. Construct a graph of the two pumps.
3. From the graph, determine the new duty point (location where the two curves cross)

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