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Bruce R. Munson. Theodore H. Okiishi. Alric P. Rothmayer
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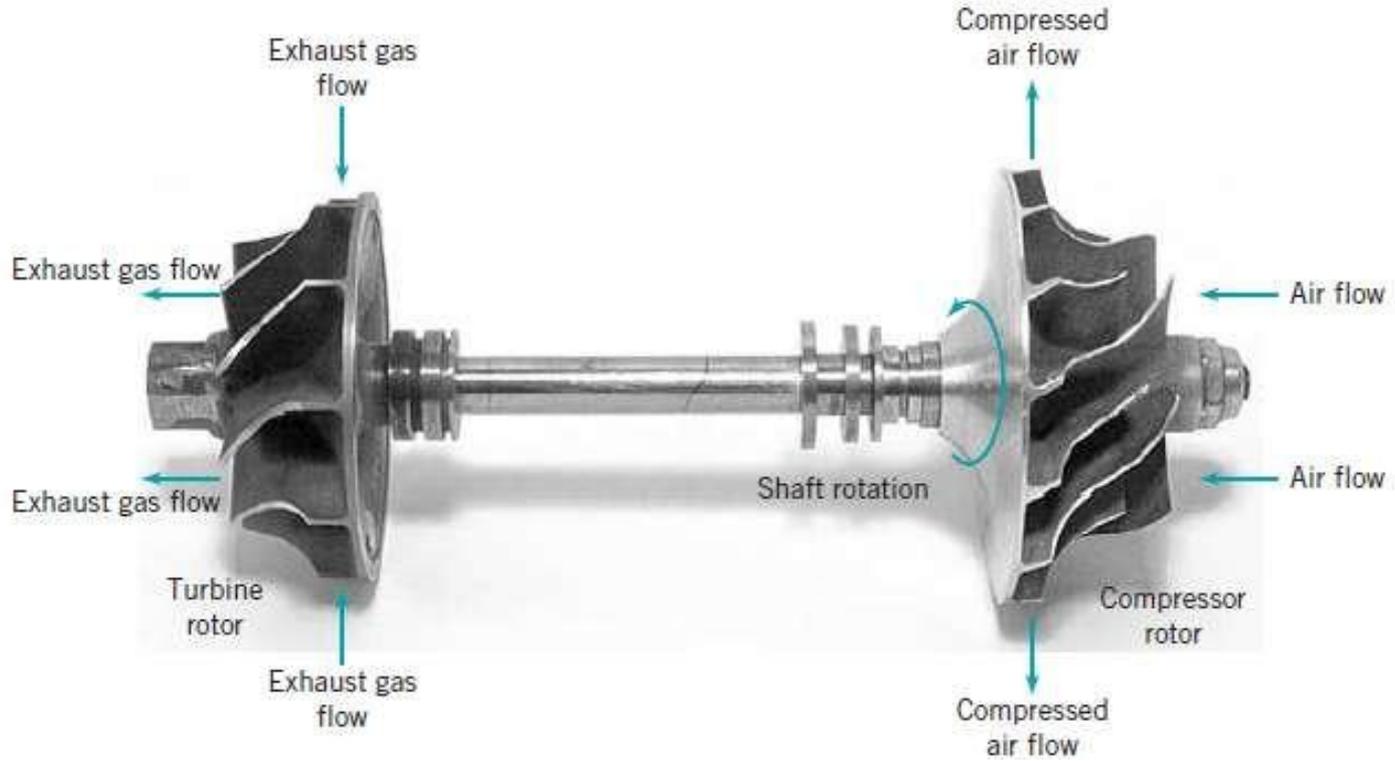
Lecture- 16

Introduction to Turbomachinery

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Outline

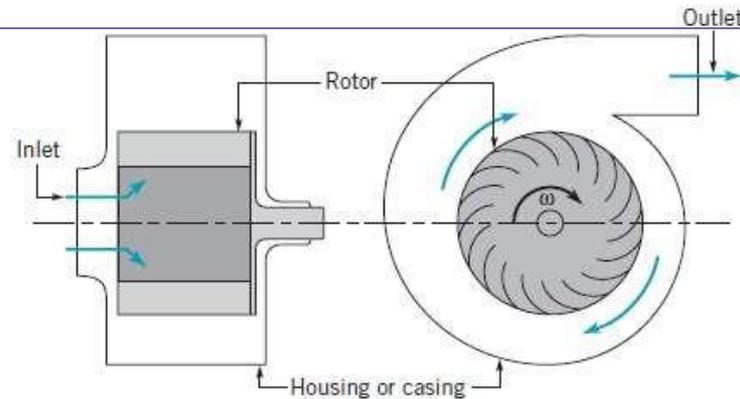
- Overview
- axial-flow, mixed-flow, or radial-flow Basic
- Basic Energy Considerations
- The Centrifugal Pump
- Pump Performance Characteristics
- Fans
- Turbines



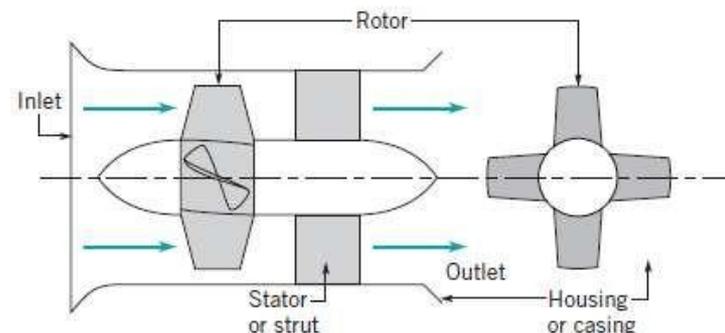
Turbomachines are mechanical devices that either extract energy from a fluid (turbine) or add energy to a fluid (pump) as a result of dynamic interactions between the device and the fluid.

While the actual design and construction of these devices often require considerable insight and effort, their basic operating principles are quite simple.

Turbomachines are classified as **axial-flow**, **mixed-flow**, or **radial-flow** machines depending on the predominant direction of the fluid motion relative to the rotor's axis as the fluid passes the blades.



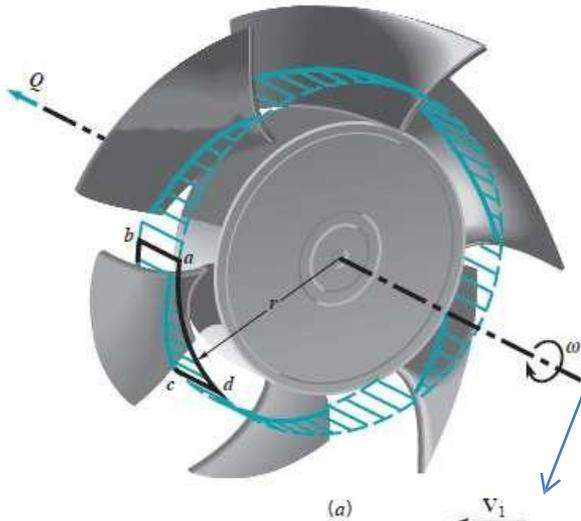
(a) Radial-flow fan



(b) Axial-flow fan



Basic Energy Considerations



Velocity triangles are obtained from:

$$\mathbf{V} = \mathbf{W} + \mathbf{U}$$

From angular momentum (moment of momentum) conservation, we can obtain:

$$T_{\text{shaft}} = -\dot{m}_1(r_1 V_{\theta 1}) + \dot{m}_2(r_2 V_{\theta 2})$$

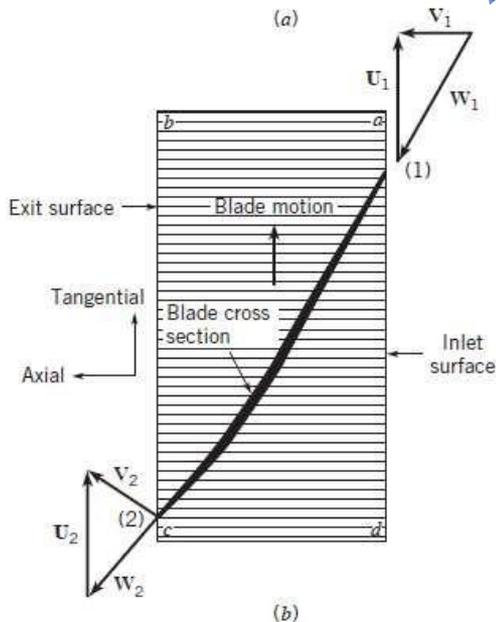
This equation is known as the ***Euler turbomachine equation***

and since the shaft power of a rotating machine is given by

$$\dot{W}_{\text{shaft}} = T_{\text{shaft}} \omega$$



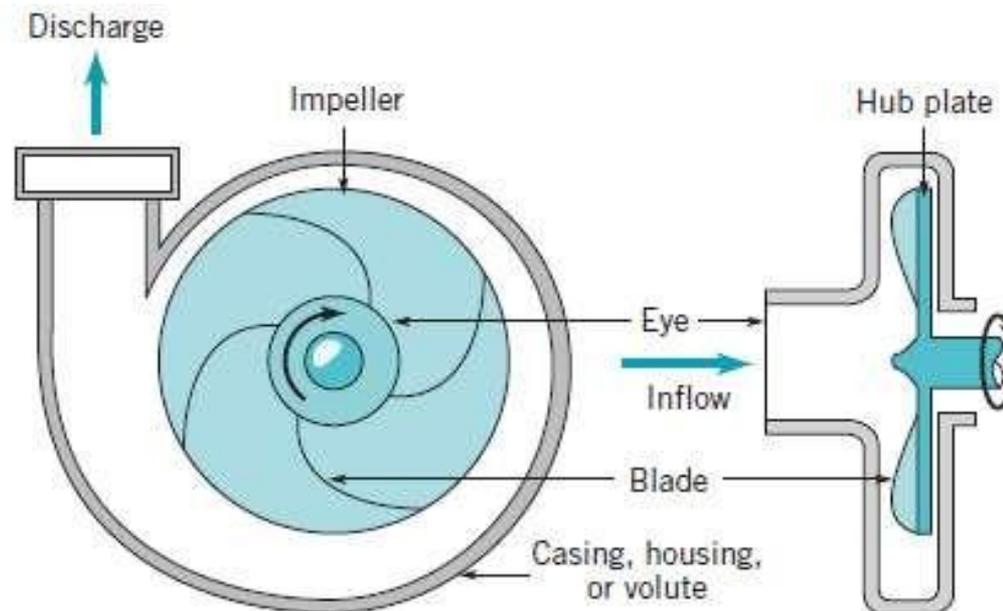
$$\dot{W}_{\text{shaft}} = -\dot{m}_1(U_1 V_{\theta 1}) + \dot{m}_2(U_2 V_{\theta 2})$$



The Centrifugal Pump

One of the most common radial-flow turbomachines is the **centrifugal pump**. This type of pump has two main components: an *impeller* attached to a rotating *shaft*, and a stationary casing, housing, or *volute* enclosing the impeller. The impeller consists of a number of *blades* (usually curved), also sometimes called *vanes*, arranged in a regular pattern around the shaft. As the impeller rotates, fluid is sucked in through the *eye* of the casing and flows radially outward.

For large centrifugal pumps, a different design is often used in which diffuser guide vanes surround the impeller. The diffuser vanes decelerate the flow as the fluid is directed into the pump casing. This type of centrifugal pump is referred to as a *diffuser pump*.



Impellers are generally of two types: (a) *Open impeller*, (b) *enclosed* or shrouded impeller.



Pump impellers can also be *single* or *double suction*. For the single-suction impeller the fluid enters through the eye on only one side of the impeller, whereas for the double-suction impeller the fluid enters the impeller along its axis from both sides.

Pumps can be *single* or *multistage*. For a single-stage pump, only one impeller is mounted on the shaft, whereas for multistage pumps, several impellers are mounted on the same shaft (stages operate in series.)

Centrifugal pumps come in a variety of arrangements, but the basic operating principle remains the same. Work is done on the fluid by the rotating blades, creating a large increase in kinetic energy of the fluid flowing through the impeller. This kinetic energy is converted into an increase in pressure as the fluid flows from the impeller into the casing enclosing the impeller.

Theoretical Considerations

For the rotating shaft, the power transferred, is given by

$$\dot{W}_{\text{shaft}} = \rho Q (U_2 V_{\theta 2} - U_1 V_{\theta 1})$$

It follows that the shaft power per unit mass of flowing fluid is

$$w_{\text{shaft}} = \frac{\dot{W}_{\text{shaft}}}{\rho Q} = U_2 V_{\theta 2} - U_1 V_{\theta 1}$$

For incompressible pump flow, the energy equation gives

$$w_{\text{shaft}} = \left(\frac{p_{\text{out}}}{\rho} + \frac{V_{\text{out}}^2}{2} + gz_{\text{out}} \right) - \left(\frac{p_{\text{in}}}{\rho} + \frac{V_{\text{in}}^2}{2} + gz_{\text{in}} \right) + \text{loss}$$



$$\frac{U_2 V_{\theta 2} - U_1 V_{\theta 1}}{g} = H_{\text{out}} - H_{\text{in}} + h_L$$

Where H
is the
total head

$$H = \frac{p}{\rho g} + \frac{V^2}{2g} + z$$

An ideal pump without losses would give a total head rise:

$$h_i = \frac{U_2 V_{\theta 2} - U_1 V_{\theta 1}}{g}$$

The actual head rise h_a is less than h_i by h_L representing the losses in the pump. An alternate expression of h_i is given by

$$h_i = \frac{1}{2g} [(V_2^2 - V_1^2) + (U_2^2 - U_1^2) + (W_1^2 - W_2^2)]$$

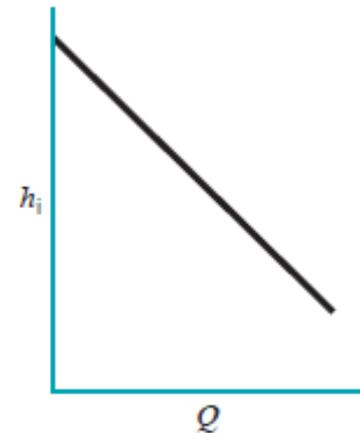
Often the fluid has no tangential component of velocity or *swirl*, as it enters the impeller. ($\alpha_1 = 90^\circ$ so that $V_{\theta 1} = 0$) and the expression reduces to:

$$h_i = \frac{U_2 V_{\theta 2}}{g}$$

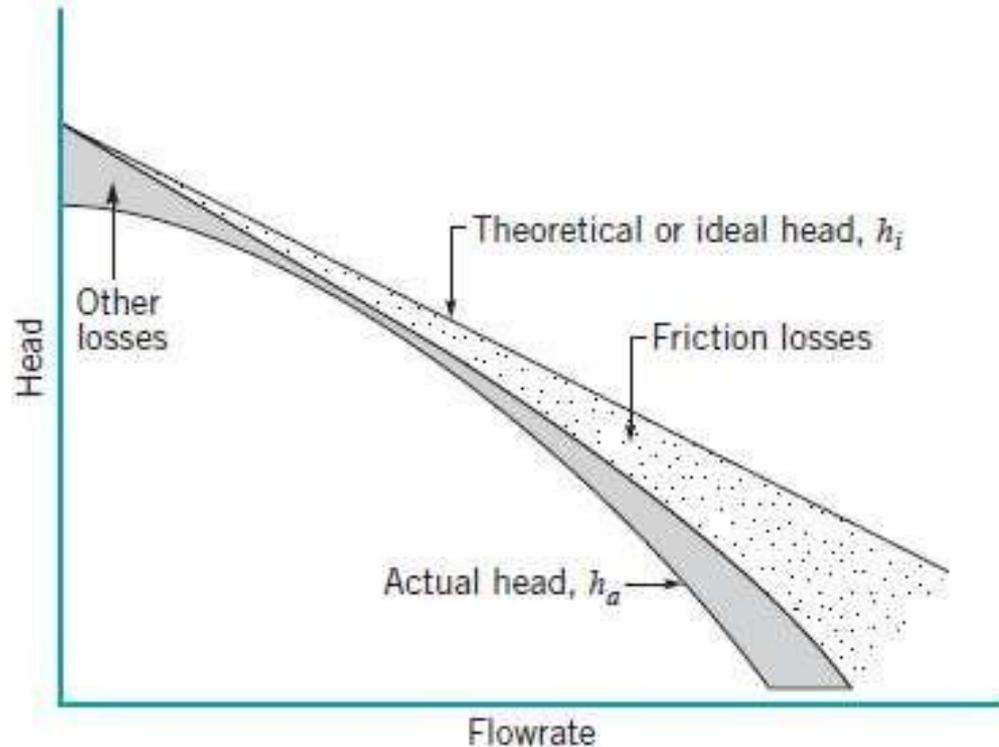
The flowrate, Q , is related to the radial component of the absolute velocity through the equation

$$Q = 2\pi r_2 b_2 V_{r2} \quad \text{And the velocity triangle gives} \quad \cot \beta_2 = \frac{U_2 - V_{\theta 2}}{V_{r2}}$$

$$\rightarrow h_i = \frac{U_2^2}{g} - \frac{U_2 \cot \beta_2}{2\pi r_2 b_2 g} Q \rightarrow$$



Blades with $\beta_2 < 90^\circ$ are called *backward curved*, whereas blades with $\beta_2 > 90^\circ$ are called *forward curved*. Pumps are not usually designed with forward curved vanes since such pumps tend to suffer unstable flow conditions. The normal range is $20^\circ < \beta_2 < 25^\circ$ and $15^\circ < \beta_2 < 50^\circ$

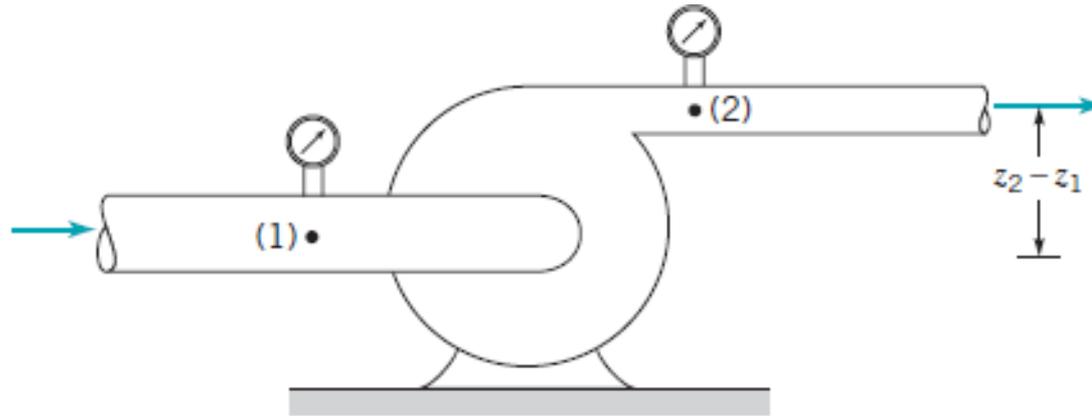


Effect of losses on the pump head–flowrate curve (backward curved blades pump)

Actual pump performance is determined experimentally through tests on the pump. From these tests, pump characteristics are determined and presented as ***pump performance curves***. It is this information that is most helpful to the engineer responsible for incorporating pumps into a given flow system.

Pump Performance Characteristics

The actual head rise, delivered by a pump to the fluid can be determined with an experimental arrangement



$$h_a = \frac{p_2 - p_1}{\gamma} + z_2 - z_1 + \frac{V_2^2 - V_1^2}{2g}$$

If the difference in elevation is neglected

$$h_a \approx \frac{p_2 - p_1}{\gamma}$$

And the power gain by the fluid is

$$\mathcal{P}_f = \gamma Q h_a$$

In addition to the head or power added to the fluid, the **overall efficiency**, is of interest, where

$$\eta = \frac{\text{power gained by the fluid}}{\text{shaft power driving the pump}} = \frac{\mathcal{P}_f}{\dot{W}_{\text{shaft}}}$$

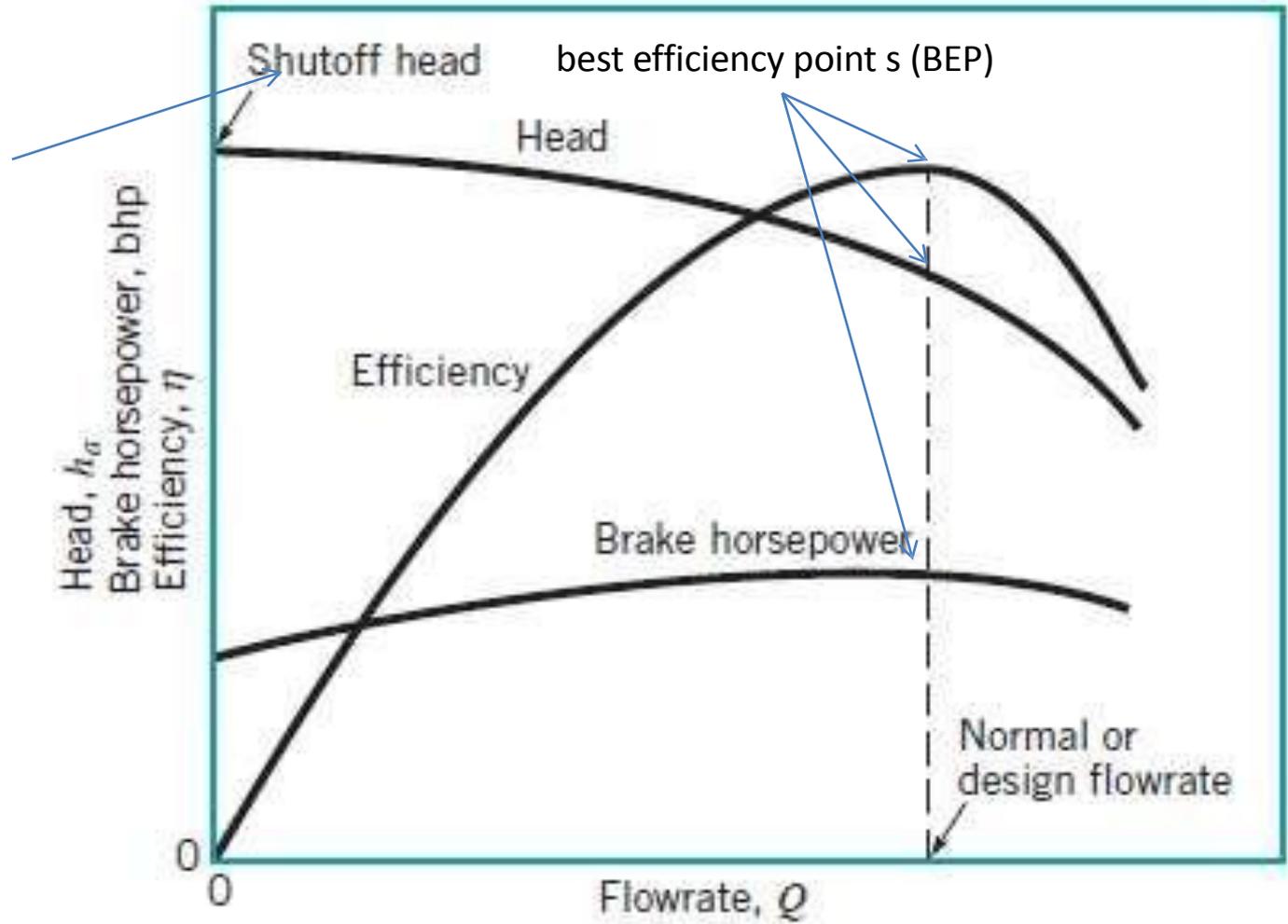
The denominator of this relationship represents the total power applied to the shaft of the pump and is often referred to as *brake horsepower* (bhp). Thus,

$$\eta = \eta_h \eta_m \eta_v$$

The diagram shows the equation $\eta = \eta_h \eta_m \eta_v$ at the top. Three blue arrows point downwards from the terms η_h , η_m , and η_v to the labels "hydraulic efficiency", "mechanical efficiency", and "volumetric efficiency" respectively.

Typical performance characteristics for a centrifugal pump of a given size operating at a constant impeller speed.

with the discharge valve closed: no flow, related efficiency is zero, and the power supplied by the pump is simply dissipated as heat.



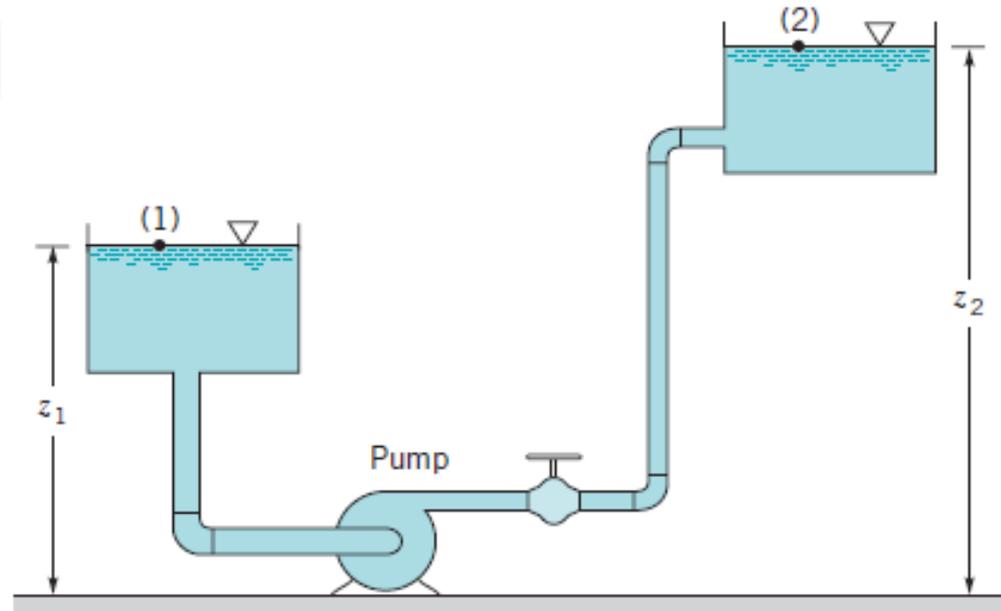
Also referred to as *capacity*

System Characteristics and Pump Selection

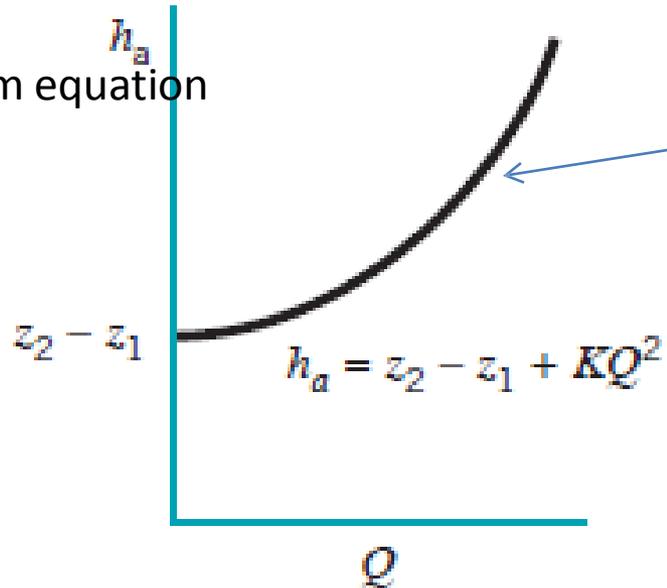
For a typical flow system in which a pump is used is shown, the energy equation applied between points 1 and 2 indicates that

$$h_a = z_2 - z_1 + \sum h_L$$

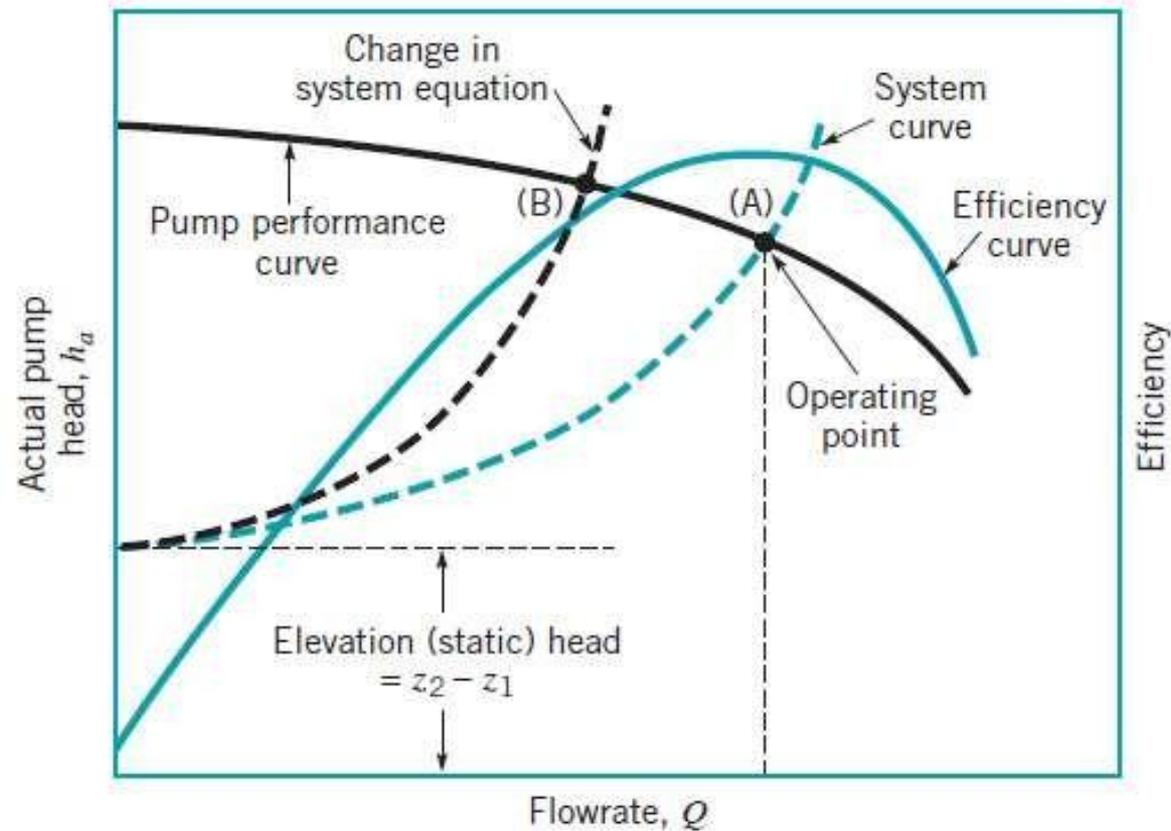
with $h_L \propto Q^2$ \Rightarrow $h_a = z_2 - z_1 + KQ^2$



system equation



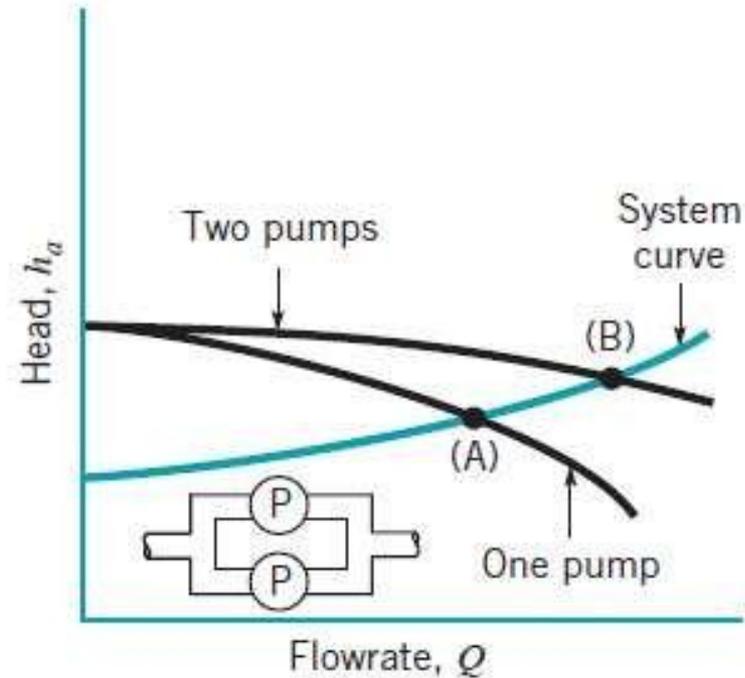
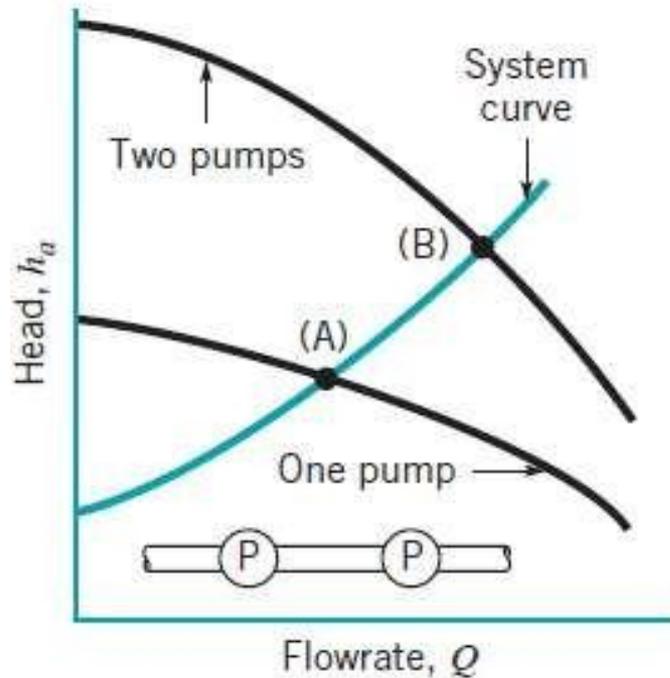
To select a pump for a particular application, it is necessary to utilize both the system curve and the pump performance curve. Their intersection represents the operating point for the system. Ideally, we want the operating point to be near the best efficiency point (BEP) for the pump.



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.

For two pumps in parallel, the combined performance curve is obtained by adding flowrates at the same head.



Specific Speed

A useful nondimensional parameter often used in characterizing the operation of pumps is the *specific speed*, defined as:

$$\frac{(Q/\omega D^3)^{1/2}}{(gh_a/\omega^2 D^2)^{3/4}} = \frac{\omega \sqrt{Q}}{(gh_a)^{3/4}} = N_s$$

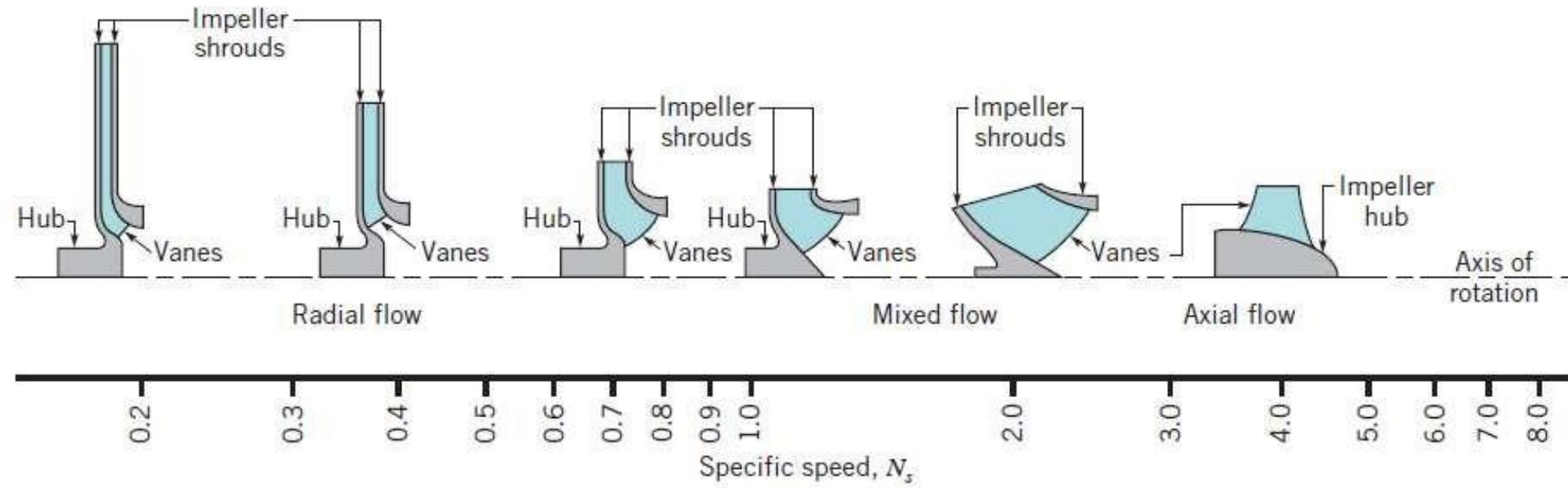
Each family or class of pumps has a particular range of values of specific speed associated with it.

For pumps with low Q and high h_a the specific speed is low compared to a pump with high Q and low h_a . Centrifugal pumps typically are low-capacity, high-head pumps, and therefore have low specific speeds.

The concept of specific speed is very useful to engineers and designers, since if the required head, flowrate, and speed are specified, it is possible to select an appropriate (most efficient) type of pump for a particular application.

centrifugal pumps are radial-flow machines that operate most efficiently for applications requiring high heads at relatively low flowrates. This head–flowrate combination typically yields specific speeds that are less than approximately .1.5

Variation in specific speed at maximum efficiency with type of pump.



Other important dimensionless parameters

In addition to the specific speed, three other nondimensional parameters are often used for engineering purposes when dealing with pumps.

The **head rise coefficient**

$$C_H = \frac{gh_a}{\omega^2 D^2}$$

The **power coefficient**

$$C_P = \frac{\dot{W}_{\text{shaft}}}{\rho \omega^3 D^5}$$

The **flow coefficient**

$$C_Q = \frac{Q}{\omega D^3}$$

Axial-Flow and Mixed-Flow Pumps

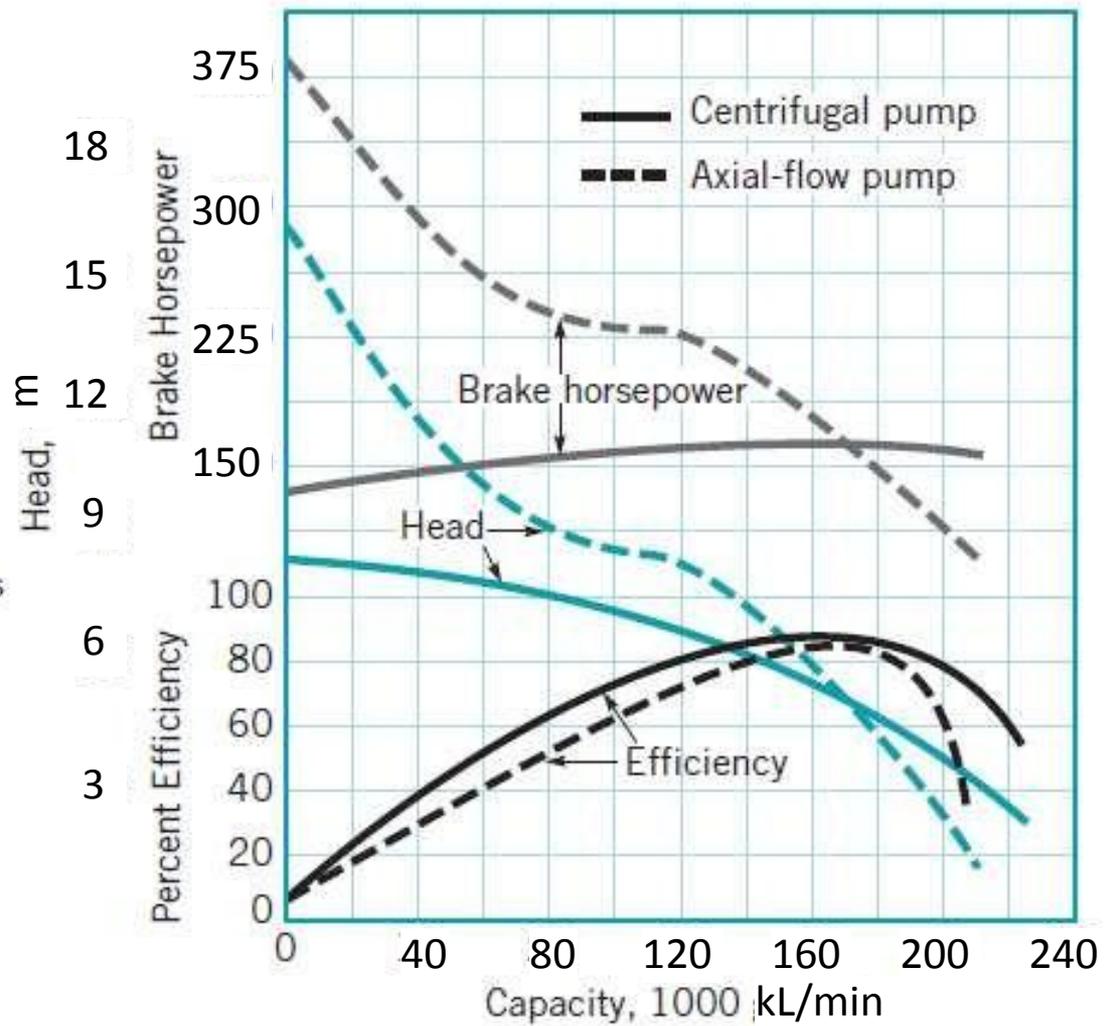
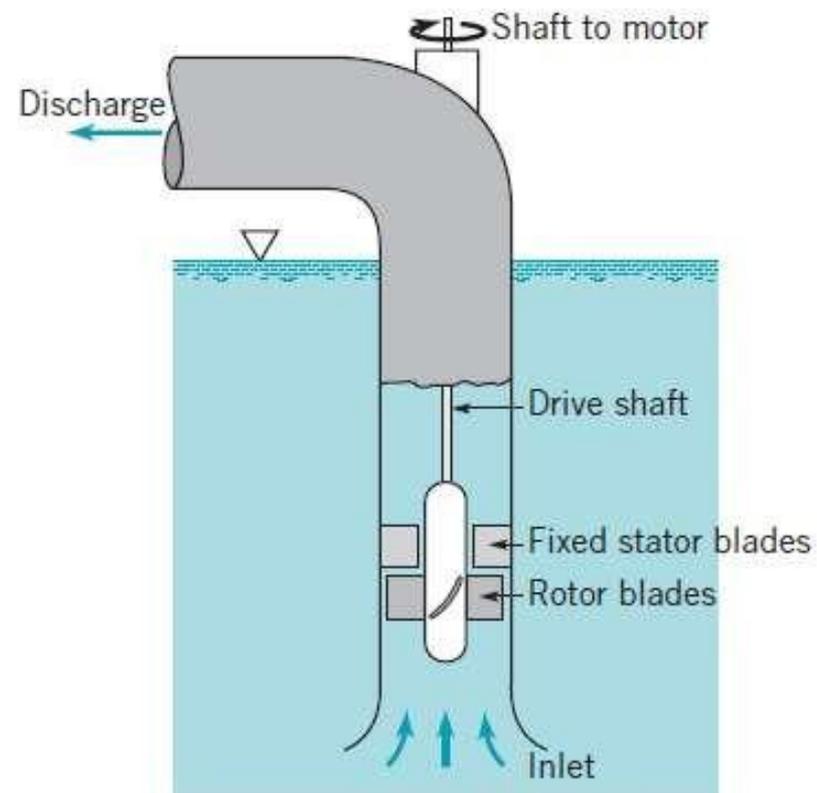
Centrifugal pumps are radial-flow machines that operate most efficiently for applications requiring high heads at relatively low flowrates. For many applications, such as those associated with drainage and irrigation, high flowrates at low heads are required and centrifugal pumps are not suitable. Axial-flow pumps usually have specific speeds in excess of .3.3

In this case, axial-flow pumps are commonly used. This type of pump consists essentially of a propeller confined within a cylindrical casing. Axial-flow pumps are often called *propeller pumps*.

Whereas the head developed by a centrifugal pump includes a contribution due to centrifugal action, the head developed by an axial-flow pump is due primarily to the tangential force exerted by the rotor blades on the fluid. The rotor is connected to a motor through a shaft, and as it rotates (usually at a relatively high speed) the fluid is sucked in through the inlet.

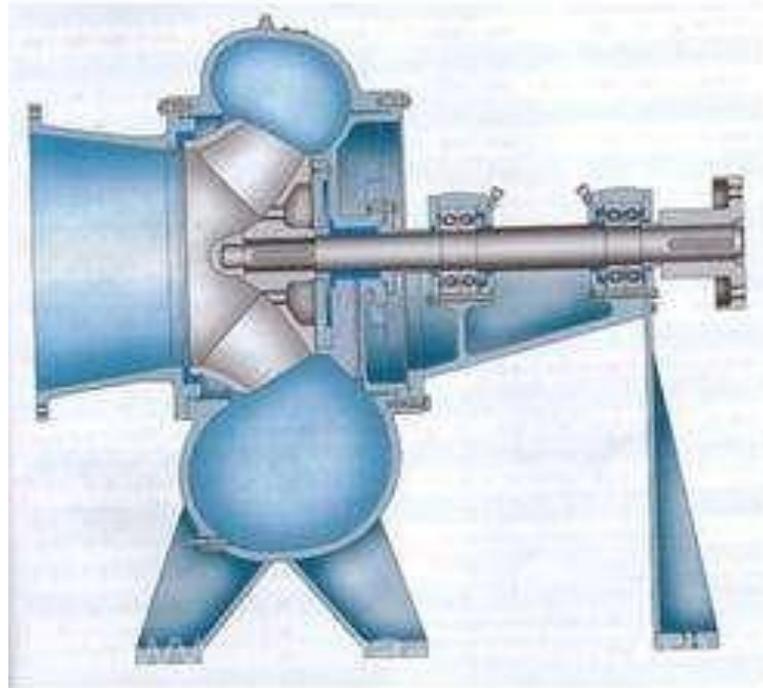
Typically the fluid discharges through a row of fixed stator (guide) vanes used to straighten the flow leaving the rotor. Some axial-flow pumps also have inlet guide vanes upstream of the rotor row, and some are multistage in which pairs (*stages*) of rotating blades (rotor blades) and fixed vanes (stator blades) are arranged in series.

The concepts that were developed for centrifugal pumps are also applicable to axial-flow pumps. The actual flow characteristics, however, are quite different.



For applications requiring specific speeds intermediate to those for centrifugal and axial-flow pumps, mixed-flow pumps have been developed that operate efficiently in the specific speed range $1.5 < N_s < 3.3$

As the name implies, the flow in a mixed-flow pump has both a radial and an axial component.



Fans

When the fluid to be moved is air, or some other gas or vapor, *fans* are commonly used. Types of fans vary from the small fan used for cooling desktop computers to large fans used in many industrial applications such as ventilating of large buildings.

Fans typically operate at relatively low rotation speeds and are capable of moving large volumes of gas.

Although the fluid of interest is a gas, the change in gas density through the fan does not usually exceed 7%. Thus, in dealing with fans, the gas density is treated as a constant, and the flow analysis is based on incompressible flow concepts.

Turbomachines used to produce larger changes in gas density and pressure than possible with fans are called *compressors*.

As is the case for pumps, fan designs include centrifugal (radial-flow fans), as well as mixed-flow and axial-flow (propeller) fans.

The analysis of fan performance closely follows that previously described for pumps. However, fan head-rise data are often given in terms of pressure rise, either static or total, rather than the more conventional head rise commonly used for pumps.

Scaling relationships for fans are the same as those developed for pumps, but replacing the head h_a with the pressure head $p_a/\rho g$:

$$\left(\frac{p_a}{\rho \omega^2 D^2} \right)_1 = \left(\frac{p_a}{\rho \omega^2 D^2} \right)_2$$

where, as before, the subscripts 1 and 2 refer to any two fans from the family of geometrically similar fans.

These equations are called the *fan laws* and can be used to scale performance characteristics between members of a family of geometrically similar fans.

Turbines

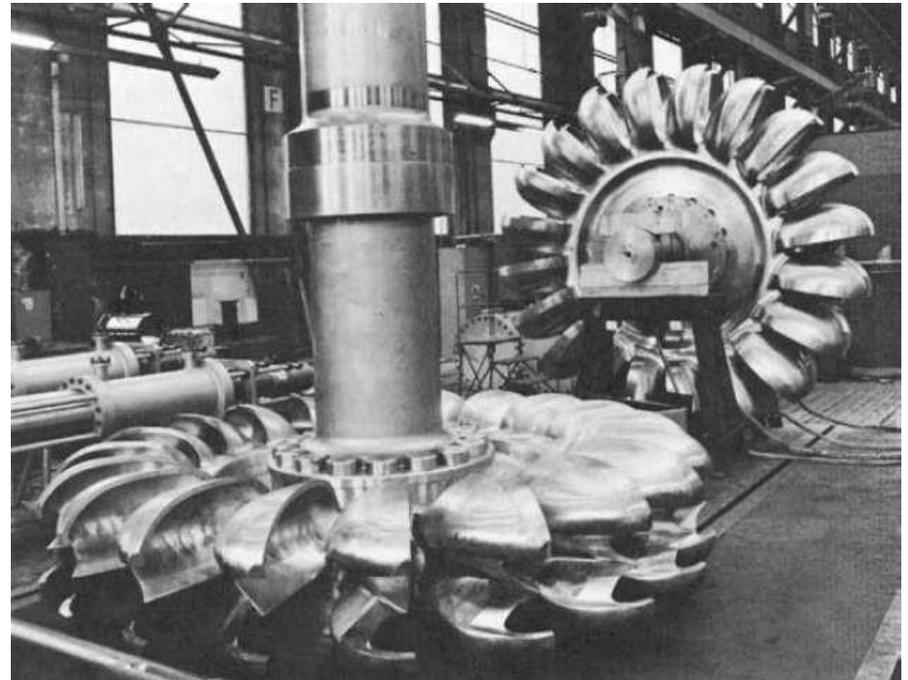
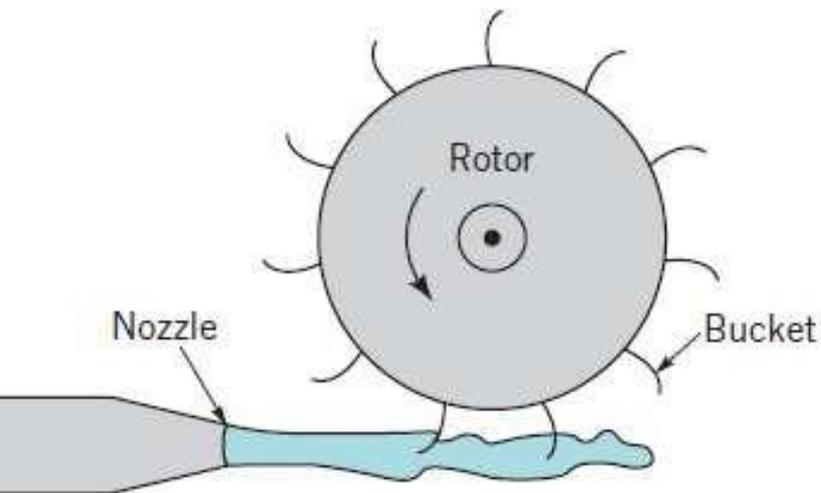
Turbines are devices that extract energy from a flowing fluid. The geometry of turbines is such that the fluid exerts a torque on the rotor in the direction of its rotation. The shaft power generated is available to drive generators or other devices.

Hydraulic turbines : the working fluid is water.

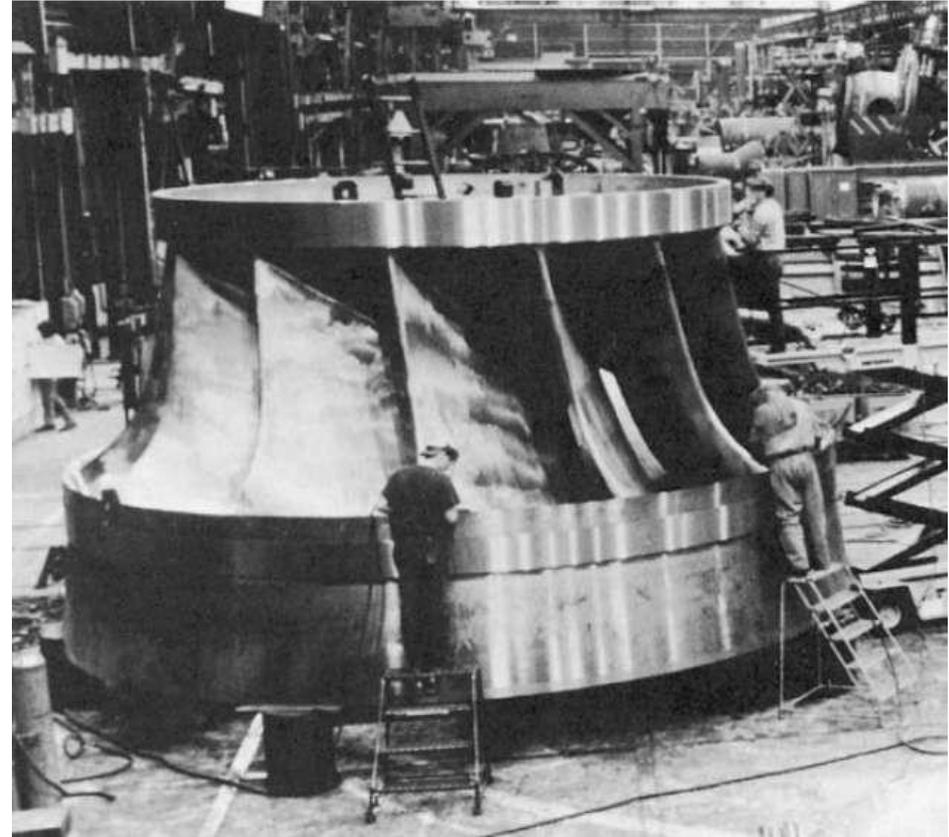
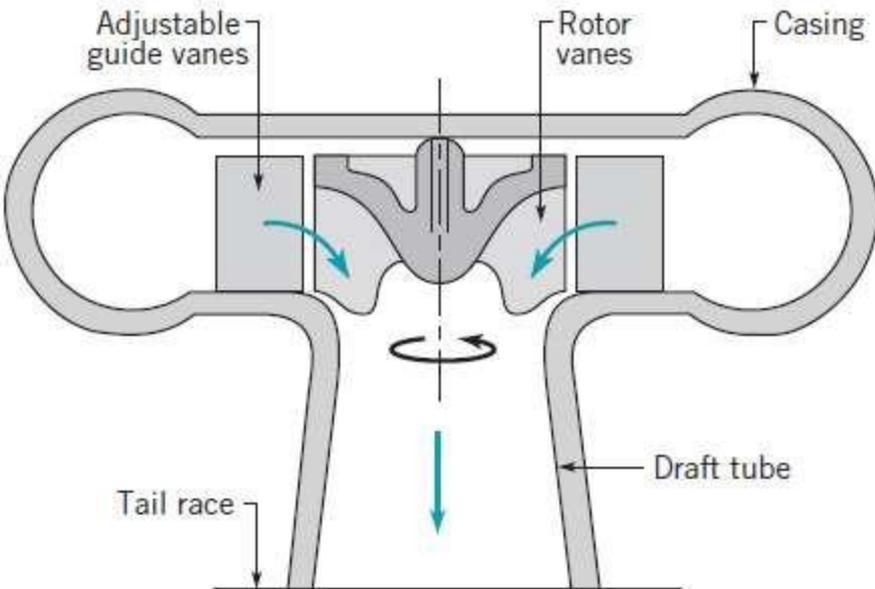
Gas turbines: the working fluid is air (+fuel); steam turbines: the working fluid is steam.

Turbines are classified into two basic types—***impulse turbines*** and ***reaction turbines***.

For hydraulic impulse turbines, the pressure drop across the rotor is zero; all of the pressure drop across the turbine stage occurs in the nozzle row. The ***Pelton wheel*** is a classical example of an impulse turbine.



For reaction turbines, on the other hand, the rotor is surrounded by a casing (or volute), which is completely filled with the working fluid. There is both a pressure drop and a fluid relative speed change across the rotor.

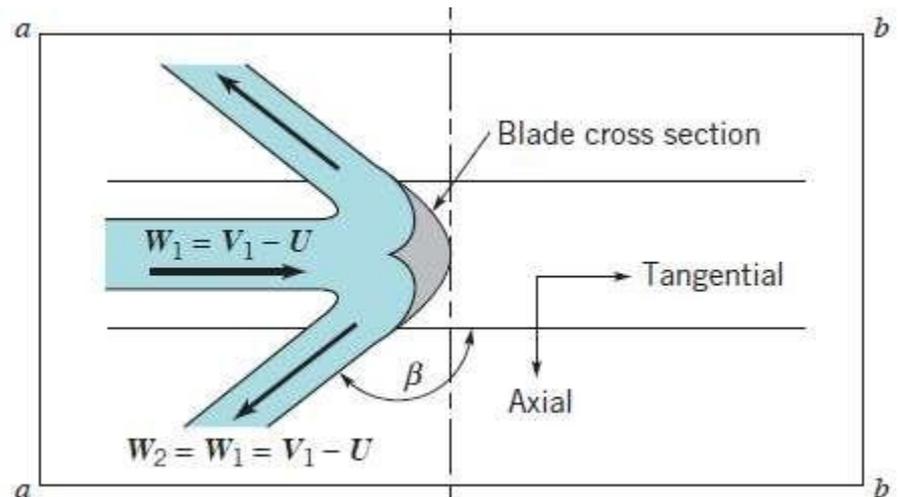
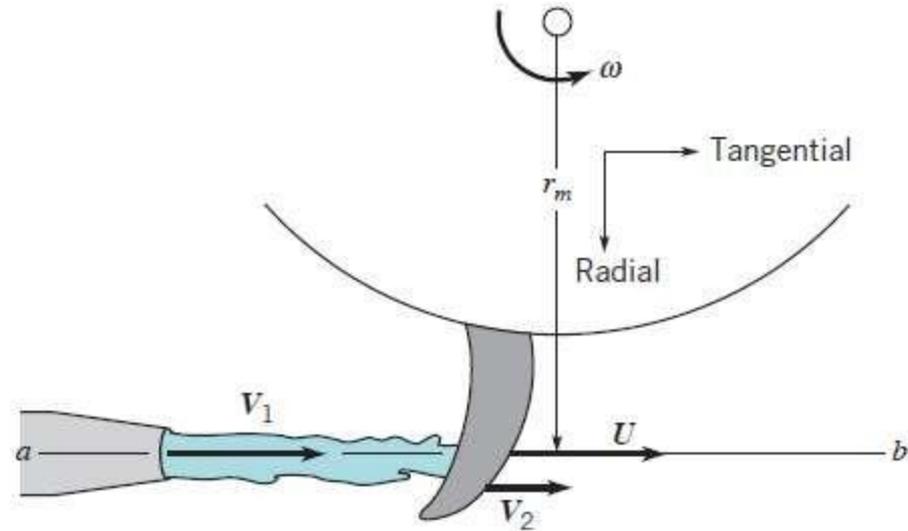
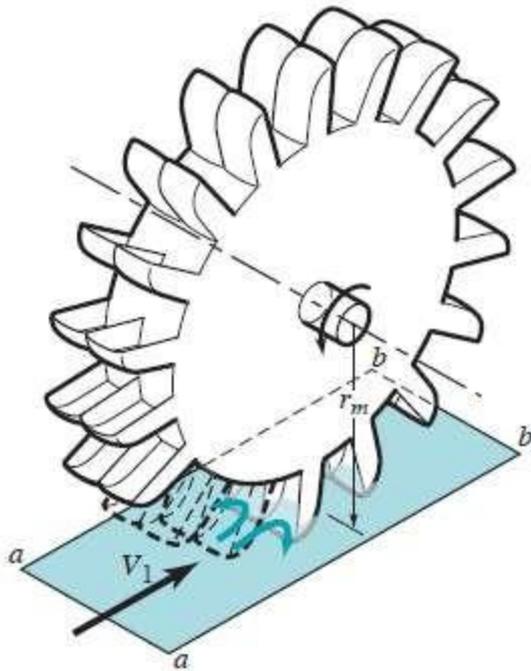


Both impulse and reaction turbines can be analyzed using the moment-of-momentum principles.

In general, impulse turbines are high-head, low-flowrate devices, while reaction turbines are low-head, high-flowrate devices.

Impulse Turbines

This type of turbines is most efficient when operated with a large head (for example, a water source from a lake located significantly above the turbine nozzle), which is converted into a relatively large velocity at the exit of the nozzle.



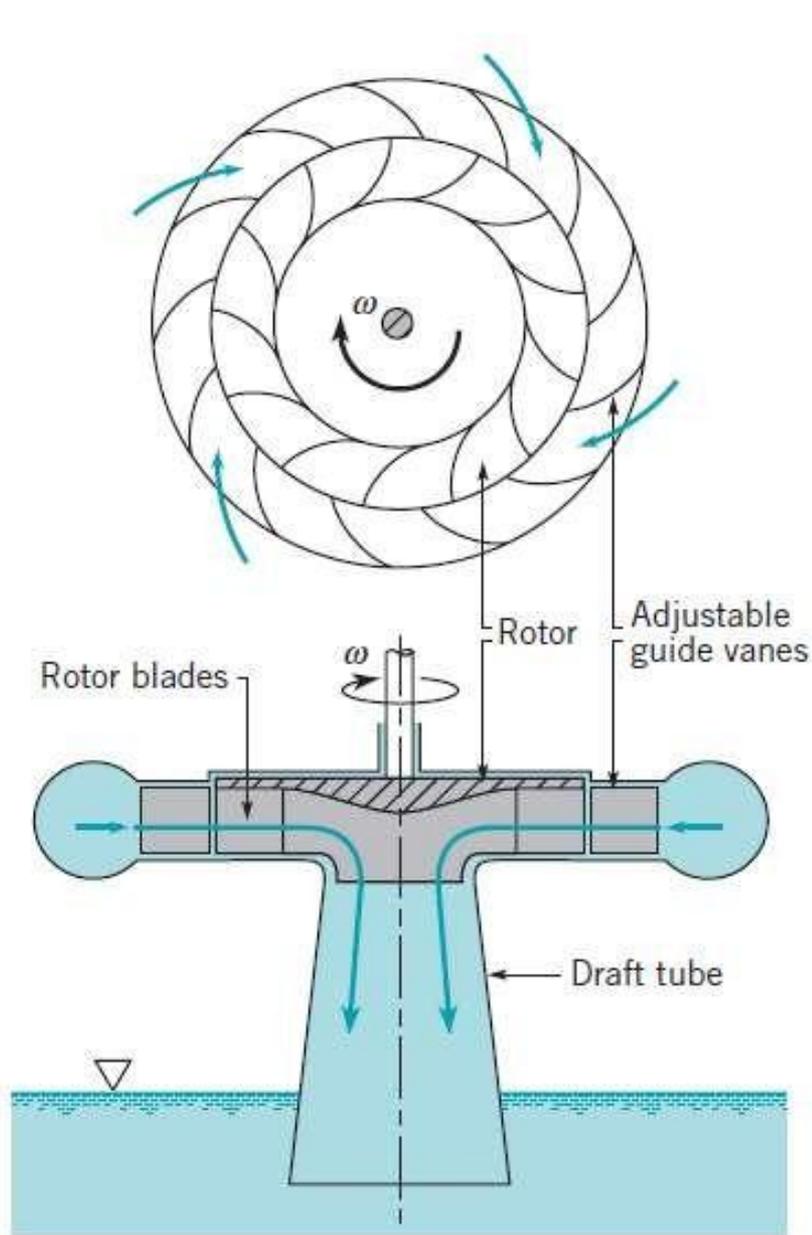
Reaction Turbines

Reaction turbines are best suited for higher flowrate and lower head situations such as are often encountered in hydroelectric power plants associated with a dammed river, for example.

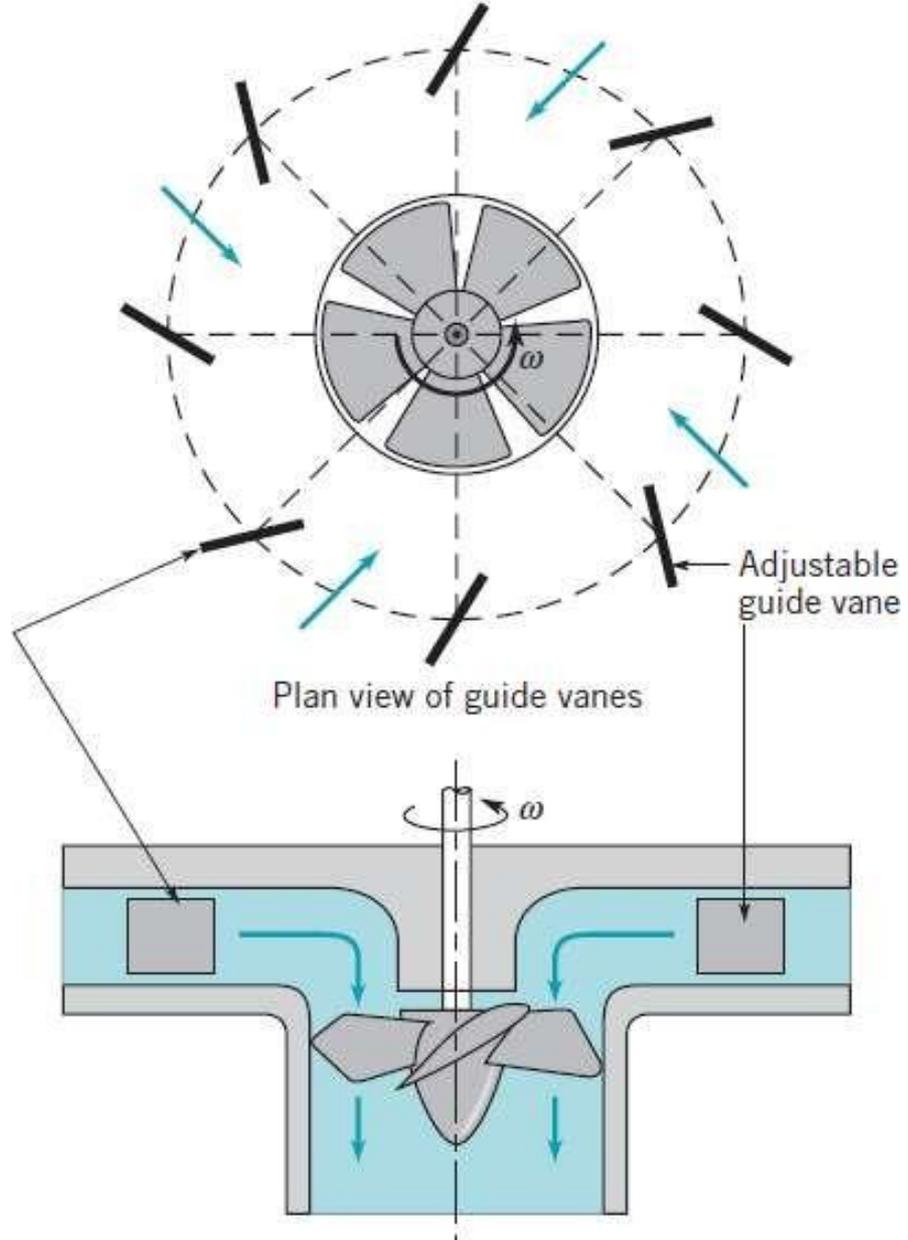
As with pumps, turbines are manufactured in a variety of configurations—radial-flow, mixed-flow, and axial-flow. Typical radial- and mixed-flow hydraulic turbines are called Francis turbines, named after James B. Francis, an American engineer. At very low heads the most efficient type of turbine is the axial-flow or propeller turbine. The Kaplan turbine, named after Victor Kaplan, a German professor, is an efficient axial-flow hydraulic turbine with adjustable blades.

The Euler turbomachine equation and the corresponding power equation for a centrifugal pump (radial) are equally valid for a radial-inflow turbine.

for an axial-flow Kaplan turbine, the fluid flows through the inlet guide vanes and achieves a tangential velocity in a vortex (swirl) motion before it reaches the rotor. Flow across the rotor contains a major axial component. Both the inlet guide vanes and the turbine blades can be adjusted by changing their setting angles to produce the best match for the specific operating conditions. For example, the operating head available may change from season to season and/or the flowrate through the rotor may vary.



Typical radial-flow Francis turbine



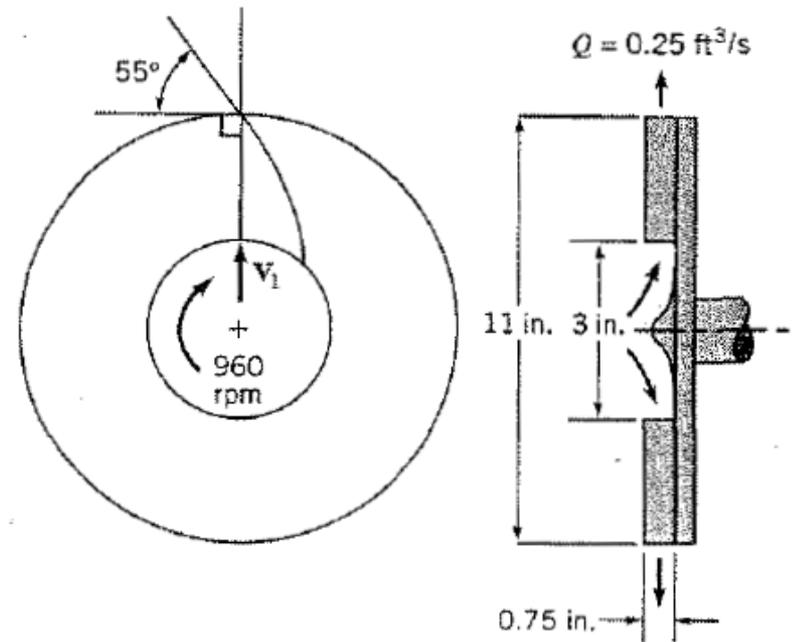
Typical axial-flow Kaplan turbine

Pumps and turbines are often thought of as the “inverse” of each other. Pumps add energy to the fluid; turbines remove energy.

Turbine efficiency is the inverse of pump efficiency:

$$\eta = \frac{\dot{W}_{\text{shaft}}}{\rho g Q h_a}$$

12.22 A centrifugal radial water pump has the dimensions shown in Fig. P12.22. The volume rate of flow is $0.25 \text{ ft}^3/\text{s}$, and the absolute inlet velocity is directed radially outward. The angular velocity of the impeller is 960 rpm . The exit velocity as seen from a coordinate system attached to the impeller can be assumed to be tangent to the vane at its trailing edge. Calculate the power required to drive the pump.



■ FIGURE P12.22.

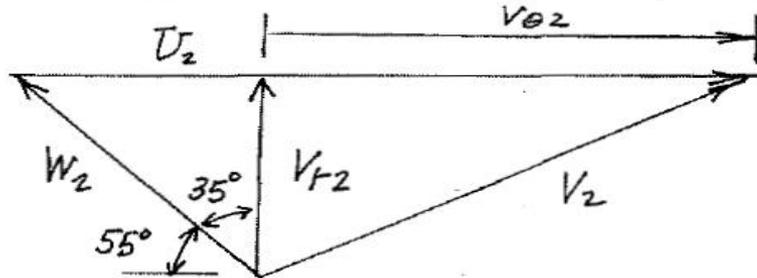
From Eq. 12.11, with $v_{\theta 1} = 0$,

$$\dot{W}_{\text{shaft}} = \rho Q U_2 V_{\theta 2} \quad (1)$$

To determine U_2 we use

$$U_2 = r_2 \omega = \left(\frac{5.5 \text{ in.}}{12 \text{ in./ft}} \right) \left(960 \frac{\text{rev}}{\text{min}} \right) \left(2\pi \frac{\text{rad}}{\text{rev}} \right) \left(\frac{1}{60 \frac{\text{s}}{\text{min}}} \right) = 46.1 \frac{\text{ft}}{\text{s}}$$

To obtain $V_{\theta 2}$ we use the exit velocity triangle shown below.



Since
and

$$V_{\theta 2} = U_2 - V_{t2} \tan 35^\circ$$

$$V_{t2} = \frac{Q}{A_2} = \frac{Q}{2\pi r_2 b_2} = \frac{(0.25 \frac{\text{ft}^3}{\text{s}})(144 \frac{\text{in}^2}{\text{ft}^2})}{(2\pi)(5.5 \text{ in.})(0.75 \text{ in.})} = 1.39 \frac{\text{ft}}{\text{s}}$$

it follows that

$$V_{\theta 2} = (46.1 - 1.39 \tan 35^\circ) \frac{\text{ft}}{\text{s}} = 45.1 \frac{\text{ft}}{\text{s}}$$

Thus, from Eq. (1)

$$\begin{aligned} \dot{W}_{\text{shaft}} &= (1.94 \frac{\text{slugs}}{\text{ft}^3}) (0.25 \frac{\text{ft}^3}{\text{s}}) (46.1 \frac{\text{ft}}{\text{s}}) (45.1 \frac{\text{ft}}{\text{s}}) \\ &= 1010 \frac{\text{ft} \cdot \text{lb}}{\text{s}} \end{aligned}$$

or

$$\dot{W}_{\text{shaft}} = \frac{1010 \frac{\text{ft} \cdot \text{lb}}{\text{s}}}{550 \frac{\text{ft} \cdot \text{lb}}{\text{s} \cdot \text{hp}}} = \underline{\underline{1.84 \text{ hp}}}$$

12.53 A certain axial-flow pump has a specific speed of $N_s = 5.0$. If the pump is expected to deliver 3000 gpm when operating against a 15-ft head, at what speed (rpm) should the pump be run?

Since

$$N_s = \frac{\omega \text{ (rad/s)} \sqrt{Q \text{ (ft}^3\text{/s)}}}{[g \text{ (ft/s}^2) h_a \text{ (ft)}]^{3/4}}$$

for $N_s = 5.0$, $g = 32.2 \text{ ft/s}^2$, $h_a = 15 \text{ ft}$, and ω

$$Q = \frac{3000 \frac{\text{gal}}{\text{min}}}{(7.48 \frac{\text{gal}}{\text{ft}^3})(60 \frac{\text{s}}{\text{min}})} = 6.68 \frac{\text{ft}^3}{\text{s}}$$

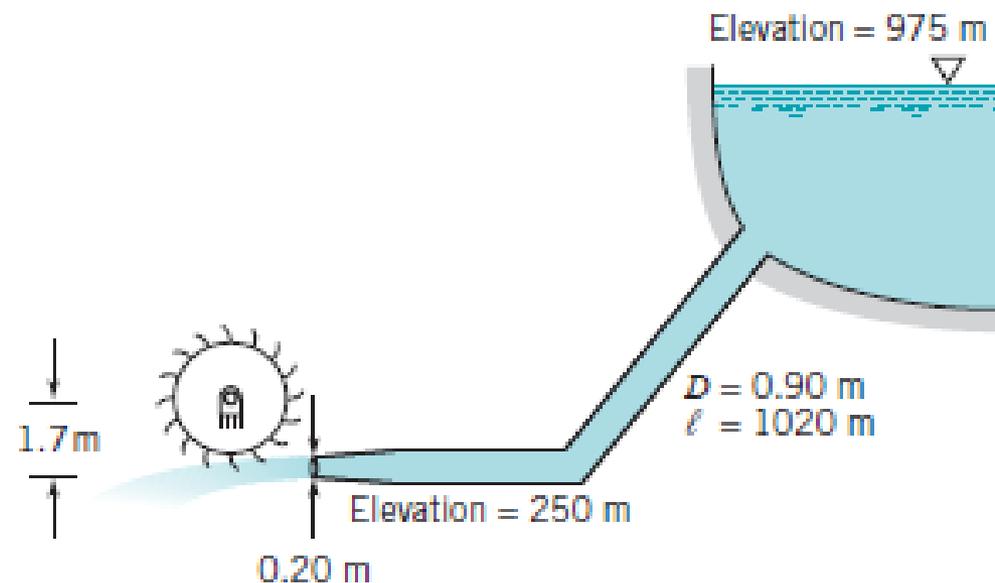
it follows that

$$\begin{aligned} \omega \text{ (rad/s)} &= \frac{(5.0) \left[(32.2 \frac{\text{ft}}{\text{s}^2})(15 \text{ ft}) \right]^{3/4}}{\sqrt{6.68 \frac{\text{ft}^3}{\text{s}}}} \\ &= 199 \frac{\text{rad}}{\text{s}} \end{aligned}$$

Hence

$$\begin{aligned} \omega \text{ (rpm)} &= \frac{(199 \frac{\text{rad}}{\text{s}})(60 \frac{\text{s}}{\text{min}})}{2\pi \frac{\text{rad}}{\text{rev}}} \\ &= \underline{\underline{1900 \text{ rpm}}} \end{aligned}$$

12.66 Water for a Pelton wheel turbine flows from the headwater and through the penstock as shown in Fig. P12.66. The effective friction factor for the penstock, control valves, and the like is 0.032



■ FIGURE P12.66

$$\dot{W}_{\text{shaft}} = \rho Q U (U - V_1) (1 - \cos \beta) \text{ or for maximum power } \beta = 180^\circ, U = \frac{V_1}{2}$$

Thus,

$$\dot{W}_{\text{shaft max}} = -\rho Q \frac{V_1^2}{2} \quad (1)$$

$$\text{But } \frac{p_0}{\rho} + \frac{V_0^2}{2g} + z_0 = \frac{p_1}{\rho} + \frac{V_1^2}{2g} + z_1 + f \frac{L}{D} \frac{V^2}{2g} \text{ where } p_0 = p_1 = 0, z_0 = 975 \text{ m, } z_1 = 250 \text{ m, and } V_0 = 0$$

Hence,

$$z_0 = z_1 + \frac{V_1^2}{2g} + f \frac{L}{D} \frac{V^2}{2g} \text{ where } A_1 V_1 = AV \quad (2)$$

$$\text{or } \frac{\pi}{4} d_1^2 V_1 = \frac{\pi}{4} D^2 V. \text{ That is } V = \left(\frac{d_1}{D}\right)^2 V_1 = \left(\frac{0.2 \text{ m}}{0.9 \text{ m}}\right)^2 V_1 = 0.0494 V_1$$

so that Eq. (2) becomes:

$$975 \text{ m} = 250 \text{ m} + \frac{V_1^2}{2(9.81 \frac{\text{m}}{\text{s}^2})} \left[1 + 0.032 \left(\frac{1020 \text{ m}}{0.9 \text{ m}}\right) (0.0494)^2 \right] \text{ where } V_1 \sim \frac{\text{m}}{\text{s}}$$

$$\text{or } V_1 = 114.3 \frac{\text{m}}{\text{s}}$$

Hence,

$$Q = A_1 V_1 = \frac{\pi}{4} (0.2 \text{ m})^2 (114.3 \frac{\text{m}}{\text{s}}) = 3.56 \frac{\text{m}^3}{\text{s}}$$

Therefore, from Eq. (1):

$$\dot{W}_{\text{shaft max}} = -(999 \frac{\text{kg}}{\text{m}^3}) (3.56 \frac{\text{m}^3}{\text{s}}) \frac{(114.3 \frac{\text{m}}{\text{s}})^2}{2} = 23.2 \times 10^6 \frac{\text{N} \cdot \text{m}}{\text{s}} = \underline{\underline{23,200 \text{ kW}}}$$