

Turbomachinery

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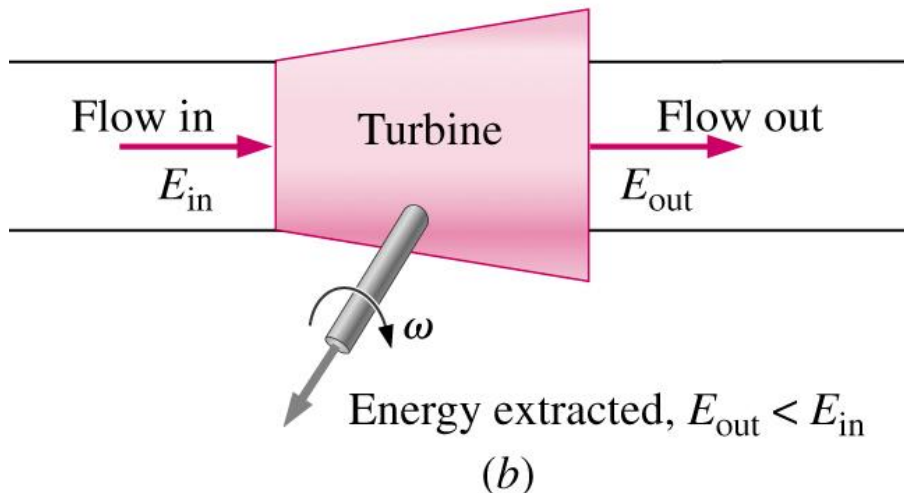
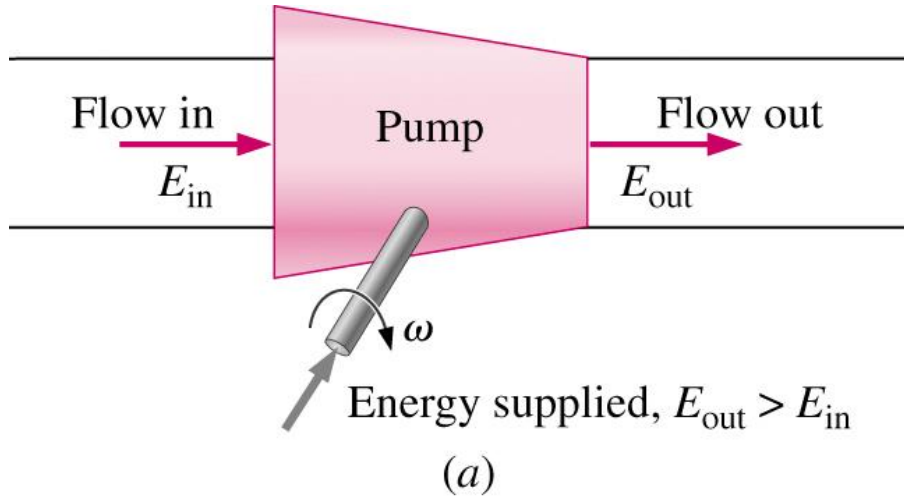
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Objectives

- Identify various types of pumps and turbines, and understand how they work
- Apply dimensional analysis to design new pumps or turbines that are geometrically similar to existing pumps or turbines
- Perform basic vector analysis of the flow into and out of pumps and turbines
- Use specific speed for preliminary design and selection of pumps and turbines

Categories



- Pump: adds energy to a fluid, resulting in an increase in pressure across the pump.
- Turbine: extracts energy from the fluid, resulting in a decrease in pressure across the turbine.

Categories

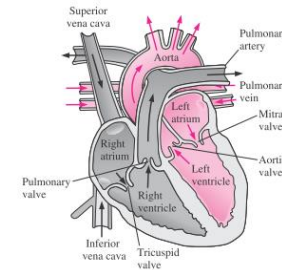
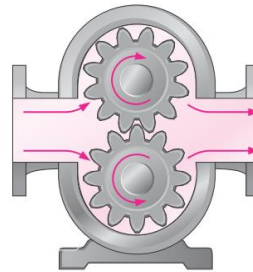
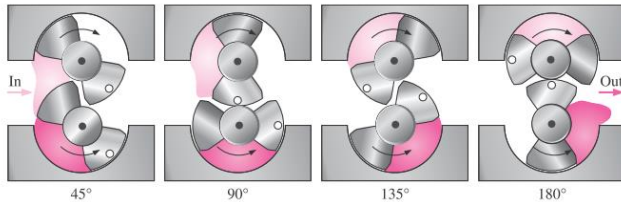
- For gases, pumps are further broken down into
 - **Fans:** Low *pressure gradient*, High *volume flow rate*. Examples include ceiling fans and propellers.
 - **Blower:** Medium *pressure gradient*, Medium *volume flow rate*. Examples include centrifugal and squirrel-cage blowers found in furnaces, leaf blowers, and hair dryers.
 - **Compressor:** High *pressure gradient*, Low *volume flow rate*. Examples include air compressors for air tools, refrigerant compressors for refrigerators and air conditioners.

	Fan	Blower	Compressor
ΔP	Low	Medium	High
Q	High	Medium	Low

Categories

■ Positive-displacement machines

- Closed volume is used to squeeze or suck fluid.
- Pump: human heart
- Turbine: home water meter

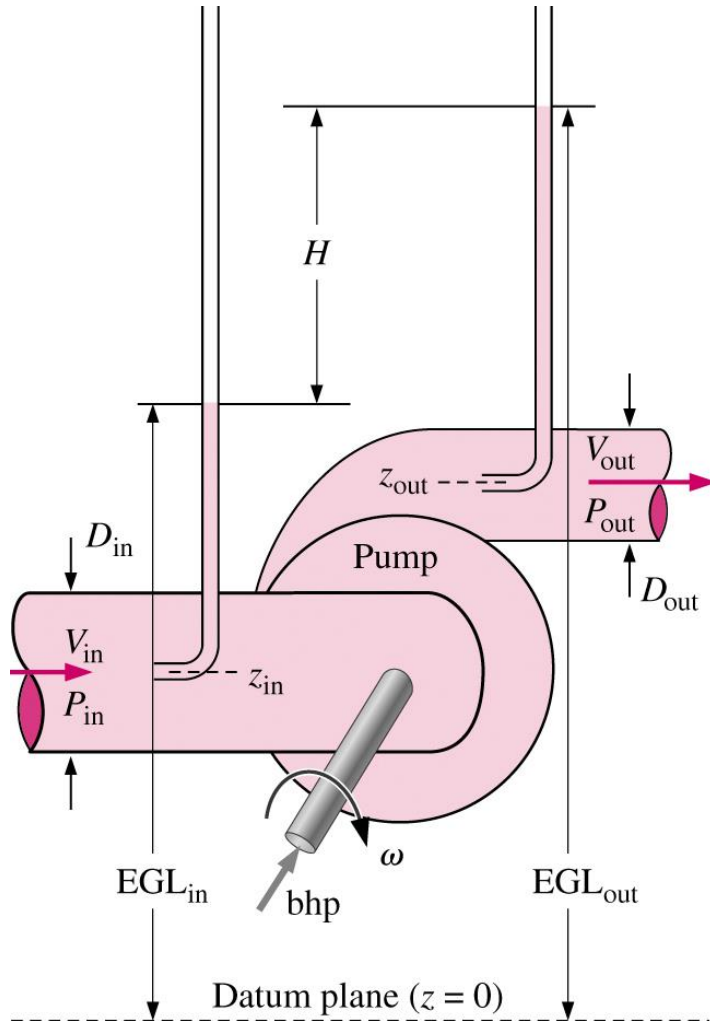


■ Dynamic machines

- No closed volume. Instead, rotating blades supply or extract energy.
- Enclosed/Ducted Pumps: torpedo propulsor
- Open Pumps: propeller or helicopter rotor
- Enclosed Turbines: hydroturbine
- Open Turbines: wind turbine



Pump Head



■ Net Head

$$H = \left(\frac{P}{\rho g} + \frac{V^2}{2g} + z \right)_{out} - \left(\frac{P}{\rho g} + \frac{V^2}{2g} + z \right)_{in}$$

$$= EGL_{out} - EGL_{in}$$

■ Water horsepower

$$\dot{W}_{water \text{ horsepower}} = \dot{m}gH = \rho gQH$$

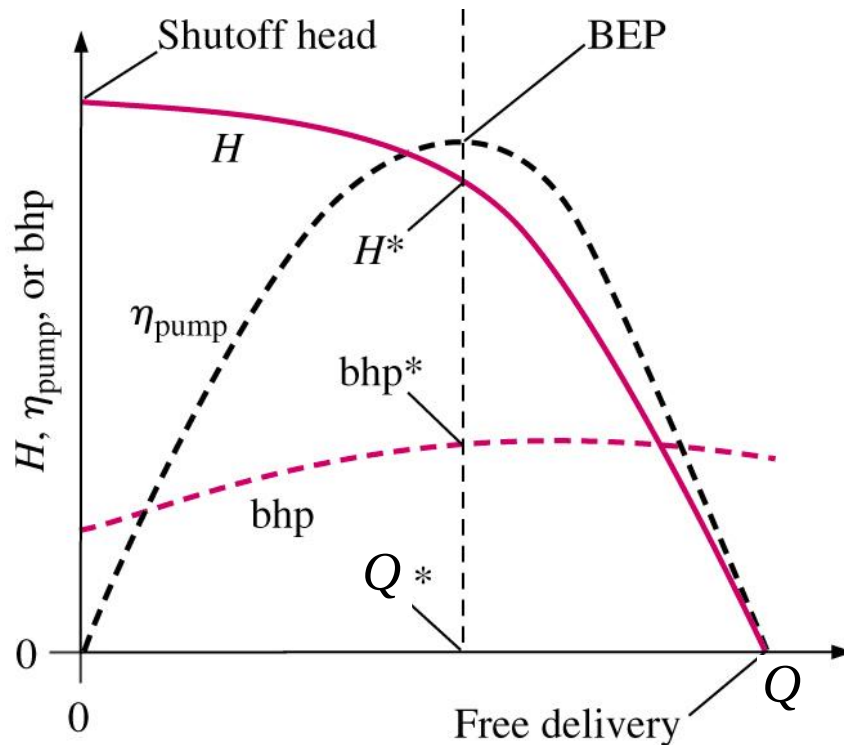
■ Brake horsepower

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

■ Pump efficiency

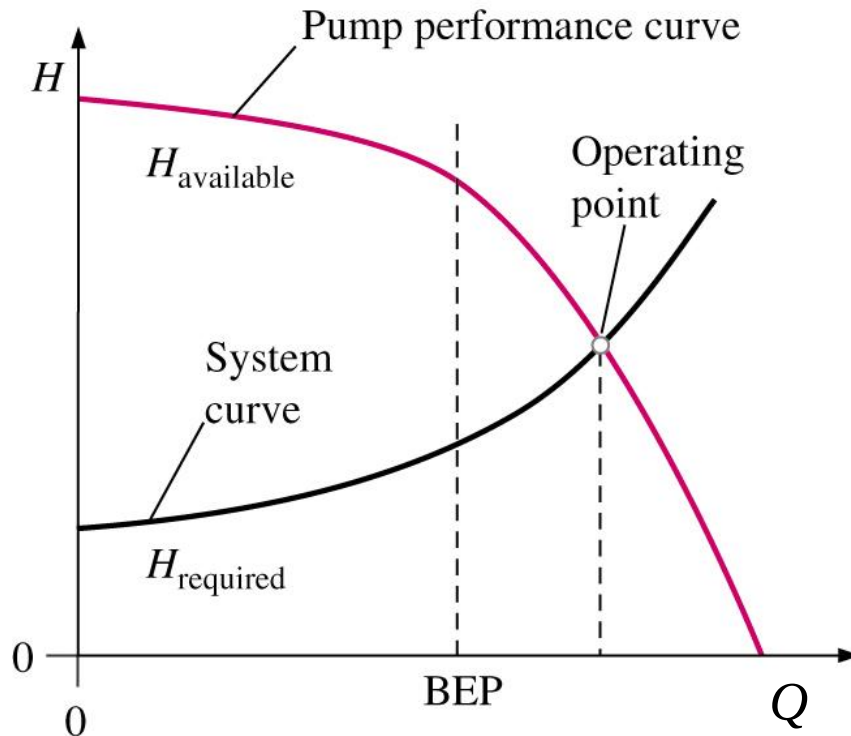
$$\eta_{pump} = \frac{\rho gQH}{\omega T_{shaft}}$$

Matching a Pump to a Piping System



- Pump-performance curves for a centrifugal pump
- BEP: best efficiency point
- H^* , bhp^* , Q^* correspond to BEP
- Shutoff head: achieved by closing outlet ($Q=0$)
Free delivery: no load on system ($H_{\text{required}} = 0$)

Matching a Pump to a Piping System



- Steady operating point:

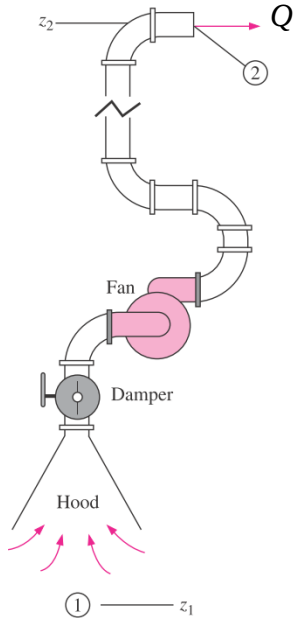
$$H_{required} = H_{available}$$

- Energy equation:

$$H_{required} = h_{pump,u} = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 V_2^2 - \alpha_1 V_1^2}{2g} + z_2 - z_1 + h_{L,total}$$

An arrow points from the 'Energy equation' text to this equation.

Example 1: Matching a Pump to a Piping System



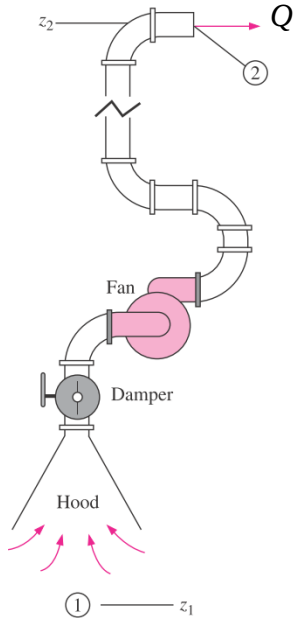
Manufacturer's performance data for the fan of Example 14-1*

Q , cfm	$(\delta P)_{fan}$, inches H_2O
0	0.90
250	0.95
500	0.90
750	0.75
1000	0.40
1200	0.0

* Note that the pressure rise data are listed as inches of water, even though air is the fluid. This is common practice in the ventilation industry.

- In a local ventilation system the duct is round and is constructed of galvanized steel. The inner diameter (ID) of the duct is $D_{in} = 9.06$ in (0.230 m), and its total length is $L = 44.0$ ft (13.4 m). There are five elbows along the duct. Roughness $\epsilon = 0.15$ mm. To ensure adequate ventilation, the minimum required Q through the duct is $Q = 600$ cfm (ft^3/min), or 0.283 m^3/s at $25^\circ C$.
- A centrifugal fan with 9.0-in inlet and outlet diameters is available. Its performance data are shown in Table.
- $K_{L,in} = 1.3$, $K_{L,damper} = 1.8$, $K_{L,elbow} = 0.21$
- Predict the operating point of this local ventilation system, and draw a plot of required and available fan pressure rise as functions of volume flow rate. Is the chosen fan adequate?

Example 1: Matching a Pump to a Piping System



Manufacturer's performance data for the fan of Example 14-1*

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Assumptions 1 The flow is steady. 2 The concentration of contaminants in the air is low; the fluid properties are those of air alone. 3 The flow at the outlet is fully developed turbulent pipe flow with $\alpha = 1.05$.

Properties For air at 25°C , $\nu = 1.562 \times 10^{-5} \text{ m}^2/\text{s}$ and $\rho = 1.184 \text{ kg/m}^3$. Standard atmospheric pressure is $P_{\text{atm}} = 101.3 \text{ kPa}$.

$$H_{\text{required}} = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 V_2^2 - \alpha_1 V_1^2}{2g} + \underbrace{(z_2 - z_1)}_{\text{negligible for gases}} + h_{L,\text{total}} \Rightarrow H_{\text{required}} = \frac{\alpha_2 V_2^2}{2g} + h_{L,\text{total}}$$

$$h_{L,\text{total}} = \left(f \frac{L}{D} + \sum K_L \right) \frac{V^2}{2g}$$

$$\text{Re} = \frac{DV}{\nu} = \frac{D}{\nu} \frac{4Q}{\pi D^2} = \frac{4Q}{\nu \pi D}$$

$$V = V_2 = 6.81 \text{ m/s}$$

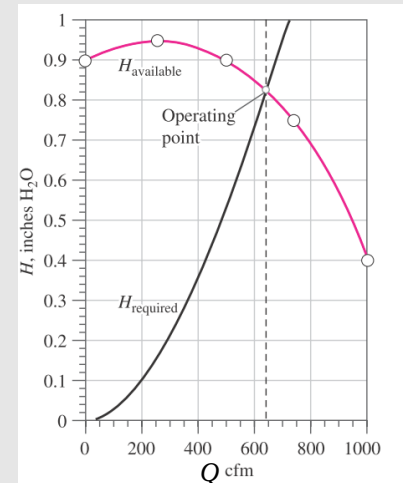
$$\text{Re} = \frac{4(0.283 \text{ m}^3/\text{s})}{(1.562 \times 10^{-5} \text{ m}^2/\text{s})\pi(0.230 \text{ m})} = 1.00 \times 10^5$$

From the Moody chart (or the Colebrook equation) at this Reynolds number and roughness factor, the friction factor is $f = 0.0209$. The sum of all the minor loss coefficients is

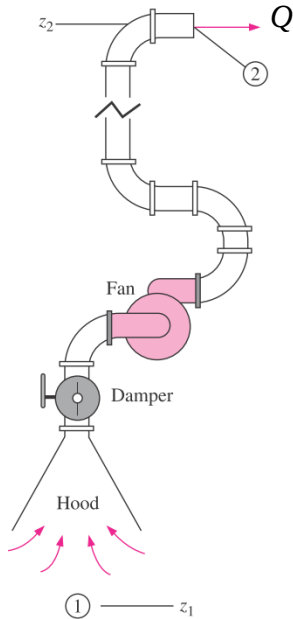
$$\text{Minor losses:} \quad \sum K_L = 1.3 + 5(0.21) + 1.8 = 4.15$$

$$\begin{aligned} H_{\text{required}} &= \left(\alpha_2 + f \frac{L}{D} + \sum K_L \right) \frac{V^2}{2g} \\ &= \left(1.05 + 0.0209 \frac{13.4 \text{ m}}{0.230 \text{ m}} + 4.15 \right) \frac{(6.81 \text{ m/s})^2}{2(9.81 \text{ m/s}^2)} = 15.2 \text{ m of air} \end{aligned}$$

$$\begin{aligned} H_{\text{required, inches of water}} &= H_{\text{required, air}} \frac{\rho_{\text{air}}}{\rho_{\text{water}}} \\ &= (15.2 \text{ m}) \frac{1.184 \text{ kg/m}^3}{998.0 \text{ kg/m}^3} \left(\frac{1 \text{ in}}{0.0254 \text{ m}} \right) \\ &= 0.709 \text{ inches of water} \end{aligned}$$



Example 1: Matching a Pump to a Piping System

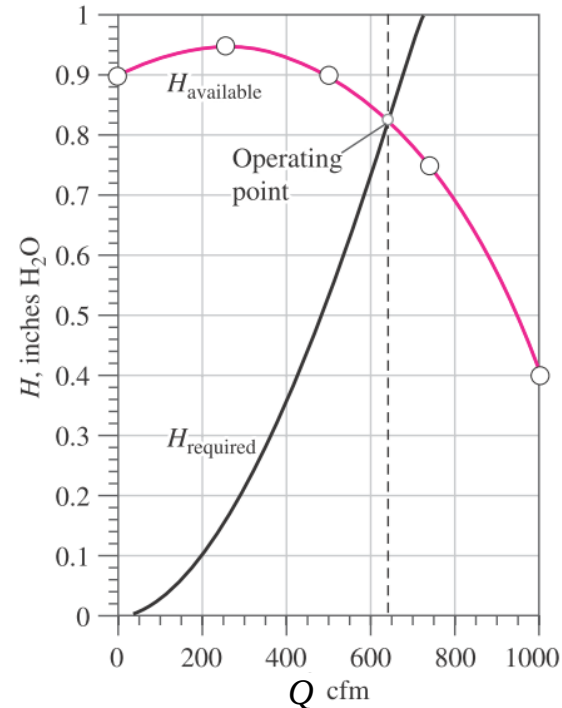


Manufacturer's performance data for the fan of Example 14-1*

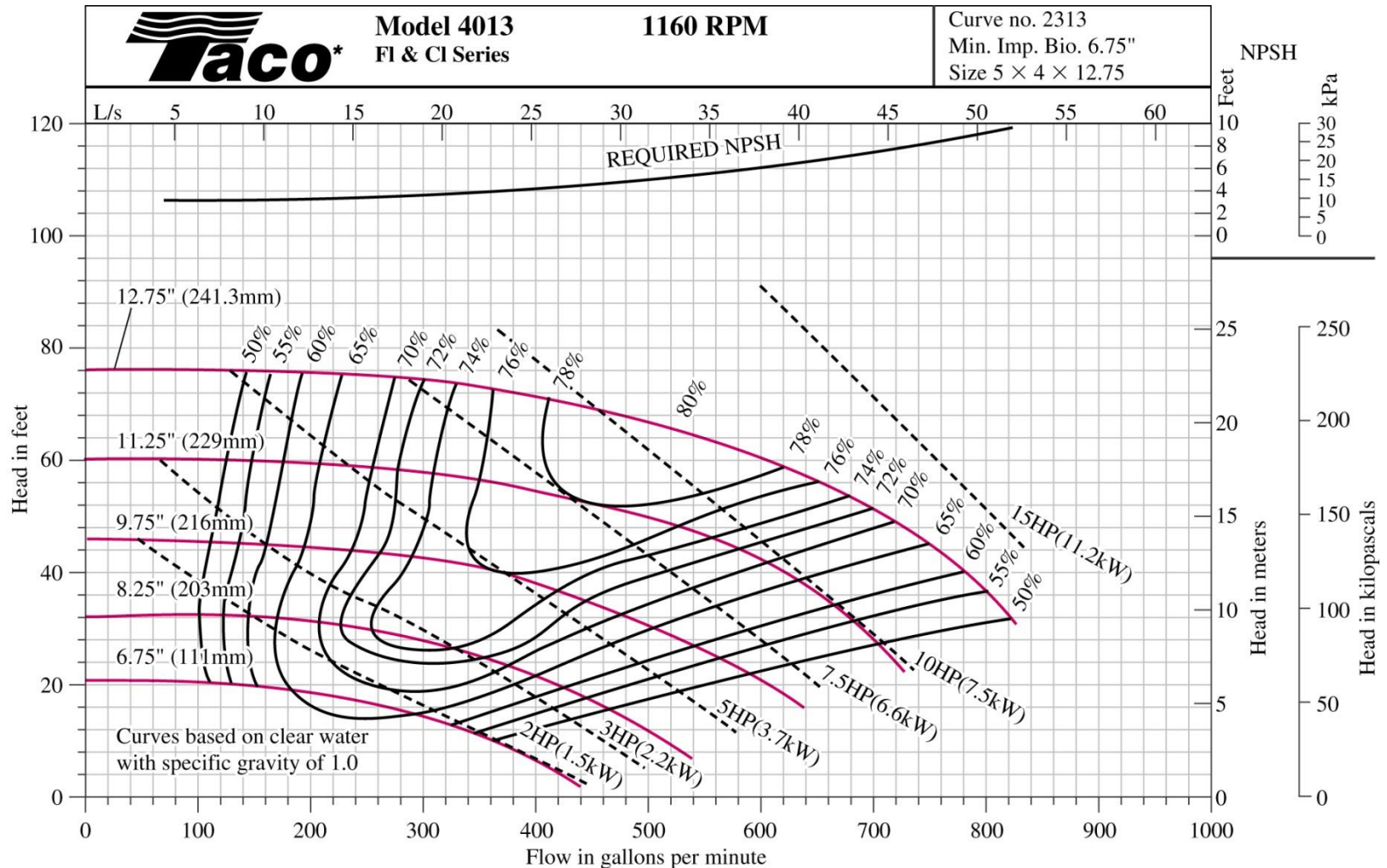
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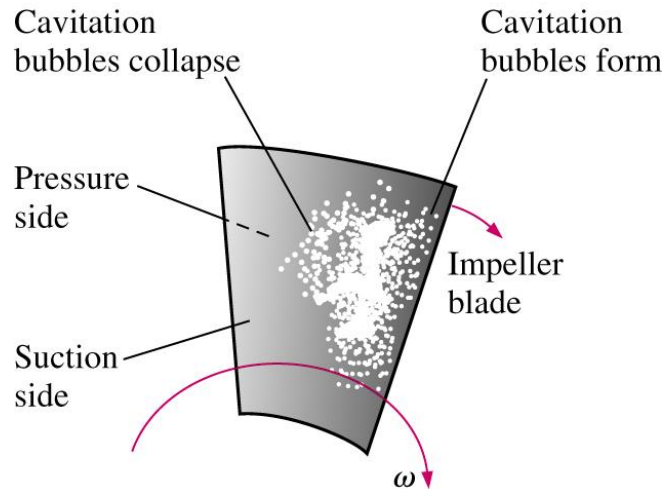
- We repeat the calculations at several values of volume flow rate, and compare to the available net head of the fan in Figure. The operating point is at a $Q \approx 650$ cfm, at which both the required and available net head equal about 0.83 inches of water.
- We conclude that the chosen fan is more than adequate for the job.



Manufacturer Performance Plot

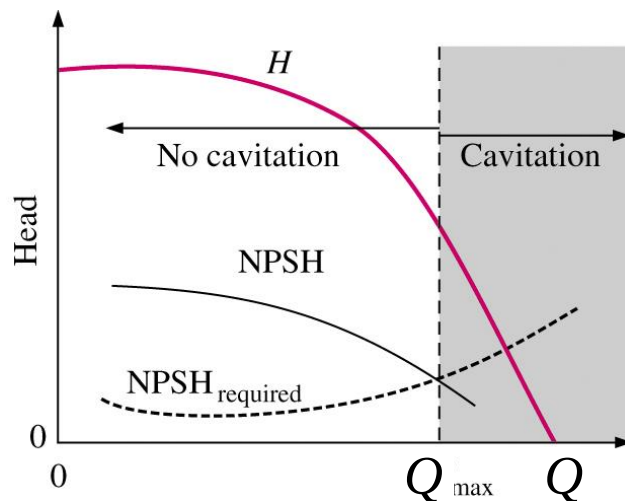


Pump Cavitation and NPSH



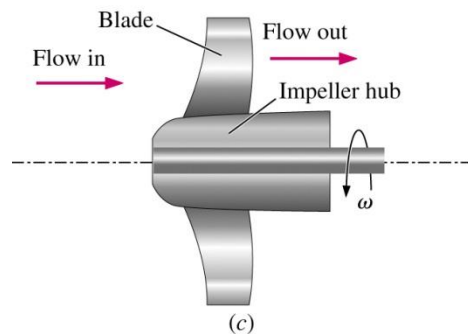
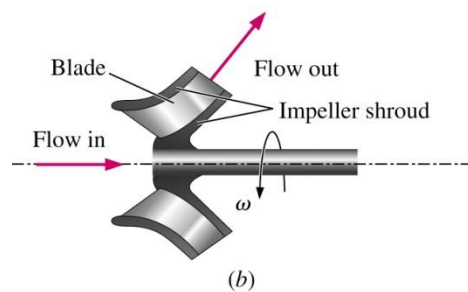
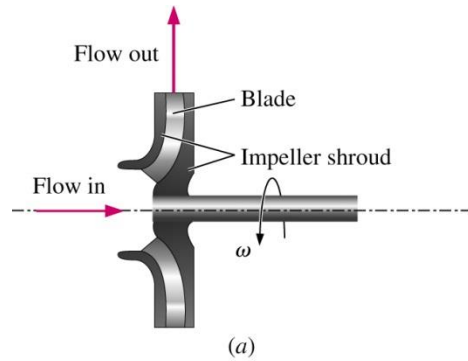
- Cavitation should be avoided due to erosion damage and noise.
- Cavitation occurs when $P < P_v$
- **Net positive suction head**

$$NPSH = \left(\frac{P}{\rho g} + \frac{V^2}{2g} \right)_{\text{pump inlet}} - \frac{P_v}{\rho g}$$



- $NPSH_{\text{required}}$ curves are created through systematic testing over a range of flow rates Q .

Dynamic Pumps



■ Dynamic Pumps include

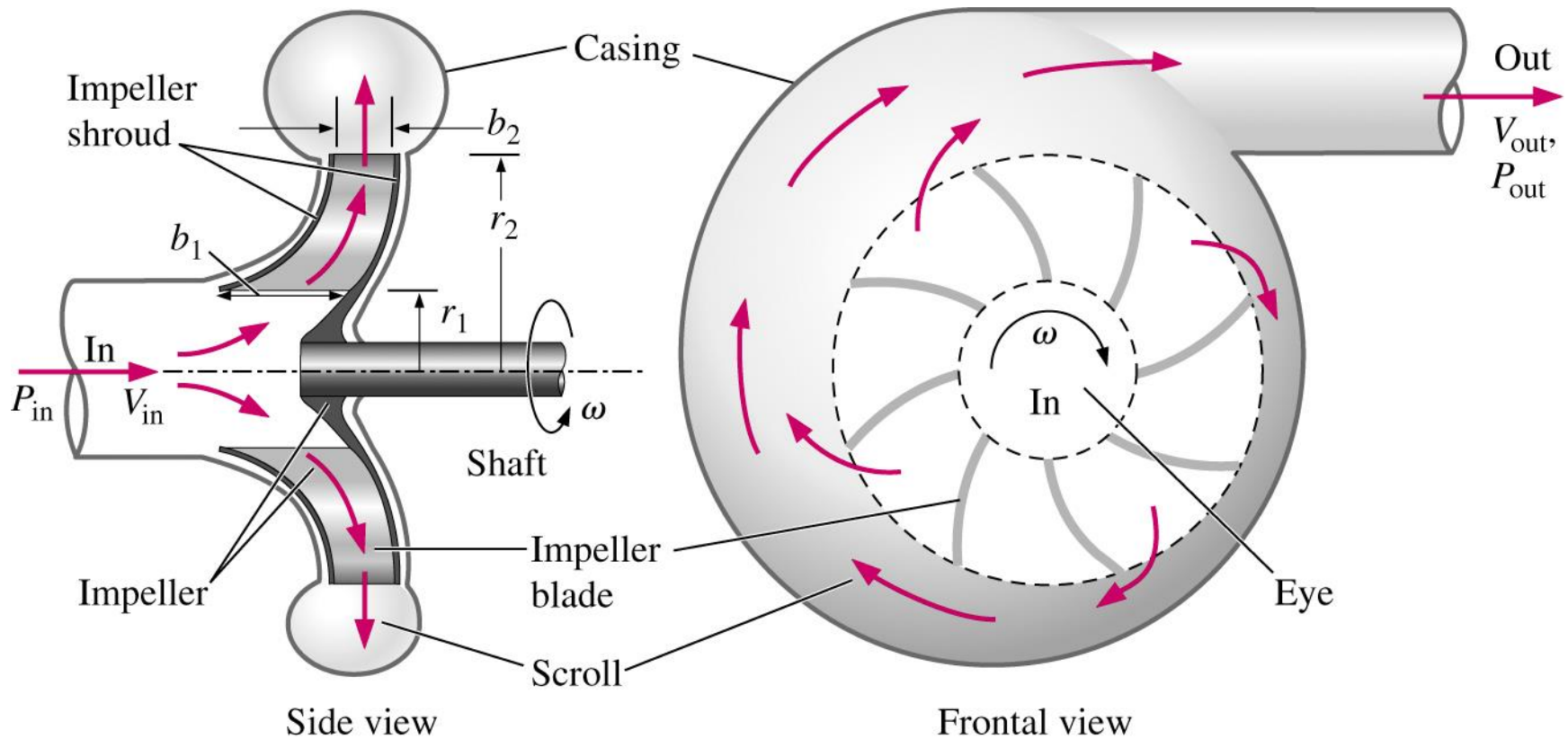
- *centrifugal pumps*: fluid enters axially, and is discharged radially.
- *mixed--flow pumps*: fluid enters axially, and leaves at an angle between radially and axially.
- *axial pumps*: fluid enters and leaves axially.

Centrifugal Pumps

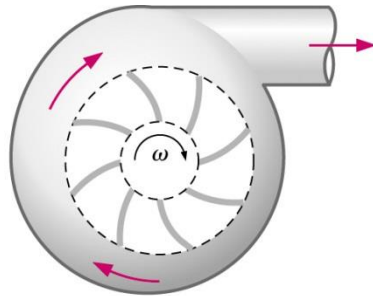


- Snail--shaped scroll
- Most common type of pump: homes, autos, industry.

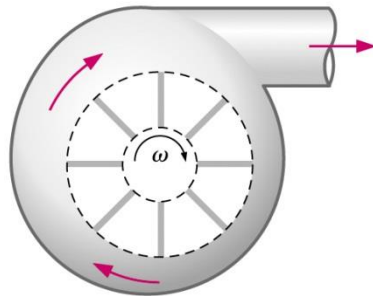
Centrifugal Pumps



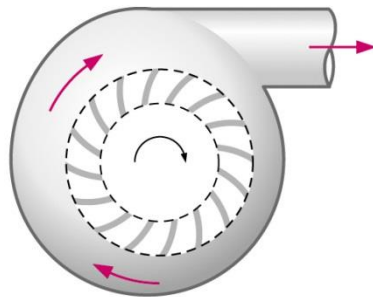
Centrifugal Pumps: Blade Design



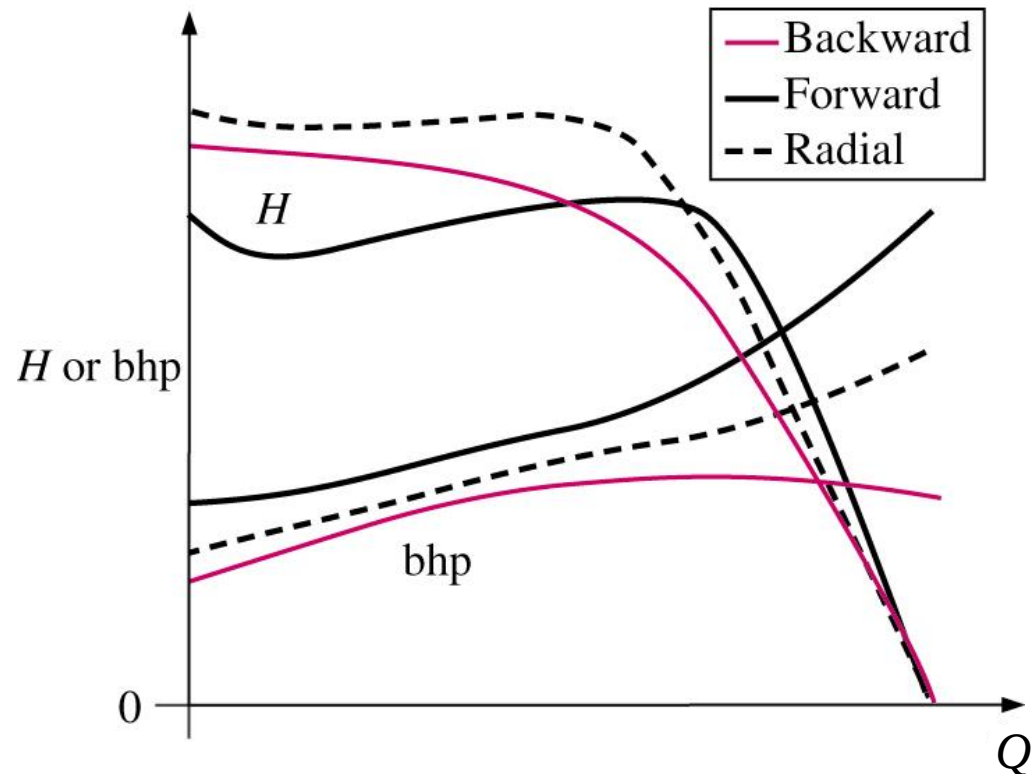
(a)



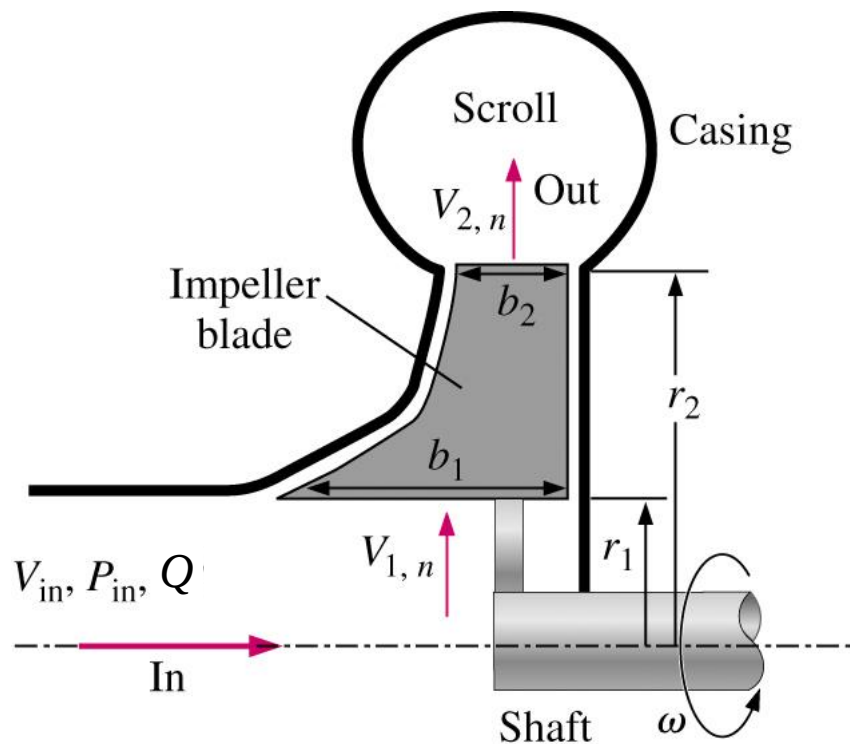
(b)



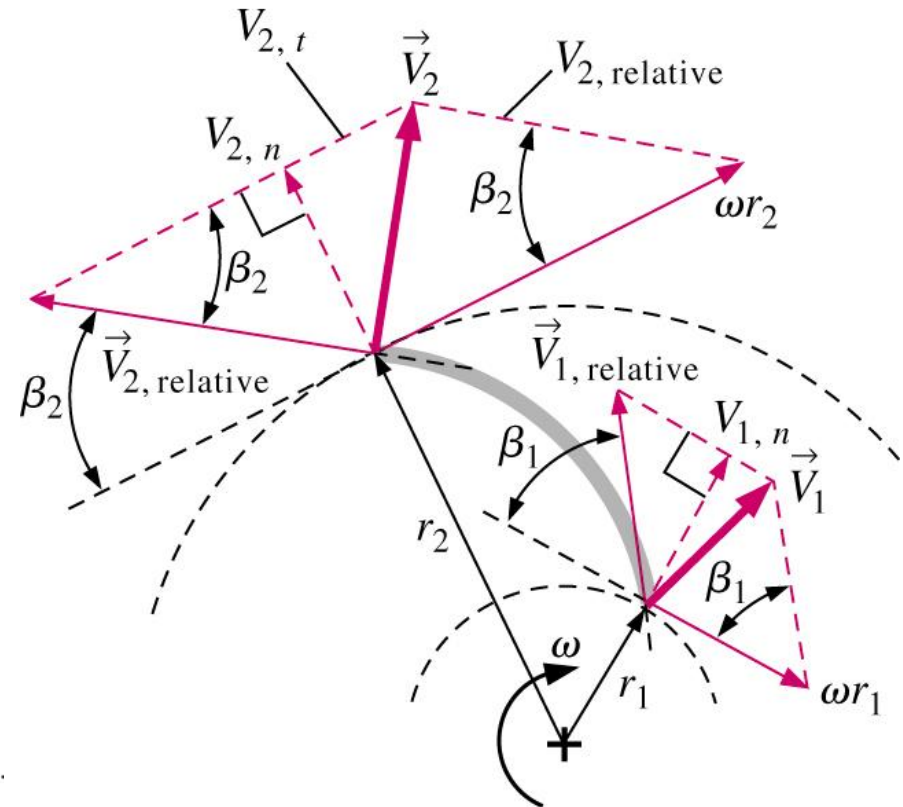
(c)



Centrifugal Pumps: Blade Design



Side view of impeller blade.



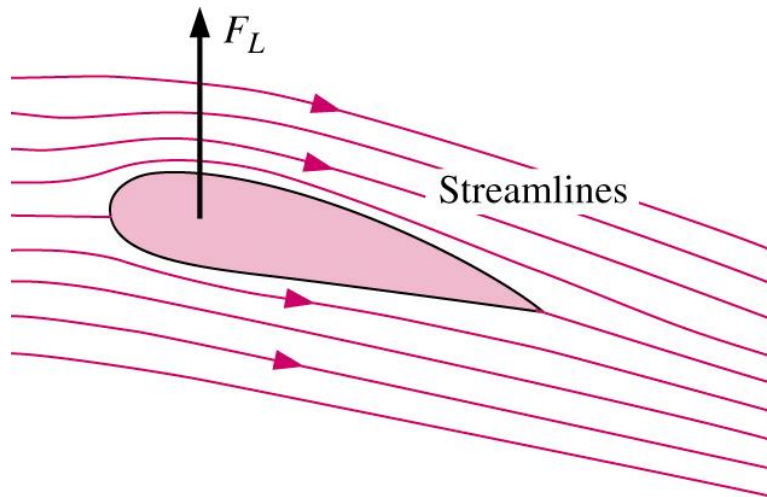
Vector analysis of leading and trailing edges.

Axial Pumps

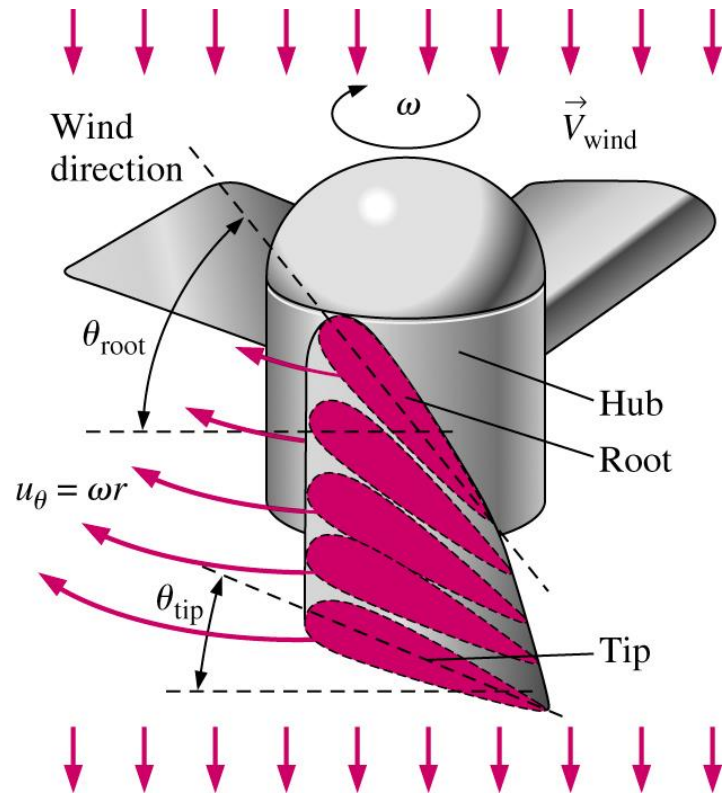


Open vs. Ducted Axial Pumps

Open Axial Pumps



Blades generate thrust like wing generates lift.



Propeller has radial twist to take into account for angular velocity ($=\omega r$)

Dimensional Analysis

- Π analysis gives 3 new nondimensional parameters

- Head coefficient $C_H = \frac{gH}{\omega^2 D^2}$

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

- Power coefficient $C_P = \frac{bhp}{\rho \omega^3 D^5}$

- Reynolds number also appears, but in terms of angular rotation

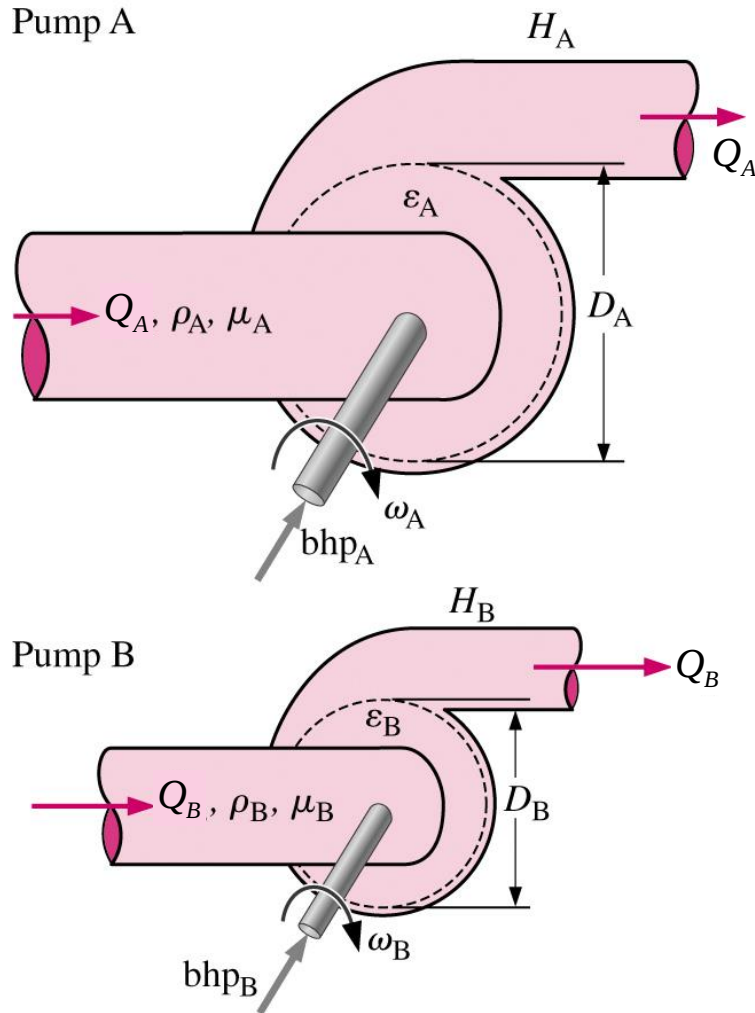
- Reynolds number $Re = \frac{\rho \omega D^2}{\mu}$

- Functional relation is

- Head coefficient $C_H = f(C_Q, Re, \epsilon/D)$

- Power coefficient $C_P = f(C_Q, Re, \epsilon/D)$

Dimensional Analysis



- If two pumps are geometrically similar, and
- The independent Π 's are similar, i.e.,

$$C_{Q,A} = C_{Q,B}$$

$$Re_A = Re_B$$

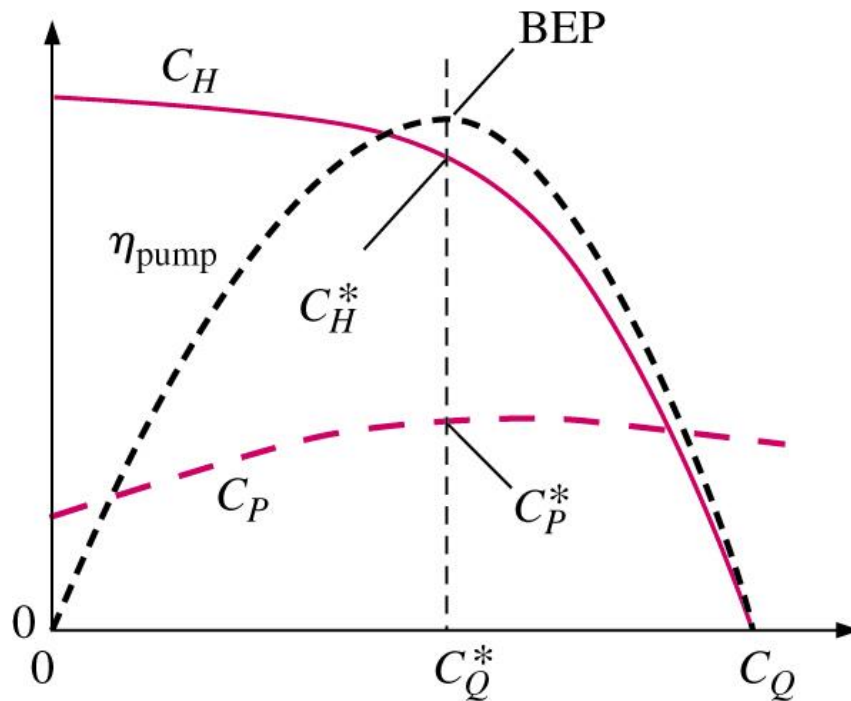
$$\varepsilon_A/D_A = \varepsilon_B/D_B$$

- Then the dependent Π 's will be the same

$$C_{H,A} = C_{H,B}$$

$$C_{P,A} = C_{P,B}$$

Dimensional Analysis



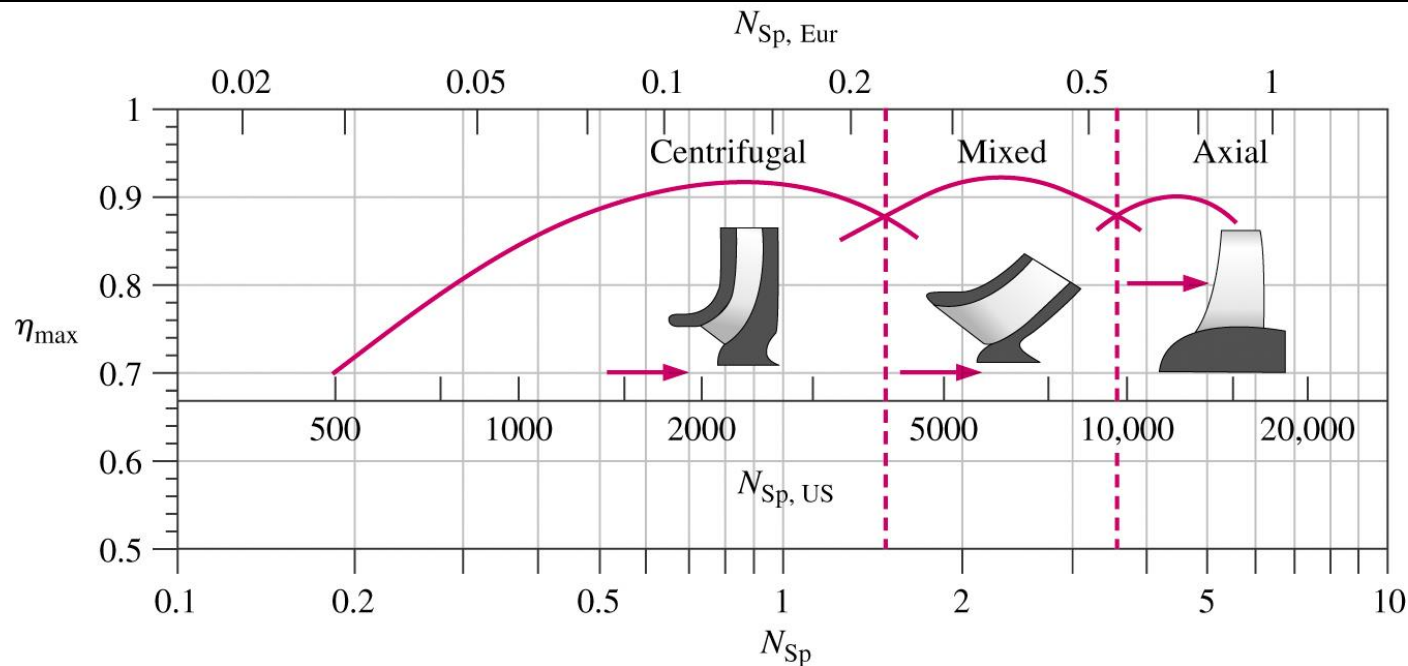
$$\eta = \frac{\rho Q g H}{b h_p} = \frac{C_Q C_H}{C_P} = f(C_Q)$$

- When plotted in nondimensional form, all curves of a family of geometrically similar pumps collapse onto one set of *nondimensional pump performance curves*

- Note: Reynolds number and roughness can often be neglected,

$$C_H \approx f(C_Q), C_P \approx f(C_Q)$$

Pump Specific Speed



Pump Specific Speed is used to characterize the operation of a pump at BEP and is useful for preliminary pump selection.

$$N_{SP} = \frac{C_Q^{1/2}}{C_H^{3/4}} = \frac{\omega Q^{1/2}}{(gH)^{3/4}}$$

Affinity Laws

- For two homologous states A and B, we can use Π variables to develop ratios (similarity rules, affinity laws, scaling laws).

$$C_{Q,A} = C_{Q,B} \Rightarrow \frac{Q_B}{Q_A} = \frac{\omega_B}{\omega_A} \left(\frac{D_B}{D_A} \right)^3$$

$$C_{P,A} = C_{P,B} \Rightarrow \frac{bhp_B}{bhp_A} = \frac{\rho_B}{\rho_A} \left(\frac{\omega_B}{\omega_A} \right)^3 \left(\frac{D_B}{D_A} \right)^5$$

$$C_{H,A} = C_{H,B} \Rightarrow \frac{H_B}{H_A} = \left(\frac{\omega_B}{\omega_A} \right)^2 \left(\frac{D_B}{D_A} \right)^2$$

- Useful to scale from model to prototype
- Useful to understand parameter changes, e.g., doubling pump speed.

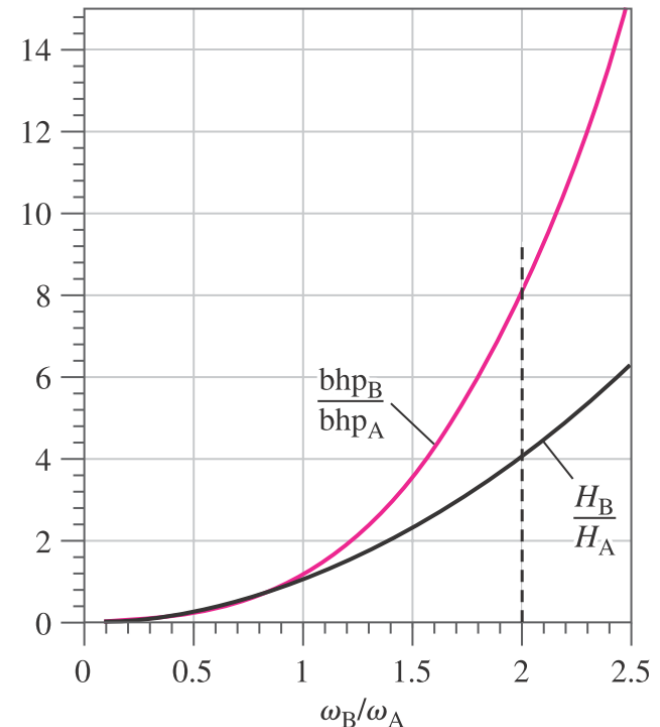
Example 2: Affinity Laws

- If we want to double the rotational speed of a pump, by approximately what factor will the motor power need to be increased? And how the total head would change?

$$\frac{bhp_B}{bhp_A} = \frac{\rho_B}{\rho_A} \left(\frac{\omega_B}{\omega_A} \right)^3 \left(\frac{D_B}{D_A} \right)^5$$

$$\omega_B = 2\omega_A; D_B = D_A; \rho_B = \rho_A \Rightarrow \boxed{\frac{bhp_B}{bhp_A} = \left(\frac{\omega_B}{\omega_A} \right)^3 = 8}$$

$$\frac{H_B}{H_A} = \left(\frac{\omega_B}{\omega_A} \right)^2 \left(\frac{D_B}{D_A} \right)^2 \Rightarrow \boxed{\frac{H_B}{H_A} = \left(\frac{\omega_B}{\omega_A} \right)^2 = 4}$$



Example 3: Affinity Laws

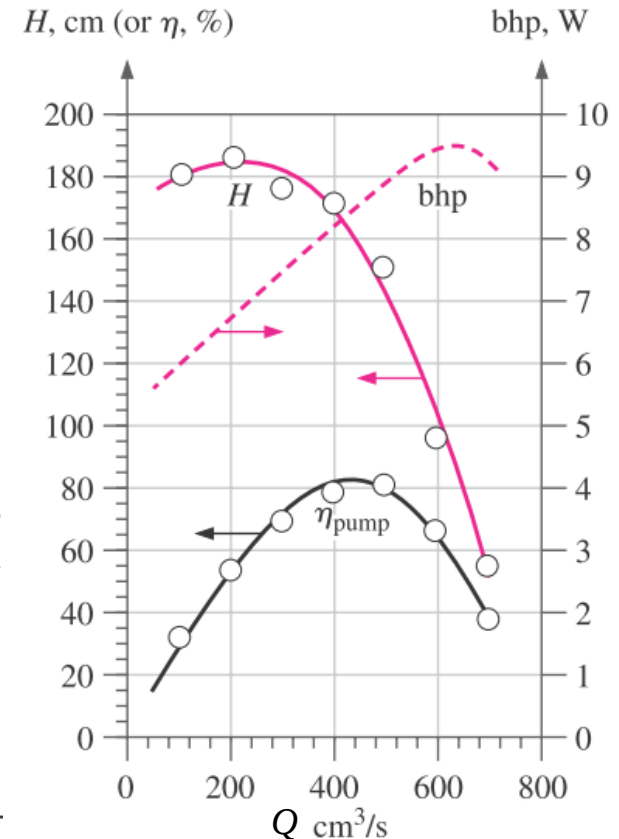
- We have a **water** pump which its impeller diameter is $D_A = 6.0 \text{ cm}$, and its performance data when operating at $\omega_A = 180.6 \text{ rad/s}$ are shown in the following table. We want to design a new product, namely a larger pump (pump B) that will be used to pump liquid refrigerant **R-134a** at room temperature. The pump is to be designed such that its best efficiency point occurs as close as possible to a volume flow rate of $Q_B = 2400 \text{ cm}^3/\text{s}$ and at a net head of $H_B = 450 \text{ cm}$ (of R-134a).

- Use the pump scaling laws to determine if a geometrically scaled-up pump could be designed and built to meet the given requirements. (a) Plot the performance curves of pump A in dimensionless form, and identify the best efficiency point. (b) Calculate the required pump diameter D_B , rotational speed ω_B , and brake horsepower bhp_B for pump B.

Manufacturer's performance data for a water pump operating at 1725 rpm and room temperature

$Q, \text{ cm}^3/\text{s}$	$H, \text{ cm}$	$\eta_{\text{pump}}, \%$
100	180	32
200	185	54
300	175	70
400	170	79
500	150	81
600	95	66
700	54	38

* Net head is in centimeters of water.



Example 3: Affinity Laws

- **Assumptions:** 1) The new pump can be manufactured so as to be geometrically similar to the existing pump. 2) Both liquids (water and R-134a) are incompressible. 3) Both pumps operate under steady conditions.
- **Properties:** At room temperature (20°C), $\rho_{\text{water}} = 998.0 \text{ kg/m}^3$, $\rho_{\text{R-134a}} = 1226 \text{ kg/m}^3$.
- **Analysis:** The best efficiency point (BEP) is around $Q = 500 \text{ cm}^3/\text{s}$ and the bhp_A at this point is:

$$\text{bhp}_A = \frac{\rho_{\text{water}} g Q_A H_A}{\eta_{\text{pump},A}} = \frac{(998.0 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(500 \text{ cm}^3/\text{s})(150 \text{ cm})}{0.81} \left(\frac{1 \text{ m}}{100 \text{ cm}} \right) \left(\frac{\text{W}\cdot\text{s}}{\text{kg}\cdot\text{m/s}^2} \right) = 9.07 \text{ W}$$

$$C_Q = \frac{Q}{\omega D^3} = \frac{500 \text{ cm}^3/\text{s}}{(180.6 \text{ rad/s})(6.0 \text{ cm})^3} = 0.0128 \quad ; \quad C_H = \frac{gH}{\omega^2 D^2} = \frac{(9.81 \text{ m/s}^2)(1.50 \text{ m})}{(180.6 \text{ rad/s})(0.06 \text{ m})^2} = 0.125$$

$$C_P = \frac{\text{bhp}}{\rho \omega^2 D^5} = \frac{9.07 \text{ W}}{(998 \text{ kg/m}^3)(180.6 \text{ rad/s})^3 (0.06 \text{ m})^5} \left(\frac{\text{kg}\cdot\text{m/s}^2}{\text{W}\cdot\text{s}} \right) = 0.00198$$

- These calculations are repeated at values of Q_A between 100 and 700 cm^3/s .
- Assuming the BEP is exactly at $C_Q^* = 0.0128$ but the true value differs slightly and is $C_Q = 0.0112$.

$$D_B = \left(\frac{Q_B^2 C_H^*}{(C_Q^*)^2 g H_B} \right)^{1/4} = \left(\frac{(0.0024 \text{ m}^3/\text{s})^2 (0.125)}{(0.0128)^2 (9.81 \text{ m/s}^2)(4.50 \text{ m})} \right)^{1/4} = 0.1 \text{ m}$$

$$\omega_B = \frac{Q_B}{(C_Q^*) D_B^3} = \frac{0.0024 \text{ m}^3/\text{s}}{(0.0128)(0.1 \text{ m})^3} = 187.5 \text{ rad/s}; \quad \text{bhp}_B = C_P^* \rho_B \omega_B^3 D_B^5 = 0.00198(1226)(187.5)^3 (0.1)^5 = 160 \text{ W}$$

