<u>PUMPS</u>

Machines that transfer energy INTO fluid system.
Pumps have a shaft and an impeller. Impeller is the rotating element.

Types of pumps

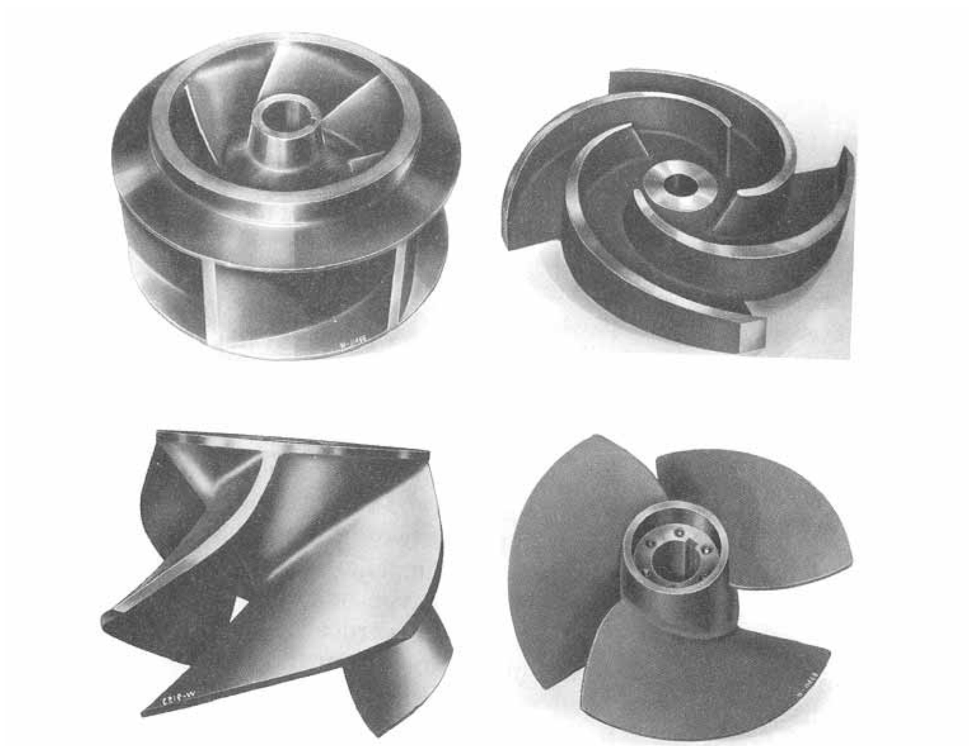
Fluid flows outward (right angle to its axis)
RADIAL FLOW

"centrifugal pump"

MIXED FLOW

Fluid flows spirally (along axis)
AXIAL FLOW

"Propeller pump"



EFFICIENCY OF A PUMP

$$\eta = \frac{power\ \ given\ \ to\ \ fluid}{power\ \ given\ to\ \ shaft} = \frac{\gamma Q h_p}{T\omega}$$

$\rightarrow$ $(brake\ power\ or\ shaft\ power)$

where T: torque exerted on the shaft

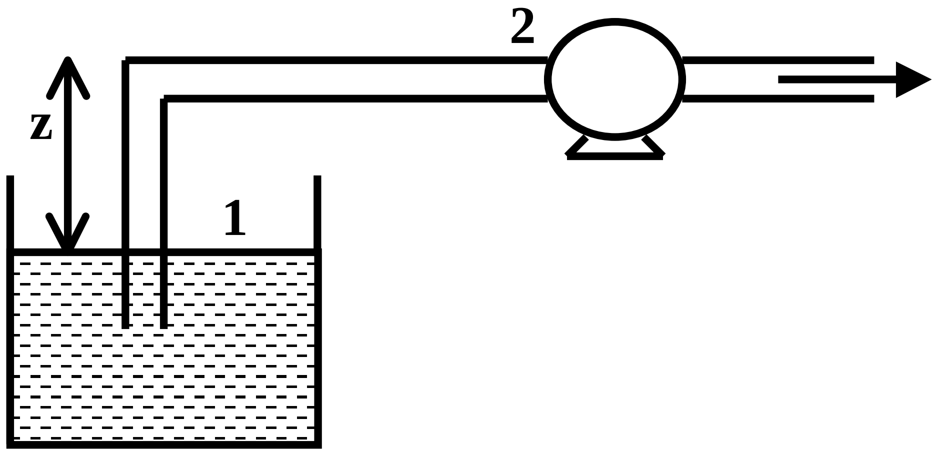
ω = rotation of the shaft in radians/sec

Note: overall pump efficiency is affected by:

- **Hydraulic losses in pump** η_h
- **Mechanical losses in the bearings, seals** η_m
- **Leakage of fluid: volumetric losses** η_v

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

CAVITATION IN PUMPS



The potential for cavitation is characterized by difference in (pressure head +velocity head) on suction side and the liquid vapor pressure head. This is NPSH

$$NPSH = \frac{p_s}{\gamma} + \frac{V_s^2}{2g} - \frac{p_v}{\gamma}$$

There is a value of NPSH that must be maintained to avoid cavitation called as $NPSH_R$(this is experimentally determined)

There is available NPSH called as $NPSH_A$

Apply energy equation between 1 and 2

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

$$\frac{p_{atm}}{\gamma} + 0 + 0 = \frac{p_s}{\gamma} + z + \frac{V_s^2}{2g} + h_L$$

$$\frac{p_s}{\gamma} + \frac{V_s^2}{2g} = \frac{p_{atm}}{\gamma} - z - h_L$$

Hence

$$NPSH_A = \frac{p_{atm}}{\gamma} - z_1 - h_L - \frac{p_v}{\gamma}$$

To prevent cavitation

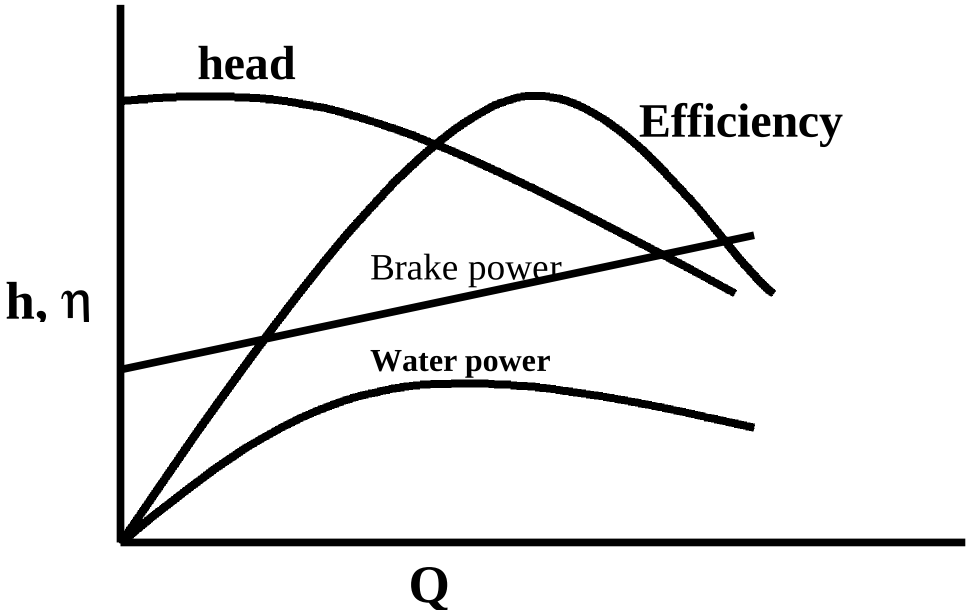
$NPSH_A \geq NPSH_R$

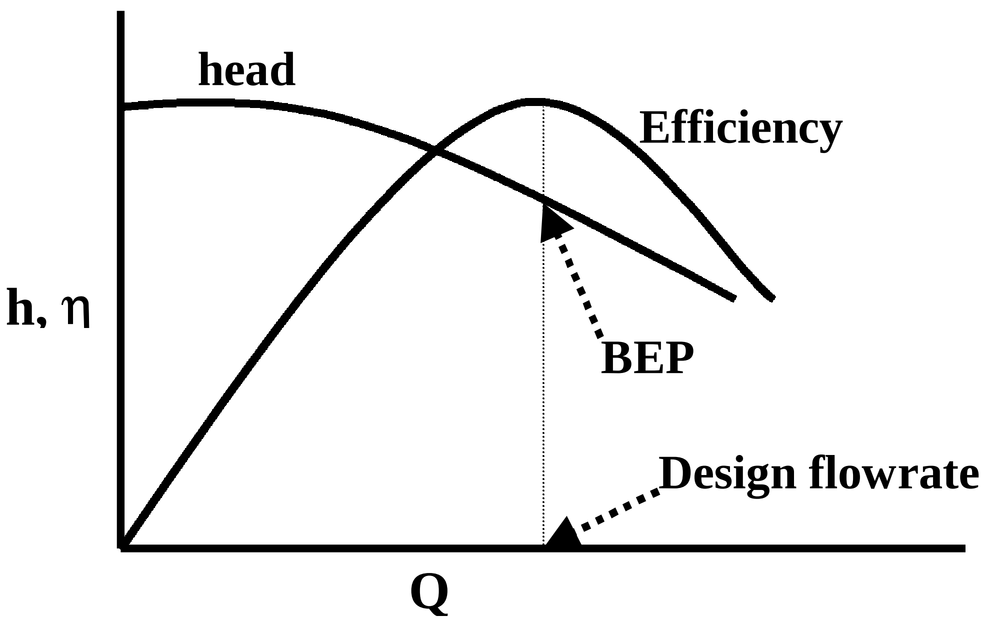
How?

- **Small z**
- **Small losses (shorter pipes)**

Note: There is a critical value of z above which pump will NOT operate without cavitation

PUMP PERFORMANCE CHARACTERISTICS





BEP: Best efficiency point

The point where pump is to be operated should be as close as possible to the BEP.

SIMILARITY LAWS FOR PUMPS

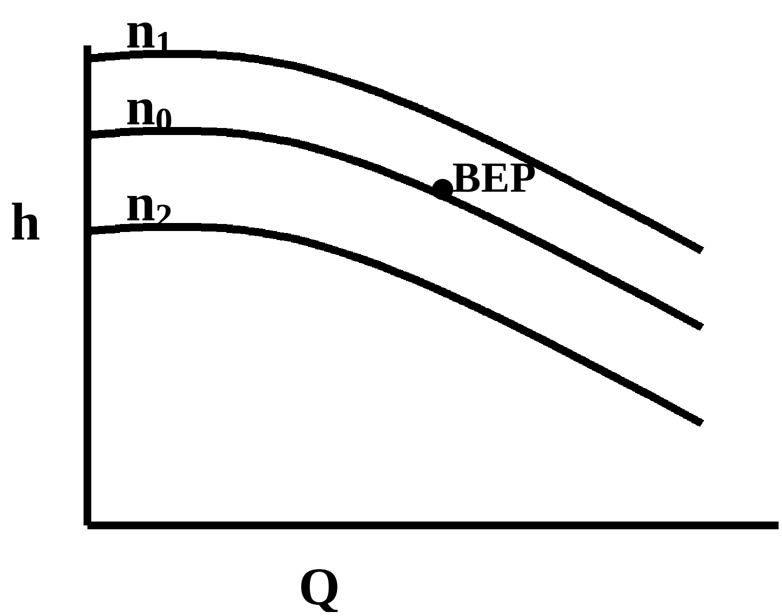
Performance prediction from a scaled test model to prototype

1. $Q \,\alpha\, nD^3$

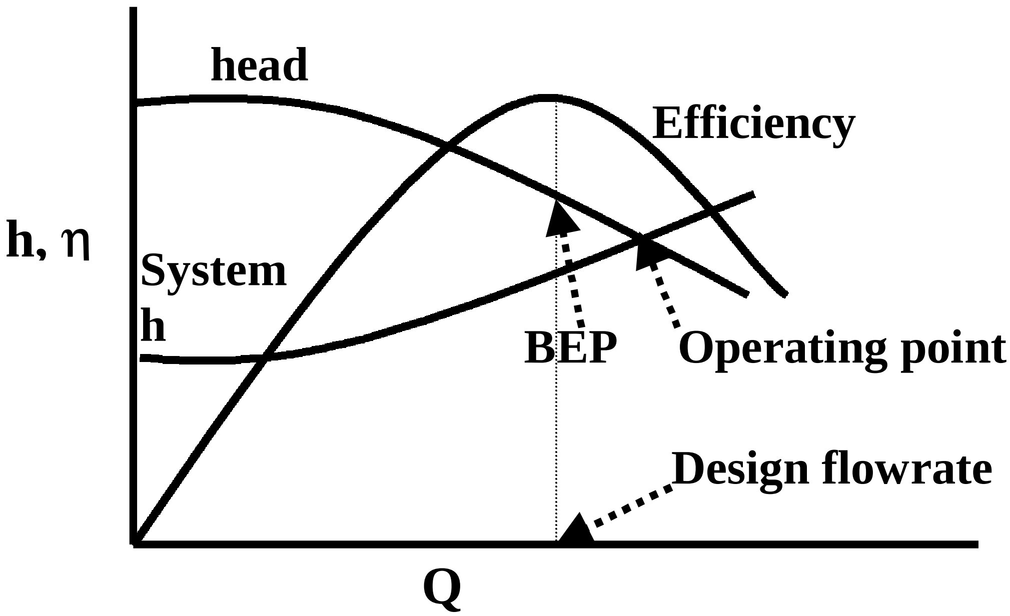
2. $h \,\alpha\, n^2 D^2$

3. $P \,\alpha\, n^3 D^5$

4. $\dfrac{1-\eta_1}{1-\eta_2} \approx \left(\dfrac{D_2}{D_1}\right)^{1/5}$



PUMP SELECTION

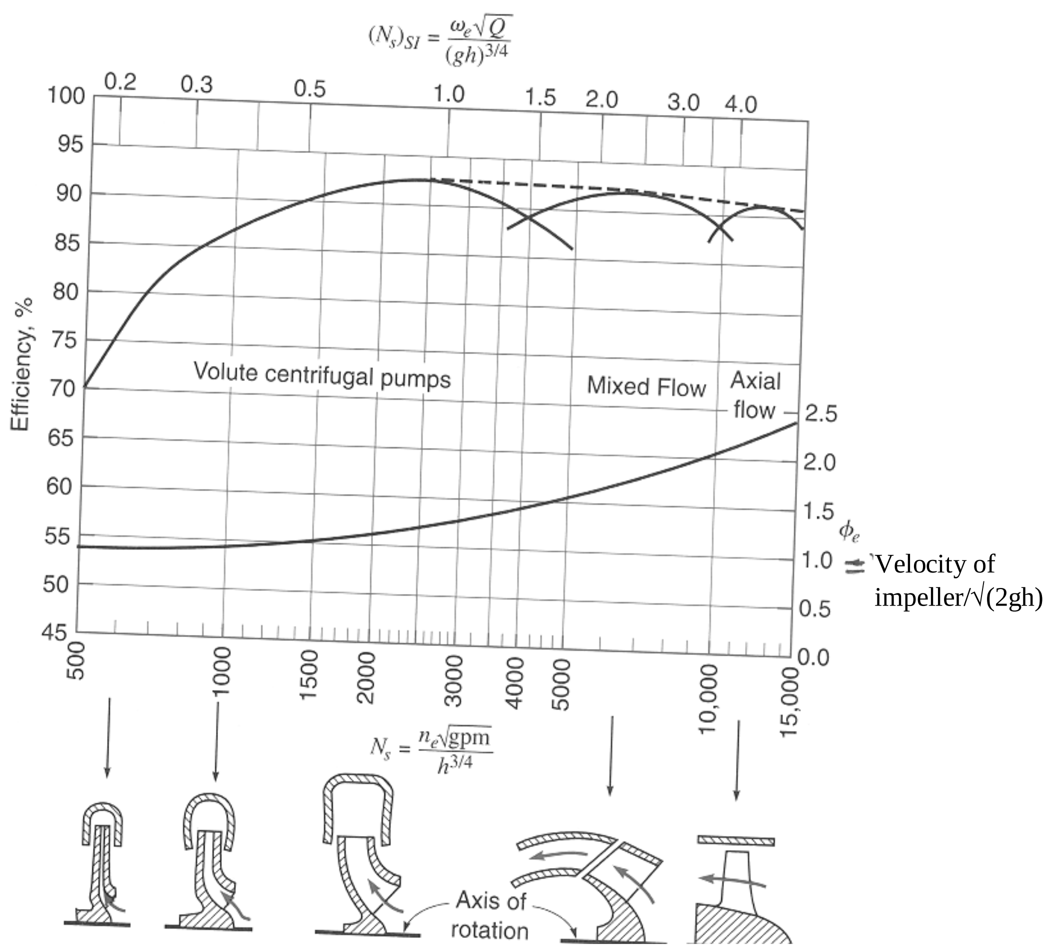


Notice: at operating point efficiency is NOT maximum

As system curve changes so will difference between BEP and operating point

<u>**SPECIFIC SPEED OF PUMPS**</u>

and w, Q, and h are the values at BEP



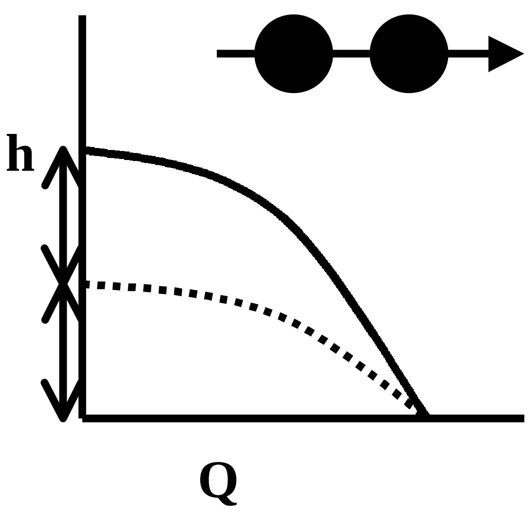
Suction Specific Speed

$$S_{sd} = \frac{\omega\sqrt{Q}}{NPSH_R^{3/4}}$$

where $NPSH_R$ *is in ft, Q in gpm,* ω *in rpm*

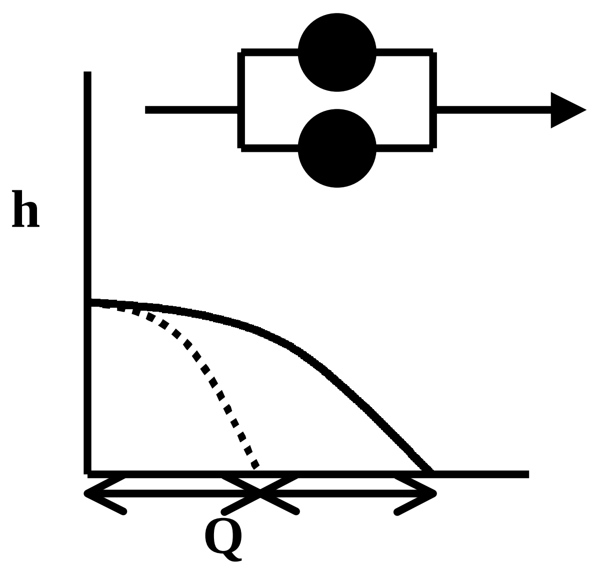
PUMPS IN SERIES AND PARALLEL

SERIES



Increase head

PARALLEL



Increase Discharge

Pumps in series and parallel should have similar pump characteristics