Lecture on

Pump

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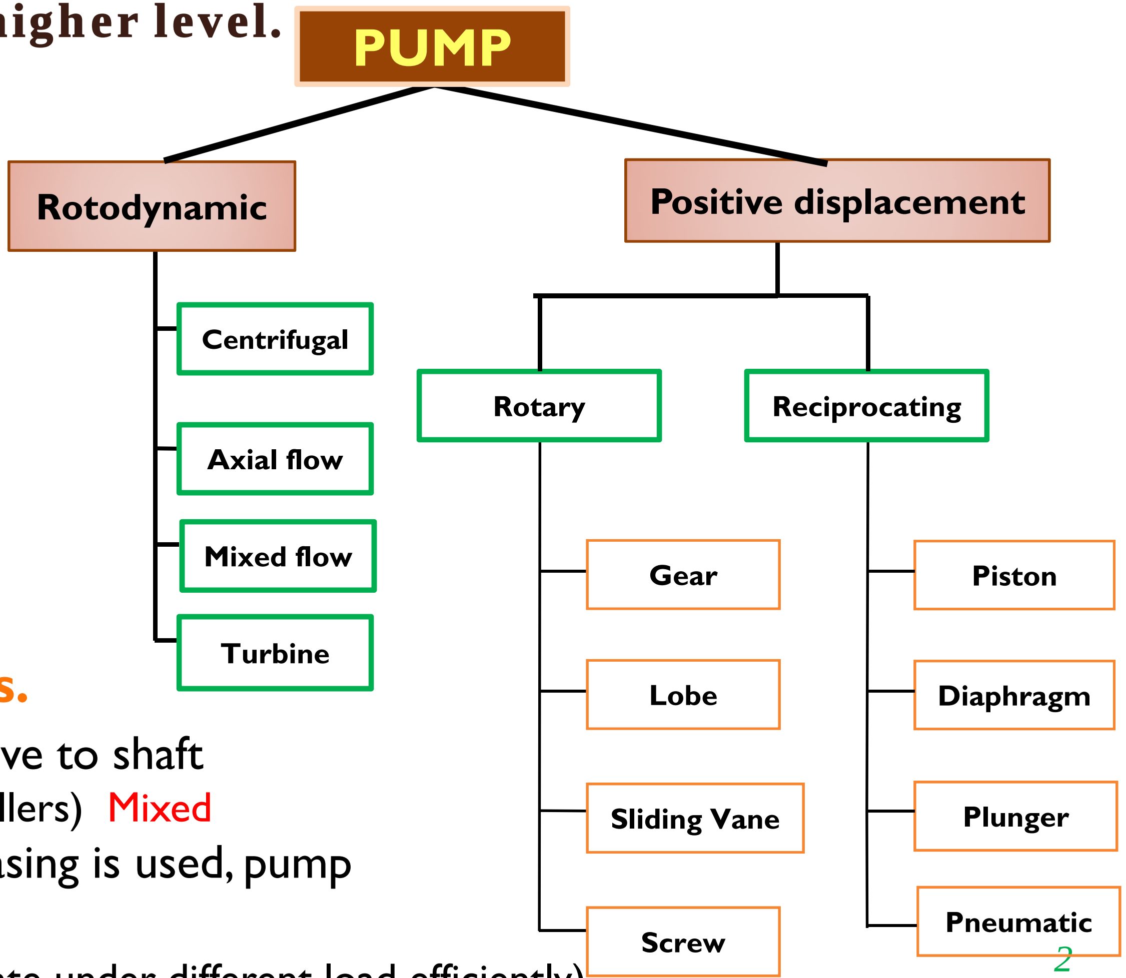
WHAT IS PUMP?

A hydrodynamic pump machine is a device which converts the mechanical energy held by a device into potential and kinetic energy in fluid.

Pumps enable a liquid to:

1. **Flow from a region or low pressure to one of high pressure.**
2. **Flow from a low level to a higher level.**
3. **Flow at a faster rate.**

Pump is the machine designed to move fluid and add energy to them.



There are two main categories of pump according to design and principle of operation:

- **Rotodynamic pumps.**
- **Positive displacement pumps.**

Based on **direction of fluid flow** relative to shaft

Radial (centrifugal pumps) Axial (boat propellers) Mixed

Single- vs **multi-stage**(diffuser type casing is used, pump impellers are connected in series)

Constant vs **variable speed** (to operate under different load efficiently)

CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Rotary:- Rotating action occurs periodically.

Gear – comprises two gears in a housing with small radial end clearances. Used in lubrication system.

Lobe- Handles solids also. Used in paper and pulp industry.

Sliding Vane – comprises number of vanes.

Screw- three screw with housing is used with housing. Can create high pressure, uniform delivery, used to transfer lubricant.

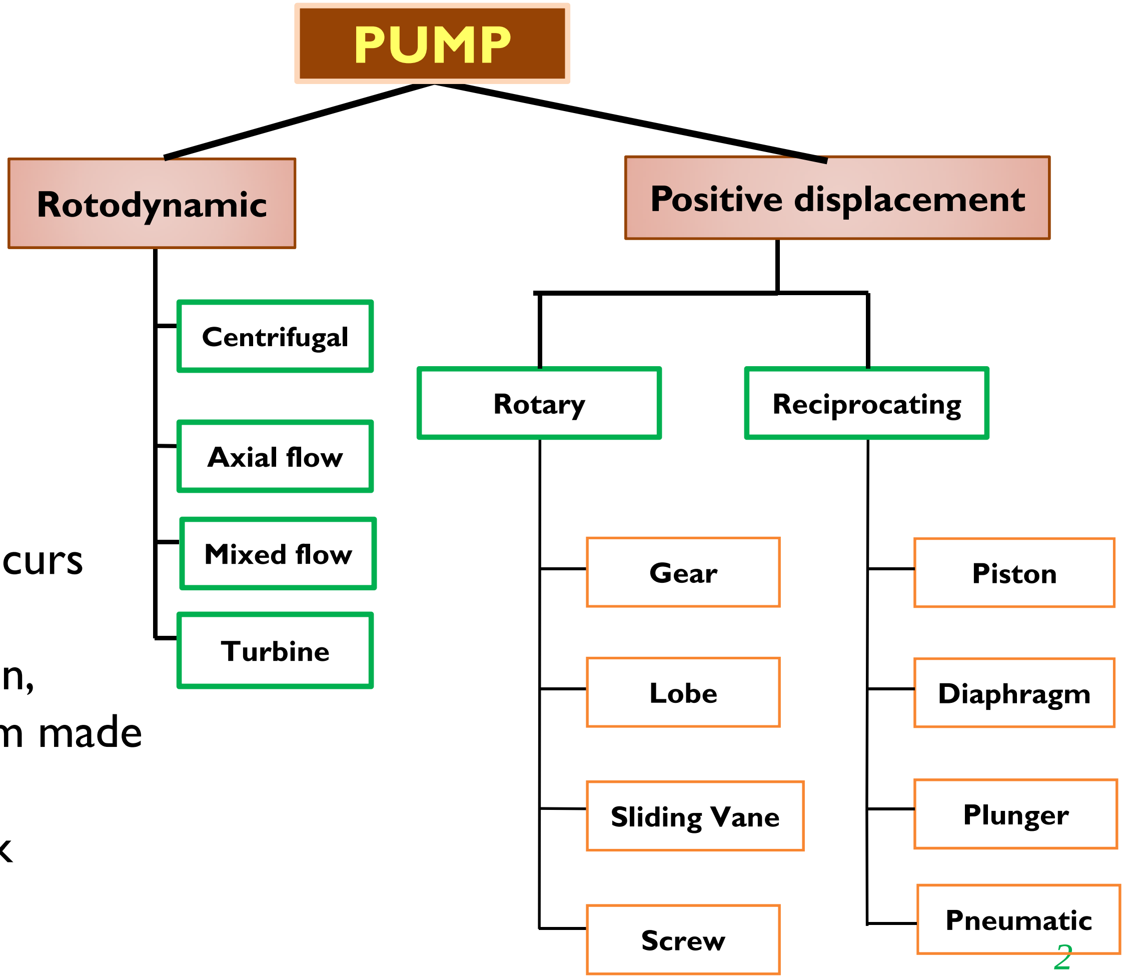
Reciprocating:- Reciprocating action occurs periodically.

Piston – comprises a cylinder and piston,

Diaphram – comprises flexible diaphram made from rubber or rubberised febric

Plunger – comprises plunger, uses crank mechanism

Pneumatic – Handles compressed air.



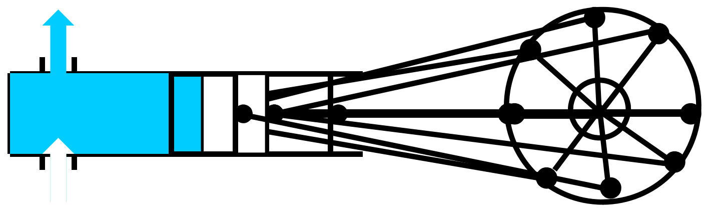
Centrifugal Pumps:

Centrifugal pumps have a rotating impeller, also known as a blade, that is immersed in the liquid. Liquid enters the pump near the axis of the impeller, and the rotating impeller sweeps the liquid out toward the ends of the impeller blades at high pressure.

Positive-displacement Pumps:

A variety of positive-displacement pumps are also available, generally consisting of a rotating member with a number of lobes that move in a close-fitting casing. The liquid is trapped in the spaces between the lobes and then discharged into a region of higher pressure. A common device of this type is the gear pump, which consists of a pair of meshing gears. The lobes in this case are the gear teeth

- A rotodynamic or non- positive displacement pump imparts velocity energy to the fluid, which is converted to pressure energy upon exiting the pump casing.
- A positive displacement pump moves a fixed volume of fluid within the pump casing by applying a force to moveable boundaries containing the fluid volume.



Differences between centrifugal pumps, reciprocating pumps and rotary pumps (relatively broad categories)

Parameter	Centrifugal Pumps	Reciprocating Pumps	Rotary Pumps
Optimum Flow and Pressure Applications	Medium/High discharge Capacity, Low/Medium Pressure, small change in pr. Diff. causes a large change in flow	Low discharge Capacity, High Pressure, pr. Fluctuation doesnot affect discharge	Low/Medium discharge Capacity, Low/Medium Pressure
Maximum Flow Rate	100,000+ GPM	10,000+ GPM	10,000+ GPM
Low Flow Rate Capability	No	Yes	Yes
Maximum Pressure	6,000+ PSI	100,000+ PSI	4,000+ PSI (pound/in^2)
Requires Pr. Relief Valve	No	Yes	Yes
Smooth or Pulsating Flow	Smooth	Pulsating	Smooth
Variable or Constant Flow	Variable	Constant	Constant
Self-priming	No	Yes	Yes
Space Considerations	Requires Less Space	Requires More Space	Requires Less Space
Costs	Lower Initial Lower Maintenance Higher Power	Higher Initial Higher Maintenance Lower Power	Lower Initial Lower Maintenance Lower Power
Fluid Handling	Suitable for a wide range including clean, clear, non-abrasive fluids to fluids with abrasive, high-solid content. Not suitable for high viscosity fluids Lower tolerance for entrained gases (vapour trapped in flowing iquid)	Suitable for clean, clear, non-abrasive fluids. Specially-fitted pumps suitable for abrasive-slurry service. Suitable for high viscosity fluids Higher tolerance for entrained gases	Requires clean, clear, non-abrasive fluid due to close tolerances Optimum performance with high viscosity fluids Higher tolerance for entrained gases

Advantages and Disadvantages of Centrifugal Pump

Advantages

Simple in construction and cheap.

Handle liquid with large amounts of solids.

No metal to metal fits.

No valves involved in pump operation.

Maintenance costs are lower.

Disadvantages

Cannot handle highly viscous fluids efficiently greater than 1000 centipoise (cP) . (1P= 0.100 $kgm^{-1}s^{-1}$)

Cannot be operated at low capacity.

Cannot be operated at high heads.

Maximum efficiency holds over a narrow range of conditions.

Cannot operate, if percentage volume of dissolved gases is greater than 5%.

Selection of Pump based on:

- ✓Pressure and capacity (discharge).
- ✓Property of liquid (viscosity, temperature, corrosiveness, grittiness i.e. erosion for suspended particle).
- ✓Initial and maintenance cost.
- ✓Pump duty.
- ✓Availability of space, size and position of locating the pump.
- ✓Speed of rotation and power required.
- ✓Standardization with respect to the type and makes of pumps already available at site.

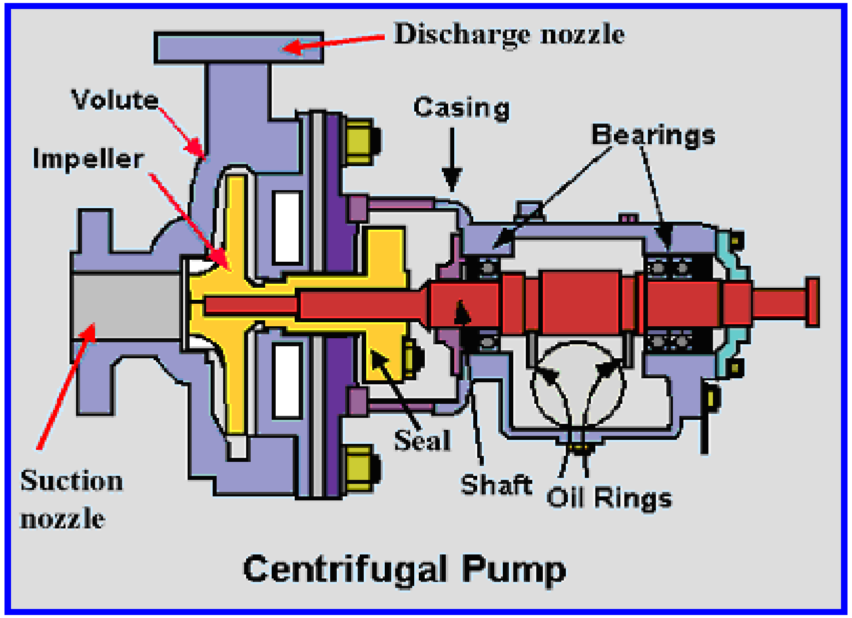
Construction of Centrifugal Pumps

Main components:

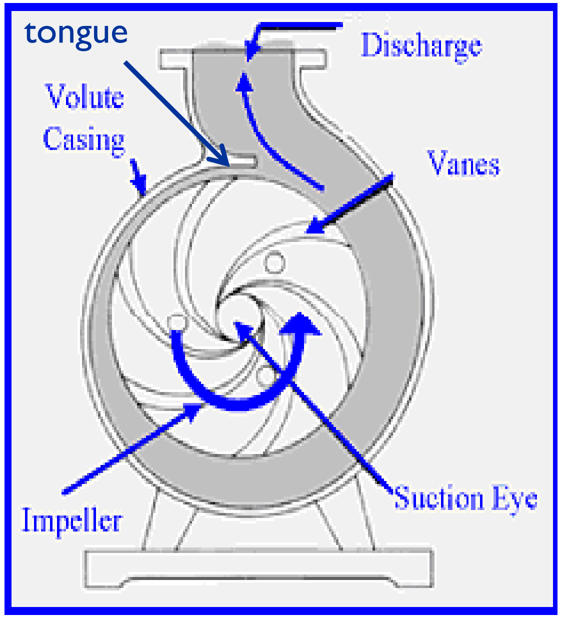
1. **Stationary componets**, casing, casing cover and bearings

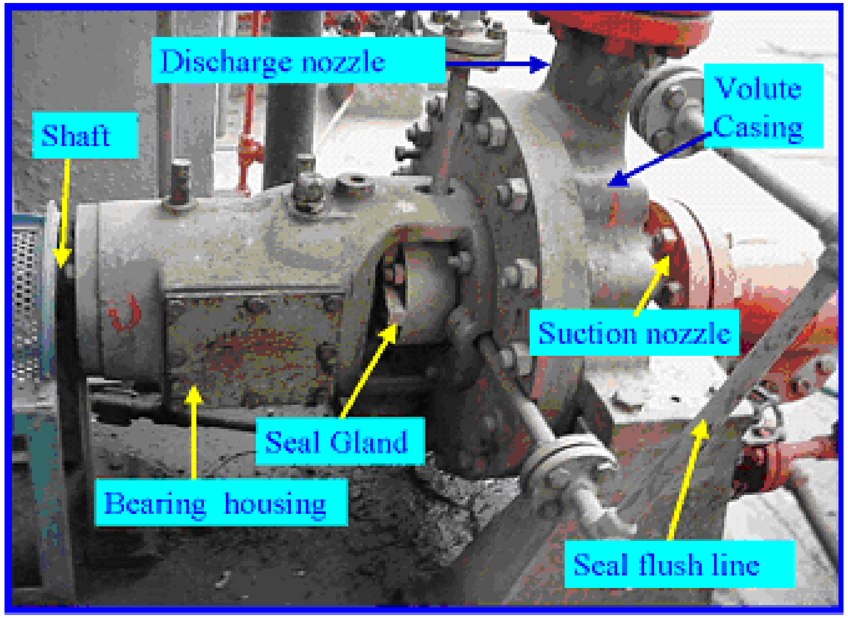
2. **Rotating components**, impeller and shaft.

Energy changes occur by virtue of impeller and volute. Liquid is fed into the pump at the center of a rotating impeller (eye) and thrown outward by centrifugal force.



The conversion of kinetic energy into pressure energy supplies the pressure difference between the suction side and delivery side of the pump



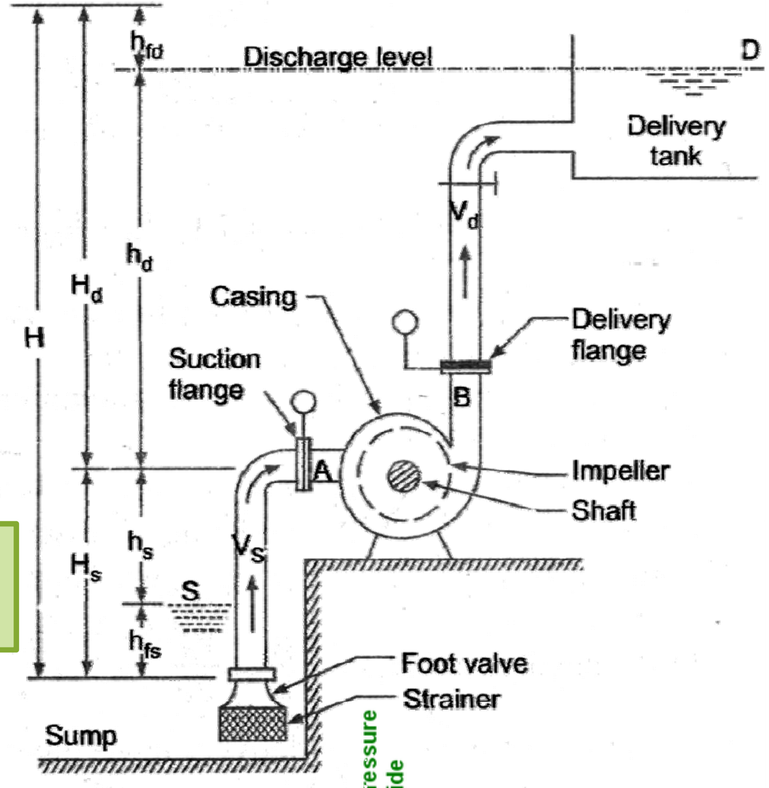


Construction of Centrifugal Pumps

3. Other accessories:

1. **Delivery pipe, strainer and foot valve-** centre pipe connects the centre (eye) of the impeller to the sump.

2. **Suction pipe, strainer and foot valve-** a regular valve is just near the pump outlet serves to control the flow of liquid into delivery pipe.



Construction wise: Integral, vertical/horizontal/diagonal split, segmented casing

Casing (stationary part)

I. Volute casings for a higher head.

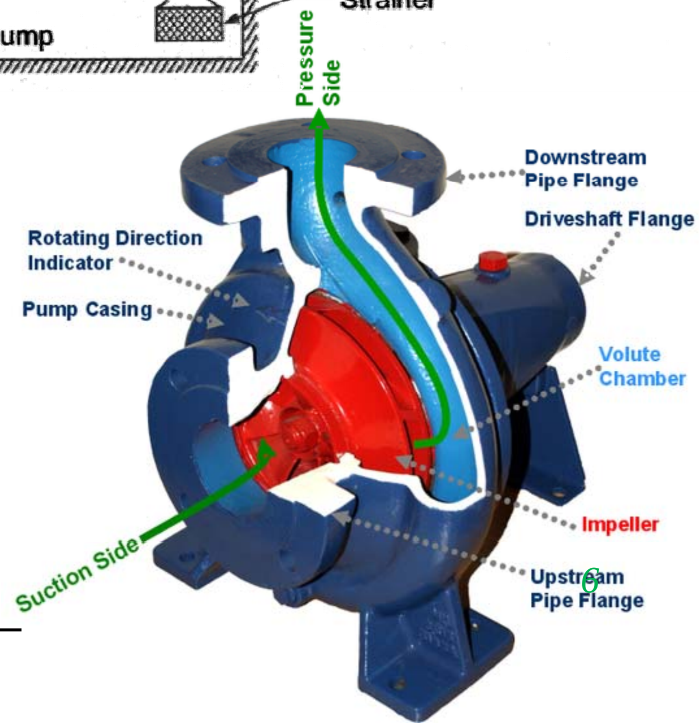
A volute or spiral casing is a curved funnel increasing in area to the discharge port.

Cross-section gradually increases from tongue towards delivery pipe, therefore velocity decreases, pressure increases. Single stage pumps are made by volute, though they have greater eddy loss, i.e. lower efficiency.

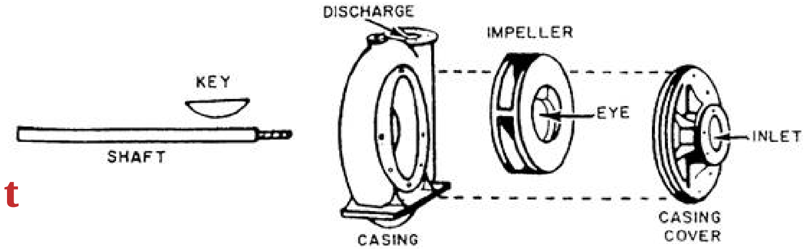
II. Circular casings for low head and high capacity.

have stationary diffusion vanes surrounding the impeller periphery that convert velocity energy (dynamic head) to pressure energy (static head). Pump fitted with guide vane is called " diffuser pump" or "turbine pump".

III. Whirlpool chamber casings annular space is provided in between volute and impeller- arrest the formation of eddies – reduce loss of energy – improve efficiency.

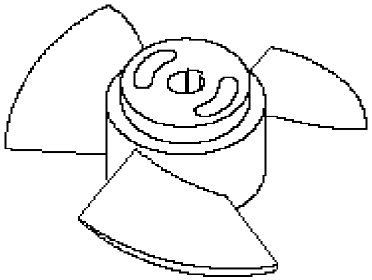


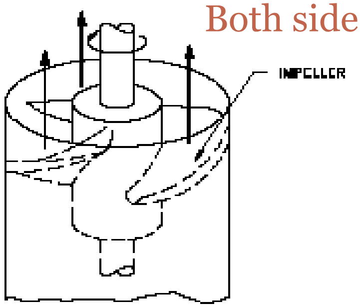
Construction of Centrifugal Pumps



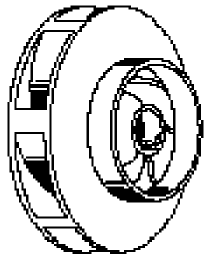
Impeller (rotating element)

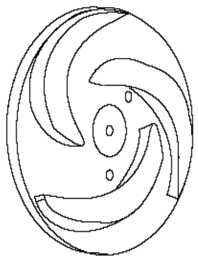
Three main categories of centrifugal pumps exist

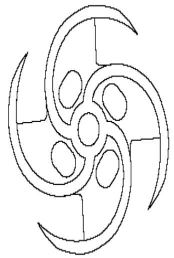




Both side shrouded (side plate) plate on one side vane







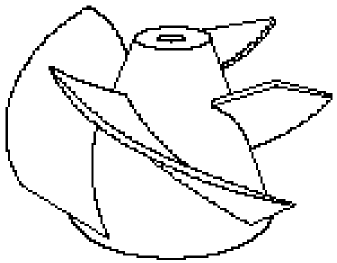
Enclosed Semi-open Open

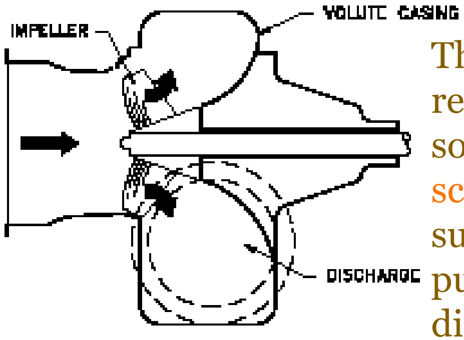
Axial flow

Head is developed by propelling or lift action of the vane on the liquid, similar to the airplane .

Radial flow

Head is developed by action of the centrifugal force upon the liquid.





This types of pumps resembles the screw , sometimes called screw impeller, suited for irrigation purpose, where high discharge in low head is required.

Mixed flow

Head is developed partly by action of the centrifugal force upon the liquid and partly by axial propulsion.

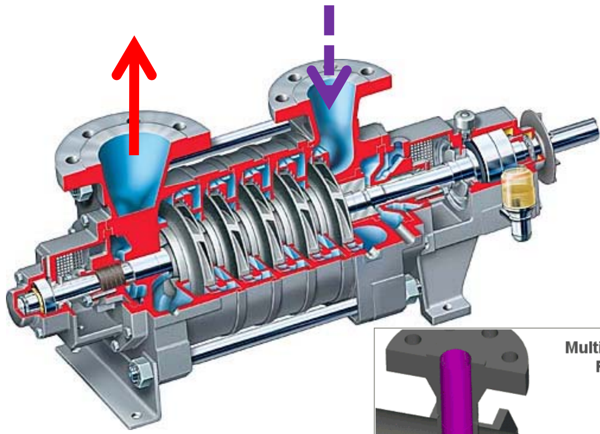
Construction of Centrifugal Pumps

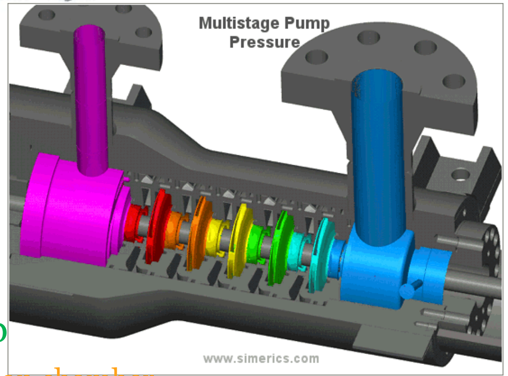
Shape and Number of Vane

Usually 6 to 12, shaped like curved, cylindrical or more complex.

Working Head and Number of Stages

Low head (upto 15 m) medium head (15 to 40 m) high head (over 40 m). Single stage pump can exceed 40 m , for more head multistage pump is required – number of impeller in series.



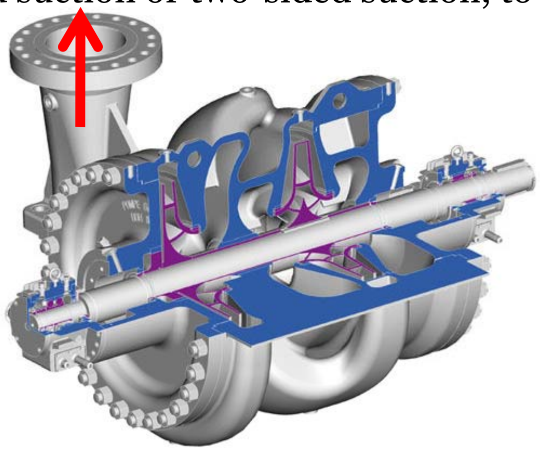


increase head

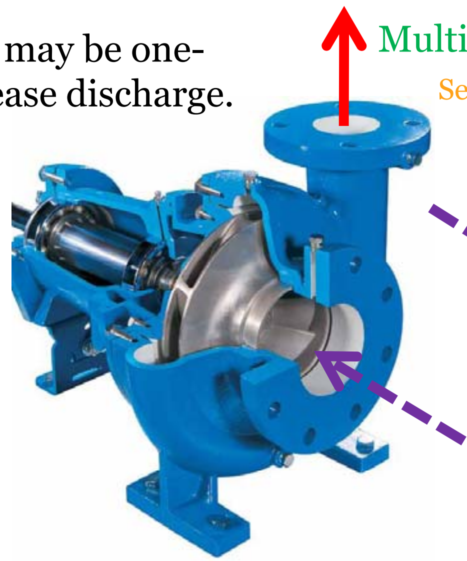
Multistage pump

Single suction and double suction

With respect to the liquid entrance , pump may be one-sided suction or two-sided suction, to increase discharge.

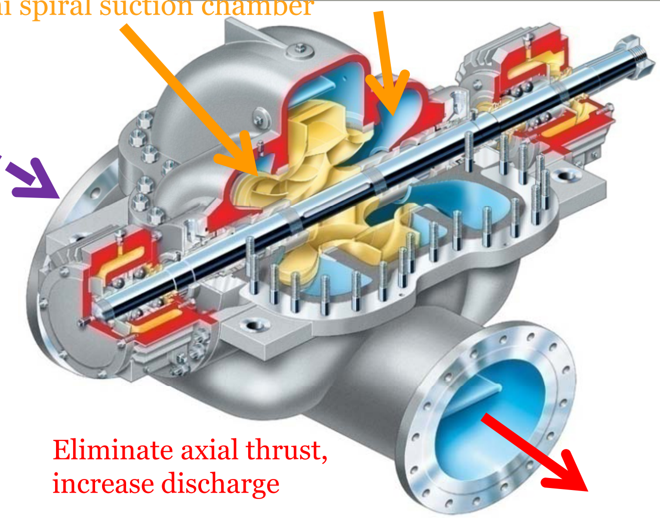


Two stage Double suction pump



Single stage single suction pump

Semi spiral suction chamber



Eliminate axial thrust,
increase discharge

Split Case Double suction pump

Shaft Position

Most pump have horizontal shaft. To accommodate space it may be vertical (for deep well, mines)

Classification of the pump may be based on liquid handled, suspended solid, viscous liquid handled, application in irrigation, boiler feed, condensate circulation, power used such as IC engine or electric motor etc.

Construction of Centrifugal Pumps

Typical Specification of a Split Case High Flow - Double Suction Centrifugal Pump

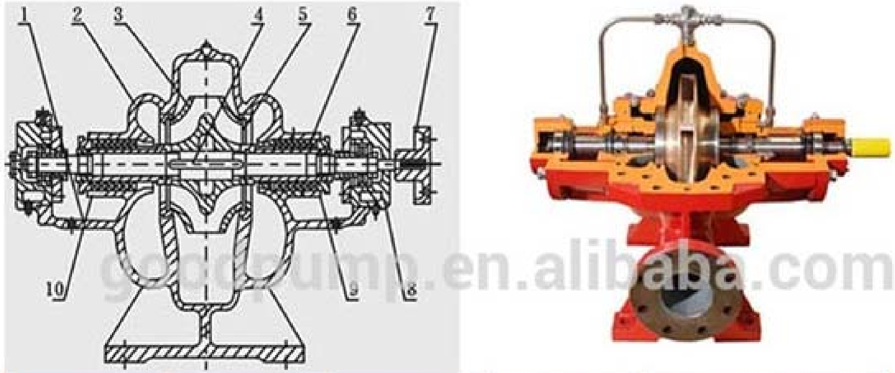
Overview

This Fire Fighting Pump is high flow rate pump single stage double suction split casing pump with high capacity and pressure. Usually temperature can not more than 80 °C. Once using check valve, the pump can be self-priming.

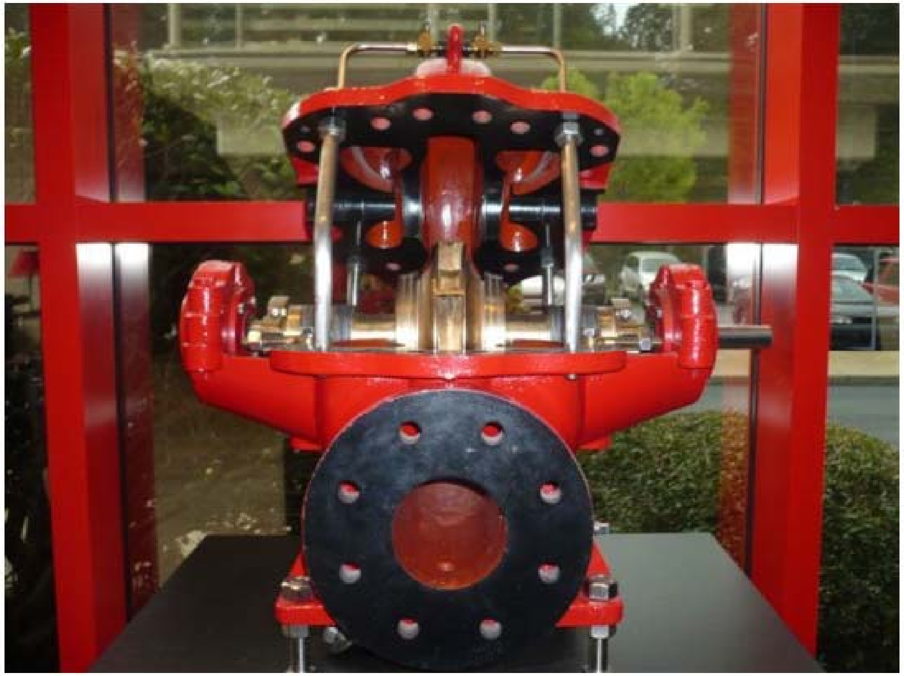
Main Material

Casing: Gray cast iron, ductile cast iron, cast steel.
Impeller: Brozne, Gray cast iron, Silicon brass, stainless steel
Shaft: Stainless steel, Carbon steel

Pump Type	Split Casing Pump
Casing Material	HT200 (Grey Cast Iron -1.5-4.3% carbon and 0.3-5% silicon plus manganese, sulphur and phosphorus) (Tensile Strength 200 σb≥/Mpa)
Impeller	SS304,SS316,Copper,HT200
Pump Sealing	Packing seal,Mechanical seal
Capacity Range	126m^3/h~8520m^3/h
Head Range	8m~140m
Inlet/Outlet diameter	Impeller 6"(150mm)~32"(800mm)
Rotary Speed	1450rmp/2950rpm/585rpm/730rpm/970rpm
NPSH(r)	2.5m~7.5m
N.weight	165kgs~2550kgs
Pump Parts	Casing,Pump cover,Impeller,Shaft, Double suction sealing ring Shaft sleeve,Bearing etc
Certificate	ISO9001:2008,CE,TUV



1	2	3	4	5
Pump Casing	Pump Cap	Impeller	Bearing	Double suction seal ring
6	7	8	9	10
Bearing Sleeve	Coupling	Bearing Boding	Stuff Cap	Stuff



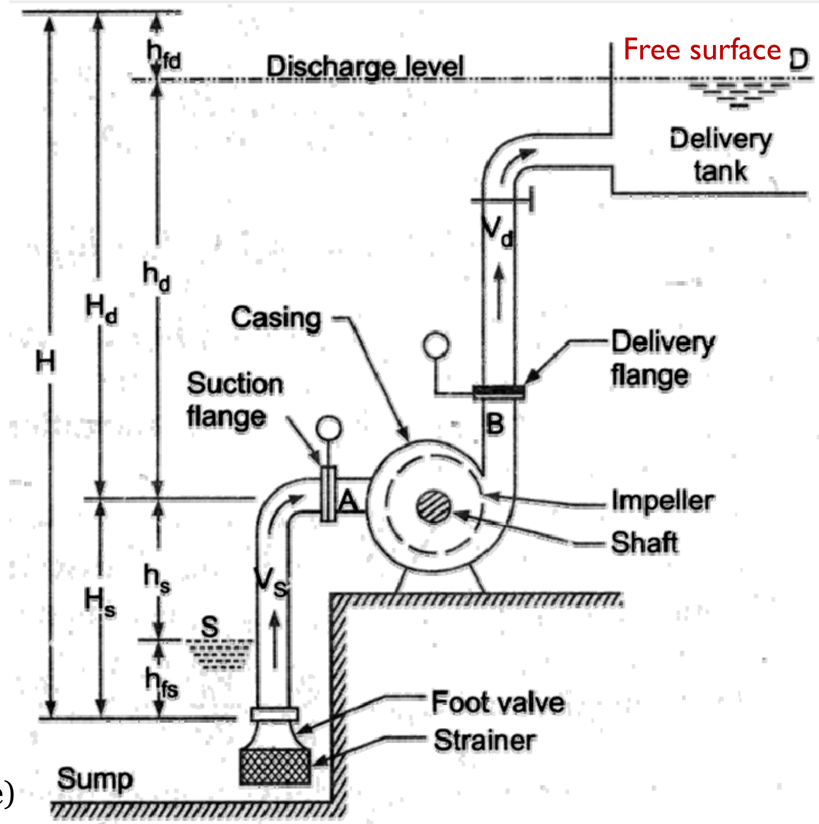
Heads of Pump:

where :

V_s = Velocity of fluid in the suction pipe.
V_d = Velocity of fluid in the delivery pipe.

h_s = Suction lift.
h_d = Delivery lift.

Total Static or vertical lift = $h_s + h_d$



(1) ***Suction Head*** *of the Pump* ($\boldsymbol{H_s}$)

$$H_s = h_i + h_{fs} + h_s + \frac{V_s^2}{2g} = \frac{P_s}{\gamma} + \frac{V_s^2}{2g}$$

where :

h_i = loss of head at inlet to suction pipe (negligible)
h_{fs} = loss of head due to friction.
$h_s + h_i + h_{fs}$ = head measured by vacuum gauge near suction flange, adjacent to the pump

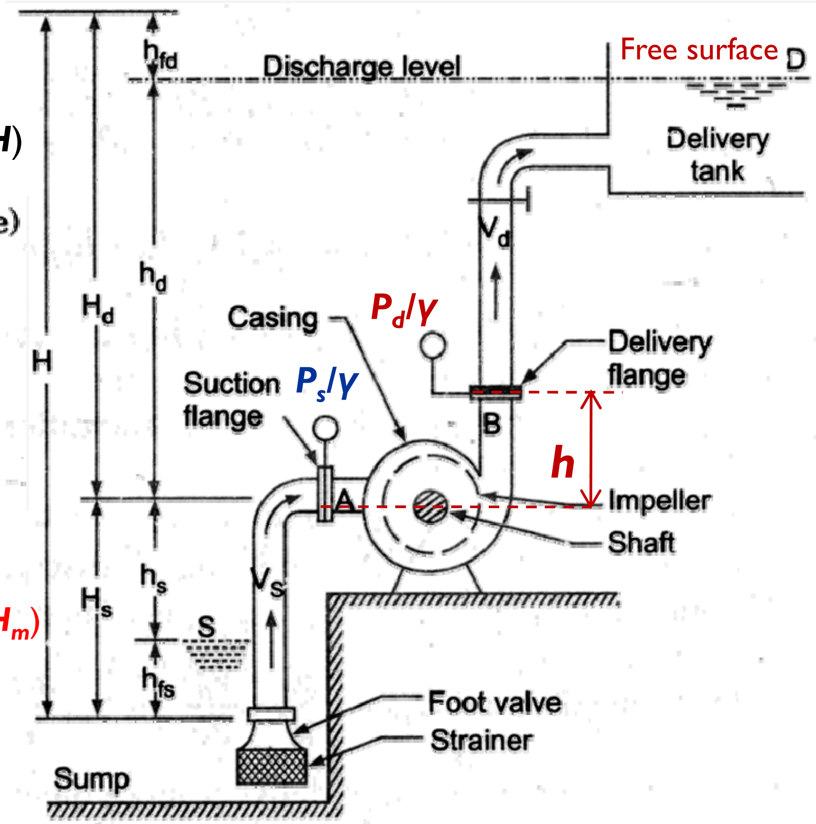
(2) ***Delivery Head*** *of the Pump* ($\boldsymbol{H_d}$)

$$H_d = h_{fd} + h_d + \frac{V_d^2}{2g} = \frac{P_d}{\gamma} + \frac{V_d^2}{2g}$$

where : $h_d + h_{fd}$ = head measured by pressure gauge near delivery flange, adjacent to the pump

Heads of Pump:

(3) ***Total External Head*** *of the Pump against which pump has to work* (***H***)



H = (suction head) + (delivery head) – (velocity head in the suction pipe)

$$= H_s + H_d - \frac{V_s^2}{2g}$$

$$= \left(\frac{P_s}{\omega} + \frac{V_s^2}{2g}\right) + \left(\frac{p_d}{w} + \frac{V_d^2}{2g}\right) - \frac{V_s^2}{2g}$$

$$= (h_i + h_{fs} + h_s) + \left(h_d + h_{fd} + \frac{V_d^2}{2g}\right)$$

Generally discharge velocity head is quite low compared to static head, hence neglected

(4) ***Manometric Head*** *is difference of total energy at inlet and exit of pump* (***H_m***)

Fluid energy at suction $P_s/\gamma + V_s^2/2g$

Fluid energy at discharge $P_d/\gamma + V_d^2/2g + h$ *where h is difference in elevation of two pressure gauge installed at inlet and outlet*

$$H_m = (P_d/\gamma + V_d^2/2g + h) - (P_s/\gamma + V_s^2/2g) = P_d/\gamma - P_s/\gamma$$

H_m is actually differences in pressure gauge reading

Another expression of H_m considering the external condition is as follows

Manometric head = static head + head losses (friction and minor) in the suction and delivery pipe + velocity head in delivery pipe

$$H_m = (h_s + h_d) + (h_{fs} + h_{fd}) + \frac{V_d^2}{2g}$$

$$= (h_s + h_{fs}) + \left(h_d + h_{fd} + \frac{V_d^2}{2g}\right)$$

$$= \left(H_s - \frac{V_s^2}{2g}\right) + H_d = H_s + H_d - \frac{V_s^2}{2g}$$

Eventually this expression equals to the Total External Head

Heads of Pump:

(5) Net Positive Suction Head ***NPSH***

NPSH **= (absolute pressure at inlet to pump) – (vapour pressure of liquid being pumped) + (velocity head in suction pipe)**

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

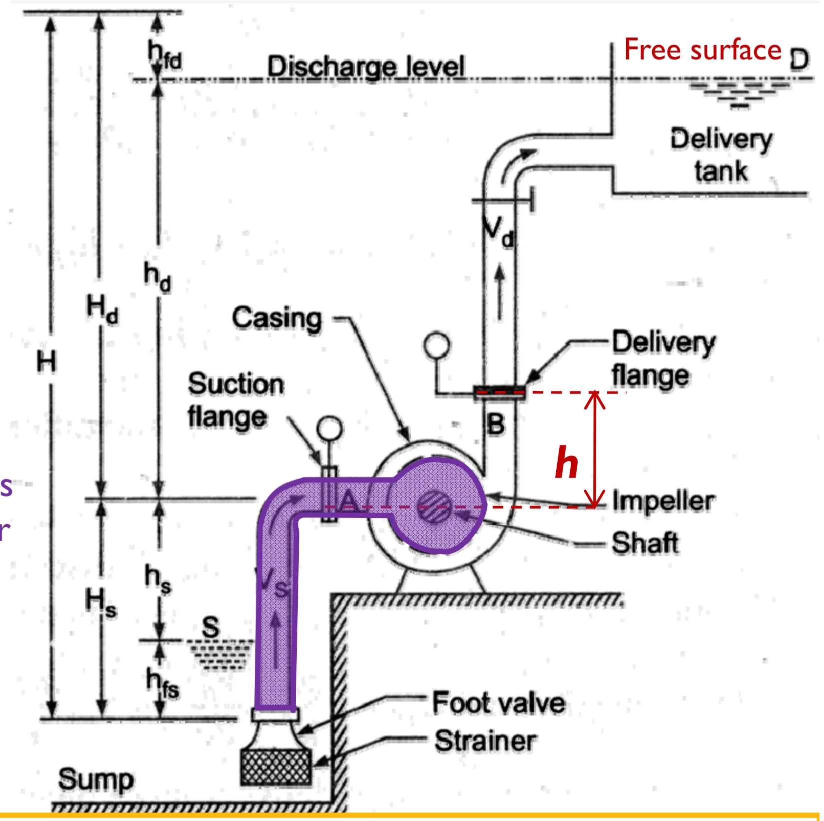
$$= \left(\frac{p_a}{w} - h_s - h_{fs} - \frac{V_s^2}{2g}\right) - \frac{p_v}{w} + \frac{V_s^2}{2g}$$

$$NPSH = (P_a/\gamma - P_v/\gamma - h_s - h_{fs})$$

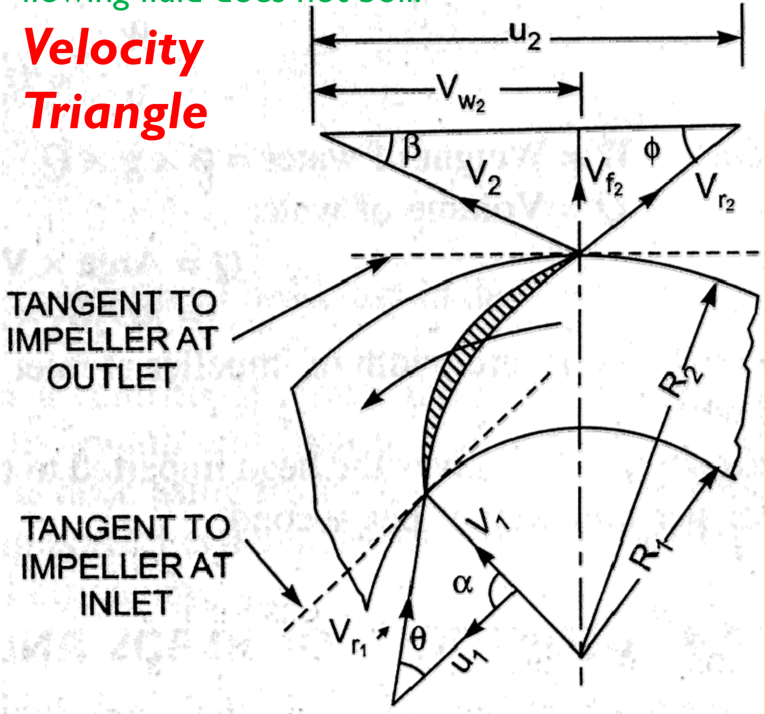
Pa=atmospheric pressure on the surface of the fluid in the suction well

NPSH is suction head at impeller eye; it represents head requires to transfer fluid from suction to the impeller.

For smooth and cavitation free operation. NPSH should have such a value so that flowing fluid does not boil.

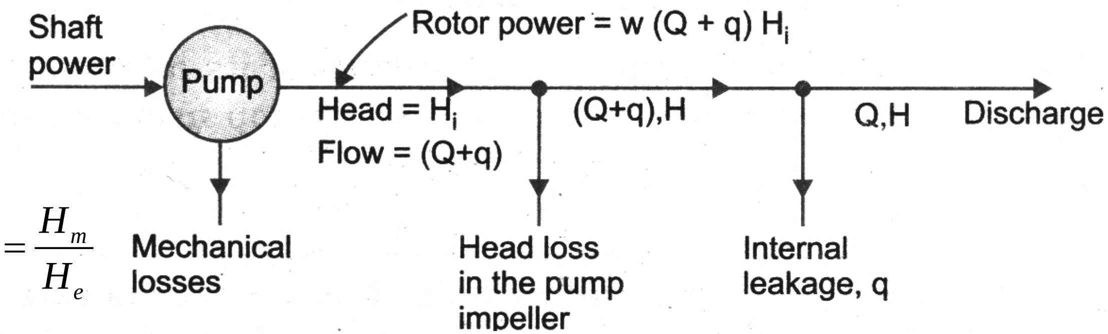


Velocity Triangle



v_1 = velocity of fluid at inlet
u_1 = velocity of the vane at inlet
v_{r1} = relative velocity of fluid at inlet
α = angle between the fluid direction and the vane motion direction
v_{w1} = velocity of whirl at inlet
θ = angle made by v_{r1} with direction of motion at inlet, inlet vane angle
v_1 = velocity of fluid at outlet
v_{f1} = velocity of flow at inlet
v_{w2} = velocity of whirl at outlet
v_{f2} = velocity of flow at outlet
β = angle between v_2 with the direction of motion of vane at outlet
ϕ = angle made by v_{r2} with vane motion direction at outlet, outlet vane angle

Efficiencies of Pump:



$\eta_m = \frac{\text{Rotor power}}{\text{Shaft power}}$ $\qquad \eta_h = \frac{H_m}{H_i}$ $\qquad \eta_v = \frac{Q}{Q+q}$

(1) **Manometric Efficiency (η_{man})** - ratio of the measured / manometric head to the head imparted by the impeller to water . Power at the impeller of the pump is more than the power given to the water at outlet of the pump.

$$\eta_{man} = \frac{H_m}{\left(\frac{V_{w2}u_2}{g}\right)} = \frac{H_m}{H_e}$$

He= Theoretical / Euler Head imparted to the fluid

The ratio of the power given to water at outlet of the pump to the power available at the impeller is called as **manometric efficiency.**

Power given to the water at outlet of the pump i.e. Pump Output $= \frac{WH_m}{1000} kW$

W= weight of the water = ρg (Q+q), if no loss mentioned, then q=0

Power at the impeller /rotor $= \frac{\text{Work done by impeller per sec}}{1000} kW = \frac{W}{g} \times \frac{V_{w2}u_2}{1000} kW$

manometric efficiency = $\boxed{\eta_{man} = \frac{gH_m}{V_{w2}u_2}}$

(2) **Mechanical Efficiency (η_m)** – Power at the shaft of the impeller of centrifugal pump is more than the power available at the impeller of the pump. The ratio of the power available in the impeller to the power at the shaft of the centrifugal pump is known as **mechanical efficiency** (95~98%).

$$\boxed{\eta_m = \frac{\frac{W}{g} \times \frac{V_{w2}u_2}{1000}}{S.P.}}$$

S. P. = Shaft Power

(4) **Hydraulic Efficiency (η_h)** – Hydraulic Losses refer to energy consumed by friction and fluid separation in the flow passage.This losses decrease the lift or head developed by the impeller. **Hydraulic efficiency** is the ratio of manometric head developed by the pump to theoretical or ideal head developed by the pump (Hi). where $H_i=\mu H_e$, **μ is slip factor of impeller**, if number of blade is infinite, then μ =1 . (for medium and higher discharge Q>0.28 m^3/s, η_v =0.98.)

$$\boxed{\eta_h = \frac{H_m}{H_i} = \frac{H_m}{\mu H_e} = \frac{\eta_{man}}{\mu}}$$

(4) **Volumetric Efficiency (η_v)** – Flow through pump is always associated with a volumetric loss due to leakage effect between the impeller and casing. Fluid leakage occurs due to pressure differential between the pressure and discharge side of the impeller, in the labyrinth seal, gland and stuffing box. **volumetric efficiency** is the ratio of actual to theoretical discharge.

$$\boxed{\eta_v = \frac{Q}{(Q+q)_i}}$$

(5) **Overall Efficiency (η_o)** –The ratio of the power output of the pump to the power input to the pump. (η_o Varies from 0.7 to 0.85)

$$\boxed{\eta_o = \left(\frac{WH_m}{1000} kW\right) \Big/ S.P.}$$

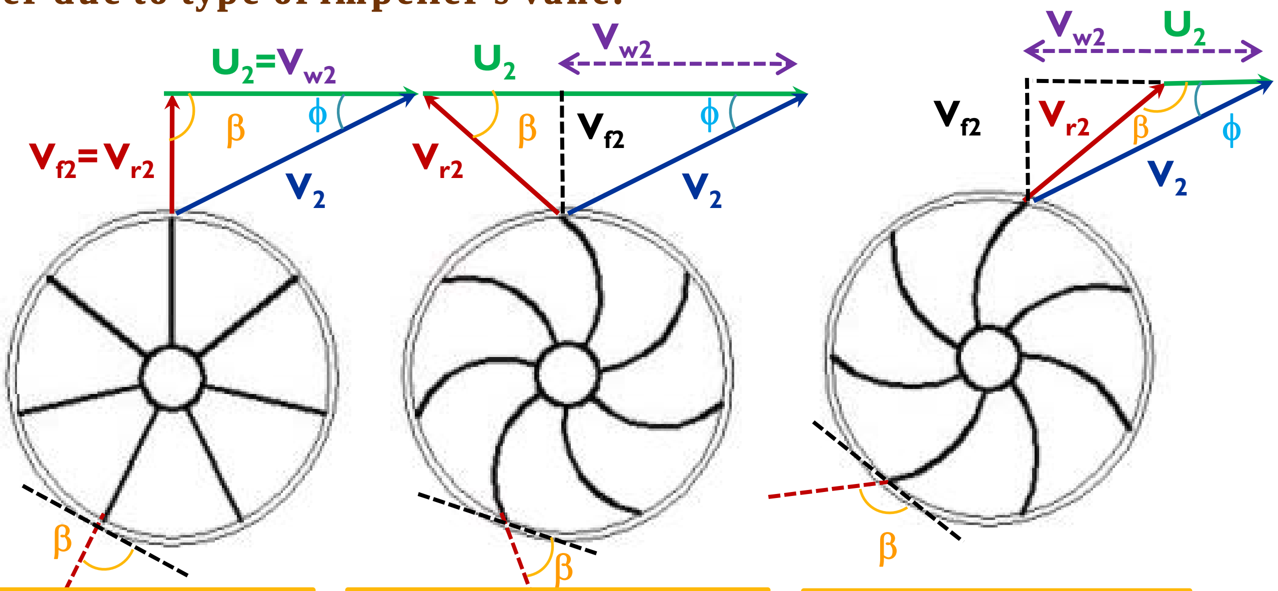
$$\boxed{\eta_o = \eta_{man} \times \eta_m}$$

$$\boxed{\eta_o = \eta_h \times \eta_v \times \eta_m}$$

Type of Impeller:

Three main categories of impeller due to type of impeller's vane:

- **Radial vanes, Fig. (a).**
- **Backward vanes, Fig. (b).**
- **Forward vanes, Fig. (c).**



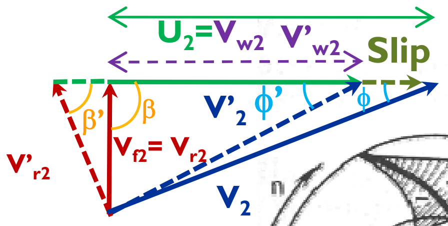
when β = 90°, radial curved vanes.

(b) when β < 90°, the Backwards curved vanes .

(c) when β > 90°, the Forwards curved vanes.

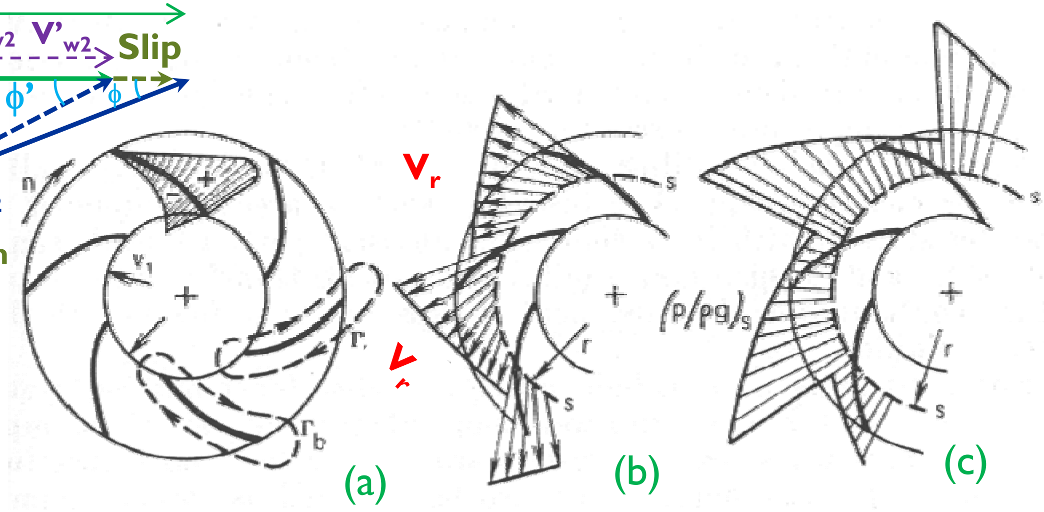
where :
V = absolute velocity of the water.
U = Tangential velocity of impeller (peripheral velocity).
V_r = relative velocity of water to the wheel.
V_f = velocity of flow.
v_w = velocity of whirl
N = Speed of impeller in (rpm).
β= angle between v_2 with the direction of motion of vane at outlet
ϕ = angle made by v_{r2} with direction of motion of vane at outlet, vane angle at outlet

*Due to inertial effect, the liquid which is trapped between the impeller vanes is reluctant to move round with the impeller. This results difference of pressure force across the vane. Therefore, high pressure developed in the leading side and low pressure on the trailing side. This difference is called **vane loading** which increases with the number of vane.*



Velocity diagram For Radial vane

where :
Fig (a) Γ_b = direction of blade circulation
Fig (b) V_r = Distribution of relative velocity in blade spaces.
Fig (c) Distribution of pressure in a certain radial section.

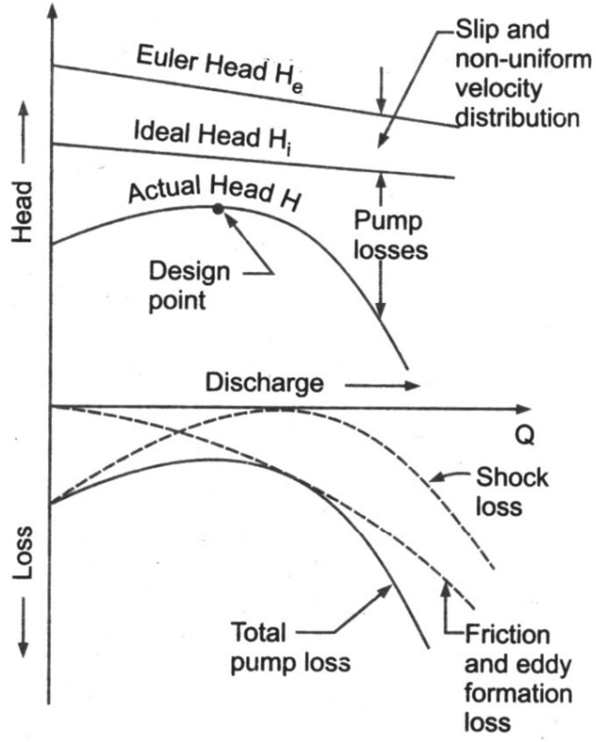


The Γ decreases V_r in working side of blade and vice versa. This forms pressure differential which is overcome by torque developed by the drive.

Slip factor:

On high pressure side: *liquid follows the blade contour it leaves blade tangentially .*
On low pressure side: *liquid leaves the vane with a certain circumferential component.*
As a result liquid leaves at an average angle β' which is less than actual geometric blade angle β.

- *Due to deviation in flow path, tangential component get reduced by (V_{w2}-V'_{w2}) which is called **slip** of the impeller.*
- *The **ideal slip coefficient** is then defined as the ratio of whirl component with the fluid deviation to the whirl component without fluid deviation*
- *Due to real fluid effect (friction and fluid separation on the wall of disc shroud and vanes) the radial velocity may not be uniform around the periphery of the impeller.*
- *The net effect of the non-uniform velocity and slip is to reduce Euler Head (H_e)*
- *Also further reduction of the losses occurs due to intake loss, friction and separation loss etc*



Loss and actual Head-Q curve

The major loss considered is shock losses at the impeller inlet caused by the mismatch of fluid and metal angles

Empirical Relations for Slip factor

$$(i)\quad \mu = 1 - \frac{\pi \sin \beta}{Z\left[1-\left(V_f/u_2\right)\cot\beta\right]}$$

(Stodolas relation)

$$= 1 - \frac{\pi}{Z} \text{ for radial } (\beta = 90°) \text{ blade}$$

$$(ii)\quad \mu = \left[A - B\left(V_f/u_2\right)\cot\beta\right]$$

$$= \left[1-\left(V_f/u_2\right)\cot\beta\right]$$

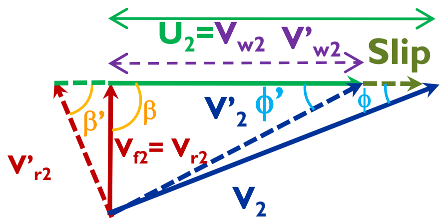
(Busemann expression)

where A and B are functions of r_2/r_1, β and Z

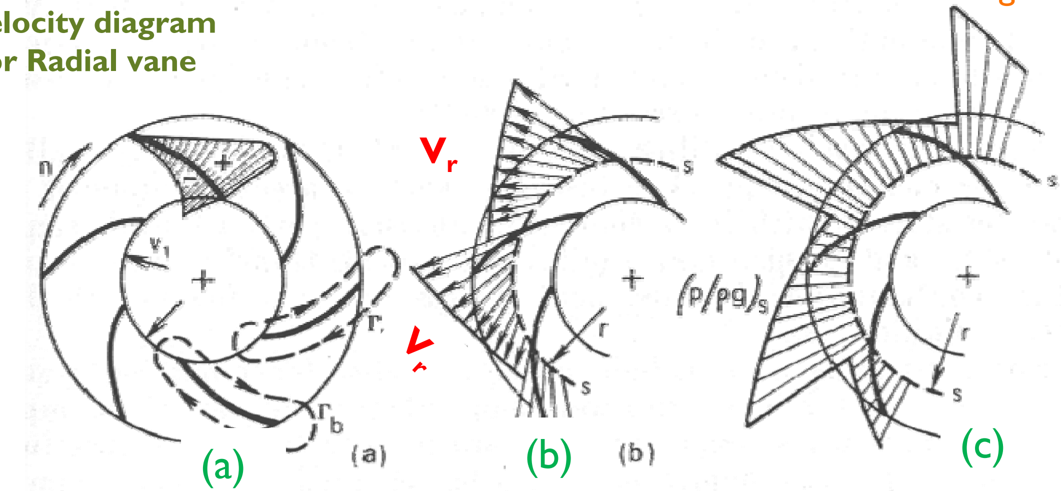
$$(iii)\quad \mu = 1 - \frac{0.63\,\pi}{Z\left[1-\left(V_f/u_2\right)\cot\beta\right]} \quad \text{(Stanitz relation)}$$

which, for radial blades, reduces to $\mu = 1 - \dfrac{0.63\,\pi}{Z}$

$$(iv)\quad \frac{1}{\mu} = 1 + \frac{1.2(1+\sin\beta)}{Z\left[1-\left(r_1/r_2\right)^2\right]}$$



Velocity diagram For Radial vane



Velocity Vector Diagram and Assumptions

Assumptions:

- ✓ Infinite number of vanes, no energy loss in impeller due to friction and eddy formation.
- ✓ Uniform velocity distribution in narrow passage between two adjacent passages.
- ✓ Fluid enters in the eye in radial/axial direction and whirl component at inlet $V_{w1} = 0$, $V_{f1}=V_1$
- ✓ No loss due to shock entry, i.e. Inlet edge of the impeller blades are parallel to the relative velocity.

The rate of change of angular momentum $m(V_{w2}r_2 - V_{w1}r_1) = \rho Q(V_{w2}r_2 - V_{w1}r_1) = \dfrac{\gamma Q}{g}(V_{w2}r_2 - V_{w1}r_1)$

Therefore, Torque = $\gamma Q / g\,(V_{w2}r_2 - V_{w1}r_1)$

Energy transfer (**E**) = torque × rotational speed in rad/sec $\dfrac{\gamma Q}{g}(V_{w2}r_2 - V_{w1}r_1)\times\omega = \dfrac{\gamma Q}{g}(V_{w2}u_2 - V_{w1}u_1)\times\omega$

Energy transfer per unit head i.e. **Euler Head (H_e)** $E/\gamma Q = (V_{w2}u_2 - V_{w1}u_1)/g$

As whirl component at inlet $V_{w1} = 0$, therefore $H_e = (V_{w2}u_2 - \cancel{V_{w1}}u_1)/g = V_{w2}u_2/g$ (=H_m, shockless entry)

Finally

$$H_e = \overset{I}{\frac{(V_2^2 - V_1^2)}{2g}} + \overset{II}{\frac{(u_2^2 - u_1^2)}{2g}} + \overset{III}{\frac{(V_{r2}^2 - V_{r1}^2)}{2g}}$$

Term I : increase in KE or dynamic head, conversation of dynamic head into static pressure is achieved through retardation in volute casing.
Term II : increase in static pressure due to centrifugal force, due to rotating fluid from one radius to another
Term III : change in KE due to retardation of fluid relative to impeller.

Head Capacity relationship

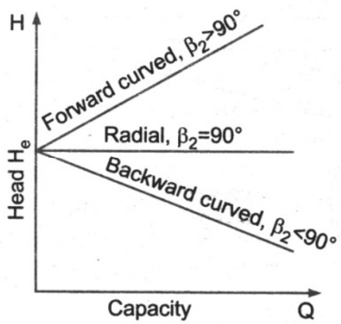
$$H_e = \frac{V_{w2}u_2}{g} = \frac{u_2}{g}\left[u_2 - V_{f2}\cot\beta\right] = \frac{u_2}{g}\left[u_2 - \frac{Q}{A_2}\cot\beta\right]$$

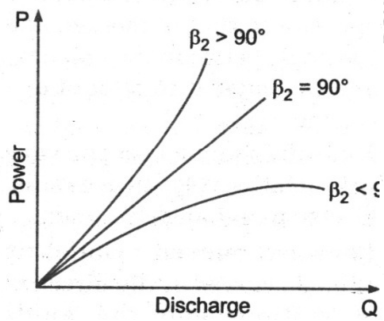
For a particular pump running at constant speed, i.e. β, A_2 and u_2=constant, So,

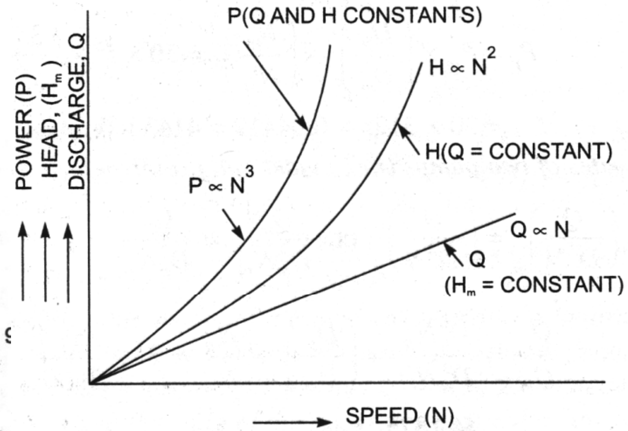
$H_e = K_1 - K_2 Q$ (Straight line)

$P \propto QH$ i.e. $P_e = A'Q - B'Q^2$

For β<90°, cot β is +ve, Q↑ H_e↓
For β=90°, cot β is 0, Q↑ H_e constant
For β>90°, cot β is -ve, Q↑ H_e ↑







(Main Characteristics Curve)

Cavitation and other criteria

Multistage Pump:

N= number of stage, then Total Head developed = n × H_m

Cavitation:

$$\sigma = \frac{V_s^{\,2}}{2g} = \frac{(H_{atm} - H_v) - H_s - h_{Ls}}{H} = \frac{NPSH}{H}$$

If Critical cavitation factor (σ_c) > σ , then cavitation start:

$$\sigma_c = 1.042 \times 10^{-3} (N_s)^{4/3}$$

Maximum suction lift for a pump is as follows, otherwise vaporization of liquid at inlet of pump take place i.e. cavitation

$$Hs = H_{atm} - H_v - \frac{V_s^{\,2}}{2g} - h_{Ls}$$

h_{Ls}= frictional head loss in suction pipe , H_s = suction pressure head in water (m), H = head developed by pump

So, for cavitation free operation $\sigma \geq \sigma_c$ or NPSH $\geq \sigma_c$H i.e. Minimum NPSH should be equal to σ_cH

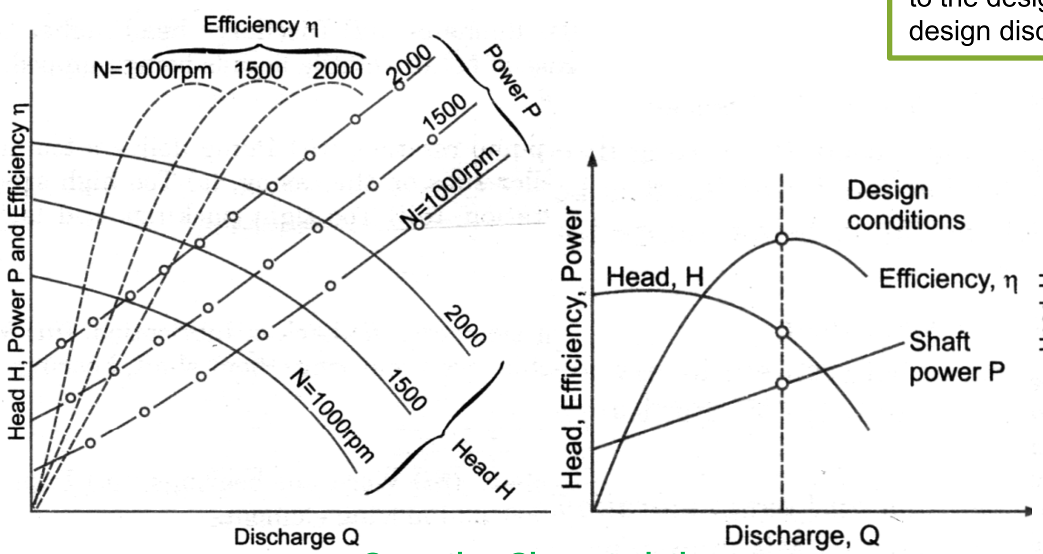
Priming

When centrifugal pump is not running for some time, the water present in the pump casing and suction pipe flows back to the sump and these spaces get filled with air. Now, when motor is switched on and pump starts running, the head developed is equal to $\boldsymbol{H_e}$ of air. Since $\rho_{air} << \rho_{water}$, the head thus generated cannot produce spontaneously the vacuum required to start the pumping action. Accordingly water cannot be sucked in along the suction pipe to reach the impeller. For making the pump deliver water, there need to make the casing, impeller and suction line free from air and fill these spaces with water. This process is called priming.

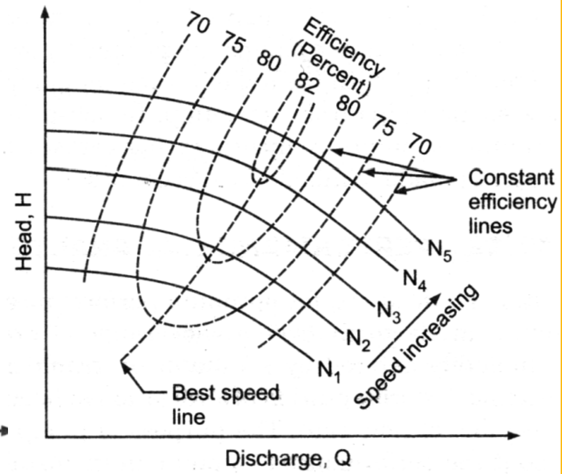
For small pump vent cock is provided to supply the water, and for self priming pump certain arrangement is done to ensure supply of water in suction pipe.

Performance characteristics of centrifugal pump

Pumps are generally designed for maximum efficiency and that occurs when the pump operates at design speed. A particular characteristics corresponding to the design speed are called operating characteristics. It helps to obtain design discharge , head corresponding to the point of maximum efficiency.



Operating Characteristics



Muschel Curve for pump

Muschel curve depicts the performance of the pump over its entire range of operation. Data for plotting these curves is obtained from main characteristics curve curves η vs Q, H vs Q. It helps to locate the region where pump would operate with maximum efficiency.

Detection of Pump Problems during operation

1. *Pump fails to start pumping (No delivery of liquid)*

(*i*) Pump may not be properly primed, (*ii*) Suction lift and delivery head too high, (*iii*) Too low speed, (*iv*) Wrong direction of pump rotation, (*v*) Clogging of impeller or strainer or suction line.

2. *Pump working but not upto capacity and pressure*

(*i*) Air leakage into pump, (*ii*) Too low speed, (*iii*) Wrong direction of pump rotation, (*iv*) Partial clogging of impeller, (*v*) Small impeller diameter, (*vi*) Discharge head higher than anticipated, (*vii*) Insufficient suction head in case of hot or volatile liquids being pumped.

3. *Pump takes too much of power*

(*i*) Too high speed, (*ii*) Wrong direction of pump rotation, (*iii*) Pump delivers too much liquid at too low head, (*iv*) Shaft is bent, impeller rubs on the casing, (*v*) Too high specific gravity of the liquid being pumped, (*vi*) Cavitation, (*vii*) Too tight packings and lack of lubrication.

4. *Pump gets overheated*

(*i*) Rubbing between stationary and moving elements, (*ii*) Lack of lubrication, (*iii*) Very high impeller speed, (*iv*) Mechanical defects such as worn out rings, bent shaft, bearings not cleaned properly.

5. *Noisy operation and pump vibrations*

(*i*) Cavitation, (*ii*) Misalignment of pump shaft, (*iii*) Worn out bearings, (*iv*) Improper foundations, (*v*) Rubbing action between stationary and moving elements.

The Affinity Law

The affinity laws for pumps/fans are used in hydraulics to express the relationship between variables involved in pump or fan performance (such as head, volumetric flow rate, shaft speed) and power. They apply to pumps, fans, and hydraulic turbines. In these rotary implements, the affinity laws apply both to centrifugal and axial flows.

The affinity laws are useful as they allow prediction of the head discharge characteristic of a pump or fan from a known characteristic measured at a different speed or impeller diameter. The only requirement is that the two pumps or fans are dynamically similar, that is the ratios of the fluid forced are the same.

Formulas for Refiguring Pump Performance with Impeller Diameter or Speed Change		
Diameter Change Only	Speed Change Only	Diameter and Speed Change
$\frac{Q_2}{Q_1}=\left(\frac{D_2}{D_1}\times\frac{N_2}{N_1}\right)$	$\frac{Q_2}{Q_1}=\left(\frac{N_2}{N_1}\right)$	$\frac{Q_2}{Q_1}=\left(\frac{D_2}{D_1}\right)$
$\frac{H_2}{H_1}=\left(\frac{D_2}{D_1}\times\frac{N_2}{N_1}\right)^2$	$\frac{H_2}{H_1}=\left(\frac{N_2}{N_1}\right)^2$	$\frac{H_2}{H_1}=\left(\frac{D_2}{D_1}\right)^2$
$\frac{P_2}{P_1}=\left(\frac{D_2}{D_1}\times\frac{N_2}{N_1}\right)^2$	$\frac{P_2}{P_1}=\left(\frac{N_2}{N_1}\right)^2$	$\frac{P_2}{P_1}=\left(\frac{D_2}{D_1}\right)^2$

Similarity relationship

$$\boxed{\frac{H}{D^2N^2}=\text{constant}}$$

$$\boxed{\frac{Q}{D^3N}=\text{constant}}$$

$$\boxed{\frac{P}{D^5N^3}=\text{constant}}$$

Specific Speed for Pump

Specific Speed $Ns=\frac{N\sqrt{Q}}{H^{3/4}}$

Specific Speed (IS code) $Ns=\frac{3.65N\sqrt{Q}}{H^{3/4}}$

Type of impeller	Slow speed radial flow	Normal speed radial flow	High speed radial flow	Mixed flow	Axial or Propeller
Specific Speed	10-30	30-50	50-80	80-160	160-500

As per BIS Code: Speed of a Geometrically similar pump capable of lifting 75 kg of water per second to a height of one meter.

Specific Speed has dimension $L^{3/4}T^{-3/2}$,

Dimensionless specific speed is $Ns=\frac{N\sqrt{Q}}{(gH)^{3/4}}$

Dimension of dimensionless specific speed $Ns=\frac{T^{-1}(L^3T^{-1})^{1/2}}{(LT^{-2}\times L)^{3/4}}$

Kirloskar 'KDS+/KDS++' Series Three Phase Pumps

Single stage, High speed, Centrifugal Monobloc with Volute Type Delivery Pumps

Features

- Can withstand wide voltage fluctuations from 300-440 volts
- Suction lift upto 7.5 metres.
- Top flat efficiency curve: minimum variation in efficiency in entire operating range.
- Efficiency at par with internationally available pumps - higher upto 10 points than minimum required by Indian Standards specifications.
- Models are with IP-44 protection. Models with IP-55 protection are also available on request.
- Class of insulation 'B' ('F' class from 7.5 kW and above).
- Designed to prevent overloading and motor burning.
- Dynamically balanced rotating parts ensure minimum vibrations.
- Replaceable wearing parts and hence longer life.
- Designed for automatic air release during priming

Applications

- Irrigation
- Industries, for clear water handling
- Water supply for domestic use in high-rise Apartments, Buildings and Hotels

Range

Head - 6 - 76 metres
Capacity - 37.0-2.0 Litres per sec.
Power Rating - 1.5 kW to 22 kW (2.0 HP to 30.0 HP)

Direction of Rotation

Anticlockwise, when viewed from suction side.

Material of Construction:

Standard supply

Impeller : Cast Iron
Delivery Casing : Cast iron
Motor body : Cast Iron
Shaft : Carbon Steel
Shaft Sleeve : Bronze

Optional supply

Impeller : Bronze
Shaft : Stainless Steel
Shaft Sleeve : Chrome Steel

Approximate performance of 'KDS+/KDS++' Series, 2 Pole, Monobloc Pumps, at rated Voltage, 50 Hz, Three Phase A.C Power Supply:-

Pump Model	Pump Rating		Pipe Size (mm)		TOTAL HEAD IN METRES																		Rated Voltage (volts)
	kW	HP	SUC.	DEL.	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	40		
					CAPACITY IN LITRES PER SECOND																		
KDS-216++	1.5	2.0	65	50		11.0	10.0	8.8	7.15	4.0													415
KDS-225++	1.5	2.0	50	40		5.3	5.18	4.9	4.75	4.5	4.25	3.9	3.56	3.1	2.25								415
KDS-314+	2.2	3.0	80	80	19.1	17.8	16.2	13.8	10.4														415
KDS-318++	2.2	3.0	80	65		13.4	12.6	11.7	10.7	9.2	7.5												.415
KDS-325++	2.2	3.0	65	50			9.2	8.8	8.4	7.9	7.4	7.0	6.4	5.8	4.9								415
KDS-335++	2.2	3.0	50	40					5.0	4.9	4.8	4.7	4.6	4.4	4.2	4.0	3.8	3.6	3.2	2.7	2.0		415
KDS-515+	3.7	5.0	100	100	32.8	31.0	28.0	24.2	19.0	12.9													400
KDS-520+	3.7	5.0	80	80	24.0	23.0	22.0	20.8	19.5	17.9	16.0	14.0	11.0										400
KDS-527++	3.7	5.0	80	65'						14.3	13.5	12.5	11.6	10.3	8.7	6.4							400
KDS-822++	5.5	7.5	100	100				27.3	25.6	24.0	22.2	20.1	17.6	14.5									400
KDS-830++	5.5	7.5	80	65						19.0	18.2	17.3	16.4	15.4	14.2	13.0	11.1						400
KDS-1030++	7.5	10.0	100	100				32.0	31.0	29.8	28.5	27.0	25.0	23.5	21.0	18.0							415
KDS-1040+	7.5	10.0	80	65			23.5	23.0	22.6	22.2	21.6	20.9	20.35	19.5	18.75	17.9	17.0	15.7	14.6	13.4	9.6		415
					18	20	24	28	30	32	36	40	44	48	52	54	56	60	64	68	72	76	
KDS-538+	3.7	5.0	65	50		8.5	8.3	7.8	7.6	7.1	5.8												400
KDS-550++	3.7	5.0	50	40							4.1	3.7	3.3	2.7	2.0								400
KDS-837	5.5	7.5	65	65				12.6	12.2	11.8	11.1	9.0											400
KDS-844++	5.5	7.5	65	65			10.7	10.3	9.9	9.5	8.4	7.1	4.7										400
KDS-852++	5.5	7.5	65	50						8.6	8.1	7.5	6.8	5.9	4.5	3.7							400
KDS-1050++	7.5	10.0	65	65			12.9	12.5	12.3	12.0	11.4	10.7	9.6	8.1	6.0								415
KDS-1065++	7.5	10.0	65	50								7.8	7.4	6.9	6.4	6.1	5.8	5.1	4.3	3.0			415
KDS-1331+	9.3	12.5	100	100	33.4	32.0	29.0	23.8	12.5														415
KDS-1348+	9.3	12.5	80	65		19.6	18.9	18.2	17.8	17.3	15.8	14.3	11.9										415
KDS-1537+	11.0	15.0	100	100	36.8	35.9	33.0	30.0	28.0	25.0	17.5												415
KDS-1555+	11.0	15.0	80	65					19.4	19.2	18.5	17.4	16.0	14.5	12.2	10.5							415
KDS-1575+	11.0	15.0	65	50											8.1	7.9	7.7	7.4	6.9	6.4	5.8	4.9	415
KDS-2050+	15.0	20.0	100	80	33.8	33.2	31.8	30.4	29.7	28.8	27.2	25.0	22.8	19.4									415
KDS-2560+	18.7	25.0	100	80										24.5	22.0	20.5	18.6	13.0					415
KDS-3068+	22.0	30.0	100	80										28.0	26.5	25.6	24.5	215	17.5	10.0			415

All pumpsets except KDS-837+, KDS-575+, KDS-2560+ and KDS-3068+ are ISI marked. KDS-318~+ can' also b~ offered with pipe size 65x50mm. Above pumpsets except KDS-2560+ and KDS 3068 can be supplied with stuffing box arrangement for GLAND PACKED or MECHANICAL SEAL against the requirement. **Note:** Performance applicable to liquid of specific gravity 1 and viscosity as of water

Sample Sheet for pump selection

Suppose ------

System Requirements:

Suction Pipe diameter: 2” (50 NB)
Delivery Pipe diameter: 1.5” (40 NB)
Motor speed – 1450 rpm
Head – 8 m to 18 m
Discharge – 8 LPS to 10 LPS

Now Select your Pump ……..