Rajasthan Institute of Engineering & Technology, Jaipur

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University Roll No. \_\_\_\_\_\_\_\_\_\_\_\_\_\_

 Branch: - …………..MECHANICAL

 Subject: - ………… TURBOMACHINES

 Set: A Solution

1. Q1. The impeller and shaft are the rotating parts of the pump that converts driver energy into kinetic energy. This inbuilt rotating impeller is the main rotating part used for pumping liquids without any complications. It provides centrifugal acceleration to the fluids. The impeller can be classified according to its mechanical construction. You can find closed, open, semi-open or vortex type. Closed impellers have a drawback and that is they come up with a lot of maintenance issues whereas the open and semi-open types of impellers do not clog, but usually need manual adjustment.

Another important rotating component is the shaft whose basic purpose is to transmit the torque that is encountered during operation. It also has to support the impeller and other rotating parts.

1. The volute or diffuser along with casing, casing cover, and bearings form the stationary parts of the centrifugal pump and are responsible for converting the kinetic energy into pressure energy. These pumps have two types of casings: volute and circular. Volute casings help to balance the hydraulic pressure created on the shaft of the pump while the circular casings are used for high capacity. Even the bearings are very useful components in the centrifugal pumps.

All these various components when put together and when they cooperate well with each other, centrifugal pumps achieve a great performance.

[Centrifugal pumps](http://www.rotechpumps.com/products/centrifugal-pumps/)are available in various different materials such as metal, plastic and ceramic. Steam turbines, high-speed electric motors and internal combustion engines all use centrifugal pumps. No valves are used in this machine.

### Applications of Centrifugal Pumps:

Centrifugal pumps are used in many applications and across many industries. Some of the commercial, municipal, industrial and scientific fields that make use of such pumps are listed below:

1. Oil and energy companies, refineries, power plants
2. Heating and ventilation, air conditioning, pressure boosting, fire protection sprinkler systems.
3. Waste water processing plants, manufacturing industries, boiler feed applications, municipal, industrial applications, irrigation, drainage and flood protection
4. Chemical and Process Industries, pharmaceuticals, cellulose, sugar refining, food and beverage production
5. Cryogenics and refrigerants.

OR

Specific speed, or Ns, is an index of impeller design that describes the relationship between the amount of head generated by the rotation of an impeller relative to the amount of flow produced by the impeller.

The formal definition of specific speed, or Ns, according to [Cameron Hydraulic Data](http://www.flowserve.com/Products/Pumps/Cameron-Hydraulic-Data-Book%2Cen_US) is:

*The speed at which an impeller, geometrically similar to the one under consideration, would run if it were reduced in size to deliver one GPM at one foot TDH.*

However, the basic idea is that specific speed is a number used to describe the design of an impeller:

* Impellers that generate a lot of head, but very little flow have very low specific speeds.
* Impellers that generate very little head but a great deal of flow have very high specific speeds.

Specific speed is calculated on the basis of flow and head at BEP according to the following formula courtesy of [Wikipedia](https://en.wikipedia.org/wiki/Specific_speed):

Common specific speed values for rotodynamic pump impellers fall between 400 and 15000.

Q2



Or

The [head of a pump](https://www.engineeringtoolbox.com/pump-energy-equation-d_631.html) can be expressed in metric units as:

h = (p2 - p1) / (ρ g) + v22/ (2 g)*(1)*

*where*

h*= total head developed (m)*

p2*= pressure at outlet (N/m2)*

p1*= pressure at inlet (N/m2)*

ρ*=   density (kg/m3)*

g*= acceleration of gravity (9.81)  m/s2*

v2*= velocity at the outlet (m/s)*

Head described in simple terms

* a pump's vertical discharge "pressure-head" is the vertical lift in height - usually measured in feet or m of water - at which a pump can no longer exert enough pressure to move water. At this point, the pump may be said to have reached its "shut-off" head pressure. In the flow curve chart for a pump the "shut-off head" is the point on the graph where the flow rate is zero

### Pump Efficiency

Pump efficiency, *η (%)* is a measure of the efficiency with which the pump transfers useful work to the fluid.

*η = Pout/ Pin  (2)*

*where*

*η = efficiency (%)*

*Pin = power input*

*Pout = power output*

Q3 Centrifugal Pumps Used in The Oil and Gas Industry

* Electric Submersible Pump (ESP) – These pumps are typically submerged entirely in the fluid to be pumped and are specifically designed to combat pump cavitation. Instead of pulling fluids, the mechanisms in these pumps push fluids, making them far more reliable and efficient than previously utilized jet pumps. Newer ESP models can also include a water and oil separator which permits water to be re-injected into the reservoir without the need to lift it to the surface, saving both time and operating costs.
* Helico-Axial Pump – Also called a Poseidon pump, this centrifugal pump uses multiple stages of impellers and vanes to move fluids. Compression is accomplished with the transfer of kinetic energy from the rotating impeller blades through circles of guide vanes in order to move fluid. This pump is often used in offshore and deep water developments.
* Deep Well Pump – The industry is beginning to lean more towards these types of pumps especially the deep well seawater lifting and fire pumps. They can have capacities of up to 2,600 cubic meters and offer added safety protection for offshore production. Many of these pumps use radial-designed impellers for smaller capacities and a semi-radial design for larger capacities, which can result in an MTBR of 25,000 hours or more.

Positive Displacement Pumps Used in The Oil and Gas Industry

* Progressive Cavity Pump – Also known as eccentric screw or single screw pumps, these types of pumps are utilized for their ability to transfer difficult liquids, such as those that containing solids or highly viscous fluids. They work by using a single screw or rotor inside a double-threaded rubber stator to build pressure and move fluid. They are mostly used in shallow wells or at the surface.
* Twin Screw Pump – This pump works by rotating to form chambers with the intermeshing of the two screws inside the pump housing. The chambers fill with fluid and move it from the suction side to the higher pressure discharge side of the pump, a process that can be reversed in some twin screw pumps. They can handle virtually any non-homogeneous fluid with any of abrasiveness, lubricity, and viscosity. Twin pumps are most often used in situations that contain high gas volume fractions and fluctuating inlet conditions.

Other Pumps Used in The Oil and Gas Industry

* Deep Well Submersible Pump – Just what it sounds like, this vertical is submerged in deep waters for the purpose of performing a number of upstream processes. The unit is oil-filled, which allows it to be reliable and long lasting while offering reduced cost in total life cycle. It operates with the use of heavy-duty impellers, dual bearings, and multiple seal options. In addition, these pumps can handle flows of up to 6,000 cubed meters, heads to 800 meters, and speeds up to 3,600 rpm.
* Chemical Process Pump – These pumps are used in the handling of harmful chemicals in many industries, including upstream oil and gas. They can convey hazardous or corrosive chemicals efficiently in order to avoid any damage occurring at the working place whether to equipment or personnel. The chemical process pump uses a combination of close coupling, heavy duty casing, specialty impellers, a sealed chamber, and other mechanisms to remove the harmful chemicals.
* Oil Skimmer Aluminum Lobe – These [rotary lobe pumps](http://empoweringpumps.com/products/new-boerger-oil-skimmer-aluminum-lobe-pump-compact-lightweight/) can be used upstream, midstream, and downstream. The pump is lightweight, compact, has a large flow range, dry run capabilities, and conveys highly viscous fluids.
* Multiple Screw Pump Line  – One of the newest introductions into the global markets, this [Multiple Screw Pump Line](http://empoweringpumps.com/products/multiple-screw-pump-line/) includes double, geared-twin, and triple screw pumps. It offers ranges to 2500 gpm / 1160 psi.  It can also handle high and low viscosities, as well as lubricating and non-lubricating liquids.

Or

In normal conditions, common centrifugal pumps are unable to evacuate the air from an inlet line leading to a fluid level whose geodetic altitude is below that of the pump. Self-priming pumps have to be capable of evacuating air (see Venting) from the pump suction line without any external auxiliary devices.

Centrifugal pumps with an internal suction stage such as water jet pumps or side channel pumps are also classified as self-priming pumps.

Centrifugal pumps which are not designed with an internal or external self-priming stage can only start to pump the fluid after the pump has initially been primed with the fluid. Sturdier but slower, their impellers are designed to move water which is far denser than air, leaving them unable to operate when air is present.[[7]](https://en.wikipedia.org/wiki/Centrifugal_pump#cite_note-7) In addition, a suction-side swing check valve or a vent valve must be fitted to prevent any siphon action and ensure that the fluid remains in the casing when the pump has been stopped. In self-priming centrifugal pumps with a separation chamber the fluid pumped and the entrained air bubbles are pumped into the separation chamber by the impeller action.

The air escapes through the pump discharge nozzle whilst the fluid drops back down and is once more entrained by the impeller. The suction line is thus continuously evacuated. The design required for such a self-priming feature has an adverse effect on pump efficiency. Also, the dimensions of the separating chamber are relatively large. For these reasons this solution is only adopted for small pumps, e.g. garden pumps. More frequently used types of self-priming pumps are side channel and water ring pumps. Another type of self-priming pump is a centrifugal pump with two casing chambers and an open impeller. This design is not only used for its self-priming capabilities but also for its degassing effects when pumping twophase mixtures (air/gas and liquid) for a short time in process engineering or when handling polluted fluids, for example when draining water from construction pits.

This pump type operates without a foot valve and without an evacuation device on the suction side. The pump has to be primed with the fluid to be handled prior to commissioning. Two-phase mixture is pumped until the suction line has been evacuated and the fluid level has been pushed into the front suction intake chamber by atmospheric pressure. During normal pumping operation this pump works like an ordinary centrifugal pump.

Simply defined, cavitation is the formation of bubbles or cavities in liquid, developed in areas of relatively low pressure around an impeller. The imploding or collapsing of these bubbles trigger intense shockwaves inside the pump, causing significant damage to the impeller and/or the pump housing.

If left untreated, pump cavitation can cause:

* Failure of pump housing
* Destruction of impeller
* [Excessive vibration](https://blog.craneengineering.net/troubleshooting-centrifugal-pumps) - leading to premature seal and bearing failure
* Higher than necessary power consumption
* Decreased flow and/or pressure

There are two types of pump cavitation: suction and discharge.

Q4.

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|  The peripheral speed at impeller outlet(given)Work input per unit weight of Water = =72.78mUnder ideal condition (without loss), the total head developed by the pump = 72.78 mAbsolute velocity of water at the outlet=23.28 m/sAt the whirlpool chamber,The velocity of water at delivery = 0.5 ´ 23.28m/sTherefore the pressure head at impeller outlet=72.78 - = 65.87mHence, we theoretical maximum lift = 65.87m |

Or

While vane pumps can handle moderate viscosity liquids, they excel at handling low viscosity liquids such as LP gas (propane), ammonia, solvents, alcohol, fuel oils, gasoline, and refrigerants.  Vane pumps have no internal metal-to-metal contact and self-compensate for wear, enabling them to maintain peak performance on these non-lubricating liquids.  Though efficiency drops quickly, they can be used up to 500 cPs / 2,300 SSU.

Vane pumps are available in a number of vane configurations including sliding vane (*left*), flexible vane, swinging vane, rolling vane, and external vane.  Vane pumps are noted for their dry priming, ease of maintenance, and good suction characteristics over the life of the pump.  Moreover, vanes can usually handle fluid temperatures from -32�C / -25�F to 260�C / 500�F and differential pressures to 15 BAR / 200 PSI (higher for hydraulic vane pumps).

Each type of vane pump offers unique advantages.  For example, external vane pumps can handle large solids.  Flexible vane pumps, on the other hand, can only handle small solids but create good vacuum.  Sliding vane pumps can run dry for short periods of time and handle small amounts of vapor.

*How Vane Pumps Work*

Despite the different configurations, most vane pumps operate under the same general principle described below.

1.  A slotted rotor is eccentrically supported in a cycloidal cam.  The rotor is located close to the wall of the cam so a crescent-shaped cavity is formed.  The rotor is sealed into the cam by two sideplates.  Vanes or blades fit within the slots of the impeller.  As the rotor rotates (*yellow arrow*) and fluid enters the pump, centrifugal force, hydraulic pressure, and/or pushrods push the vanes to the walls of the housing.  The tight seal among the vanes, rotor, cam, and sideplate is the key to the good suction characteristics common to the vane pumping principle.

2.  The housing and cam force fluid into the pumping chamber through holes in the cam (*small red arrow on the bottom of the pump*).  Fluid enters the pockets created by the vanes, rotor, cam, and sideplate.

3.  As the rotor continues around, the vanes sweep the fluid to the opposite side of the crescent where it is squeezed through discharge holes of the cam as the vane approaches the point of the crescent (*small red arrow on the side of the pump*).  Fluid then exits the discharge port.

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| *Advantages** Handles thin liquids at relatively higher pressures
* Compensates for wear through vane extension
* Sometimes preferred for solvents, LPG
* Can run dry for short periods
* Can have one seal or stuffing box
* Develops good vacuum
 | *Disadvantages** Can have two stuffing boxes
* Complex housing and many parts
* Not suitable for high pressures
* Not suitable for high viscosity
* Not good with abrasives

  |

*Applications*

* Aerosol and Propellants
* Aviation Service - Fuel Transfer, Deicing
* Auto Industry - Fuels, Lubes, Refrigeration Coolants
* Bulk Transfer of LPG and NH3
* LPG Cylinder Filling
* Alcohols
* Refrigeration - Freons, Ammonia
* Solvents
* Aqueous solutions

SET 2

Q1

: When the pump is started there will be no flow until the pressure rise in the impeller is more than or equal to the manometric head. In other words the centrifugal head should be greater than the manometric head. Therefore, minimum starting speed is the speed of centrifugal pump at which centrifugal head is equal to manometric head.

For minimum starting speed condition, *centrifugal head=manometric head*

$$\frac{\left(U\_{2}^{2}-U\_{1}^{2}\right)}{2g}=H\_{m}=\frac{η\_{ma}U\_{2}V\_{u2}}{g}$$

$$\frac{\left[\left(\frac{πD\_{2}N}{60}\right)^{2}-\left(\frac{πD\_{1}N}{60}\right)^{2}\right]}{2}=η\_{ma}\left(\frac{πD\_{2}N}{60}\right)V\_{u2}$$

$$\frac{\left(\frac{πN}{60}\right)^{2}\left[D\_{2}^{2}-D\_{1}^{2}\right]}{2}=η\_{ma}\left(\frac{πN}{60}\right)D\_{2}V\_{u2}$$

$$\frac{πND\_{2}^{2}\left[1-\frac{D\_{1}^{2}}{D\_{2}^{2}}\right]}{120}=η\_{ma}D\_{2}V\_{u2}$$

$$\frac{πND\_{2}\left[1-\frac{D\_{1}^{2}}{D\_{2}^{2}}\right]}{120}=η\_{ma}V\_{u2}$$

Then minimum starting speed in rpm is,

$$N\_{min}=\frac{120η\_{ma}V\_{u2}}{πD\_{2}\left[1-\frac{D\_{1}^{2}}{D\_{2}^{2}}\right]}$$

OR

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| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| The axial flow or propeller pump is the converse of axial flow turbine and is very similar to it an appearance. The impeller consists of a central boss with a number of blades mounted on it. The impeller rotates within a cylindrical casing with fine clearance between the blade tips and the casing walls. Fluid particles, in course of their flow through the pump, do not change their radial locations. The inlet guide vanes are provided to properly direct the fluid to the rotor. The outlet guide vanes are provided to eliminate the whirling component of velocity at discharge. The usual number of impeller blades lies between 2 and 8, with a hub diameter to impeller diameter ratio of 0.3 to 0.6.The Figure 37.1 shows an axial flow pump. The flow is the same at inlet and outlet. an axial flow pumps develops low head but have high capacity. the maximum head for such pump is of the order of 20m.The section through the blade at X-X (Figure 37.1) is shown with inlet and outlet velocity triangles in Figure 37.2.

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| Figure 37.1 A propeller of an axial flow pump |

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| Figure 37.2 Velocity triangles of an axial flow pump |

AnalysisThe blade has an aerofoil section. The fluid does not impinge tangentially to the blade at inlet, rather the blade is inclined at an angle of incidence(i) to the relative velocity at the inlet  . If we consider the conditions at a mean radius  then

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where  is the angular velocity of the impeller.Work done on the fluid per unit weight = For maximum energy transfer ,  , i.e  .Again , from the outlet velocity triangle,

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|  |

Assuming a constant flow from inlet to outlet

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Then, we can writeMaximum energy transfer to the fluid per unit weight

|  |  |
| --- | --- |
|  | (37.1) |

For constant energy transfer over the entire span of the blade from hub to tip , the right hand side of Equation (37.1) has to be same for all values of  . It is obvious that  increases with radius  , therefore an equal increase in must take place, and since  is constant then  must increase. Therefore , the blade must be twisted as the radius changes.Matching of Pump and System CharacteristicsThe design point of a hydraulic pump corresponds to a situation where the overall efficiency of operation is maximum. However the exact operating point of a pump, in practice, is determined from the matching of pump characteristic with the headloss-flow, characteristic of the external system (i.e. pipe network, valve and so on) to which the pump is connected.Let us consider the pump and the piping system as shown in Fig. 15.18. Since the flow is highly turbulent, the losses in pipe system are proportional to the square of flow velocities and can, therefore, be expressed in terms of constant loss coefficients. Therefore, the losses in both the suction and delivery sides can be written as

|  |  |
| --- | --- |
|  | (37.2a) |

|  |  |
| --- | --- |
|  | (37.2b) |

where,  is the loss of head in suction side and  is the loss of head in delivery side and *f*is the Darcy's friction factor,  and  are the lengths and diameters of the suction and delivery pipes respectively, while  and are accordingly the average flow velocities. The first terms in Eqs. (37.1a) and (37.1b) represent the ordinary friction loss (loss due to friction between fluid ad the pipe wall), while the second terms represent the sum of all the minor losses through the loss coefficients  and  which include losses due to valves and pipe bends, entry and exit losses, etc. Therefore the total head the pump has to develop in order to supply the fluid from the lower to upper reservoir is

|  |  |
| --- | --- |
|  | (37.3) |

Now flow rate through the system is proportional to flow velocity. Therefore resistance to flow in the form of losses is proportional to the square of the flow rate and is usually written as

|  |  |
| --- | --- |
|  = system resistance =  | (37.4) |

where *K*is a constant which includes, the lengths and diameters of the pipes and the various loss coefficients. System resistance as expressed by Eq. (37.4), is a measure of the loss of head at any particular flow rate through the system. If any parameter in the system is changed, such as adjusting a valve opening, or inserting a new bend, etc., then *K*will change. Therefore, total head of Eq. (37.2) becomes,

|  |  |
| --- | --- |
|  | (37.5) |

The head *H*can be considered as the total opposing head of the pumping system that must be overcome for the fluid to be pumped from the lower to the upper reservoir.The Eq. (37.4) is the equation for system characteristic, and while plotted on *H-Q*plane (Figure 37.3), represents the system characteristic curve. The point of intersection between the system characteristic and the pump characteristic on *H-Q*plane is the operating point which may or may not lie at the design point that corresponds to maximum efficiency of the pump. The closeness of the operating and design points depends on how good an estimate of the expected system losses has been made. It should be noted that if there is no rise in static head of the liquid (for example pumping in a horizontal pipeline between two reservoirs at the same elevation),  is zero and the system curve passes through the origin. |

Q2.

The principle of similarity is a consequence of nature for any physical phenomenon. By making use of this principle, it becomes possible to predict the performance of one machine from the results of tests on a geometrically similar machine, and also to predict the performance of the same machine under conditions different from the test conditions. For fluid machine, geometrical similarity must apply to all significant parts of the system viz., the rotor, the entrance and discharge passages and so on. Machines which are geometrically similar form a homologous series. Therefore, the member of such a series, having a common shape are simply enlargements or reductions of each other. If two machines are kinematically similar, the velocity vector diagrams at inlet and outlet of the rotor of one machine must be similar to those of the other. Geometrical similarity of the inlet and outlet velocity diagrams is, therefore, a necessary condition for dynamic similarity.

Let us now apply dimensional analysis to determine the dimensionless parameters, i.e., the π terms as the criteria of similarity for flows through fluid machines. For a machine of a given shape, and handling compressible fluid, the relevant variables are given in Table 3.1

Table 3.1 Variable Physical Parameters of Fluid Machine

|  |  |
| --- | --- |
| Variable physical parameters | Dimensional formula |
|   |  |
| *D*= any physical dimension of the machine as a measure of the machine's size, usually the rotor diameter | L |
| *Q*= volume flow rate through the machine | L3 T -1 |
| *N*= rotational speed (rev/min.) | T -1 |
| *H*= difference in head (energy per unit weight) across the machine. This may be either gained or given by the fluid depending upon whether the machine is a pump or a turbine respectively | L |
| =density of fluid | ML-3 |
|  = viscosity of fluid | ML-1 T -1 |
| *E*= coefficient of elasticity of fluid | ML-1 T-2 |
| *g*= acceleration due to gravity | LT -2 |
| *P* = power transferred between fluid and rotor (the difference between *P* and *H* is taken care of by the hydraulic efficiency  | ML2 T-3 |

In almost all fluid machines flow with a free surface does not occur, and the effect of gravitational force is negligible. Therefore, it is more logical to consider the energy per unit mass *gH*as the variable rather than *H*alone so that acceleration due to gravity does not appear as a separate variable. Therefore, the number of separate variables becomes eight: *D, Q, N, gH, ρ, µ, E*and *P*. Since the number of fundamental dimensions required to express these variable are three, the number of independent π terms (dimensionless terms), becomes five. Using Buckingham's π theorem with *D, N*and ρ as the repeating variables, the expression for the terms are obtained as,

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We shall now discuss the physical significance and usual terminologies of the different π terms. All lengths of the machine are proportional to *D*, and all areas to D2. Therefore, the average flow velocity at any section in the machine is proportional to . Again, the peripheral velocity of the rotor is proportional to the product *ND*. The first π  term can be expressed as

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Or

The Specific Speed of a Centrifugal Pump is the speed in r.p.m. at which a similar model of the Pump would need to run when of such a size as to deliver unit quantity against unit head.

Each type of pump (Radial Flow, Mixed Flow, Axial Flow, etc.) has it's own characteristic value of . 'Similar' includes both dynamic and geometric similarity, i.e., similar velocity triangles, with all relevant velocities proportional to each other.

The **speed** of an object is the magnitude of its velocity (the rate of change of its position).

A **pump** is a device used to move fluids, such as liquids, gases or slurries.

but

and

Since

where  Constant.

But  and  are by definition unity.

Therefore the Specific Speed:

### Notes

a)  is based on the values of , , and  at the design point, i.e. at maximum efficiency.

b)  is not dimensionless and will have different values in the different measuring systems (in the foot/slug/second system  is in r.p.m.,  is in feet and  is in gallons/second).

The Dimensions of  are:

Thus  could be made dimensionless by dividing by  and it would still be a Constant.

For example, **Addison's** Shape Number is:

c) Comparison of  and  (Imperial Units)

Thus for a particular machine:

d) Specific Speeds for Differing pump types (Imperial Units) Centrifugal Pumps Type Specific Speed

* Radial Flow 800 - 2000
* Mixed Flow 2000 - 4000
* Axial Flow 4000 - 8000 and Screw Pump
* Propeller Pump 8000 - 16,000 (High  and Low )

Q3.

 The peripheral speed at impeller outlet

(given)

Work input per unit weight of

Water =

=72.78m

Under ideal condition (without loss), the total head developed by the pump = 72.78 m

Absolute velocity of water at the outlet

=23.28 m/s

At the whirlpool chamber,

The velocity of water at delivery = 0.5 ´ 23.28m/s

Therefore the pressure head at impeller outlet

=72.78 -

= 65.87m

Hence, we theoretical maximum lift = 6

Or

It is defined as the head against which centrifugal pump has to work. Basically it is summation of suction head+delivery head+frictional head loss in suction pipe +frictional head loss in delivery pipe+velocity head at outlet.

Hm=hs+hd+hfs+hfd+Vd^2/2g

The margin of pressure over vapor pressure, at the pump suction nozzle, is Net Positive Suction Head (NPSH). NPSH is the difference between suction pressure (stagnation) and vapor pressure. In equation form:

NPSH = Ps ‑ Pvap

Where:

NPSH = NPSH available from the system, at the pump inlet, with the pump running

Ps = Stagnation suction pressure, at the pump inlet, with the pump running

Q4

The characteristic curves for a Centrifugal Pump are plotted from a constant speed or on a capacity basis. There is no means of altering the Guide Vane angles as in a Turbine and the only control is the Delivery Valve. If only one motor speed is possible then there will be only one performance curve.

### Iso-efficiency Curves

If the driving speed of the motor can be vared, then tests of the performance at several speeds can be carried out and a chart drawn to show performance at all possible operating conditions.

OR

There are two common types of hydraulic cylinders- single acting and double acting.

But what is the difference between the two?

Simply put, a double acting cylinder has both an A and a B Port.

This means oil enters the cylinder via the A port, which pushes the piston down. When the control calls for the piston to retract, oil is diverted to the B port, which then pushes the piston up. A double acting cylinder uses hydraulic power to both extend and retract. You can tell if a cylinder is double acting by looking at the number of ports. See the image below as an example.

In contrast, a single acting cylinder has an A port for running the piston down by pressure from the pump but is retracted with an internal spring.

See the image below as an example.

A double acting cylinder has pressing and pulling power and is generally controlled with a joy stick control. A single acting cylinder only has pressing power.