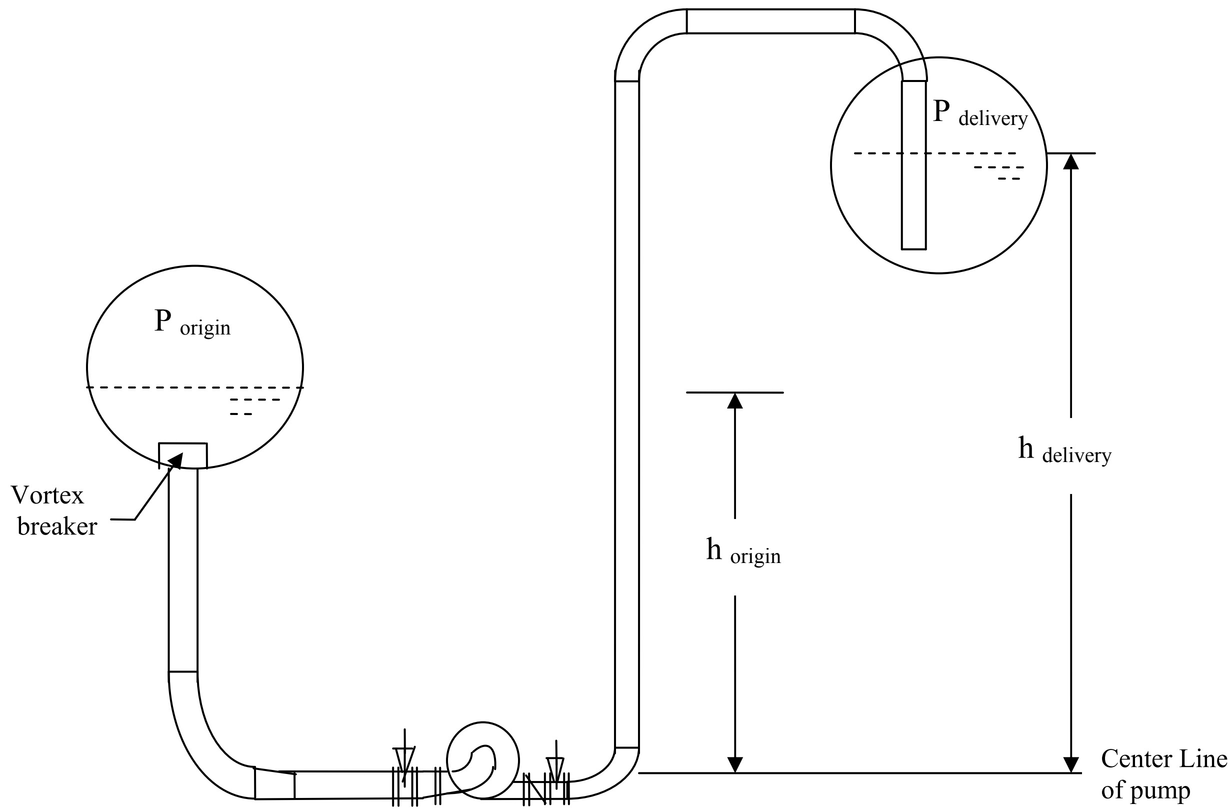
System Hydraulic Design of Liquid or Water Pumping Circuit

We will address Newtonian liquids in this course. Non-Newtonian liquids, such as slurries or suspensions, may produce widely different results in pipe flow frictional pressure drop and pump performance depending on their fluid characteristics.



If a natural flow should occur from an origin (or a flow source) at a pressure P_{origin} to a delivery (or a flow sink) at a pressure $P_{delivery}$ and no additional energy such as pumping energy is required, then the combination of (a) the difference in pressure from origin to delivery, ΔP and (b) the difference in potential energy or the static energy, ΔP_s, must overcome the fluid's frictional energy loss, ΔP_f, to piping, flow elements and equipment along the flow path. In other words,

$$\Delta P + \Delta P_s \geq \Delta P_f \qquad \text{(for natural flow without pumping)} \qquad (1)$$

where,

$$\Delta P \quad = \text{system pressure drop} = P_{origin} - P_{delivery}\text{, psi} \qquad (2)$$

P_{origin} = pressure at the flow origin point, psig

$P_{delivery}$ = pressure at the delivery point, psig

ΔP_s = static pressure change due to change in elevation, psi

$$\Delta P_{s\ psi} = \left[\frac{(\rho_{lb/CF})(h_{ft})}{144}\right]_{origin} - \left[\frac{(\rho_{lb/CF})(h_{ft})}{144}\right]_{delivery}$$
$$= \left[\frac{(G)(h_{ft})}{2.31}\right]_{origin} - \left[\frac{(G)(h_{ft})}{2.31}\right]_{delivery} \qquad (3)$$

ΔP_f = frictional pressure drop for suction and discharge piping, psi.

$$\Delta P_{f\ psi} = \left(f\frac{L_{ft}}{D_{ft}} + \sum_i K_i\right)\frac{(\rho_{lb/CF})(\upsilon_{ft/sec})^2}{(2)(g_c)(144)} \qquad (4)$$

ρ = flowing density, pounds per cubic foot, lb/CF;

G = flowing specific gravity relative to water @60°F = $\rho_{lb/CF}$ /62.37

h = elevation, ft

g_c = gravitational constant, 32.18 ft/sec^2

υ = pipe velocity, fps

f = Darcy friction factor

D = inside diameter of the pipe, ft

L = Pipe straight length, ft

K = flow resistance coefficient for flow elements or equipment

On the other hand, if the calculated ΔP_f at a target flow rate is greater than $\Delta P + \Delta P_s$, then flow will not occur:

$$\Delta P + \Delta P_s < \Delta P_f \qquad \text{(fluid will not flow)} \qquad (5)$$

In order to move the fluid, an additional energy, such as a pumping energy, would be needed to move fluid from the origin to the delivery point. If a pump is used to supply the needed energy, then a pumping energy term can be added to Eq. 5 reflecting an overall mechanical energy balance:

$$\Delta P + \Delta P_s + \Delta P_p = \Delta P_f \qquad \text{(for pumped flow)} \qquad (6)$$

where

ΔP_p = pressure increase due to pump, psi

System Head

The **System Head** is defined as:

$$\text{System Head} = -\Delta P - \Delta P_s + \Delta P_f$$
$$= P_{delivery} - P_{origin} + [\frac{(\rho)(h)}{144}]_{delivery} - [\frac{(\rho)(h)}{144}]_{origin} + \Delta P_f \quad (7)$$

We will use the term System Head to describe the requirement of **Pump Head** in order to deliver a flow at a target flow rate of q_{gpm} from an origin a delivery point.

Furthermore, the System Head involves the term of frictional pressure drop ΔP_f (for lines, valves and fittings, equipment) which is a function of flow rate and approximately proportional to the square of the flow rate. Thus, the System Head can be further re-written, as an approximation:

$$\text{System Head} = K_1 + (K_2)(q_{gpm})^2 \quad (8)$$

where

$$K_1 = P_{delivery} - P_{origin} + [\frac{(\rho)(h)}{144}]_{delivery} - [\frac{(\rho)(h)}{144}]_{origin}$$

= a constant and not a function of flow rate representing the Pressure Head, pressure difference between the origin and delivery points, $P_{delivery} - P_{origin}$, and the Static Head $[\frac{(\rho)(h)}{144}]_{delivery} - [\frac{(\rho)(h)}{144}]_{origin}$.

$K_2 =$ proportionality constant to calculate the ΔP_f if $(q_{gpm})^2$ is known.

System Curve

The **system curve** is a plot of System Head vs. system flow rate. Fig. 1 depicts a sample overall system curve showing a system head vs. capacity. The system head of Eq. (7) is graphically represented by the following three components:

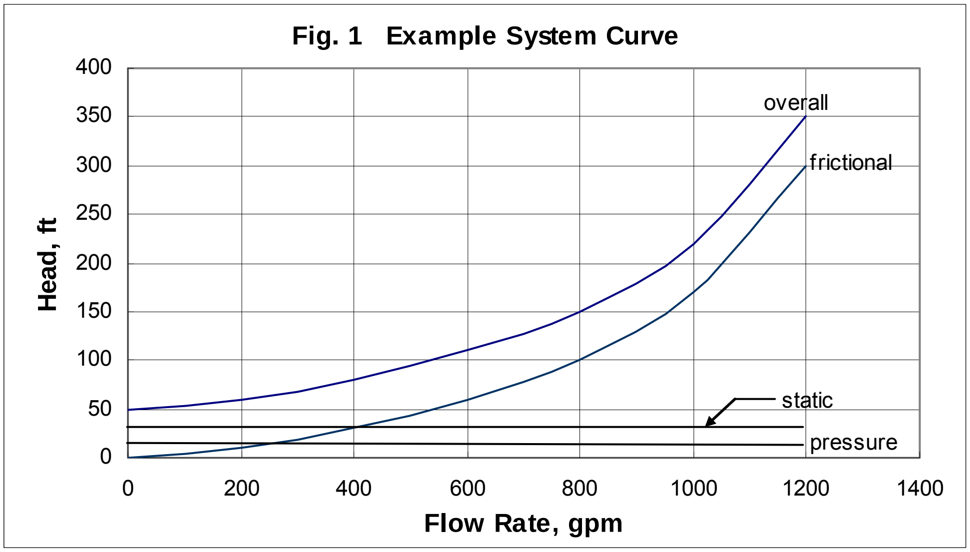
$$\text{Pressure Head} = (P_{delivery} - P_{origin})_{psi} = \left.\frac{(2.31)(P_{delivery} - P_{origin})_{psi}}{G}\right|_{ft} \quad (9)$$

$$\text{Static Head} = \left.\left([\frac{(\rho)(h)}{144}]_{delivery} - [\frac{(\rho)(h)}{144}]_{origin}\right)\right|_{psi} = (h_{delivery} - h_{origin})\Big|_{ft} \quad (10)$$

$$\text{System Frictional Pressure Drop} = \Delta P_f\big|_{psi} = \left.\left(\frac{(2.31)(\Delta P_f|_{psi})}{G}\right)\right|_{ft} \quad (11)$$

Unit of head can be in either the pressure (e.g., psi) or height of liquid column (e.g., feet) with latter being typically used in pump industry.

If there is a control valve installed in the pump discharge piping, the pressure drop is added to the system pressure drop. The resulting system curve can be caused to shift up or down due to throttling or opening the control valve.



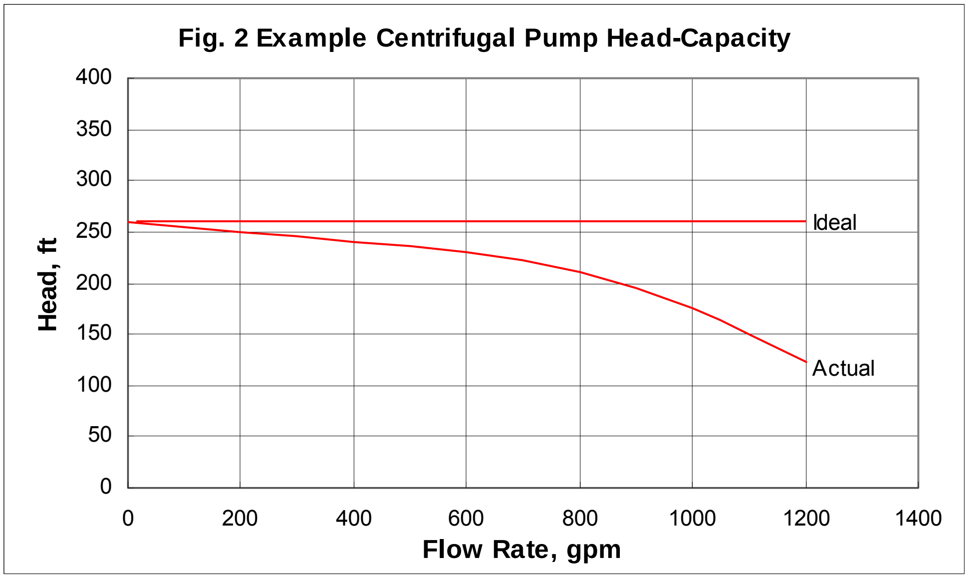
Pump Curve

A **Pump Curve** is a plot of **Pump Head** (increase of fluid pressure) vs. **Capacity** (the pumping flow rate). There are two major types of pump that can be used to increase the fluid pressure: **centrifugal** and **positive displacement**.

A centrifugal pump uses spinning impellers to accelerate the fluid to a high velocity (kinetic energy), then slows down the velocity in the diffuser (or volute) converting it to head (potential energy). On the other hand, the positive displacement (PD) pump admits fluid to be pumped to a suction volume, compresses the trapped fluid to a higher pressure using a piston, plunger, or diaphragm, and then discharges it out of the suction volume to the discharge pipe where the pressure of the compressed fluid equals the pressure at the discharge pipe. A PD pump does not need to use fluid velocity to achieve pressure.

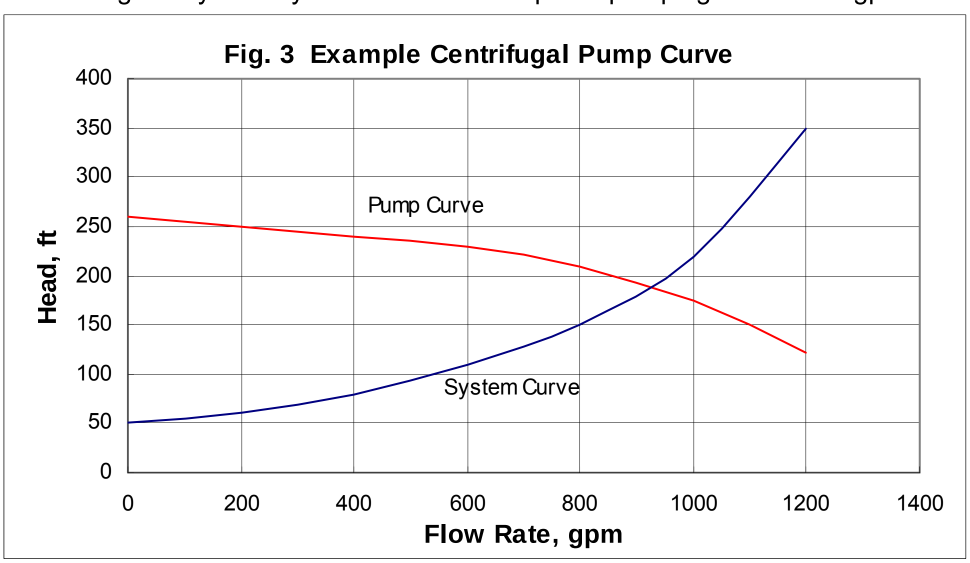
- **Centrifugal Pumps**

  If a centrifugal pump is used to supply the energy to fluid and to move the fluid at a target flow rate q_{gpm}, the pump shall impart energy to fluid and deliver a **Total Developed Head** or **Total Differential Head (TDH)** or **Pump Head** so that it is able to balance out the system head corresponding to the q_{gpm}. Ideally, the TDH would be constant for a centrifugal pump in the range of capacity that the pump is designed for. However, due to factors involved in pump internal design, the centrifugal pump is not able to deliver a constant head within the range of flow rates that it was design for. Fig.2 illustrates a sample centrifugal pump's ideal and actual **head-capacity curves.**



Centrifugal pump generates a head (in height of fluid column) which is independent of fluid density. When the density of the pumping fluid is changed, the pump head is constant at the constant pumping rate, yet the **differential pressure** is changed according to the fluid density.

Fig. 3, as an example, shows a centrifugal pump head-capacity curve superimposed on the system curve shown in Fig 1. The centrifugal pump will operate at the intersection of pump curve and the system curve. Fig. 3 indicates that this pump is capable of overcoming the system hydraulic resistance up to a pumping rate of 920 gpm.



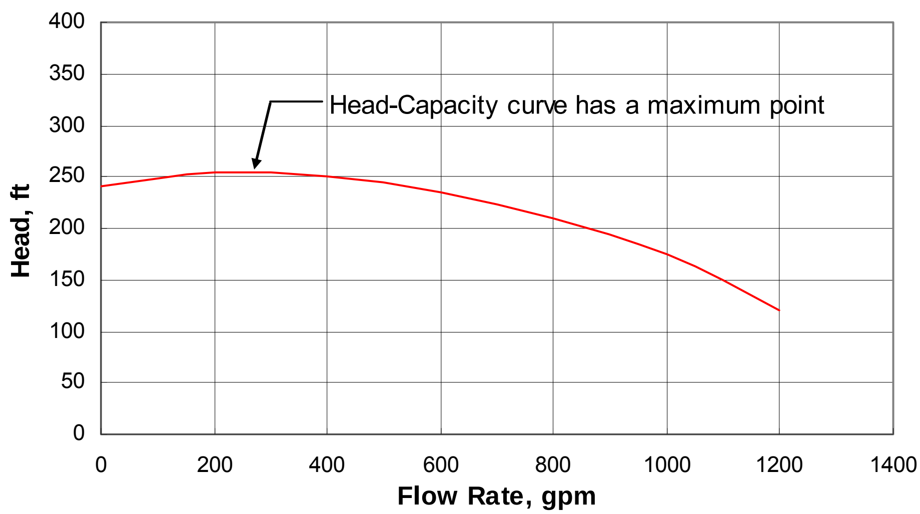
Centrifugal pump will operate at only one operating point and this operating point is the intersection of the system curve and the pump curve. Typically, a control valve is installed at pump discharge piping to be used as a device to effect a change of the pump curve by changing the system pressure drop ΔP_f during a normal operation. The control valve can throttle (to increase the system head) or open more (to reduce the system head) to help maintain a stable overall system head in case a perturbation of pressure head or static head occurs. For example, a level control valve is installed in the pumping circuit to maintain the liquid level of the pump suction vessel. In this situation, the movement of the control valve, causing the system curve to move higher or lower, will result in shifting the operating point to a decreased or increased flow rate, achieving the flow control.

Drooping

In certain low-specific-speed centrifugal pump designs, when operating in a region of the pump curve, the pump becomes unstable, i.e., head, capacity, and power fluctuate. This phenomenon is called **Drooping** and is indicated in its head-capacity curve where there is a maximum point on the curve, i. e., a common head at two different capacities (Fig. 4). Thus, when operating near the maximum point, a change in the pumping system resistance, such as the suction or discharge vessel pressure, the pump doesn't know which capacity to operate.

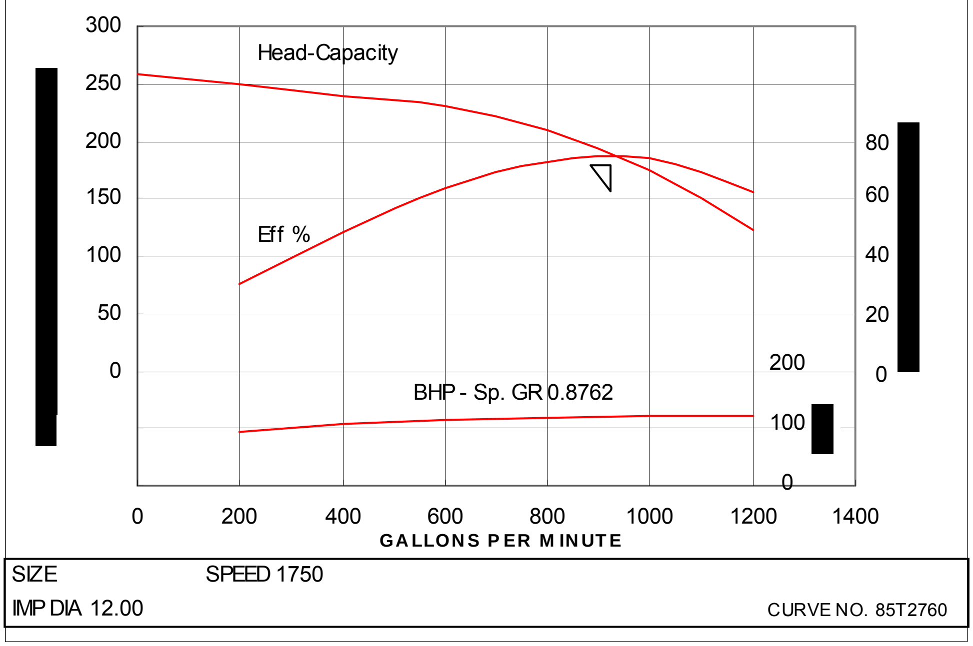
Do not install drooping-curve pumps in parallel as one pump may increase in pumping rate in drooping and throw the other pump into operating below minimum flow.

Fig. 4 Example Centrifugal Pump Drooping Curve



In addition to the head-capacity relationship for the hydraulics, each set of centrifugal pump performance curve typically also shows pump mechanical efficiency, the required net positive suction head, and the brake horsepower vs. capacity. Accompany parameters for the pump are impeller size and speed.

Fig. 5 Example Centrifugal Pump Curve



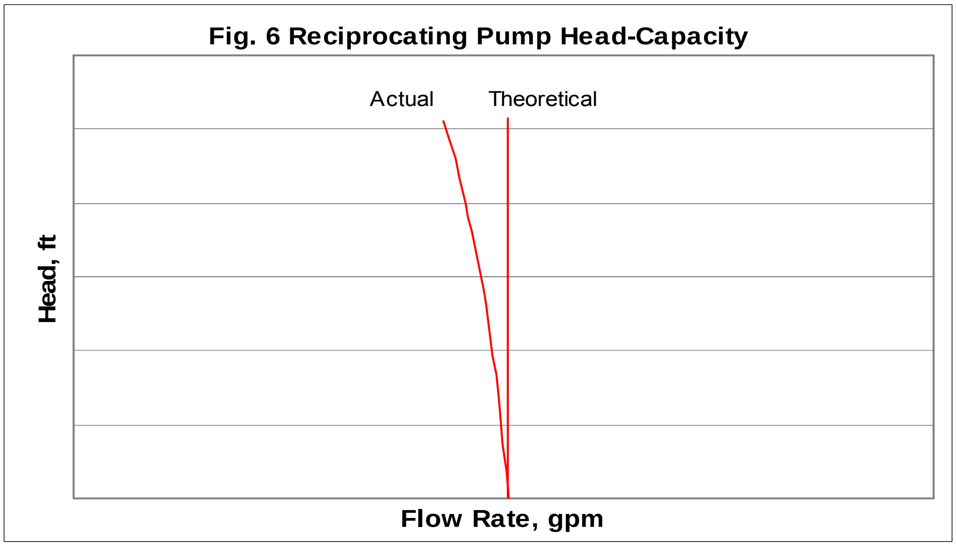
Each pump curve is unique to a pump and every centrifugal pump will operate on its own characteristic pump curve.

- **Positive Displacement (PD) Pumps**
  A PD pump will deliver the pumped fluid at a constant flow rate given the suction volume and the speed of the pump. Thus, a PD pump can be considered as a constant volume machine given the suction volume and the speed of the pump.

  A PD pump will also always deliver the pumped fluid against a discharge pressure because it will force the pumped fluid into the discharge system and is theoretically unaffected by the pressure provided it is within the capability of the pump driver and the pump mechanical design limits (design pressure of casing, maximum piston loading, etc.). The actual pumped volume is less due to slip or inefficiency. Fig.6 illustrates a typical PD pump **head-capacity curve** on both the ideal and actual situations.

  There are variations in offering the PD technology: reciprocating, plunger, diaphragm, screw, rotary, metering, etc.

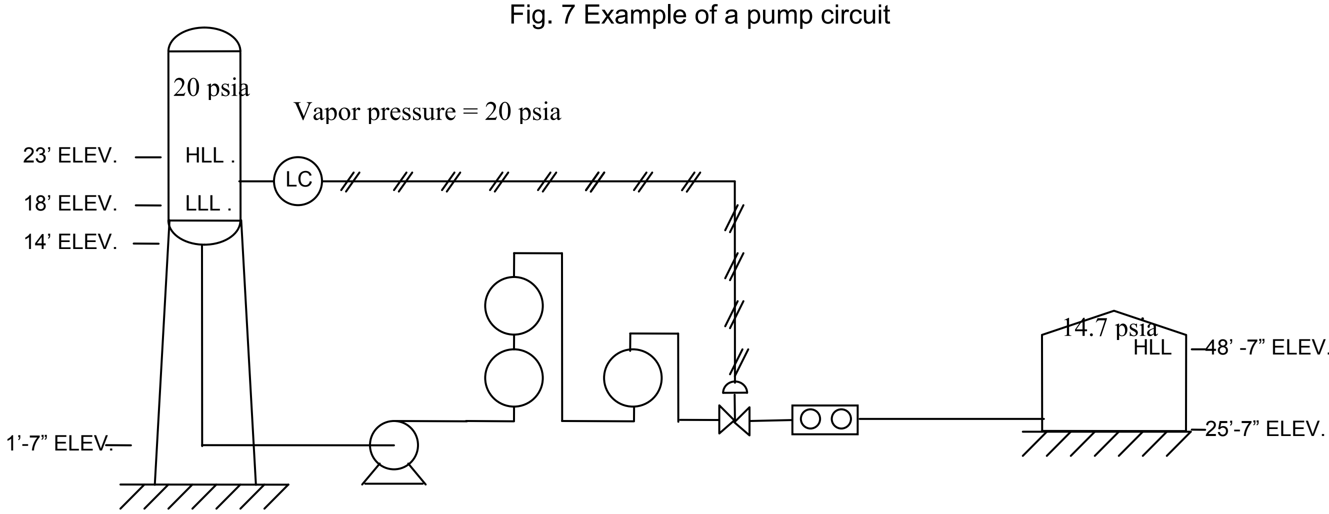
  Typical differential pressure/capacity range: 500 to 6000 psi/10-600 gpm for commercial reciprocating pumps for in, gas, chemical, or pipeline applications. Other PD pumps, such as diaphragm or metering, have lower heads and capacities.



Hydraulics of the Pumping Circuit

Fig. 7 illustrates an example pumping circuit where a pump is used to send a boiling liquid from a column, through three heat exchangers, one level control valve, and one air cooler to a storage tank. Hydraulic parameters and their values, either given or by calculation, involved in defining the pump system are listed below.

Fig. 7 Example of a pump circuit



	Sample Input or calc.
• **Origin Pressure**	
Fluid to be pumped typically flows from a reservoir (e. g., a tank, a vessel, etc.) through a pipe segment before it flows to the pump. This pipe segment is referred to as the suction piping. The origin pressure can be taken as the pressure at the entrance to the suction piping. However, the pressure at the entrance of the suction piping normally is not measured. Instead, the pressure at the top of the reservoir and the elevation of the liquid level of the reservoir are normally known. Therefore, for convenience, the reservoir top pressure can be taken as the origin pressure. Adding the suction static head and subtracting the frictional pressure drop through the suction piping, the pressure at the pump suction can be calculated.	20 psia

• **Delivery Pressure**	
Pumped fluid flows to a destination (e. g., another reservoir such as a tank or vessel, a point in a pipework, etc.) of given pressure and elevation. If the pumped fluid exits the discharge piping in a vapor space above the liquid space in the reservoir, then the delivery pressure can be the pressure in that vapor space. If the pumped fluid exits the discharge piping in the liquid space of the reservoir, then the delivery point can be taken as the liquid level. In this case, the delivery pressure is the pressure in the vapor space and the delivery elevation is the elevation of the liquid level. ***Design Tip*** *ELEVATION OF LIQUID LEVEL AT DELIVERY FOR PUMP HEAD:* *It is prudent to consider the highest operating liquid level for hydraulic calculation for pump sizing.*	14.7 psia

• **Static Head**	
Static head is a measurement of pressure in terms of the weight of column of fluid to be expressed in feet. Note that the static head is relative to the pump centerline for either the suction or the discharge piping so that the static heads typically can be taken as numerically positive quantities.	

	Sample Input or calc.
• **Suction Static Head**	
Static head of the suction piping from the elevation of origin to that of pump centerline. Example shown is taking the bottom of vessel's bottom head for a conservative NPSH consideration.	(14'-0")-(1'-7")=12.4 ft

• **Line Loss**	
Line loss refers to the frictional pressure drop at the target flow rate for pipes plus fittings and valves in either the suction piping or the discharge piping.	

• **Suction Line Loss**	
Line loss for the suction piping.	0.1 psi

• **Suction Pressure**	
Suction pressure is the pressure at the entrance of the pump. Refer to the pressure balance table (Table 1) for calculation.	20+12.4/2.31*0.928-0.1=24.9 psia =10.2 psig

• **Suction Head, Total Suction Head (TSH)**	
The term Suction Head or Total Suction Head (TSH) expresses the pump suction pressure in terms of height of liquid column. A practical use of TSH is to subtract the vapor pressure of the pumping fluid from it to arrive at the Net Positive Suction Head available (NPSHA) at the pump impeller eye. See NPSH discussions later. $TSH_{ft} = \frac{(144)(P_{origin,psia})}{\rho_{lb/CF}@T,P} + \text{SuctionStaticHead} - \frac{(144)(\Delta P_{f,\ suctionpiping,psi})}{\rho_{lb/CF}@T,P}$ (12)	144X20/57.88 +12.4 -144X0.1/57.88 = 61.9 ft

• **Suction Static Lift**	
Certain pump, such as diaphragm pumps, are capable of drawing liquid from an liquid with a level at an elevation that is lower than that for the centerline of the pump. Instead of having a suction static head for the pump, these pumps are designed to operate within a specified Suction Static Lift. Suction static lift takes a positive value. $\text{Suction Static Lift}_{ft} = \text{Elevation}_{\text{pump centerline},\ ft} - \text{Elevation}_{\text{liquid level},\ ft}$ (13)	

	Sample Input or calc.
• **Suction Lift or Total Suction Lift (TSL)**	
$$TSL_{ft} = \frac{(144)(P_{origin,psia})}{\rho_{lb/CF}} - SuctionStaticLift - \frac{(144)(\Delta P_{f,\ suctionpiping,psi})}{\rho_{lb/CF}} \quad (14)$$	

• **Equipment Pressure Drop**	
Equipment pressure drop refers to the pressure drop associated with equipment that present in the piping. Typical equipment items include flow meter, heat exchangers, air coolers, control valves, fired heaters, and others (such as filters, strainers, flow straightener, etc.).	Control Valve =10 psi Heat Exch =15 psi

• **Margin**	
Margin in pressure drop can be allowed for pump head calculation for a new design to account for design uncertainties until they are firmed up. Design uncertainties may include actual line routing, actual equipment pressure drop, or allowance for future capacity expansion or line fouling etc. To calculate the pump head for an existing pumping system, where lines have been laid and equipment has been installed and tested, actual pressure surveys should be examined and judicially incorporated into the calculation and not to further allow margin. Otherwise, a pump, through barely adequate for the current project, can be deemed inadequate in head and requires modification or replacement due to inclusion of margin.	7 psi Selected to use for this example

• **Discharge Static Head**	
Static head of the pump discharge piping from the elevation of pump centerline to that of delivery. If delivery is not at the highest elevation, take the highest elevation on the pump discharge to ensure that there is sufficient discharge pressure to overcome the highest static head loss.	47 ft

• **Discharge Line Loss**	
Line loss for the discharge piping.	19 psi

• **Discharge Pressure**	
Discharge pressure is the pressure at the pump discharge nozzle. Refer to the pressure balance table (Table 1) for calculation.	14.7+18.883+19+10+ 15+0+0+7=84.6 psia =69.9 psig

	Sample Input or calc.
• **Discharge Head**	
Discharge Head is the discharge pressure expressed in head.	2.31x84.6/0.928 =211 ft

• **Differential Pressure**	
Pump differential pressure is the pressure difference between the inlet and outlet nozzles. $\text{Differential Pressure}_{psi} = \text{Total Dynamic Head}_{psi} = \text{P@Discharge}_{psig} - \text{P@Suction}_{psig}$ $\text{Differential Pressure}_{psi} = \frac{(\rho_{lb/CF} @ \text{T, P})(\text{Pump Head}_{ft})}{(144)} = \frac{(\text{G})(\text{Pump Head}_{ft})}{(2.31)}$ (15) where G = specific gravity of fluid at T,P relative that of water at 60°F.	84.6-24.9= 59.7 psi

• **Pump Head / Total Head / Differential Head / Total Differential / Head / Total Developed Head / Total Dynamic Head / TDH**	
Pump head or total head is the pump differential pressure expressed in head unit, as a measure of pressure rise due to pump. $\text{Differential Head}_{feet} = \text{Total Dynamic Head}_{feet} = \frac{(144)(\text{Differential Pressure}_{psi})}{(\rho_{lb/CF} @ \text{T, P})} = \frac{(2.31)(\text{Differential Pressure}_{psi})}{(\text{G})}$ (16)	2.31x59.7/0.928 =148.6 ft

• **Shutoff Head (or Dead Head)**	
Shut-off head is the centrifugal pump head at the zero pumping rate. This is the highest head on the head-capacity curve.	260 feet in Fig. 3

Pressure Balance

Table 1 is a pressure balance table summarizing the corresponding heads and pressures.

Table 1 - Pressure Balance

	FT	PSI	PSIA
ORIGIN PRESSURE			**20**
+ STATIC HEAD (PSI = FT*S.G.*.433)	**12.4**	4.982	
- LOSS (LINE +OTHER)		**0.1**	
PUMP SUCTION PRESSURE			24.9
DELIVERY PRESSURE			**14.7**
+ STATIC HEAD (PSI = FT*S.G.*.433)	**47**	18.883	
+ LINE LOSS		**19**	
+ ΔP CONTROL VALVES		**10**	
+ ΔP EXCHANGERS		**15**	
+ ΔP FURNACES		**0**	
+ ΔP ORIFICES		**0**	
+ ΔP OTHERS *		**7**	
PUMP DISCHARGE PRESSURE			84.6
DIFFERENTIAL PRESSURE			
DISCHARGE PRESSURE			84.583
- SUCTION PRESSURE			24.882
TOTAL PUMP DIFF. PRESS.		59.701	
PUMP HEAD (PSI*2.31/S.G.)	148.63		

Pressures for the suction piping tabulation starts out from the origin pressure to the pump suction pressure. Pressures for the discharge piping tabulation starts out from the delivery pressure and work reversely to the pump discharge pressure. The differential pressure or the pump head is finally calculated.

For the suction piping, the static head is the elevation of origin minus the elevation at pump center line. For the discharge piping, its static head is the elevation of delivery minus the elevation at the pump centerline.

The elevation of origin is shown to be taken from the bottom of the bottom head so that it will be more conservative for pump head and NPSHA calculation. The elevation of the delivery is taken as the high liquid level for the tank so that the calculated pump would be able to deliver the flow up to that liquid level in the tank.

Pumps Process Specification Sheet

Accurate process data is specified in a **pump data sheet** to define the fluid and the conditions under which a pump is to meet its hydraulic requirement. The vendor supplied performance expected of the pump and driver are also documented in the **pump data sheet** for your evaluation. It is necessary to consider such normal operating conditions as startup, shutdown, pumpout, winter nights, hot summer days, emergency situations, long term changes in fluid density and viscosity, etc. before specifying a pump.

	Illustrated with sample values
• **Normal Operating Temperature**	
Flowing temperature of the pumping liquid.	260°F

• **Sp. Gr. @ P, T**	
Pumping liquid's specific gravity at the flowing pressure and temperature relative to that of water at 60°F (62.3688 lb/CF). This parameter helps convert the pump head to pump differential pressure, or vice versa.	G = 57.77/62.3688 = 0.928

• **Viscosity @ P, T**	
Pumping liquid's viscosity at the flowing pressure and temperature. Viscosity could reduce a centrifugal pump's capacity, head, and efficiency, generally more impact in this order, due to increased internal friction. Positive displacement pumps (e. g., screw or gear pumps) are more efficient for viscous fluids. Consider using PD pumps when viscosity exceeds 50 centistokes. Consults the original pump manufacturer for the viscosity correction curves for high viscosity application if the original pump curves are developed based on water.	0.23 cP

• **Vapor Pressure @ P, T**	
Pumping liquid's vapor pressure at the flowing pressure and temperature. This figure normally is obtained from charts or tables for a single pure chemical component, such as propane, water, etc. or from a process simulation program for fluid mixtures of multiple chemical components. The pressure of a **boiling liquid**, such as those from a vapor-liquid separator, distillation column's a reboiler, a distillation column side or bottom draw, etc., is its vapor pressure because the liquid is at vapor-liquid equilibrium with the vapor exiting the vapor-liquid separator, traveling upward to the tray above the reboiler, traveling upward to the drawing tray, respectively.	20 psia

	Illustrated with sample values
• **NPSH Available**	
One unique characteristic of centrifugal pumps is the Net Positive Suction Head. The **net positive suction head available** (**NPSHA**) for centrifugal pump in a given service is the available pressure at the lowest-pressure point in the pump (i.e., TSH, which take place at the eye of the impeller where fluid velocity is highest inside the pump) **above** the vapor pressure of the pumping fluid. NPSHA indicates how much energy that is available throughout its travel inside the pump to guarantee the pumping fluid to remain in the liquid form. If the NPSHA falls below NPSHR, then the liquid vaporization (boiling or flashing) occurs at the lowest-pressure point with subsequent collapse of the bubbles as pressure rises. Cavitation is the collapse of vapor-filled cavities (bubbles) in a liquid on the surface of metal. Cavitation may cause pump vibration and mechanical damage (metal erosion or pitting). User will specify NPSHA. The NPSHA should be provided sufficiently high to be equal to or exceed the anticipated **net positive suction head required** (**NPSHR**) for the specific pump (i.e. characteristics of pump) being considered throughout its expected capacity range. Manufacture provides NPSHR. The NPSHA can be expressed either in unit of psi or feet: $NPSHA = Total\ suction\ head - Vapor\ pressure$ $NPSHA_{psi} = (P@Origin_{psia} + StaticHead_{psi} - \Delta P_{f\ psi}) - VaporPressure_{psia}$ $NPSHA_{ft} = (P@Origin_{psia} - VaporPressure_{psia} - \Delta P_{f\ psi})(\frac{144}{\rho_{lb/CF@T,P}}) + StaticHead_{ft}$ (17) $NPSHA_{ft} = (P@Origin_{psia} - VaporPressure_{psia} - \Delta P_{f\ psi})(\frac{2.31}{G}) + StaticHead_{ft}$	NPSHA=(20-20-0.1) X2.31/0.928 +12.4 =12.2 ft or NPSHA= 61.9-2.31X20/0.928 =12.2 ft before subtracting margin

Design or Operating Tip

> *NPSHA for BOILING LIQUIDS:*
> *For services in which the pumping liquid's origin pressure is the same as its vapor pressure, the increase in the origin pressure will not increase the NPSHA as their effects will be cancelled. These systems are commonly encountered in the bottom product pump circuit in a distillation column. Liquid leaving the column bottom is normally in equilibrium with the vapor to the stage immediately above; and the column bottom pressure is the vapor pressure of the column bottom product.*
> *In this situation, the option is either to increase the static head (from the minimum operating liquid level to the elevation of impeller's eye) or to find another pump to meet the NPSHA*

Tabulated below is a sample NPSH calculation:

Fig. 5 NPSHA Calculation

NET POSITIVE SUCTION HEAD		
	FT	PSI
STATIC HEAD	12.4	4.982
+ [(ORIG. PR.)-(VAP PR.)]*2.31/S.G. IN FT	0.0	0
- LINE LOSS (FT=PSI*2.31/S.G.)	0.2	0.1
- ALLOWANCE (for a new design)	**3.0**	1.2053
AVAILABLE NPSH (NPSHR)	9.2	3.7
PUMP REQU'D. NPSH (WATER)		

	Illustrated with sample values
• **Normal Capacity**	
The pumping rate in gpm for normal operation.	920 gpm

• **Rated Capacity**	
The pumping rate in gpm specified for purchase. The rated capacity is the point at which pump manufacturer will perform the hydraulic test. Normally, rated capacity has a safety margin of 5% (for charge pumps, product pumps), 10% (for reflux pumps or reboiler circulation pumps) to 20% (level-controlled and fire-heater charge pumps) higher than the normal capacity in order to meet the flow requirement in actual operation due to possible uncertainty in flow rates used in process design or to provide some flexibility in case of operational deviation from the process design parameters.	920x1.05 = 966 gpm for product pump

• **Rated Suction Pressure**	
The Rated Suction Pressure is the suction pressure calculated based on the Rated Capacity.	24.9 psia =10.2 psig

• **Rated Discharge Pressure**	
The Rated Discharge Pressure is the discharge pressure calculated based on the rated capacity.	90.1 psia = 75.4 psig

• **Minimum Suction Pressure**	
The minimum suction pressure that is possible (or available) in all possible modes of pump operation to be caused by minimum liquid level or and/or pressure at the suction vessel.	20 psia

	Illustrated with sample values

• **Maximum Suction Pressure**	
The maximum suction pressure that is possible (or available) in all possible modes of pump operation to be caused by maximum liquid level or and/or pressure at the suction vessel. This pressure is used by the centrifugal pump vendor to calculate the maximum discharge pressure and maximum power once they come up with a pump for recommendation. The calculated maximum discharge pressure shall be lower than the casing design pressure for safety. The calculated maximum power helps specifying driver horsepower.	40.0 psia =25.3 psig

• **Maximum Discharge Pressure (for PD pumps)**	
This pressure can be the maximum pressure to be allowed at the pump discharge so that the pressure at any point of the pump circuit does not exceed the equipment maximum allowable operating pressure. Installation of PSV at pump discharge at specified set pressure sets the Maximum Discharge Pressure to protect the pump discharge circuit from overpressure. This pressure is typically specified for PD pump.	85.3 psia =100.0 psig

• **Maximum Differential Pressure (for PD pumps)**	
Maximum Discharge Pressure minus Minimum Suction Pressure. This pressure normally is for the PD pump vendor to evaluate their mechanical design requirement.	(100+14.7)-20 = 94.7 psi

• **Differential Pressure**	
Use the differential pressure figure from the pressure balance calculation. The pressure drop margin used in the pressure balance calculation plus the control valve action (open or close more from its design specification) shall provide sufficient flexibility to meet the actual pumping circuit hydraulic demand.	59.7 psi

• **Differential Head or Total Developed Head**	
Use the differential head figure from the pressure balance calculation. Manufactures of centrifugal pump provide head-capacity curve to indicate the pump performance. Therefore, the head provide a convenient compression and selection of a pump.	148.8 ft

	Illustrated with sample values
• Hydraulics Horsepower	
Assuming an incompressible flow behavior for the liquid, the theoretical or hydraulic horsepower that is required to drive the pump is:	59.7x966/1714.28 =33.6 hp
$$\text{Hydraulic horsepower} = \frac{(\text{Differential Head}_{ft})(W_{lb/hr})}{(3600)(550)}$$ $$\text{Hydraulic horsepower} = \frac{(\text{Differential Pressure}_{psi})(\text{USgpm})}{(1714.28)} \quad (18)$$ $$\text{Hydraulic horsepower} = \frac{(G\ @T,P)(\text{Differential Head}_{ft})(\text{USgpm})}{(3960)}$$	

• Brake Horsepower	
The actual horsepower consumed (required) by the pump.	33.6/0.7 = 40 Bhp
$$\text{Brake Horsepower Required} = \text{Bhp} = \frac{(\text{Hydraulic Horsepower})}{(\text{Pump Efficiency}_{\%}/100)}$$ $$\text{Brake Horsepower Required} = \text{Bhp} = \frac{(\text{Differential Pressure}_{psi})(\text{USgpm})}{(1714.17)(\text{Pump Efficiency}_{\%}/100)} \quad (19)$$ $$\text{Brake Horsepower Required} = \text{Bhp} = \frac{(G\ @T,P)(\text{Differential Head}_{ft})(\text{USgpm})}{(3960)(\text{Pump Efficiency}_{\%}/100)}$$ The remaining portion of the actual horsepower that did not consumed by the pump (i.e., drive the fluid and increase its pressure) puts in heat to the pumping fluid, resulting a fluid temperature rise. The amount of heat imparted to the fluid is $$Q_{p\ Btu/hr} = (2545)(\text{Bhp})\left(\frac{100 - \text{Pump Efficiency}_{\%}}{100}\right) \quad (20)$$ $$= (2545)\frac{(\text{Differential Head}_{ft})(\text{USgpm})}{(3960)}\left(\frac{100 - \text{Pump Efficiency}_{\%}}{\text{Pump Efficiency}_{\%}}\right)$$	

• Pump Efficiency	
Pump mechanical efficiency is normally function of pump size: Large pump is generally more efficient than two small pumps in parallel. The efficiency figure is pump specific and is determined by pump vendor. The discharge specific speed figure, Ns, can help validate vendor's efficiency claims.	70%

	Illustrated with sample values
• **Minimum continous flow**	
Operating centrifugal pump at low flow rates can create suction hydraulic instability, causing erosion. A minimum flow rate per pump specification shall be maintained when operating a centrifugal pump. This figure is pump-specific and is determined by pump vendor.	gpm

• **Pump Speed**	
This is the rotating speed for a centrifugal pump or the frequency of strokes for a PD pump. The rotating speed relates to the shape of impeller (radial, mixed, or axial flow) used for the pump which, in turn, relates to the head and capacity via discharge specific speed, Ns.	1750 rpm

• **(Discharge) Specific Speed,** Ns	
The centrifugal pump specific speed or discharge specific speed (a dimensionless quantity) is defined by this formula: $$Ns = (s_{rpm}) \frac{q_{gpm}^{1/2}}{H_{ft}^{3/4}} \quad (22)$$ where s = impeller rotational speed, revolutions per minute, rpm q = pumping rate, US gallons per minute, gpm H = differential head, feet, ft Ns relates impeller geometry and helps define the shape of impeller: 4,000 or less - radial flow 10,000 or higher - axial flow Ns also relates to pump's efficiency. Efficiency typically peaks at Ns of 2,000 to 3,000. Thus, Ns can be used to guide pump speed selection to optimize pump life cycle cost. Radial flow pumps normally find applications in refinery or petrochemical processes. Avoid too low a Ns where drooping can occur. Impeller diameter can be trimmed or can be replaced up to the maximum size that the casing allows. Axial flow pumps are suitablemfor low head high flow applications, such as cooling water, evaporator, or crystallizer. Head rise is more steep at lower capacity than the radial counterpart, thus, operating at best efficiencity point or beyond is recommended. Axial flow implellers can not be trimmed.	(1750)(920^0.5) /(148.8^0.75) = 1,246

	Illustrated with sample values

• **Suction Specific Speed,** Nss	
The centrifugal pump suction specific speed (a dimensionless quantity) only deals with inlet and is defined by this formula: $$Nss = (s_{rpm})\frac{q_{gpm}^{1/2}}{NPSHR_{ft}^{3/4}} \quad (21)$$ The higher the Nss, the lower the stage efficiency and stronger tendency for internal circulation (suction recirculation), which leads to flow instability. Most users try to keep $Nss < 11{,}000$ [2].	(1750)(920^0.5) /(10^0.75) = 9,440

• **Driver**	
The horsepower required of the pump driver (a motor or a steam turbine) delivering the Bhp to the pump can be established by knowing the driver's mechanical efficiency: $$\text{Driver's Horsepower requirement} = \frac{\text{Pump Bhp}}{\left(\frac{\text{Driver Efficiency}_{\%}}{100}\right)} \quad (23)$$ or $$\text{Driver's kW requirement} = \frac{(0.7457)(\text{Pump Bhp})}{\left(\frac{\text{Driver Efficiency}_{\%}}{100}\right)} \quad (24)$$	

• **Electric Motor**	
If the driver is an electric motor, then the kVA requirement from a power feeder to run a motor driver is then: $$kVA = \frac{(0.7457)(\text{Pump Bhp})}{\left(\frac{\text{Driver Efficiency}_{\%}}{100}\right)\left(\frac{\text{Motor's Power Factor}_{\%}}{100}\right)} \quad (25)$$ and the ampere is $$\text{Ampere}_{\text{3-phase}} = \frac{(0.7457)(\text{Pump Bhp})}{(\sqrt{3})(\frac{V_{volt}}{1000})\left(\frac{\text{Driver Efficiency}_{\%}}{100}\right)\left(\frac{\text{Motor's Power Factor}_{\%}}{100}\right)} \quad (26)$$ The amperes drawn by the motor in a three-phase 480 Volts installation becomes: $$\text{Ampere}_{\text{3-phase,480V}} = \frac{(0.7457)(\text{Pump Bhp})}{(\sqrt{3})(0.48)\left(\frac{\text{Driver Efficiency}_{\%}}{100}\right)\left(\frac{\text{Motor's Power Factor}_{\%}}{100}\right)} \quad (27)$$	

	Illustrated with sample values

• **Steam Turbine**	
There are two types of steam turbine (ST) that can drive a pump: condensing (steam is fully condensed at the exhaust) or back pressure (steam is not condensed at the exhaust). The steam usage m can be calculated once the enthalpy change $h_{in} - h_{out}$ is established.	

$$\text{Work needed from ST} = \frac{(\text{Pump Bhp})}{\left(\frac{\text{ST Driver Efficiency \%}}{100}\right)}$$

$$= \left(m_{steam\ lb/hr}\right)\left(h_{in,Btu/lb} - h_{out,Btu/lb}\right)\left(\frac{hp}{2545.1\frac{Btu}{hr}}\right) \quad (28)$$

where,

m = steam usage

h = enthalpy of steam, which can be obtained from a stable table.

Centrifugal Pump Affinity Laws

Lack of actual pump performance data the Affinity Laws can be helpful in estimating the future performance by extrapolating from the known performance. Use these laws for speed and diameter change.

Change in → Known Quantity ↓	**Impeller Diameter**	**Speed**	**Diameter and Speed**
Q_1	$Q_2 = Q_1\left(\frac{d_2}{d_1}\right)$	$Q_2 = Q_1\left(\frac{n_2}{n_1}\right)$	$Q_2 = Q_1\left(\frac{d_1}{d_2}\right)\left(\frac{n_2}{n_1}\right)$
H_1	$H_2 = H_1\left(\frac{d_2}{d_1}\right)^2$	$H_2 = H_1\left(\frac{n_2}{n_1}\right)^2$	$H_2 = H_1\left[\left(\frac{d_1}{d_2}\right)\left(\frac{n_2}{n_1}\right)\right]^2$
Bhp_1	$Bhp_2 = Bhp_1\left(\frac{d_2}{d_1}\right)^3$	$Bhp_2 = Bhp_1\left(\frac{n_2}{n_1}\right)^3$	$Bhp_2 = Bhp_1\left[\left(\frac{d_1}{d_2}\right)\left(\frac{n_2}{n_1}\right)\right]^3$

Q = capacity, H = head, Bhp = brake horsepower; subscript 1= known, subscript 2 = future.

Selection of Pump Type for the Service

There are many type of pumps commercially available. It is best to match the pump to the application. General guidelines for selecting pumps between the major types of commercial pumps are highlighted below:

	Centrifugal	Reciprocating
Viscosity	Typically <100 cSt (500 SSU) Up to 8,000 SSU capable; but lose efficiency at high viscosity (200 to 500 SSU)	Probably more economical in power usage on high viscosity fluids; Rotary pumps are more appropriate.
Fluids with Entrained Gas	Can lose suction by gas bubbles accumulating at the eye of impeller;	
Emulsion	Bad choice because of pump's agitating action	Minimizes emulsion formation
Pulsating Flow	None-pulsating flow.	Likely; to be minimized by using pulsating dampers
Best Location to Install	Gravity feed from bottom of tank or vessel	
Head vs. Capacity		Low flow, high head
Efficiency vs. Capacity	To operate from 70% to 110% rated capacity or best efficiency point	Remains reasonably efficient within rated capacity.
Pump Discharge Specific Speed	Centrifugal – High head, low flow $1{,}500 < Ns < 3{,}000$ Axial - Low head, high flow[3] $5{,}000 < Ns < 20{,}000$	
Driver Speed	Normally 1,800 3,600	600 A gearbox can be used to speed up; but can easily cost more than pump
Reliability	High	Typically lower than centrigugal.
Costs	Widely available; Low initial cost	Maintenance cost usually exceeds centrifugals due to more moving parts.

Design Considerations

- **Centrifugal pump – Minimum Flow Bypass**
  The minimum flow bypass line shall return the pumped flow to the suction vessel or upstream of a heat removal device before the actual pump suction to avoid temperature built-up as discussed previously. There are several design options to provide the minimum flow bypass for the centrifugal pumps:

- **Centrifugal pump – Provide a flow control valve on the minimum flow bypass**
  A flow recycle line is installed from pump discharge to suction. This line is normally closed when pump is operating at the design flow rate. When the flow rate through pump is below the manufacturer specified minimum flow rate, the flow control valve on a bypass line shall open to allow certain pumped flow to recycle to the pump to maintain a minimum pumped flow rate.

- **Centrifugal pump – Provide a pressure control valve on the minimum flow bypass**
  In addition to flow control method to maintain the minimum flow is installing a pressure sensing device at the centrifugal pump discharge line. The pressure control is cost effective and makes use of the centrifugal pump head-capacity curve. Thus, the pump discharge pressure controller will open the control valve in the minimum flow bypass line at a discharge pressure above the set point and will close at a pressure below the set point.

  The pressure control scheme can not be used if pump suction is not stable and a variation of it could result in a discharge pressure that will cause the pump to operate below the minimum flow. For example, in a normal operation, a decrease in discharge pressure due to a reduction in origin pressure or the suction drum level (resulting in low discharge pressure) will make pump to gain head in order to return to the constant discharge pressure control set point in such way that the required head gain will cause pump to operate below the minimum flow or cause the pump to go to shutoff.

- **Centrifugal pump – Connect a pressure safety valve at pump outlet**
  Another method to effect a discharge pressure control for a centrifugal pump is to install pressure safety valve (PSV) at the pump discharge. When the pump discharge pressure exceeds the set pressure of the PSV, the PSV opens sending potion of the flow to recycle, building up the pumping flow.

  PSV may either stay open or simmer when the pressure returns to below its set pressure either due to leak because of imperfect reseat or because the pressure is not below the blowdown pressure (typically below the set pressure) to cause a full closure.

- **Variable speed drive**
  In addition to running the bypass to vary pump's net forward flow rate, changing the pumping speed can also change the pumping or the net forward flow without the use of recycle. The range of pump capacity change shall correspond to the range of best efficiency. A pumping circuit with a system curve dominated by the frictional pressure drop, instead of one that is dominated by the pressure head or static head, makes a better candidate for centrifugal variable speed pump.

  The impact of the entire range of speed on pump performance, such as NPSHR at low speed, maintenance cost at high speed, shaft critical speed, or electrical demand, shall be examined to ensure safe and reliable operation.

- **Cavitation**
  Pressure at the centrifugal pump inlet shall be adequate to accelerate the liquid to a high velocity at the entrance of impeller without causing the pressure there to fall below liquid's vapor pressure to cause cavitation. Provide adequate NPSHA to satisfy pump's NPSHR to avoid cavitation.

- **Flashing**
  When pumping a boiling liquid, locate the control valve at the downstream of heat exchangers or air coolers to avoid flashing through valve.

- **Vortex Breaker**
  Entrained gas bubbles, due to the natural swirling action of fluid before entering the suction piping, may reducing the NPSHA and not desirable for centrifugal pump operation.

  Installing a **vortex breaker** immediately before the suction piping can prevent the gas entrainment to the pump. Every pump suction vessel should have a vortex breaker.

- **Parallel pumps**
  Combined capacity is by adding capacities for each centrifugal at any given head.

  When two or more centrifugal pumps are pipe parallel to a common suction header, one pump may seem to take more flow form the other pump(s). This is caused by uneven suction pressures at the impeller inlets of various pumps. Symmetrical suction piping will minimize this problem. Low suction header pressure, flat pump curves (worse yet, drooping curve) can produce pronounced effect due to minor inlet piping un-symmetry.

  Positive displacement pumps are more suited for parallel configuration.

- **Pumps in series**
  Final head is by adding heads for each centrifugal at the given pumping rate.

- **Suction Piping - Centrifugal**
  Maximize NPSHA by having inlet piping one size larger than the pump suction nozzle.

  Reduce gas bubble formation or accumulation in the suction piping of a centrifugal pump. Avoid flow swirl and turbulence at the pump inlet. Use of **reducers** at pump suction downstream of valves and fittings can improve inlet velocity profile, If an reducer is used in a horizontal line, an eccentric reducer (with flat side on the top) shall be selected to avoid gas accumulation.

  Do not to trap gas bubbles in the suction piping. Trapped bubbles effectively reduce suction piping size resulting in higher suction friction loss and lower NPSHA. Piping layout shall have **continuous-rising slope** (in lift operation) or **continuous-decreasing slope** (if liquid level in suction vessel is high than the pump) to avoid trapping of gas.

- **Suction Piping - Recip**
  Consider using lower velocity for the suction line sizing for the recip. pump than the centrifugal counterpart in order to dampen out the pulsation whenever possible.

How much does a pumping system cost?

Piping cost is typically 20 to 40% of the project capital outlay.

US Department of Energy report in 1998 indicated that pumping cost consumed approximately 25% electrical energy in the industrial motor applications in the US. The Hydraulic Institute have also estimated that pump motors consumed approximately 20% of the world electric power. Thus, incentives exist to optimized the pump horsepower via designing a low first cost piping system and selection of efficient pumps and motor drivers.

While there are countless combination of sizing for pipes and pumps for a given pumping circuit, a spreadsheet or a computer hydraulic simulation software can be a useful tool to allow a quick what-if analysis to assist in estimating the capital expenditure (CAPEX). You should also consider the annual operating expenditure (OPEX) to get an idea of a life-cycle cost. The life-cycle cost is CAPEX plus the net present value of OPEX based on the useful period and the discount rate specified in your project.

Some economic considerations for evaluating **life cycle cost** of a pumping circuit are listed as reference examples:

- Eliminate continuous use of minimum flow bypass line for centrifugal pumps.
- Evaluate the use of the adjustable speed drive to reduce pumping energy (see Affinity Law for speed vs. Bhp) in response to flow reduction during any period of the life time of the pumping circuit, instead of dissipating energy through throttling the control valve.
- Choose suitable lines sizes. Consider allowance for possible future expansion, increase in frictional pressure drop over time or different modes of process operation, velocity limits to help minimize corrosion or erosion, etc.

References

1. www.edstech.com
2. Nelson, W. E., "Important information that's not on the pump curve, but should be …", *Fluid Handling Systems*, P.25, September 2001.
3. Hydraulic Institute , "Pump FAQ's", *Fluid Handling Systems*, P.29, September 2001.