EXPERIMENT (5)

PUMP CHARACTERISTICS AND PRESSURE DROP IN PIPE
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Introduction

The most common pump in chemical industry is the centrifugal pump, used for moving liquids of low viscosity from one place to another. It is simple in design, convenient to use, and can be built for a very wide range of capacity and pressure conditions.

There are numberless variations in details of the design of individual pumps, but the underlying principle is the same in all of them. A rotating element, called the impeller, slings liquid outward and tangentially. The force thus applied increases the momentum of the liquid. This energy input to the liquid manifests itself as a pressure increase on the discharge side of the pump. The impeller revolves in a chamber called the casing. Liquid flows to and enters the pump through a pipe, where it is delivered to the center of impeller. The liquid slung off the impeller enters a space surrounding the impeller, called the volute, and is carried off in a tangential direction, to the discharge, where it exits through another pipe. For a discussion of the design and operation of centrifugal pumps see McCabe et al., (1993), Church (1994), Karassik and Carter (1960), Worthington Pump (1971), Yedidiah (1977).

Pump designs vary according to the capacity and pressure requirements, and the nature of the fluid pumped. The fluid may, for example, be corrosive, have suspended solids, be very near its boiling point, or it may present other problems. The required flow rate may be a few gallons per hour, or hundreds of gallons per minute. The discharge pressure may be only a few pounds per square inch, or it may be dozens of psi. The engineer chooses the right pump from among many designs, basing his decision on the performance specifications supplied by the manufacturer.

It may happen, however, that a pump is available, but the performance specifications are missing. Or it may be that the engineer, for some reason, questions the accuracy or the applicability of the specifications available to him. For example, specifications are ordinarily based on tests using water as the liquid pumped. The engineer may question whether the specifications are valid for a liquid of considerably higher viscosity. It may then be necessary for him to test the performance of the pump for himself.

This is the situation the student is faced with in this experiment. The student has a pump for which there is a performance curve based on water. He will test the pump with a solution of ethylene glycol, which has considerably higher viscosity. Further, the pump is circulating the solution through a system of pipe which the student will check to compare with calculated values for pressure drop. His observations will enable him to estimate the true present condition of the piping system (which may or may not be badly corroded).

Background

* Pump performance curves

Figures 3-36A, B, C represent typical centrifugal pump performance curves, where head, brake horsepower, pump efficiency and Net Positive Suction Head (N.P.S.H.) are plotted against the capacity. These figures demonstrate the relationship among these performance parameters and are supplied by the manufacturer. The following describes the performance parameters shown in Fig. 3-36A, B, C, (Ludwig, 1995).
Capacity:

Capacity is the rate of liquid or slurry flow through a pump. This is usually expressed as gallon per minute (GPM). For proper selection and corresponding operation, a pump capacity must be identified with the actual pumping temperature of the liquid in order to determine the proper power requirements as well as the effects of viscosity.

Pumps are normally selected to operate in the region of high efficiency, and particular attention should be given to avoiding the extreme right side of the characteristic curve where capacity and head may change abruptly.

Total Head:

Total head is the pressure available at the discharge of a pump as a result of the change of mechanical input energy into kinetic and potential energy. This represents the total energy given to the liquid by the pump. Head, previously known as total dynamic head, is expressed as feet of fluid being pumped.

The total head read on the pump curve is the difference between the discharge head (the sum of the gauge reading on the discharge connection on the pump outlet, for most pumps corrected to the pump centerline, plus the velocity head at the point where the gauge is attached) and the suction head (the sum of the suction gauge reading corrected to the pump centerline and the velocity head at the point of attachment of the suction gauge). Note that the suction gauge reading may be positive or negative, and if negative, the discharge head minus a minus suction (termed lift) creates an additive condition.

The total head is shown on the curves of Figure 3-36A. This head produced is independent of the fluid being pumped and is, therefore, the same for any fluid through the pump at a given speed of rotation and capacity.

![Figure 3-36A. Typical centrifugal pump curves. (Adapted by permission, Allis-Chalmers Mfg. Co.)](image)
Figure 3-36B. Typical performance curve showing NPSH in convenient form. (By permission, Crane Co., Deming Pump Div.)
Figure 3-36c. Illustrates exact same pump casing and impellers at two different shaft speeds. (By permission, Goulds Pumps, Inc.)
Through conversion, head may be expressed in units other than feet of fluid by taking the specific gravity of the fluid into account.

\[
(\text{Head in feet of a liquid}), \ H = (\text{psi}) \ (2.31) \div \text{SpGr}.
\]

for any fluid

Note that psi (pounds per square inch) is pressure on the system and is not expressed as absolute unless the system is under absolute pressure. Feet are expressed as head, not head absolute or gauge. Note the conversion of psi pressure to feet of head pressure.

\[
(\text{head in ft}), \ H = (\text{psi}) \ (144/\rho)
\]

where \( \rho = \text{fluid density}, \ lb/cu\ ft \)

\[
lb/\text{sq in.} = 2.31 \text{ ft of water at SpGr = 1.0}
\]

\[
lb/\text{sq in.} = 2.31 \text{ ft of water / SpGr of liquid = ft liquid}
\]

\[
1 \text{ in. mercury} = 1.134 \text{ ft of water} = 1.134/\text{SpGr liquid, as ft liquid}
\]

For water, SpGr = 1.0 at 62°F, (for general use it can be considered 1.0 over a much wider range).

\textbf{Example 1: Liquid Heads}

If a pump were required to deliver 50 psig to a system, for water, the feet of head on the pump curve must read,

\[
2.31 \times (50) = 115.5\ ft
\]

For a liquid of SpGr 1.3, the ft of head on the pump curve must read,

\[
115.5/1.3 = 88.8\ ft\ of\ liquid.
\]

For a liquid of SpGr 0.86, the ft of head on the pump curve must read,

\[
115.5/0.86 = 134.2\ ft\ of\ liquid.
\]

If a pump were initially selected to handle a liquid where SpGr = 1.3 at 88.8 ft, a substitution of light hydrocarbon where SpGr = 0.86 would mean that the head of liquid development by the pump would still be 88.8 feet, but the pressure of this lighter liquid would only be \( 88.8\times[(2.31)/(0.86)] \) or 44.8 psi. Note that for such a change in service, the impeller seal rings, packing (or mechanical seal) and pressure rating of casing must be evaluated to ensure proper operation with a very volatile fluid. For other examples, see Figure 3-37.

The total head developed by a pump is composed of the difference between the static, pressure and velocity heads plus the friction entrance and exit head losses for the suction and discharge sides of the pump. Refer to Figures 3-38 and 3-39.

\[
H = h_d - h_s
\]

Where \( h_d \) is the discharge head and \( h_s \) is the suction head which are defined in Figure 3-38 and 3-39.
FIGURE 3-37: Comparison of columns of various liquids to register 43.3 psig on pressure gauge at bottom of column.
Figure 3-38. Suction head system.

Figure 3-39. Suction lift system.
Static Head

This is the overall height to which the liquid must be raised.

For Figure 3-40A

Discharge static head: H
Suction static head: L (actually \(-L\))
Total system static head: \(H + L\);
actually \(H - (-L)\)  

For Figure 3-40B

Discharge static head: H (from centerline of pump)
Suction static head: S, (actually \(+S\))
Total system static head: \(H - S\); or \(H - (+S)\)  

Pressure Head

For Figure 3-40C

Discharge pressure head = 100 psig
Suction pressure head = 0 psig
Total pressure head = 100 \((-0)\) = 100 psig
\(= 100(2.31)^* = 231\) ft of water

Note: The totals are differentials and neither gauged nor absolute values.

*Applies to water only. For the other fluids use appropriate specific gravity conversion.

For Figure 3-40D

Discharge pressure head = 100 psig
Suction pressure head = \(+50\) psig (=64.7 psia)
Total pressure head = 100 \((-50)\) = 50 psig
not gauge or absolute
\(50 (2.31) = 115.5\) ft of water
The sign of \( h_s \) when a suction lift is concerned is negative, making 
\[ H = h_d - (-h_s) = h_d + h_s. \]

The three main components illustrated in the examples are:

1. Static head
2. Pressure head
3. Friction in piping, entrance and exit head losses

A pump is acted on by the total forces, one on the suction (inlet) side, the other on the discharge side. By subtracting (algebraically) all the suction side forces from the discharge side forces, the result is the net force that the pump must work against. However, it is extremely important to recognize the algebraic sign of the suction side components, that is, if the level of liquid to be lifted into the pump is below the pump centerline, its algebraic sign is negative (-). Likewise, if there is a negative pressure or vacuum on the liquid below the pump centerline, this works against the pump and it becomes a negative (-).

**Note:**

In the lab experiment, the total head which is

\[
\text{Total head} = \text{total discharge head} - \text{total suction head}
\]

would be calculated as follows:

\[
total \ head \ (ft \ fluid) = (P_{\text{outlet}} - P_{\text{inlet}}) \ (in \ psi) \times \frac{2.31 \ ft \ of \ water}{psi} \ \text{SpGr of liquid}
\]

\[ P \ (\text{absolute pressure, psi}) = P' \ (\text{gauge pressure, psig}) + 14.7 \]

\[ 1 \ psi = 1 \ lb/in^2 = 2.3067 \ ft \ water = 2.31 \ ft \ water \]

In the lab experiment, \( P_{\text{outlet}} \) and \( P_{\text{inlet}} \) are measured by gauges (i.e. psig). \( (P_{\text{outlet}} - P_{\text{inlet}}) \) is the difference in the pressure across the pump which is the pressure developed by the pump.
Brake horsepower (BHP):

The horsepower curves (HP) shown in Fig. 3-36A, B, C represent the Brake Horsepower (BHP). It represents the power required by the pump, which must be transmitted from the driver through the drive shaft through any coupling, gear-box, and/or belt drive mechanism to ultimately reach the driven shaft of the pump.

\[ \text{BHP} = \text{horsepower (energy) input to the pump} \]
\[ \quad = \text{horsepower output from the motor (driver) - losses in transmission from the motor to the pump shaft through coupling, gear-box, and/or belt drive, etc.} \]

In the lab experiment, BHP can be calculated as follows:

\[
\text{BHP} = \frac{\text{Torque (lb} \cdot \text{ft)} \times N \text{ (rpm)} \times 2\pi \text{ (rad/rev.)}}{60 \text{ (sec/min)} \times 550 \text{ (lb} \cdot \text{ft/sec. HP)}}
\]

(Where, 550 is a conversion factor)

Torque is defined in the appendix (reading materials). In the lab experiment torque can be measured as:

\[
\text{Torque} = \text{spring reading (lb}_f) \times \text{Torque arm (ft)}
\]
\[
= \text{spring reading (lb}_f) \times 1.5 \text{ ft.}
\]

Note: Unless specifically identified otherwise, the BHP values read from a manufacturers performance curve represent the power only for handling a fluid of viscosity about the same as water and a specific gravity the same as water, i.e., SpGr = 1.0. To obtain actual BHP for liquids other than water, correction for specific gravity and viscosity should be made (Ludwig, 1995).

Water or Liquid Horsepower:

Water or liquid horsepower is the energy delivered by pump to the liquid being pumped (i.e., the energy received by the liquid). It represents the horsepower output from the pump. In the lab experiment, it can be calculated as follows:

\[
\text{Liquid horsepower (HP)} = \frac{\text{total head (ft liq.)} \times Q \text{ (GPM)} \times \text{SpGr}}{3960 \left( \frac{\text{ft liq. GPM}}{\text{HP}} \right)}
\]

Where total head (ft liq.) = \( P_{\text{outlet}} - P_{\text{inlet}} \) (psi) \( \times \)
\[
2.31 \left( \frac{\text{ft water}}{\text{psi}} \right) \times \text{SpGr}
\]

Hence,
Liquid horsepower (HP) = \frac{\Delta P \text{ (psi)} \times Q \text{ (GPM)}}{3960} \times 2.31

Pump Efficiency:

\text{Pump efficiency (\%\%)} = \frac{\text{liquid HP (delivered to fluid)}}{\text{Brake HP (delivered to pump shaft)}} \times 100

\Delta P \ (\text{psi}) \times Q \ (\text{GPM})

\text{1715} \times BHP

Overall efficiency (Pump & Motor):

\text{overall efficiency} = \frac{\text{liquid HP (energy delivered by pump to fluid)}}{\text{electrical HP (energy supplied to input side of the pump's driver (motor))}}

Note: Knowing the BHP and pump efficiency, liquid HP can be determined. Therefore, in typical pump performance curves, liquid HP curve is not provided.

NPSH (Net Positive Suction Head (Ludwig, 1995)):

NPSH curves, shown in Fig. 3-36A, B, C (previously mentioned), are the required NPSH (NPSH\text{R}). The available value of NPSH of the system (NPSH\text{A}; which should be considered in a pump installation) must be greater by a minimum of two feet than the required NPSH (i.e. NPSH\text{R}). The following provide more information about NPSH and its importance (Ludwig, 1995).

Net positive suction head (in feet of liquid absolute) above the vapor pressure of the liquid at the pumping temperature is the absolute pressure available at the pump suction flange, and is a very important consideration in selecting a pump which might handle liquids at or near their boiling points, or liquids of high vapor pressures.

Do not confuse NPSH with suction head, as suction head refers to pressure above atmospheric. If this consideration of NPSH is ignored the pump may well be inoperative in the system, or it may be on the border-line and become troublesome or cavitating. The significance of NPSH is to ensure sufficient head of liquid at the entrance of the pump impeller to overcome the internal flow losses of the pump. This allows the pump impeller to operate with a full "bite" of liquid essentially free of flashing bubbles of vapor due to boiling action of the fluid.

The pressure at any point in the suction line must never be reduced to the vapor pressure of the liquid. Both the suction head and the vapor pressure must be expressed in feet of the liquid, and must both be expressed as gauge pressure or absolute pressure. Centrifugal pumps cannot pump any quantity of vapor, except possibly some vapor entrained or absorbed in the liquid, but do not count on it. The liquid or its gases must not vaporize in the eye/entrance of the impeller (this is the lowest pressure location in the impeller).
For low available NPSH (less than 10 feet) the pump suction connection and impeller eye may be considerably oversized when compared to a pump not required to handle fluid under these conditions. Poor suction condition due to inadequate available NPSH is one major contribution to cavitation in pump impellers, and this is a condition at which the pump cannot operate for very long without physical erosion damage to the impeller.

Cavitation of a centrifugal pump, or any pump, develops when there is insufficient NPSH for the liquid to flow into the inlet of the pump, allowing flashing or bubble formation in the suction system and entrance to the pump. Each pump design or "family" of the dimensional features related to the inlet and impeller eye area and entrance pattern requires a specific minimum value of NPSH to operate satisfactorily without flashing, cavitating, and loss of suction flow.

Under cavitating conditions a pump will perform below its head-performance curve at any particular flow rate. Although the pump may operate under cavitation conditions, it will often be noisy because of collapsing vapor bubbles and severe pitting, and erosion of the impeller often results. This damage can become so severe as to completely destroy the impeller and create excessive clearances in the casing. To avoid these problems, watch the following situations:

1. Have NPSH$_A$ available at least 2 feet of liquid greater than the pump manufacturer requires under the worst possible operating conditions (see pump curves Figures 36A, B, C) with pump curve values for NPSH expressed as feet of liquid handled. These are the pump's required minimum NPSH$_R$. The pump's piping and physical external system provides the available NPSH$_A$.

   $$NPSH_A \text{ must be } > (NPSH_R + \text{ at least } 2 \text{ ft. of liquid, presumably more})$$

2. Internal clearance wear inside the pump.
3. Plugs in suction piping system (screens, nozzles, etc.).
4. Entrained gas (non-condensable).
5. Deviations or fluctuations in suction side pressures, temperatures (increases), low liquid level.
6. Piping layout on suction, particularly tee-intersections, globe valves, baffles, long lines with numerous elbows.
7. Liquid vortexing in suction vessel, thus creating gas entrainment into suction piping. Figure 3-43 suggests a common method to eliminate suction vortexing. Since the forces involved are severe in vortexing, the vortex breaker must be of sturdy construction, firmly anchored to the vessel.
8. Nozzle size on liquid containing vessel may create severe problems if inadequate. Liquid suction velocities, in general, are held to 3-6.5 feet per second. Nozzle losses are important to recognize by identifying the exit design style (see Chapter 2, Ludwig, 1995). Usually, as a guide, the suction line is at least one pipe size larger than the pump suction nozzle.
The NPSH\textsubscript{A} available from or in the liquid system on the suction side of a pump is expressed (corrected to pump centerline) as (see the reading materials):

\[ \text{NPSH}_A = S + (P_a - P_v) \left( 2.31 / \text{SpGr} \right) - h_{SL} \]

Where \( P_a \) (psi) represents the absolute pressure in the vessel (or atmospheric) on the liquid surface on the suction side of the pump.

\( P_v \) (psi) represents the absolute vapor pressure of the liquid at the pumping temperature (i.e. the vapor pressure of the liquid inside the pump).
\( h_{sl} \) is the suction line, valve, fitting and other friction losses from the suction vessel to the pump suction flange (For the lab experiment you may neglect it).

\( S \), which is suction static head or static lift, may be \(+\) or \(-\) depending on whether suction static head \(+\) or static lift \(-\) is involved in the system.

This available value of \( \text{NPSH}_A \) (of the system) must always be greater by a minimum of two feet and preferably three or more feet than the required \( \text{NPSH} \) stated by the pump manufacturer or shown on the pump curves in order to overcome the pump's internal hydraulic loss and the point of lowest pressure in the eye of the impeller. The \( \text{NPSH} \) required by the pump is a function of the physical dimensions of casing, speed, specific speed, and type of impeller, and must be satisfied for proper pump performance. The pump manufacturer must always be given complete suction conditions if he is to be expected to recommend a pump to give long and trouble-free service.

As the altitude of an installation increases above sea level, the barometric pressure, and hence \( P_a \) decreases for any open vessel condition. This decreases the available \( \text{NPSH} \).

Figure 3-36A (previously mentioned) represents a typical manufacturer's performance curve. The values of \( \text{NPSH}_R \) given are the minimum values required at the pump suction. As mentioned, good practice requires that the \( \text{NPSH}_A \) available be at least two feet of liquid above these values. It is important to recognize that the \( \text{NPSH}_R \) and Suction Lift Values are for handling water at about \( 70 \_F \). To use with other liquids it is necessary to convert to the equivalent water suction lift at \( 70 \_F \) and sea level.

Total Suction Lift (as water at \( 70 \_F \)) = \( \text{NPSH}_A \) (calculated for fluid system) - 33 feet. The vapor pressure of water at \( 70 \_F \) is 0.36 psia.

**Example 2: Suction Lift**

What is the Suction Lift value to be used with the pump curves of Figure 3-36A (previously mentioned), if a gasoline system calculates an \( \text{NPSH} \) of 15 feet available:

Total Suction Lift (as water) = 15 - 33 = -18 feet. Therefore, a pump must be selected which has a lift of at least 18 feet. The pump of Figure 3-36A is satisfactory using an interpolated Suction Lift line between the dotted curves for 16 feet and 21 feet of water. The performance of the
pump will be satisfactory in the region to the left of the new interpolated 18-foot line. Proper performance should not be expected near the line.

If the previous system were at sea level, consider the same pump with the same system at an altitude of 6000 feet. Here the barometric pressure is 27.4 feet of water. This is $34 - 27.4 = 6.6$ feet less than the sea level installation. The new NPSH$_A$ will be 15 ft - 6.6 ft = 8.4 feet available. Referring to the pump curve of Figure 3-36A it is apparent that this pump cannot do greater than 21 feet suction lift as water or 12 feet NPSH$_R$ of liquid (fluid).

Total Suction Lift as water = 8.4 - 33 = -24.6 feet. The pump curves show that 21 feet suction lift of water is all the pump can do, hence the 24.6 feet is too great. A different pump must be used which can handle this high a suction lift. Such a pump may become expensive, and it may be preferable to use a positive displacement pump for this high lift. Normally lifts are not considered reasonable if over 20 feet.

- Developing the pump performance curves

If there are no gauges at the outlet and the inlet of the pump and the BHP and the pump efficiency are unknown, how the pump performance curves can be developed? (See McCabe et. al, 1993, page 190).

Bernoulli equation over the pump itself as shown below, gives

\[
\frac{P_a}{\rho} + \frac{g_z}{g_e} + \frac{V^2}{2g_c} + \eta_{\text{input}} = \frac{P_b}{\rho} + \frac{g_z}{g_e} + \frac{V^2}{2g_c} + h_f
\]

where $\eta_{\text{input}}$ is the work done on the liquid by the pump ($w_{\text{output from the pump}}$)

\[
P_b - P_a = \rho \left( \frac{V^2 - V^2}{2g_c} + \eta_{\text{input to the pump}} \right)
\]
All terms on the right hand side of the equation are known except $\eta \mu_{\text{input}}$ which needs to be calculated.

To calculate $\eta \mu_{\text{input}}$, Bernoulli equation needs to be written for the whole system (i.e. the tanks and the pump between the points c and d) as shown in the figure above.

$$\frac{P_c}{\rho} + \frac{gZ_c}{g_c} + \frac{V_c^2}{2g_c} + \mu \mu_{\text{input}} = \frac{P_d}{\rho} + \frac{gZ_d}{g_c} + \frac{V_d^2}{2g_c} + h_f$$

$\frac{V_c^2}{2g_c}$ can be neglected where the diameter of the tank (at c) is larger than that of the pipe (at d). Hence,

$$\eta \mu_{\text{input}} = (Z_d - Z_c) \frac{g}{g_c} + \frac{V_d^2}{2g_c} + h_f$$

Calculating $\mu \mu_{\text{input}}$, $(P_b - P_a)$ can be evaluated. The total head then is estimated as mentioned before:

$$TDH = (P_b - P_a) \text{psi} \times 2.31 \text{ ft/SpGr ft of liquid}$$

By doing the calculation above at different flow rates, the total head curve can be developed.
- Trend of the pump performance curves obtained using the lab experiment set-up.

The trend of the pump performance curves obtained by the experimental data collected using the lab experiment set-up would be more or less comparable to the following trends:

Figure 3-58. Typical curves showing the effect on a pump designed for water when pumping viscous fluids. (By permission, Pico-a-Pump, 1959, Allis-Chalmers Mfg. Co.)
The performance curves shown previously in Fig. 3-36A cannot be fully developed in the experiment because data collected is not enough. Hence, only the following part of the performance curve would be obtained.

Total head (TH or TDH) (ft. liq.) can be plotted against capacity (GPM) which represents the head produced by the impeller diameter used. BHP, pump efficiency and NPSHR are plotted as point intercepts on the TDH curve at corresponding GPM. Short intercept lines can be drawn to show the trend and values of these parameters measured at each GPM value.

![Graph showing Head (TDH) vs GPM]

* Motor performance curves

The motor performance curves are developed by plotting the current (I) to the motor, motor shaft rotational speed (rpm), motor efficiency (% $\eta_{motor}$) and the power factor against the load. Such curves demonstrate the relationship among the performance parameters. To develop the motor performance curves, the following should be considered:
- The motor performance curve is based on the name plate voltage (nameplate is a metal tag mounted on the motor).

- load abscissa (x-axis) ranges from 0 to 1.5 load. This means that the x-axis represents the load fraction which can be calculated as follows:

\[
\text{load fraction} = \frac{\text{output HP from the motor to the shaft}}{\text{nameplate HP (rated HP)}} = \frac{BHP}{2}
\]

where, the lab experiment nameplate HP (rated HP) for the motor is 2 HP.

**Current supplied to the motor:**

The current curve shown in a performance curve represents the current that would have been supplied if the voltage had been at the rated voltage on the motor nameplate.

In the lab experiment, the current supplied to the motor can be measured by a clamp-on ammeter. For each load (i.e. load fraction), this can be done by measuring the current through each wire of the three wires connected to the motor (i.e. three phase motor) and taking the average.

Accordingly, the measured (observed) current has to be corrected to that of the nameplate value as follows:

\[
I_{\text{corrected for the nameplate voltage}} = I_{\text{measured}} \times \frac{E_{\text{measured}}}{E_{\text{rated (i.e. nameplate)}}} = \frac{I_{\text{measured}} \times E_{\text{measured}}}{E_{\text{rated}}}
\]

where \(E_{\text{measured}}\) is the measured voltage by a voltmeter and \(E_{\text{rated}}\) is the rated voltage on the nameplate which is equal to \(\frac{230}{220}\) volt in this experiment (check this number on the motor nameplate).

**Motor shaft rotational speed (rpm):**

The rpm of the motor shaft is measured by a tachometer. An average of few measurements should be taken at each load.

**Motor efficiency:**

\[
\%\eta_{\text{motor}} = \frac{\text{HP output from the motor}}{\text{HP input to the motor}} \times 100
\]

\[
\text{HP output from the motor} = BHP = \frac{\text{Torque (lb ft)} \times N (\text{rpm})}{5252.1}
\]

\[
\text{HP input} = \frac{W_{\text{measured}}}{746}
\]
where, \( W_{measured} \) is measured by a wattmeter in watt and 746 is a converter factor.

\[ 1 \text{ HP} = 746 \text{ watt} \]

Hence,

\[ \% \eta_{motor} = \frac{BHP}{(W_{measured}/746)} \times 100 \]

**Power factor (Pf):**

As discussed in the Appendix, the power factor (pf) is evaluated as follows:

\[ W = E I \cos(\theta) \sqrt{\phi} \]

where \( \cos(\theta) \) is the power factor, \( \phi \) is the number of phases (i.e. 3 for the lab experiment), \( E \) is the voltage measured by a voltmeter and \( I \) is the current measured by an ammeter.

\[ Pf = \frac{W_{measured \text{ (watt)}}}{E_{measured \text{ (volt)}} I_{measured \text{ (amp)}} \sqrt{3}} \]
* The trend of the motor performance curves

The following figure shows more or less the trend of the performance curves for the parameters discussed previously.
Friction Factor Chart

Friction factor chart is a plot of fanning friction versus the Reynolds number for a range of values of the roughness parameter ($\varepsilon/D$). Pipe roughness $\varepsilon$ represents the condition of the pipe surface and may be considered as the characteristic of the height of projections from the pipe wall, hence it has a dimension of length. It has been known that in turbulent flow a rough pipe gives a larger friction factor for a given Reynolds number than a smooth pipe does. However, in laminar flow (Re < 2300), the friction factor is not a function of pipe roughness unless $\varepsilon$ (pipe roughness) is so large that the measurement of the diameter becomes uncertain. The roughness of a pipe is usually represented by the relative roughness ($\varepsilon/D$) which is the ratio of the pipe roughness to the diameter of the pipe.

It has been found that all clean, new commercial pipes seem to have the same type of roughness (new steel or wrought iron, $E=0.00015$ ft.) and that each material of construction has its own roughness characteristic. Old, foul and corroded pipe can be very rough, and the character of the roughness differs from that of clean pipe. Hence the source of roughness in a pipe would be the manufacturing process, material of construction, fouling, corrosion, etc.

Friction factor chart based on the experimental measurements

Friction factor chart is developed by plotting fanning friction factor or Darcy/Blasius friction factor against Reynolds number on log-log scale. Fanning friction factor chart is also called Moody chart.

The friction factor is the ratio of the wall shear stress to the product of the velocity head ($V^2/2g_c$) and the density (McCabe et al., 1993).

Friction head loss:

$$h_f = 2f \left( \frac{L_{total}}{D} \right) \frac{V^2}{g_c}$$

Fanning friction factor:

$$f = \frac{g_c h_f}{2V^2 \left( \frac{L_{total}}{D} \right)}$$

Darcy friction factor is defined by

$$h_f = f_D \left( \frac{L_{total}}{D} \right) \frac{V^2}{2g_c}$$

Hence, $f_D = 4f$. 

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In general, students must identify which friction factor is used to evaluate the head lost. Fanning friction factor \((f)\) is used in this experiment.

where \(f = \) fanning factor, \(f_D = \) Darcy factor, \(h_f = \) friction loss \((lb_f \text{ ft/} lb_m)\), \(V = \) velocity and

\[
L_{\text{total}} = L_{\text{pipe}} + L_{\text{equiv}}.
\]

where \(L_{\text{equiv}}\) is the length equivalents of the various fittings, values, etc. knowing the piping system properly as described in the lab manual, \(L_{\text{total}}\) can be estimated (see McCabe et al., 1993, Perry et al., 1984, etc.)

Hence at each liquid flow rate through the piping system, fanning friction factor \((f)\) can be evaluated by measuring friction head loss and liquid velocity (where Reynolds number can be evaluated too) as:

- **Measuring friction head \((h_f)\) loss:**

\(h_f\) represents the energy dissipated in the system. Hence, by measuring the pressure drop across the piping system, \(h_f\) in \(lb_f \text{ ft/} lb_m\) unit can be determined as:

If \(\Delta P\) is measured in psi:

\[
h_f \left(lb_f \text{ ft/} lb_m\right) = \left(P_{\text{pipe inlet}} - P_{\text{pipe outlet}}\right) \frac{lb_f}{in^2} \times 144 \frac{in^2}{ft^2} \times \frac{1}{SpGr \times 62.42 \frac{lb_m}{ft^3}}
\]

If \(\Delta P\) is measured in ft of water:

\[
1 \text{ psi} = 1 \text{ lb/in}^2 = 2.31 \text{ ft of water}
\]

\[
h_f \left(lb_f \text{ ft/} lb_m\right) = \frac{\Delta P \left(ft \text{ of water}\right)}{2.31} \times \frac{144}{SpGr \times 62.42}
\]

- **Reynolds number \((Re)\)**

\[
Re = \frac{\rho V D}{\mu}
\]

the velocity of the liquid is measured by measuring the volumetric flow rate using the rotameter and calculating the inside cross section area of the pipe.

Plotting \(f\) versus \(Re\) on log-log scale, yields the fanning friction factor chart (Moody chart).
• Friction factor chart based on the theoretical calculations.

Friction factor chart can be developed based on the theoretical calculation using the empirical correlations proposed in the literature as follows:

- For laminar flow, \( f = \frac{46}{Re} \left( \frac{\varepsilon}{D} = 0 \right) \)

- For turbulent flow in a smooth pipe,

where \( \varepsilon \) is the roughness of the pipe (length dimension).

Friction factor can be calculated by one of the following correlations

\[
\frac{1}{\sqrt{f}} = 4.06 \log \left( Re \sqrt{f} \right) - 0.6 \hspace{1cm} \text{or} \hspace{1cm} \frac{1}{\sqrt{f}} = 4.0 \log_{10} \left( Re \sqrt{f} \right) - 0.40
\]

(Welty et al., 1984, eq. 14-11 and 14-12, respectively)

\[
\sqrt{f} = -4 \log_{10} \left( 1.256 / Re \sqrt{f} \right)
\]

(Perry et al., 1994, page 5-24, eq.5.59)

By using \( Re \) values within the experimental range, \( f \) curve for smooth pipe can be developed.

- For turbulent flow in a rough pipe (new steel or wrought iron pipe, \( \varepsilon = 0.00015 \) ft. regardless of the diameter of the pipe), one of the following correlations can be used

\[
\sqrt{f} = -4 \log \left( \frac{\varepsilon}{3.7D} + 1.256 / Re \sqrt{f} \right)
\]

for \( 4 \times 10^4 \leq Re \leq 10^8 \) and \( 0 \leq \varepsilon/D \leq 0.05 \) (Welty et al., 1984, eq. 14-15a)

\[
\frac{1}{\sqrt{f}} = -3.6 \log \left[ \frac{6.9}{Re} + \left( \frac{\varepsilon}{3.7D} \right)^{10/7} \right]
\]

(Perry et al., 1994, eq. 5-58)

By using \( \varepsilon/D = 0.00015 \) and the \( Re \) values within the experimental range, \( f \) curve for new steel pipe can be developed.

• \((\varepsilon/D)\) of the experimental piping system

The ratio of the roughness over the pipe diameter for the piping system used in the experimental set-up can be evaluated using the following correlation
\[
\frac{1}{\sqrt{f}} = -4 \log \left( \frac{\varepsilon}{3.7D} + \frac{1.256}{Re \sqrt{f}} \right)
\]

By using the measured values of \( f \) and \( Re \) (experimental data), averaged \( (\varepsilon/D) \) can be estimated.

- The optimum operating flow rate of a pump for a piping system

There is an optimum operating flow rate a pump can deliver for each piping system. This depends on the friction head loss and the total head (total dynamic head). Plotting friction head loss against flow rate (GPM) shows that the head loss increases as the flow rate in the piping system increases, whereas the total head decreases as the flow rate increases as shown in the pump performance curves. Therefore, the optimum operating flow rate is the GPM at which the friction head loss equivalent to the total head. This can be obtained by plotting the experimental friction head loss for a piping system on the pump performance curves. The flow rate (GPM) at the intercept of the friction head loss with the total head is the optimum value. Another way to do it is by plotting both the total head and the friction loss head against the flow rate (GPM) on the same graph. The intercept of the two curves is the optimum flow rate as shown below:
Equipment Description

A 1 X 1 1/2 - 6 (nominal size designation) Goulds centrifugal pump is used to circulate liquid from a tank through a network of pipe, and back. The pump has a 4-1/2 inch impeller. The pump is driven by a 2-horsepower Louis Allis motor which is suspended by a pair of bearings from its end bells. This converts the motor to a dynamometer. A lever arm projecting from the motor housing prevents the housing from turning. The force needed to restrain it is indicated by a spring balance. The torque developed by the motor and applied to its rotating shaft is thus determined. The torque-arm is 18 inches long, and the spring scale indicates the force in pounds. The rotational speed of the motor shaft is measured (it varies somewhat, around 3520 rpm). From these data the horsepower delivered to the pump at any moment can be computed readily. The electrical energy input to the motor is measure with a voltmeter, an ammeter and a wattmeter so that the efficiency and power factor of the motor under various loads can be determined.

There are pressure gauges on the suction and on the discharge sides of the pump. These, along with the rotameters in the discharge line for measuring flow rates enable calculating the hydraulic horsepower output of the pump. Since the observed flow rates will cover a wide range, there are two flow meters; one for low rates and the other for high rates of flow. To obtain very low discharge pressures, the pumped liquid can be returned directly to the storage tank. Otherwise the liquid is pumped through the piping system.

The pressure drop through the piping system is measured by gauges on the inlet and outlet of the pipe system. For very low pressures, which cannot be read accurately on the gauges, there is a manometer between the inlet and outlet. This manometer consists of two glass tubes, each of which serves as a stand-pipe to indicate the head of liquid at the point to which it is connected. The tubes are connected at the top, where air pressure is equal to both tubes. Thus the actual head is not measured, but rather this arrangement permits measuring the difference in head, which is the item of interest. See the Appendices for details.

The flow rate from the pump is measured by either of two meters, as shown in Figure 1. For low flow rates, up to 2 gpm, the rotameter is used. Viscosity has a negligible effect on the readings of a rotameter in the upper 80% of its range and for viscosities of 20 centipoise and less. Readings must be corrected for fluid density.

At higher flow rates the magnetic meter is used. The magnetic meter, as presently installed, indicates flow as 0 to 100 percent of full scale. That is, 0 to 90 gpm. The scale is linear. The accuracy is excellent.

The variable measured by the magnetic meter is the average linear velocity of the fluid flowing through it. Since the cross-sectional area is constant, the variable indicated is the actual volumetric flow rate. No correction, such as that used in the rotameter, is needed.

The meter is a short section of plastic tube clamped between two flanges in the pipeline carrying the fluid to be measured. On opposite sides of the tube are coils of wire carrying an electric current. These establish a steady magnetic field within the tube. The meter is suitable only for fluids which are electrical conductors, such as water and other polar liquids. Since the fluid is conductive (i.e., it has a good content of free electrons) a voltage is induced. The magnitude of the voltage indicates the velocity of the fluid. The voltage is measured by two electrodes in the
wall of the tube in a plane normal to the lines of magnetic flux. The arrangement is shown in the figure below, copied from the manufacturer's bulletin.

![Diagram showing the principle of operation of a magnetic flowmeter.](image)

**Figure 1-6 Principle of Operation**

1.3 Principle of Operation

Magnetic flowmeters can only be used to measure conductive liquids.

Operation of a magnetic flowmeter is explained by Faraday's Law, which simply states "the voltage induced across any conductor as it moves at right angles through a magnetic field is proportional to the velocity of that conductor". In a magmeter, the liquid is the conductor. As illustrated in Figure 1-6, a magnetic field is constructed throughout the entire cross section of the flow tube.

Faraday's Formula:

\[ E = v \times B \times d \]

Where:

- **E** = The voltage that is generated by the flow of conductive liquid through the magnetic field of the flowmeter.
- **v** = The average velocity of the liquid through the cross section of the flow tube in the flowhead.
- **B** = The strength of the magnetic field generated by the field coils.
- **d** = The distance between the electrodes which detect the signal voltage (E) that is generated.
Faraday's law states that the emf induced in a conductor, in volts is equal to the rate of change in magnetic flux:

$$\text{emf} = -\frac{d\Phi}{dt}$$

where \(\Phi\) is the flux in webers and \(t\) is time. Thus, in a given magnetic field the induced voltage is proportional to the velocity of the conductor through the field. Further, if a number of conductors connected in series pass through the field, the emf is also proportional to that number. The voltage induces by a flowing liquid is, of course, very small. The secret of the effectiveness of the magnetic flow meter is in the electronic circuitry that transforms the voltage into a reliable signal. The meter is calibrated by the manufacturer.

The pipe loop, consists of 1-inch schedule 40 iron pipe and fittings, as shown in Figures 1, 2, and 3. Figures 4 and 5 are the pump performance curves supplied by the manufacturer for the pump used in the system.

The system is filled with ethylene glycol solution (in water), the concentration of which may vary from time-to-time during the course of the semester. The properties of ethylene glycol are tabulated in Tables 1 and 2. Table 1 gives the density and viscosity of ethylene glycol in water as a function of temperature and concentration. The density of the solution is measured with a hydrometer and cylinder.

A tachometer for measuring motor speed, a clamp-on ammeter for measuring motor current, a hydrometer and cylinder for measuring liquid density, a lockout padlock and key, and any other small items needed for carrying out this experiment are available from the lab technician.

The pressure drop through the pipe loop is measured by spiral wound Bourdon type pressure gauges at the inlet and outlet of the loop, which are accurate over their whole range. It is the difference between the two pressure indications that is important in this experiment. If the gauge indicating the pressure from the loop indicates a pressure that is too low for easy readability, the pressure indication of both gauges can be raised by restricting the flow of liquid returning to the reservoir.

At low flowrates through the pipe loop the pressure drop may be on the order of 2 psi or less. The Bourdon gauges are not suitable for measuring such small differences accurately. The "inverted manometer" is used instead. This device requires some care if accurate readings are to be obtained. See the "Measurements of Small Pressure Drops" below. Notice that the liquid in the manometer is ethylene glycol solution, not water. Its reading must be corrected for the density of the fluid.

Measurement of Small Pressure Drops

The laboratory manual points out that for small pressure drops through the piping system the regular pressure gauges at the inlet and outlet cannot be read accurately enough to give meaningful results. Instead two standpipes are used to indicate the pressure, in inches of fluid, at the inlet and outlet of the system. A schematic of the arrangement is indicated in Figure 1. It should be obvious that with the system at rest, the liquid levels in the standpipes will be exactly
the same as the level in the tank.

It will also be apparent that if an air bubble is trapped at a high point in the pipe, the level in the standpipe will NOT be the same as the level in the tank. The amount of error in the standpipe indication will depend on the size of the bubble, and the hydrostatic pressure in the system at the point where bubble is located. The error introduced by a bubble is therefore indeterminate, but certainly not insignificant since it can vary from a fraction of an inch to several inches. Bubbles within the pipe cannot be seen, but their presence is revealed when the system is at rest and the standpipe levels do not agree with each other, nor with the level in the tank.

Before making measurements of small pressure differences, the experimenter should stop the pump and open the valves to the standpipes to make sure the static readings are correct. If they are not, the system and connecting lines should be purged of air before proceeding with the measurements. This is done as follows. (See Figure 2).

Close the valves to the standpipes, A and B. Start the pump and throttle the flow to a low rate (1 to 2 gpm). Adjust the valve on the return line (just above the tank) so that the pressure indication on the return line pressure gauge (PI) is 5 to 10 psi.

Make sure valve C is open, so that the standpipe is open to the atmosphere. Hold a beaker at the outlet from valve C and open valve A to allow fluid to purge the line to the standpipe. Close valve A. Repeat the procedure with the other standpipe, which has a valve to the standpipe, shown as B. This too should be opened to vent air bubbles and allow liquid to flush them out.

Finally, stop the pump and open valves C, A and B to make sure they both indicate the same level. If they do not, repeat the purge procedure until they do.

The two standpipes are interconnected at the top, so that the air pressure above the liquid is the same on each side. When pressure differences are being measured valves C should be closed. The pressure drop through the pipe system will be small at low flow rates, but the hydrostatic pressure may be quite high (enough to force liquid out of the standpipes). Thus the necessity for keeping the valves closed, except when taking readings.

Leave valves A and B closed except when measuring pressure drop through the system.
Figure 1: Standpipe Schematic

Figure 2
Operation Procedure

Notice that valves on the discharge side of a centrifugal pump (NEVER the suction side) must be used to control the rate of flow. The valves on the suction side of any pump, centrifugal or otherwise, must always be left wide open, unless the pump is to be removed for service for some reason. (That will not be the case in your experiment). Close the valves on the pump discharge line and start the pump motor. Then open the valve to either the high or low flow rate meter and adjust the flow rate to the desired level. Readings should be taken in the top 80% of the range of the rotameter. That is, for example, approximately 100, 80, 60, 40 and 20 percent of the maximum rate for that meter. Measurements using the magnetic meter should be taken at several increments (at least five) across the range, up to the maximum flow obtainable. In order to load the motor and pump to the highest level possible, a reading should be taken with fluid being returned directly to the reservoir, largely bypassing the pipe loop.

The following steps outline the operating procedure:

1. Close the valve at the pump discharge line. (Caution: Valve on the suction side of the pump must always remain open)
2. Close the valve at the bypass line from the pump to the feed tank.
3. Note down the pressure gauge reading (static head) at the pump inlet.
4. Open the valve A, B, and C (refer to Figure 2, appendix 3). If the liquid level in both stands pipes are not the same air bubbles are trapped in the line. Remove these air bubbles by following the procedure as given in Appendix 3.
5. Close the valve C, let the valve A & B remain open.
6. Open the valve at the inlet of the rotameter and adjust the flow at the desired level. Take the following measurements:
   a) flowrate
   b) suction and discharge pressure
   c) wattmeter reading

   Run the high flow set up where the point the wattmeter pegs. Then for succeeding flow rates, you will have to compute watts by the following formula:
   \[ \text{Watts} = \text{volts} \times \text{amps} \times \sqrt{3} \]

   Do this for each line (wire) to the motor and then average your results.
   d) voltage
   e) current drawn by the motor (average of all three wires, ammeter scale should be 0-20 mp)
   f) force extension on spring by the motor
   g) inlet and outlet pressure of the pipe loop
7. Vary the flow and take the measurements at least five different flow rates.
8. Close valves A & B and the valve at the inlet of the rotameter.

9. Open line valve at the inlet of magnetic flow meter and adjust the flow at the desired level. Take at least five measurements within full ranges of the meter. Use pressure gauges for measuring the pressure drop across the pipe line.

10. Now open the valve at the bypass line. This will return the fluid directly to the reservoir mainly by passing the pipe line. Use magnetic flow meter for measuring the flow.

11. Adjust the valve at the inlet of magnetic flow meter such that flow rate is more than maximum flow obtained during the flow through the pipe line. Take all the above measurements (except forces drop of course). Take measurements at least five different flow rates across the ranges from this flow rate up to maximum flow possible.

12. Note down the electrical and mechanical measurements when motor is disconnected from the pump. For this step procedure as given in appendix 4 should be followed.

13. Note down the specific gravity and temperature of the fluid in the feed tank at the beginning and at the end of the experiments.
ASSIGNMENT

OBJECTIVES:
You are requested to define focused objectives for this experiment which represent the task(s) (i.e., the assignments) to be performed.

NOTE: The purpose of the enclosed list of assignments and discussions is to provide you with guidance and ideas. It is not meant to be followed as is nor to propose an equivalent amount of work. You may define only one objective or one task that, based on its results and findings, you can give the reader an important message that reflect the purpose of the experiment.

PURPOSE:
In a separate subsection you will need to state, based on your defined objective(s), the purpose of the experiment which poses the problem to be solved and the scope.

Note: The above apply for the pump as well as for the pressure drop in the pipe system.
ASSIGNMENT:

For pre-Lab proposal, create flow diagram (P&ID - Process and Instrumentation Diagram) showing process lines, components, instrumentation, etc. correctly labeled (i.e. valves type, pump types, etc.). (DO NOT SIMPLY COPY THE ONES PROVIDED IN THE MANUAL WHICH ARE USED ONLY AS A GUIDELINE).

1. Develop the pump performance curves for the pump used in this experiment over a wide range of flow rates.
   a. Show the total head, horsepower input (BHP), efficiency and NPSH as a function of the capacity (GPM) on one graph.
   b. By using the parameters mentioned in (a) above, develop the pump performance curves similar to those shown in Figures 3-36A, B, C. Discuss the result and the trend of the curves.

2. Develop the motor performance curves for the motor used in this experiment over a wide range of flow rates, including no load (i.e. disconnect the motor and the pump shafts; Appendix 4 gives the lockout procedure to be used in disconnecting the pump to enable no load conditions to be measured).
   - show the current supplied to the motor, motor shaft rotational speed, motor efficiency and the power factor as a function of the load as a function of the load on one graph.
   Discuss the results and the trend of the curves.

3.  a) Develop the friction factor chart (log $f$ vs. log Re) for the experiment's piping system over a wide range of flow rates.
   b) Compare the experimental friction factor chart with the calculated one for smooth pipe and for new steel pipe.
   c) Estimate the $\varepsilon/D$ of the experiment's piping system.
   Discuss the results and the trend of the curves and the values obtained.

4. Determine the optimum flow rate delivered by the pump used to the experiment's piping system (neglect all the friction head losses outside the loop of the piping system (Figures 2 and 3 used in this experiment).

5. Discuss whether the viscosity and density of the glycol solution affect the performance of the pump, as compared to the performance reported by the manufacturer using water. If there was a significant difference, state how the performance was affected. (see the reading materials)
RESULTS AND DATA ANALYSIS OUTLINE:

Present the direct and the indirect results that represent the assignment in the form of equations, tables, values, and figures. Meanwhile, provide interpretations of the results (i.e. provide a thorough discussion of findings). The results should be discussed in the light of uncertainties of measurements (i.e. identifying the sources and magnitudes of the experimental errors if any), statistical analysis of experimental data, significance and meaningful of any calculated numbers, trend of the experimental data and models, etc. Data analysis may include, but not limited to, the following discussion outline:

Assignment 1:

a. Discuss the developed pump performance curves. Discuss the experimental data used to develop such curves. Does the curves follow the right trend as discussed previously? State the trend of each curve with explanation (i.e. the total head increase or decrease with flow rate pumped by the pump and if it does, at what GPM value; do the total head decrease or increase noticeably, why such behavior occurs, etc.). For total head calculation, did you account for the static ‘discharge’ and suction heads? Did you neglect them in your calculation? Why? Are they significant or not? If you neglect them, what impact do you expect on the results, trend, etc.? Discuss the relationships among these curves. In other words, do they depend on each other? How?

b. State why performance curves similar to that presented by figures 4 and 5 cannot be produced with what you have collected of experimental data. What do you need to develop such curves? Can the graph of performance curves (Fig. 4 and 5) be used for another different piping systems, etc.? Explain.

Assignment 2:

Discuss the developed motor performance curves and their experimental data obtained. Does the experimentally obtained curves follow the proper trend as discussed previously? State the trend for each curve with explanation in the same manner mentioned in Assignment (1) above. Discuss the relationships among these curves. Do they depend on each other? How and why?

Assignment 3:

a. - Discuss the developed friction factor chart and the results, used to evaluate it. Does it follow the right trend. Have you considered the friction head losses outside the piping system loop (identify the items outside the piping system loop that friction head loss would occur). Does this affect the results and the trend? How much? Explain.

- For laminar flow regime compare the trend of the experimental $f$ against $Re$ on log-log scale with the theoretical one (i.e., is $f$ a function of $Re$, roughness or both?). Explain. Does the roughness affect the friction factor or not. If the roughness is so large, what kind of effect are you expecting to get. What is the slope of $f$ vs. Re (on log-log scale). Does this value make sense? If not, why and what should it be? Explain.
- For turbulent flow regime, discuss the relationship between the friction factor and the Reynolds number and the roughness. Identify the sources of roughness inside a pipe.

b. Discuss the comparison of the experimental friction factor curve with the calculated ones for smooth pipe and for new steel pipe. Is this comparison reliable? What do the results tell regarding the age of the piping system, the degree of the roughness. What is it about the effects of the experimental errors (identify them if any) on such comparison and conclusion?

c. Is the evaluated ε/D for the experimental piping system low or high. What does this mean? Is the equation used suitable for the lab experimental system? Is this agreeable with the conclusion deduced in (b). If ε/D is high based on your judgment, identify what makes it high.

Assignment 4:

Explain and discuss how the optimum flow rate is obtained. Is this an optimum flow rate for the lab experimental system only or for any other piping system? Explain. What do you get if you operate the pump at flow rate lower or higher than the optimum? How does the optimum flow rate affect the pump selection or design (i.e. what will happen if the pump size is over estimated or under estimated for a process)?

How does the negligible friction head losses affect the obtained optimum flow rate? Explain and discuss your expectation and judgment based on identifying the parts that would produce some friction loss compared to that accounted for in the piping system.

Discuss also how the discharge and suction static head affect the optimum flow rate evaluation. Have you accounted for these or not? If not, is this a valid assumption? Why didn't you account for them, etc.?

Assignment 5:

Explain and discuss the effect of the fluid viscosity and the density based on the information provided in the appendices regarding this subject. Compare the developed performance curves with that reported by the manufacturer for water. Based on the reading material provided, you should be able to distinguish between the experimental errors and the effect of viscosity and density when you make a comparison (i.e. developed and manufactured performance curves).
References


FIGURE 1: Flow schematic for pressure drop in pipe & pump & motor performance
Figure 2
Isometric - WP Loop
Figure 3
Pipe Loop

Plan

Elevation

33'-9"

18'-10"

REV. 11/14/80

1-11-75
Figure 5
Goulds Pump Perf. Curve 1750 rpm

1750 RPM.

1750 RPM.
Figure 6
Density of Glycol solutions

Densities of Aqueous Ethylene Glycol Solutions (percent by weight)

Degrees Centigrade

Density, Grams Per ml

Pounds Per Gallon

Temperature, Degrees Fahrenheit
Figure 7
Viscosities of Glycol Solutions

Degrees Centigrade

Temperature, Degrees Fahrenheit

Viscosity, Centipoises

-100 -50 0 50 100 150 200

-70 -60 -30 -10 10 30 50 70 90

100% BY WEIGHT GLYCOL
95%
90%
80%
70%
60%
50%
40%
30%
20%
10%
0%
-100%