EXPERIMENT NUMBER 7
PERFORMANCE TEST OF A CENTRIFUGAL PUMP

OBJECTIVE
The primary objectives of this experiment is to measure the performance of a centrifugal pump and compare the results to the manufacturers specifications. Secondary objectives are to familiarize the student with the characteristics of a centrifugal pump and to introduce the student to the homologous scaling relationships. In addition, this experiment gives the student further exposure to the use of computerized data acquisition systems. Additional objectives may be specified by the instructor.

REFERENCES
White, F.: Fluid Mechanics, McGraw-Hill

INTRODUCTION
The most common machine for causing liquids to flow through piping systems is the centrifugal pump. The centrifugal pump is well suited to situations requiring moderate to high flowrates and modest increases in head (or pressure). Typical applications include municipal water supply systems, circulating water heating and cooling system for buildings, pumps in dishwashers and clothes washing machines and the cooling water circulating pump in an automobile engine. For very high pressure, low flow applications, positive displacement pumps are more suitable. Positive displacement pumps move fluids with pistons, gears or vanes and flowrate is a function of rotational speed and has little if any dependence on the pressure rise. A common application of a positive displacement pump is that used to supply high pressure oil for hydraulic actuators such as those on a large earth moving machines.
A sketch of a typical centrifugal pump is shown in Figure 7.1. Fluid which flows into the impeller at the inner radius is given significant angular momentum and kinetic energy as it flows radially outward. After the fluid leaves the outer radius of the impeller, it is diffused or slowed down resulting in a significant increase in pressure.

The actual head rise (H) produced by a centrifugal pump is a function of the flowrate (Q). It is possible to determine the head-flow relationship by appropriate selection of the geometry of the impeller blades. Normally, pumps are designed so that the head decreases with increasing flow since such a design results in a stable flowrate when the pump is connected to a piping system. A typical head flow curve for a pump is shown in Figure 7.2.

If the mechanical energy equation is applied between two points in a piping system on opposite sides of the pump, the result is:

\[
H_p + \frac{V_1^2}{2g} + \frac{P_1}{\gamma} + z_1 = \frac{V_2^2}{2g} + \frac{P_2}{\gamma} + z_2 + h_l
\]

\(H_p\) is the pump head, and \(h_l\) is the total head loss in the associated piping. If section 1 is located at the pump inlet and section 2 at the pump outlet, the \(h_l\) term in the mechanical energy equation is zero. If there is include piping, either upstream or downstream, the \(h_l\) term may be greater than zero.

For a given pump operating at a given rotational speed, there is only one operating point where the geometry of the impeller blades is optimum. This fact, combined with other losses results in an efficiency which is a function of the flowrate. The efficiency is defined as the ratio of the fluid work to the shaft power input to the pump:

\[
\eta = \frac{\gamma Q H_p}{P_{\text{shaft}}}
\]

The performance of centrifugal pumps is only partially determined by analysis - testing is usually required. The performance of a pump is dependent on the impeller and casing geometry, the size of the pump, the rotational speed and the properties of the flowing fluid.

Fortunately, it is not necessary to vary all of these parameters in order to determine the performance of pumps. Two geometrically similar pumps which have flowrates adjusted so that the ratio of tangential to radial fluid velocities are the same are said to be homologous. More specifically, two pumps are said to be homologous if they are geometrically similar and the following two dimensionless parameters are the same:
\[ \Pi_1 = \frac{Q}{ND^3} \]
\[ \Pi_2 = \frac{gH}{N^2D^2} \]

N is the angular speed and D is the diameter. The diameter is a measure of the length scale of the pump. A larger diameter implies that all the other dimensions of the pump are proportionally larger. The diameter used is usually the outer diameter of the impeller.

With these relationships, it is possible to estimate the performance of one pump with one diameter by testing another pump with a different diameter.

It is also possible to determine the effect of a change in angular speed. If a pump is tested at for example at 1800 rpm, then it is possible to estimate the performance at 3600 rpm.

The scaling permitted by the homologous pump relationships isn't perfect. For example, the absolute roughness height of the casing may be a function of the manufacturing process and so the relative roughness might be a function of scale. This problem, together with other loss effects means that errors of a few percent may result from the application of the homologous machine relationships.

**THE EXPERIMENTAL APPARATUS**

The apparatus consists of a pump, motor, flow control valve, shaft torque meter, pressure transducers at the inlet and the outlet and two inline paddle wheel flowmeters as shown in Figure 7.3. Only one of the flowmeters is shown in Figure 7.3. This is a 2" unit which measures all the flow that goes through the pump. Unfortunately, this flowmeter is unreliable at flowrates of less than about 15 gpm. Fluid velocity in the pipe is insufficient to overcome friction in the paddle wheel. However, there is a smaller, 3/4" flowmeter in the system which can be used at lower flows. This unit is shown in the system schematic for the Pipe Friction Experiment, Experiment 3. It only measures the
flowrate through the three smallest pipes used for that experiment. To make use of it, the valves in the test rig must be set so that all of the pump flow goes through these smaller pipes. This small flowmeter can then be used at low pump flowrates. This flowmeter is not valid at flows greater than 20 gpm.

The pump is a Price Pump Co. Model B-15, size 1-1/2x2x6. The size information indicates a 1-1/2 inch outlet pipe and a 2 inch inlet pipe with a nominal maximum impeller diameter of 6 inches. Our pump has a 5.88 inch impeller. Manufacturer's data for this pump is contained in Figures 7.4 and 7.5. The motor is a two speed type that can operate at either a nominal 1800 or 3600 rpm depending on the setting of the connected control. The pressure transducers and the two flowmeters are connected to an IBM PC controlled data acquisition system. Each of the devices which are connected to the DAS has a voltage output which is proportional to the magnitude of the measured variable. See the above information on the validity of the flowmeter data.

The shaft torque meter is a shaft which twists in torsion in linear proportion to the applied torque. This twist is measured by a scale on the rotating device. The torquemeter can be read using a strobotachometer flashing at the shaft rotational speed. The strobotach can also be used to measure the shaft rotational speed.

Both the inlet and outlet pressure taps are located on two inch schedule 40 plastic pipe which has an inside diameter of 2.067 inches. The actual outlet diameter of the pump is 1.610 inches and there is a sudden expansion to 2.067 inches and then short section of 2.067 pipe before the outlet pressure tap. You may want to evaluate these sources of head loss when using the mechanical energy equation to evaluate the pump head.

A bourdon type pressure gage is also connected to the outlet pressure tap. This gage can be used to check the accuracy of the outlet pressure transducer. The accuracy of the inlet pressure transducer can be checked by applying the Bernoulli equation between the water surface in the supply tank and the location of the inlet pressure tap. Since the pressure and velocity at the water surface are both zero, the Bernoulli equation takes the form:

\[ z_1 = \frac{P_{t1}}{\gamma} + \frac{V_{t1}^2}{2g} \]

where \( z_1 \) is the elevation of the water surface relative to the inlet pressure tap and \( P_{t1} \) and \( V_{t1} \) are the pressure and velocity in the pipe at the inlet pressure tap.

Part of the same flow system is a tank placed on a mechanical scale as shown in Figure 1 of the pipe friction experiment. Using appropriate valves, the flow can be diverted to this weigh tank. Together with a timer, this system can be used to calibrate the 2" paddle wheel flow meter.
Figure 7.4 - Head vs. flow for Price Pump Model B15, 1 1/2 x 2 x 6, at 3600 and 1800 rpm. Manufacturers data sheet.

Figure 7.5 - Efficiency vs. flow for Price Pump model B15, 1 1/2 x 2 x 6, with 5.88" dia. impeller for 1800 and 3600 rpm. Based on mfg. data.
REQUIRED TEST DATA

The electrical output of the pressure transducers and the flowmeter might drift in time. Furthermore, the output of the electrical devices might be affected by loose connections. As a result, it is desirable to perform some initial calibration runs to check out these devices.

Turn on the data acquisition system. At the C> prompt, type CD\DAS and then ACQUIRE. At the prompt for a configuration file, type PUMPCAL.CNF. Enter a group name at the prompt and enter 0 for the sampling rate. The data acquisition system is now set up to take the calibration data. Take a first run (with the pump still not running). The program will prompt for the readings of the bourdon gage, the water weight and the time to collect the water. Read the bourdon gages and type in the value. For this run, the amount of water collected is zero and type in any non-zero time. Take another run with these same conditions.

The remaining calibration runs are taken with the pump running. The instructor will explain how to turn on the pump and operate the weigh tank. Take four runs, two at the maximum possible flow and two at flow about half the maximum flow. NOTE: During these runs, the flowrate will change during the run since the head in the upstream dump tank will vary. Take the computer data about halfway through the run by watching the volume of water in the supply tank. You will have to practice each run (without using the computer) to determine when to take the data.

When you are done taking the data, print out the results and save them on a floppy disk.

On the printout, compute the flowrate from the weigh tank data. Compare the weigh tank data to the paddle wheel flow meter data. It should agree to within 5%. Compare the bourdon gage reading to the pressure transducer reading. They should agree within 5%. If the agreement is within these limits, you may assume the original calibration is valid. If it is moderately more, you will have to factor in some correction factors into your results (when you reduce the data at a later time). If the deviation is very large, check with the instructor, the equipment may be defective.

After completing the calibration runs, you are ready to take the pump performance data. You are to take data at a range of flows for both 1800 and 3600 rpm.

Run ACQUIRE again. Use the configuration file PUMPEXP.CNF. For each pump speed, take runs for at least eight settings of the flow control valve from a flow just above shutoff to the maximum flow. Make sure the flows are taken at fairly uniform increments of flow. Take extra runs if needed. At each valve setting, take more than one run to minimize the effects of data scatter. For each run, it will be necessary to read the torquemeter and obtain the shaft speed using the strobotachometer. For runs at the same control valve setting, it will not be necessary to read the strobotachometer and torquemeter more than once but you must type them into the computer for each run. When done, print out your results and save them on a floppy disk. At some point in the run process, you should measure and record the temperature of the water. This completes the data taking.
REQUIRED RESULTS

Our main objective in this experiment is to compare the measured performance to the manufacturers specifications (see attached sheets). A secondary objective is to evaluate the validity of the homologous pump scaling relationships.

The manufacturers specifications are for 1800 rpm and 3600 rpm. You will notice that the test data values of rpm vary and are generally less than the manufacturers numbers. We will correct for these small variations using the homologous pump relationships. The corrected values of head and flow can be computed from:

\[
Q_{\text{corr}} = Q_{\text{test}} \left( \frac{N_{\text{mfg}}}{N_{\text{test}}} \right) \\
H_{\text{corr}} = H_{\text{test}} \left( \frac{N_{\text{mfg}}}{N_{\text{test}}} \right)^2
\]

where \( N_{\text{mfg}} \) is the rpm in the manufacturers data.

Create a plot of \( H \) vs \( Q \) showing both the manufacturers data and your corrected data for 3600 rpm. Create a similar plot for 1800 rpm.

Next, you are to prepare a plot of efficiency vs. flow for the 3600 rpm data. To do this you must compute the shaft power. This can be computed from:

\[
P_{\text{shaft}} = 0.008727 T N
\]

Where the torque \( T \) is in in-lbf and the speed \( N \) is in rpm. \( P_{\text{shaft}} \) then has units of ft-lbf/sec. The efficiency is then:

\[
\eta = \frac{Q H}{P_{\text{shaft}}}
\]

where \( Q \) is in ft\(^3\)/sec. For this equation, the original data values of \( H \) and \( Q \) are used (not the corrected values). Prepare a plot of \( \eta \) versus corrected \( Q \) for both your data and the manufacturers specifications. Since you corrected the data using the homologous relations, the efficiency should be the same at both the original operating point and the corrected point which corresponds to the manufacturers data.

Finally, you are to use the homologous relationships and the 1800 rpm data to predict the H-Q performance at 3600 rpm. This process is the same as above. The resulting data should be plotted on the 3600 rpm curves prepared previously. Thus at 3600 rpm, you should have three curves - the manufacturers data, your 3600 rpm data and your 1800 rpm data corrected to 3600 rpm.

In the discussion, you should at a minimum address the following:

a) Causes of deviations between data and manufacturers specifications
b) Accuracy and characteristics of the instrumentation
c) Validity of homologous pump scaling in this situation
d) Validity of the test procedure and possible improvements.

OPTIONAL RESULTS

The instructor will specify one or more of the following options for this experiment.
Option I - Uncertainty of Pump Flow and Head

For this experiment, you are to produce a curve of pump head versus flowrate of the water. Noting that the pump head, \( H_p \) is the negative of \( w_s \), the work term in the mechanical energy equation, the mechanical energy equation can be rearranged to obtain an expression for

\[
H_p = \frac{P_2 - P_1}{\gamma} + \frac{V_2^2 - V_1^2}{2g} + z_2 - z_1
\]

\[
H_p = \frac{P_2 - P_1}{\gamma} + \left[ \frac{Q}{\pi D_2^2 / 4} \right]^2 - \left[ \frac{Q}{\pi D_1^2 / 4} \right]^2 + z_2 - z_1
\]

For a run in the middle of the data range, you are to determine the uncertainty in the pump head at the higher speed (3600) and also the uncertainty in the corresponding flowrate, \( Q \). You will first have to correct your data for certain systematic errors:

1. Zero offset in the pressure transducers. These can be estimated from a static run (pump not running) when both pressures are simply due to the static head of the water. (measure)

2. Pressure loss in the discharge line between the pump and the pressure transducer. (analysis)

You will need to determine both the Bias limit and the Precision limit for each of the measured variables.

Precision uncertainties:

The only major precision uncertainties are associated with the pressure and the flowrate measurements. The precision uncertainties of the pressures and the flowrate can be determined by taking repeated measurements at a fixed test condition. This is an auxiliary test and can be performed using the data acquisition system and the data taking configuration file. Instead of taking the data manually, use 10 samples per second. From these data you can determine the sample standard deviation of the measurements.

Bias uncertainties:
Bias uncertainties are associated with all the terms in the mechanical energy equation above.

For pressures, the manufacturers specifications of accuracy can be used to estimate the bias uncertainty. This is +0.5% of full scale. For the inlet pressure transducer, full scale is 15 psig and for the outlet full scale is 100 psig.

For the flowrate, the bias uncertainty can be obtained by plotting the flowmeter flow versus the true flow (from the weigh tank measurements). Fit a best fit line to obtain a linear function which relates the true value versus the reading. You can obtain the accuracy limits by drawing straight lines parallel to the best fit line such that all the data is enclosed between the parallel lines. (see the 300 text). You can treat the accuracy limits as the bias limit of the measurement. Note we are double counting the precision error by this method but this is a common approach in uncertainty analysis.

You will have to measure the change in elevation and estimate the uncertainty. You will have to make estimates of the uncertainties in the pipe diameters.

For each measured variable, you should be able to determine the uncertainty. For example, for Q:

\[ w_o = \sqrt{B_o^2 + (tS_o)^2} \]

When all this is done, you should be able to state that  \( H_p = H_p \pm w_{H_p} \) and  \( Q = Q \pm w_Q \). See if the manufacturers data overlaps the error bands produced by this uncertainty analysis.

**Option 2 - Stability of Pump Flow and Head**

In some parts of the flow range, centrifugal pumps may tend to surge - that is show moderate variations of flow and head over time even though valves and the electric power input are constant. The objective of this option is to determine how pronounced these effects are over the range of pump operation.

We will examine this problem by selecting four different values of flow and then taking multiple measurements of flow and output pressure over a period of time at each selected flow. We can then take the standard deviation of the pressure and flow and see if these values depend on flow.

This problem is complicated by the fact that the signal for outlet pressure is rather noisy anyway. First, there is a lot of electrical noise in the signal which has been partially damped out by an electrical low-pass filter. There is also variability in the flow and head...
due turbulence in the piping. Consequently, the effect that we are seeking may be
masked to some extent.

We will make measurements at max flow, close to zero flow and two values of flow
equally spaced in between. You will need to set up a configuration file for the acquire
program. This file will be similar to the file for the experiment (pumpexp.cnf) except that
you will not configure the channels for the manual inputs. The only channels of interest
will be the pump outlet pressure and the mass flowrate. The calibration constants
(a0,a1 and a2) can be set to zero for all other channels.

Set the flow to close to zero flow but allow a very small flow to dribble through in order
to keep the pump cool. Load acquire and the configuration file that you have created.
Set the sampling rate at 10 samples per second and take 200 samples. Save the data
on a file. Restart ACQUIRE and take data for the next flow and then repeat for the
remaining two values of flow.

Load each of the files into a spreadsheet and find the standard deviation of the flowrate
($S_{\text{mdot}}$) and the outlet pressure ($S_{P2}$) for each file. Record the mean value of the flow
and the outlet pressure.

Next, make a plot of $S_{\text{mdot}}$ and $S_{P2}$ versus flowrate.

Examine this curve and see if the standard deviations show any clear variation with
flowrate.