EXPERIMENT NUMBER 6
PERFORMANCE TEST OF AN IMPULSE TURBINE

OBJECTIVES

The primary objective of this experiment is to determine the characteristics of a small impulse turbine and to compare these characteristics to those predicted by theory. Optional objectives may be specified by the instructor.

REFERENCES

Engr 304 text (Potter and Wiggert: Mechanics of Fluids, Prentice Hall.)
White, F. Fluid Mechanics, McGraw-Hill
Engr. 300 text (Wheeler and Ganji: Introduction to Engineering Experimentation, Prentice Hall.)

INTRODUCTION

Approximately 12% of the electric power in the United States is generated from the energy contained in falling water. Hydroelectric power plants produce electricity at lower cost than any other source and produce essentially no atmospheric pollution. Hydroelectric power is not, however, completely harmless from an environmental standpoint. The construction of dams will flood previously available land and have a significant effect on the flow in downstream rivers. Nor is Hydroelectric power completely safe. Dams do occasionally collapse, causing considerable destruction and loss of life in the downstream regions.

Hydroelectric power is generated by transporting water from a lake or reservoir at a high elevation through a pipe called a penstock to a water turbine. The water is then discharged into a downstream river or lake. The water turbine drives an electric generator (Figure 1).

Water turbines are generally placed into two main categories - reaction turbines and impulse turbines. In the reaction turbine, some of the available energy in the entering water is converted into kinetic energy in a stationary
nozzle called a wicket gate (Figure 2). The remaining fluid energy is converted into kinetic energy in the rotating part of the turbine called the runner (impeller). At the outlet of the runner, the water pressure and velocity are very low so almost all the fluid energy has been converted to shaft work. In the United States, most hydroelectric power is generated using a type of reaction turbine called a Francis turbine because this type of turbine is most economical for the moderate head drop (100-500 ft) found at most US hydroelectric sites.

In situations found in very mountainous places where very high values of head (1000 ft or more) are available, another type of turbine called the impulse turbine becomes practical (Figure 3). Impulse turbines are also simpler to construct using unsophisticated manufacturing techniques and were more popular in times past and in developing countries. In the impulse turbine, all the energy available in the fluid is converted into kinetic energy in a nozzle. The resulting jet strikes a double cup shaped blade on a rotor creating a force which turns the rotor (Figure 3). The most common impulse turbine is called a Pelton Wheel.

A small Pelton wheel is set up in the laboratory. This turbine has all the elements of full scale turbines. The impulse turbine has two main features making it desirable for a laboratory demonstration. First, the student can readily see the flow and how the turbine operates. Secondly, the analysis of an impulse turbine is simpler than that of a reaction turbine.

**THEORETICAL BASIS OF THE IMPULSE TURBINE**
The analysis is discussed in the 304 text in some detail. A brief summary is given here. As shown in the 304 text, the tangential force on a moving cascade of vanes (Figure 4) is given by:

\[ F_t = F_x = \rho Q(V_j - u)(1 - \cos \theta) \]  

(1)

where \( V_j \) is the jet velocity, \( u \) is the tangential velocity of the buckets, \( Q \) is the water flowrate, \( \theta \) is the vane turning angle (typically about 175° for a Pelton Wheel) and \( \rho \) is the fluid density. The tangential bucket velocity \( u \) is simply \( r \omega \) where \( r \) is the distance from the shaft to the bucket center and \( \omega \) is the angular velocity of the shaft. The power generated is simply the force times the velocity or:

\[ P_{wr} = F_t u = \rho Q u (V_j - u)(1 - \cos \theta) \]  

(2)

The ideal efficiency is the power obtained above divided by the energy available in the jet which is \( QV_j^2/2g \):

\[ \eta_{ideal} = u(V_j - u)(1 - \cos \theta)/(V_j^2/2) \]  

(3)

The maximum power and efficiency is obtained when the water leaving the bucket is as close to zero as possible (in a stationary reference frame). This can be demonstrated by differentiating equation 2 wrt \( u \) and setting the resulting expression to zero (to maximize the power). The resulting maximum power is obtained when \( u = V_j/2 \). For an ideal turbine then the maximum power is:

\[ P_{wr max} = \rho QV_j^2 / 4(1 - \cos \theta) \]

The maximum ideal efficiency is then \((1 - \cos \theta)/2\) which for typical values of \( \theta \) of 175° is an efficiency close to 1. That is, almost all the available energy in the fluid jet is converted to shaft power in the turbine. For shaft speeds different from the optimum speed, however, the ideal efficiency is less than 1.

This analysis neglected any losses in the buckets and the efficiency calculated is an ideal efficiency. To evaluate the performance of an actual turbine, we need to consider the ratio of the actual power output (as measured) to the available energy in the water supply. Since the nozzle is part of the turbine system, the losses in the nozzle must also be considered. The efficiency of an actual turbine is the ratio of the actual power output to the available energy in the fluid upstream of the nozzle:

\[ \eta_{actual} = P_{wr meas} / \gamma QH_t \]

where \( H_t \) is the total head upstream of the nozzle \((V^2/2g+P/\gamma+z)\). When operating at the optimum angular speed, the actual efficiency of large Pelton wheels is high, on the order of 85-90%.

THE EXPERIMENTAL APPARATUS
The test apparatus consists of a water storage tank, a pump and orifice meter and the Pelton wheel turbine as shown in Figure 5. The orifice meter has been calibrated and the flowrate \( Q \) through the turbine can be determined by measuring the orifice pressure drop and looking up the flowrate in Figure 6. The Pelton wheel has a variable area nozzle. At the inlet to the nozzle is a pressure gage which can be used to measure the inlet pressure which can be used to determine the jet velocity as shown below. The power output of the turbine goes directly into a dynamometer known as a Prony Brake which in this case physically resembles an automotive disc brake. This is the most unsophisticated of laboratory dynamometers and converts the power of the turbine to heat by direct friction. The force \( F_D \) applied to the dynamometer can be measured (in Newtons) on an attached scale.

**TEST PROCEDURE**

You are to measure to the performance of this turbine as a function of shaft speed at a fixed value of the inlet flowrate \( Q \) and inlet head \( H_t \). Set the valve used to vary the nozzle area to a position specified by the instructor. Undo the knob that adjusts the dynamometer load so that the disk turns freely. Open the valve just upstream of the pressure gage. Close all the valves on the horizontal pipes of the flow friction experiment. Close the valves to the upper weigh tank and to the lower dump tank. Turn on the pump to high speed. Make sure that the load reading on the dynamometer scale is zero using the zero adjuster.

You are to make a series of at least seven runs from no load where the speed is a maximum to maximum load where the speed is close to but not exactly zero. The first run is the no load run. The load is adjusted by turning the adjusting knob on the dynamometer caliper.

For each run, measure the inlet pressure and the \( \Delta h \) across the orifice, the dynamometer load and the turbine rotational speed. The rotational speed is determined

![Figure 5 - Pelton wheel test setup](image)
with a strobotachometer. At the very low rpm's, it may be necessary to count the revolutions and measure time with a timer. It should be noted that the inlet pressure and the orifice $\Delta h$ will be approximately constant for all runs. In reducing the data, you should use an average value of each.

**REQUIRED RESULTS**

You are to prepare plots of both measured and ideal torque, power and efficiency as a function of rotational speed.

The torque output of the dynamometer is the measured force times the effective radius on the disk at which the friction force is applied. This effective radius $R$ is 0.083 m. The torque $T$ is:

$$ T = F_D R $$

The power is this torque times the the angular velocity:

$$ P_{wr} = T\omega = F_D R\omega $$

The actual efficiency is determined from:

$$ \eta_{actual} = \frac{P_{wr \text{ meas}}}{\gamma QH_t} $$

In order to determine the actual efficiency of the turbine system, it is necessary to determine the total head at the inlet to the nozzle. The following analysis applies to the sketch of the actual system in Figure 7. The pressure is measured at a point in the inlet pipe where the diameter is $D_1$ and the area is $A_1$. We will assume that the inlet total head occurs at section 2, which has a larger diameter. The mechanical energy equation can then be applied between point 1 and 2:

$$ V_1^2 / 2g + P_1 / \gamma + z_1 = V_2^2 / 2g + P_2 + z_2 + h_{1-2} = H_1 + h_{1-2} $$

We can define $z_1 = 0$ since section 1 is at the same elevation as the nozzle. $V_1 = Q/A_1$.

Between sections 1 and 2 is a sudden expansion which results in a significant loss. The loss due to a sudden expansion is given by:

$$ h_{\text{exp}} = \frac{V_1^2}{2g} (1 - A_1 / A_2)^2 $$

Ignoring losses other than the sudden expansion loss, equation 4 can be used to compute the inlet total head. By applying the mechanical energy equation between sections 2 and $j$ and neglecting losses, it can be shown that $V_j = \sqrt{2gH_j}$. Actual nozzles have some friction in them. The actual jet velocity will be less than calculated above but we will not try to estimate this difference here but will examine it in the discussion.

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**Figure 7 - Inlet to flow nozzle (top view)**

- Water Supply from Pump
- Shutoff Valve
- Pressure Gage
- Nozzle
- Variable Area Nozzle

$D_1 = 0.94''$
$D_2 = 1.55''$
In order to determine the theoretical power, it is necessary to know the distance from the turbine axis to the center of the bucket. For the test turbine, this distance is 1.93 inches.

**Figure 6** - Orifice meter calibration

**OPTIONAL RESULTS**

The instructor may specify one or more of the following options for this experiment.

**Option I - Uncertainty Analysis of Power and Efficiency**
There are a number of major error sources in this experiment - in fact this experiment uses some instrumentation which is not highly accurate. In this option, you are to estimate the errors in the instruments and compute the uncertainty estimates for to the measured power and efficiency. To do this, you will need to obtain uncertainty estimates for the angular velocity, inlet total head, flowrate and torque.

One measurement is of very high accuracy - the measurement of angular speed using the strobotachometer. The flashing of the light is very accurate - it is controlled by a quartz crystal clock. There is some error in determining that the mark on the disk is stationary but the error from this source is negligible compared to other errors in the experiment. You can neglect uncertainty in measurement of the angular speed.

There are two sources of uncertainty in the determination of the inlet pressure to the turbine - the uncertainty in the reading of the gage and the uncertainty in the correction due to the minor losses between the gage and the turbine inlet.

The gage is not a precision gage - it probably has a bias uncertainty (B) of ±5% of full scale. There may be some precision error in the measurement. You can estimate this by observing any oscillations in the gage during a test. For practical purposes, you will average these oscillations visually but this averaging process is itself rather uncertain. You can estimate the precision limit, P, as 10% of the maximum oscillations. If, for example, the oscillations are ±1 psi, then the precision limit will be ±0.1 psi. The bias and precision errors can be combined to get the uncertainty, w, in the pressure measurement according to the formula:

\[ w_{\text{press}} = (B^2 + P^2)^{1/2} \]  

When computing the inlet head from the measured pressure, it is also necessary to know the the head loss due to the sudden expansion and the fluid velocity. Assume that the head loss uncertainty is ±20% of its value. The uncertainty in the fluid velocity is a function of the uncertainty in the pipe diameter at section 1 and also the fluid flowrate. Neglect the uncertainty in the pipe diameter and assume that the percent uncertainty in the velocity is the same as the percent uncertainty in the flowrate as discussed later. The uncertainty in the inlet head can then be obtained by applying the following equation to Eq. 4 presented earlier:

\[ w_{H_i} = \left( \sum_{i=1}^{n} W_{x_i} \left( \frac{\partial H_i}{\partial x_i} \right)^2 \right)^{1/2} \]

The measurement of the flowrate has a number of uncertainties. First, there are both bias and precision uncertainties in the measurement of the manometer height. Secondly, there is uncertainty in the calibration curve that is supplied.
The manometer height has a bias uncertainty, $B_h$, of at least ±0.2 inches due to difficulty in discerning the top of the mercury column and problems in setting the initial zero. There is also a problem due to oscillations in the height. This precision uncertainty, $P_h$, can be estimated as for the pressure gage above. The bias and precision errors for the manometer height can be combined using Eq. 5 above.

There is some uncertainty in the calibration of the orifice (the supplied calibration curve). Assume that this curve has a bias uncertainty, $B_c$, of ±2% of reading; There is some precision uncertainty, $P_c$, associate with reading the curve - assume this to be ±0.05 lb./sec. The bias and precision errors for the calibration can be combined using Eq. 5 above.

We now need to combine the manometer height and calibration uncertainties to obtain the overall uncertainty in the flowrate. The value of the flowrate is essentially the product of manometer height and the calibration curve. Based on the evaluation of uncertainties for product solutions given in the Engr. 300 text, the overall uncertainty in the flowrate can be obtained from:

$$ \frac{w_m}{m} = \left[ \left( \frac{w_h}{h} \right)^2 + \left( \frac{w_c}{c} \right)^2 \right]^{1/2} $$

where $C$ is the estimated flowrate from the curve. The percent uncertainty in $Q$ is essentially the same as the uncertainty in $m$ since the fluid density is known to high accuracy.

The most uncertain measurement is that of the shaft torque. There is a bias uncertainty in the moment arm (location of the center of the frictional force on the disk), there is bias uncertainty in the calibration of the spring scale and there is a large precision uncertainty in reading the scale due to oscillations in the reading. Assume that the bias uncertainty in the moment arm is ±5%. Let us assume that the bias uncertainty is 3% of the reading. You can estimate the precision error in the scale reading in the same manner as for the pressure gage. The uncertainty in the force reading can be obtained using the appropriate form of Eq. 5 above. The uncertainty in the torque can then be found using an equation similar to Eq. 7 above.

You now have enough information to compute the uncertainty in the power and efficiency using equations similar to Eq. 7 above based on the equations for power and efficiency.

**Option 2 - Determination of Maximum Available Power**

The available power to run the turbine is the energy in the fluid upstream of the turbine, which is determined from $\gamma QH_t$. For our system, $H_t$ depends on the flowrate due to losses in the connecting piping and the characteristics of the pump. In real turbine system, losses in the penstock would produce a similar effect. After completing the data
taking for the regular experiment you are to take some supplementary data. Take at least 6 data pairs (inlet pressure at the gage and orifice manometer Ah) with varying positions of the nozzle flow control valve. Vary the position of the control valve from wide open(100%) to about 10%. You must then perform computations to determine available power versus flowrate. Plot the data and determine the maximum available power.

**Option 3 - Determination of Maximum Efficiency based on Jet Velocity**

The efficiency of a turbine is usually based on the power output and the available fluid power at the inlet and that is what we have done here. Normally losses in the nozzle are only a few percent of the available fluid power but in the case of our small turbine, the losses in the nozzle are very large due to the small passage sizes and the resulting turbine efficiency is rather low. The effect of nozzle friction is to reduce the jet velocity. We could make an estimate of the efficiency of the rest of the system (fluid-bucket interaction, shaft bearings and windage) if we could obtain a better estimate of the actual jet velocity.

As noted in the Theory section, the peak power occurs when the bucket velocity is approximately 1/2 the jet velocity. (In real turbines, the peak efficiency occurs when the bucket velocity is slightly less than 1/2 the jet velocity but we will not consider this fact here). We can then look at the power versus speed curve for our data and find the angular speed for maximum power. We can use this speed and the radius to the buckets to compute u, the bucket velocity. The estimated jet velocity is then 2u.

Recompute the efficiency based on this new estimate for jet velocity. The equivalent total head based on this velocity is $$H_j = \frac{V_j^2}{2g}$$. Make a plot of efficiency versus speed and compare it to the ideal efficiency based on this same jet velocity.