Experiment No. 4

HYDRAULIC IMPULSE TURBINE
(PELTON WHEEL)

I. OBJECTIVE

The performance characteristics of a hydraulic impulse turbine (Pelton wheel) will be investigated. The device will be operated over three different hydraulic heads utilizing four nozzle openings. For each constant hydraulic head and nozzle position, the results will be presented in the form of performance curves of the torque, flow rate, hydraulic-power, brake-power and efficiency as a function of the turbine wheel speed. Additionally, the brake horsepower and efficiency will be graphically presented as a function of the speed ratio for each nozzle position.

II. BACKGROUND

Hydraulic impulse turbine (Pelton wheel) converts the kinetic energy of water into shaft power. Water jet from a nozzle strikes the turbine blades causing the wheel to turn. The source of energy is a high head of water such as obtained by construction of a dam. The process is shown schematically in Fig. 4-1.

The water behind the dam provides an elevation head (potential energy) which results in a combination of pressure and velocity heads at the pressure pipe. The jet of water at the exit impinges upon the vanes of the water wheel, imparting some of its kinetic energy and delivering tangential thrust.

Pelton made an improvement in the blades by devising a shape that would split the water stream into two portions, providing canceling force components in the wheel’s axial directions. See Fig. 4-2. Most practical impulse turbines are Pelton wheels.

The total head of the water - consisting of its pressure and kinetic head components - becomes velocity head as the fluid becomes a moving stream at the nozzle. The total head according to Bernoulli’s equation is:

$$H = \frac{P_1 + \frac{V_1^2}{2g}}{\gamma} = \frac{Q}{A}$$

(4-1)

where

- $$P_1$$ = pressure
- $$V_1$$ = velocity of the fluid
- $$\gamma$$ = fluid specific weight
- $$g$$ = gravitational acceleration
- $$Q$$ = volumetric flow rate
- $$A$$ = outlet pipe internal cross-sectional area

By Bernoulli’s principle, if friction is disregarded, the jet velocity ($$V_j$$) can be written according to Eq. 4-2
\[
\frac{V_j^2}{2g} = \frac{P_1}{\gamma} + \frac{V_1^2}{2g}
\]  

(4.2)

Since \( \gamma = \rho g \)

where \( \rho \) = fluid density

Eq. 2 can be written in the form:

\[
V_j^2 = \frac{2P_1}{\rho} + V_1^2
\]  

(4.3)

or

\[
V_j = \sqrt{(2gH)^{1/2}}
\]

(4.4)

However, in the nozzle there are considerable friction losses. The losses depend on the opening of the nozzle, which can be adjusted. The friction losses are accounted for in Eq. 4 by a coefficient \( C_v \)

\[
V_j = C_v \sqrt{(2gH)^{1/2}}
\]  

(4-5)

Speed Ratio

The peripheral speed of the wheel (U) is at the radius (R), where the water jet impinges the vanes (see Fig. 1). The dimensionless speed ratio \( \phi \) of the wheel is defined as the ratio between the speed and the theoretical velocity of the jet, when friction losses are disregarded.

\[
\phi = \frac{U}{(2gH)^{1/2}}
\]  

(4-6)

The efficiency of the Pelton wheel is limited by several factors. These factors can be generally divided into three parts

- a) ineficiency due to nozzle friction
- b) ineficiency due to wheel friction
- c) other mechanical losses

Nozzle inefficiency results from the resistance to flow in the piping system and the turbulence created as the liquid leaves the nozzle. The wheel efficiency results from the inability of the turbine blades to completely convert the kinetic energy into shaft power. This loss is in the form of the energy carried away by the water leaving the turbine. The power losses, \( P_t \) result from friction in the shaft bearings as well as viscous friction losses.

In the fluid mechanics literature you can find an analytical derivation, showing that the maximum efficiency of an impulse turbine is obtained when the speed of the wheel (U) is half of the velocity of the jet (Vj).

\[
U = 0.5 V_j
\]  

(4.7)
Since the speed ratio $\phi$ considers only the ideal velocity, we can expect deviation from the theoretical maximum efficiency at $\phi = 0.5$.

The efficiency of the total process is the ratio of the output brake-power of the Pelton wheel to the hydraulic power, based on the hydraulic head in the pipe (before the friction in the nozzle reduces it when the jet is generated). The output brake power is,

\[ \text{Brake power} = T \omega \quad \text{where} \quad T = F L \quad (4-8) \]

where $T$ = measured output brake-torque
$\omega$ = shaft rotation speed in rad/sec
$F$ = brake-force measured by a scale
$L$ = length of the brake lever arm

The efficiency is thus

\[ \eta = \frac{T \omega}{Q \gamma H} \quad (4-9) \]

Where,
$Q \gamma H$ = hydraulic power available

The maximum efficiency is achieved when most of the kinetic energy of the jet is transferred into shaft power. In horsepower units the hydraulic horsepower is HHP and the brake horsepower on the shaft is BHP. The efficiency $\eta$ is

\[ \eta = \frac{\text{BHP}}{\text{HHP}} \quad (4-10) \]

III. EQUIPMENT

![Diagram of Impulse turbine (Pelton wheel)](Fig. 4-1. Impulse turbine (Pelton wheel))
The equipment is an impulse turbine with a jet of fluid issuing from a nozzle. The jet impinges on vanes of the turbine wheel (Pelton wheel), thus generating power as the runner rotates, see Fig. 4-1.

IV. PROCEDURE

The apparatus to be used in the experiment is a Pelton wheel test bed, see Fig. 4-1. The working fluid of the test bed is supplied with a total head by an electric pump. Water is pumped through a pipe fitted with a pressure gauge through a nozzle whose opening can be continuously varied from fully open to fully shut by means of a needle valve. The needle travel is marked. In this way, a partial opening can be gauged.

The water jet emanating from the nozzle impinges on a Pelton wheel inside housing. The wheel is connected by shaft to a brake dynamometer arrangement, in which a brake shoe is manually tightened around a rotating drum. A lever arm on the shoe pushes against a force gauge, allowing for the calculation of the output torque. The rotational speed of the disc, as it slips against the brake, can be measured by a hand tachometer applied to fixture on the wheel hub.

The water jet impinging against the Pelton wheel. Then the water is collected by the wheel housing and passes into the front trough. As the water flows back to the pump inlet, it flows over a V-notch cascade flow meter. The volumetric flow rate of water through the system is thus gauged by the height of water at the V-notch.

The test bed will be characterized for three pump heads. For each constant head, the nozzle will be set in four positions: fully, three-quarters, half, and one-quarter open. For each nozzle setting, the system flow rate will be noted, and the wheel speed will be varied by manually adjusting the brake dynamometer. The wheel will be allowed to spin freely, to spin at four slower speeds, and finally will be completely stopped. At each point, brake force and speed will be noted.

V. ANALYSIS

Some of the most important information a user of such a Pelton wheel apparatus would want is torque and efficiency variation with speed and efficiency variation with system capacity.

A typical torque-speed plot is shown at the top of Fig. 4-2. As can be seen, the plot has two speed scales: wheel speed in RPM, and the non-dimensional speed ratio. It is possible to use the two scales simultaneously because, for a given pressure head, the scales are essentially proportional. (Although the velocity head component of the total head does vary with nozzle setting, it changes the total head figure by only a few percent in this experiment). For each of the three total heads there will be a torque-speed graph, consisting of four plots corresponding to each nozzle setting tested.

The second graph in Fig. 4-2 is an efficiency-speed plot. Similar to the torque-speed graphs, these figures also make use of the dual speed scale The speed ratio scale allows us to gauge the effect of jet velocity overestimation on the location of the crests of the efficiency curves. Each efficiency graph will again contain four plots, one for each nozzle setting.

The final graph in Fig. 3 represents line of constant efficiency in a capacity-speed regime. These charts are constructed by collecting points of equal efficiency from the efficiency-speed charts. This would correspond to drawing straight, horizontal lines across the efficiency-seed
rate is constant for constant head and nozzle setting, these points are laid out as shown in Fig. 4.3. The isoefficiency lines combine data taken from four sets of experiments for various nozzle settings.

**Equipment data:**
- Pelton wheel effective radius: 2.0 in.
- Prony brake effective arm: 6.0 in.
- Outlet pipe: 2 in., Schedule 40.

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![Graph of efficiency, power, and torque vs. speed ratio](image)

**Fig. 4-2** Impulse turbine characteristics
Fig. 4-3 Example of drawing of isoefficiency lines for impulse turbines

Use the points of a constant efficiency (horizontal line on the $\eta$-$N$ graph) and transfer the points to the corresponding capacity line on the $Q$-$N$ curve.