KING FAHD
UNIVERSITY OF PETROLEUM & MINERALS

ME 316: Thermofluids Laboratory

Experiment # 6

PELTON IMPULSE TURBINE

1) OBJECTIVES

a) To introduce the operational principle of an impulse hydraulic turbine.
b) To study the performance characteristics of the Pelton impulse turbine.

2) INTRODUCTION

Hydraulic turbines are used to convert fluid energy into mechanical power. Such turbines are widely used in hydroelectric installations and also in energy storage systems. Small hydraulic turbines are also used for energy recovery in desalination plants. The two main types of hydraulic turbines are the reaction and impulse types. In the reaction type, part of the pressure drop occurs in the stationary guide vanes and the other part occurs in the turbine runner, which is completely filled with water. On the other hand, an impulse turbine first converts the high pressure through a nozzle into a high-velocity jet, which then strikes the turbine vanes at one position as they pass by as shown in Figure 1.

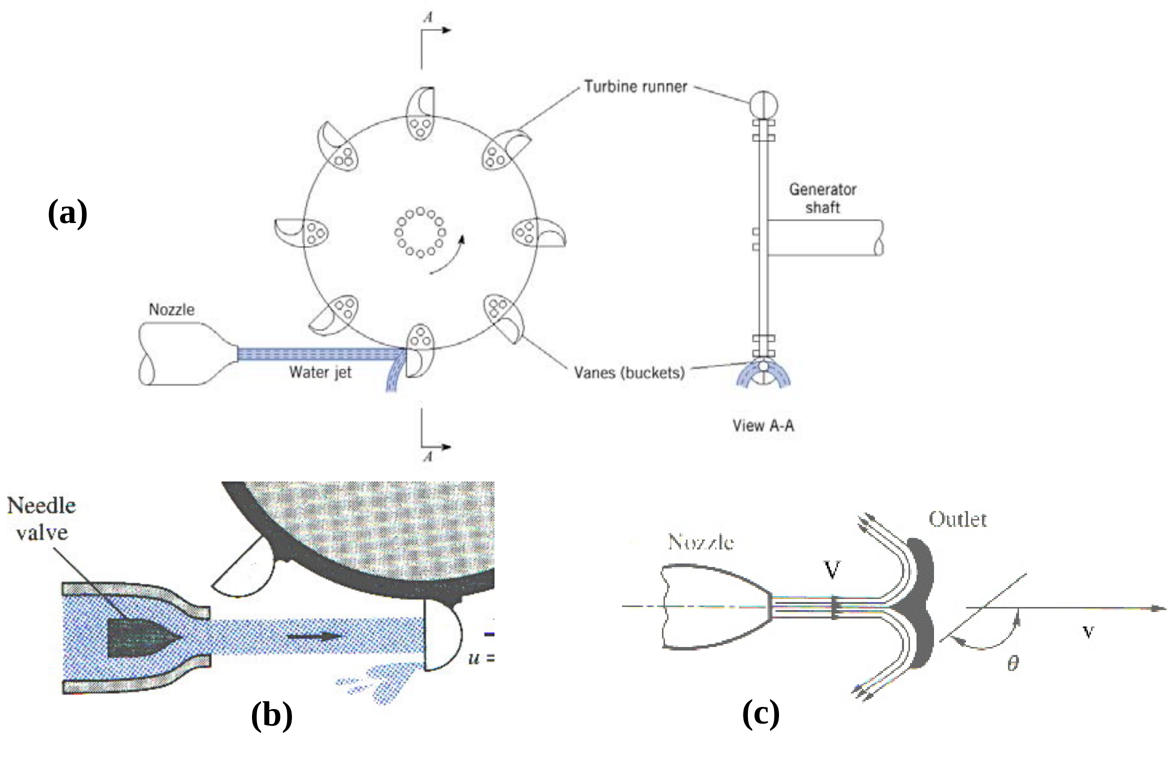

Figure 1. Schematic diagram of the impulse turbine: a) side view of turbine runner and jet; b) section view of the needle valve; c) plan view of the bucket.

Impulse turbines are used in cases of high head and relatively low output power. In these turbines, the high pressure is confined to the nozzle, which converts the head to a high-velocity water jet at atmospheric pressure. The nozzle flow rate is controlled by a needle valve as shown in Figure 1b. The jet strikes the turbines vanes (buckets) and imparts a force due to the change in the fluid linear momentum. The buckets are of a symmetrical double-outlet shape, as shown in Figure 1c, in order to eliminate the thrust on the turbine shaft and bearing.

3) THEORETICAL BACKGROUND

Consider the case of a jet of liquid of velocity V strikes a curved vane moving with velocity v as shown in Figure 2. The vane deflects the liquid stream through an angle θ and V_r represents the velocity of the jet relative to the vane. The fluid leaves the vane with an absolute velocity V_e as shown in the figure. Assuming that F_x represents the x-component of the force exerted by the vane on the fluid, one can apply the momentum equation to obtain

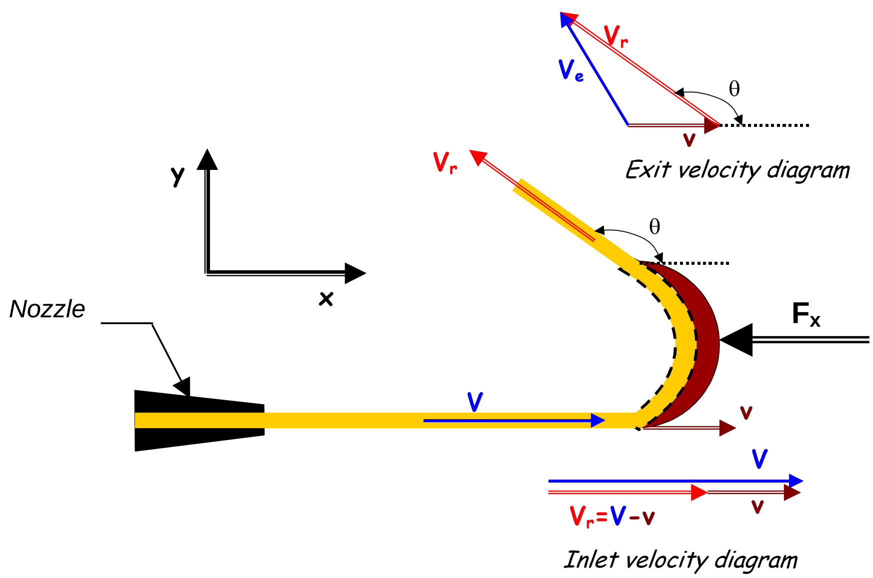

Figure 2. Schematic sketch showing the velocity diagrams at inlet and exit of a moving vane.

$$-F_x = \dot{m}\left(V_{ex} - V\right) \quad \Rightarrow \quad F_x = \rho Q\left(V - V_{ex}\right) = \rho Q\left[V - \left(V_r\cos\theta + v\right)\right]$$

$$or \qquad F_x = \rho Q\left(V - v\right)\left(1 - \cos\theta\right) \tag{1}$$

The force acting on the vane is equal and opposite of F_x. The mechanical power generated in this process is obtained from

$$P = F_x.v = \rho Q\, v\left(V - v\right)\left(1 - \cos\theta\right) \tag{2}$$

In the case of the Pelton wheel, the y-component of the force vanishes because the vane is symmetrical (see Figure 1). According to equation (2), the maximum power is obtained when θ=180°, however, because of design considerations θ ≈165° (the main reason is to avoid having the outgoing fluid impacting the next bucket).

For a perfect nozzle, the entire head would be converted to kinetic energy and this results in $H = V^2/2g \quad or \quad V = \sqrt{2gH}$. However, because of friction losses in the nozzle, the jet velocity becomes

$$V = C_v \sqrt{2gH} \tag{3}$$

where C_v is the velocity coefficient of the nozzle that normally ranges between 0.92 and 0.98. The theoretical impulse turbine efficiency can be expressed as

$$\eta_{th} = \frac{Output\ mechanical\ \ power}{Input\ fluid\ power} = \frac{\rho Q v (V - v)(1 - \cos\theta)}{\gamma Q H}$$

Now, by introducing the peripheral-velocity factor ϕ such that $\phi = v/\sqrt{2gH}$ and using equation (3), one can write the above equation in the form

$$\eta_{th} = 2\phi\,(C_v - \phi)(1 - \cos\theta) \tag{4}$$

The value of ϕ that maximizes the theoretical efficiency can be obtained from

$$d\eta_{th}/d\phi = 0 \quad \Rightarrow \quad 0 = C_v - 2\phi \quad \Rightarrow \quad \phi = C_v/2 \approx 0.47 \tag{5}$$

By assuming typical values for C_v and θ (C_v = 0.94 and θ = 165°), one can obtain a typical maximum value for the theoretical efficiency using equation (4).

$$[\eta_{th}]_{max} \simeq 2 \times 0.47\,(0.94 - 0.47)(1 - \cos 165^\circ) \simeq 86\% \tag{6}$$

The actual maximum efficiency for the Pelton impulse turbine is less than the above value because of mechanical losses, fluid friction on the vane surface, back splashing and the non-uniformity of the flow adjacent to the bucket. The maximum efficiency for this turbine may be slightly below 80%. The variation of the theoretical efficiency with the speed of rotation can be plotted using equation (4) as shown in Figure (3).

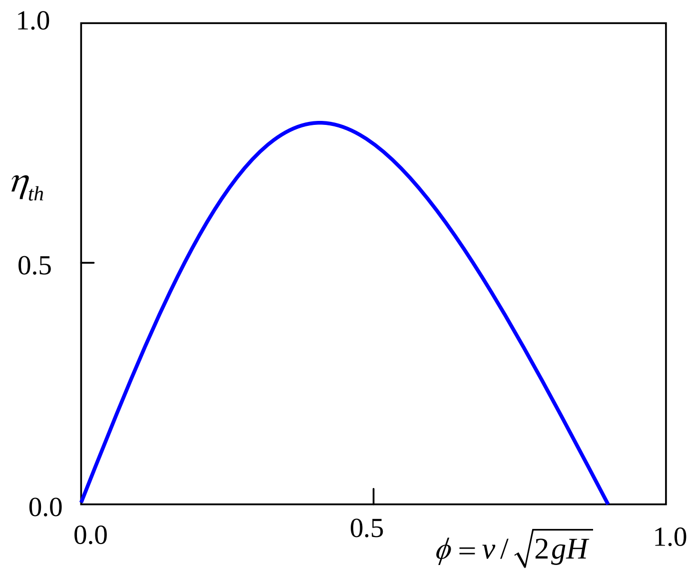

Figure 3. Variation of theoretical efficiency with the peripheral-velocity factor.

4) <u>APPARATUS</u>

The apparatus to be used is the TeQuipment Pelton impulse turbine (model # H19). Figure 4 represents a general view of the apparatus, which consists of the following:

Figure 4. A general view of the nozzle flow apparatus

i) <u>Hydraulic Bench:</u> The bench consists of a water reservoir, a measuring tank and a pump.

ii) Pelton Impulse Turbine: The turbine wheel has 16 vanes (buckets) and is mounted on a shaft that is supported by a bearing on one side and the other side covered by a Plexiglas window.

iii) Nozzle: The fluid enters into the turbine casing through a convergent nozzle equipped with a needle valve for flow rate control.

iv) Pressure gage: The static pressure immediately before the nozzle is measured by a pressure gage.

v) Dynamometer: The apparatus is equipped with a dynamometer for measuring the turbine output torque.

vi) Stroboscope: A stroboscope is used for measuring the turbine speed of rotation.

5) TEST PROCEDURE AND CACULATIONS

a) Test Procedure

1. Ensure that the apparatus is leveled.
2. Adjust the needle valve to the fully closed position and the dynamometer load to the full load.
3. Start the pump of the hydraulic bench.
4. Open the needle valve gradually until reaching approximately the half-open position.
5. Record the reading of the pressure gage and use the hydraulic bench for measuring the flow rate by taking the readings of volume $\forall$ and time t.
6. Reduce the dynamometer load gradually until the turbine speed reaches 100 rpm and record the dynamometer load, F. Use the stroboscope for measuring the turbine speed.
7. Increase the turbine in increments of 100 rpm and record the dynamometer load in each step until reaching the maximum speed (approximately 700 rpm) at which the dynamometer load is reduced to zero.
8. Adjust the needle valve opening to the fully open position and repeat steps 5→7.

b) Calculations

1. Calculate the flow rate, $Q = \forall / t$.
2. Determine the turbine total head from the equation, $H = \frac{p}{\gamma} + \frac{V^2}{2g}$. Since the velocity V in the inlet pipe is very small in comparison with the jet velocity, the velocity head ($V^2/2g$) can be neglected and one can write $H \approx p/\gamma$.
3. For every speed, the torque can be obtained from the dynamometer reading. Torque (N.m) = dynamometer reading × 0.0045
4. Calculate the input fluid power from the equation $P_{fluid} = \gamma QH$.
5. Calculate the output power or brake power from the equation B.P. $= T\omega$.
6. Calculate the actual turbine efficiency from the equation $\eta_{act.} = P_{fluid} / B.P.$
7. Determine the peripheral-velocity factor ϕ from the equation $\phi = \omega R / \sqrt{2gH}$, where R is the radius of the turbine runner (R= 43 mm).

6) PRESENTATION OF RESULTS

1. It is required to plot the actual turbine efficiency $(\eta_{act.})$ versus the peripheral-velocity factor (ϕ) for the two valve openings.
2. Determine the maximum efficiency and the value of ϕ at which it occurs for each opening.
3. Plot the theoretical efficiency curve on the same graph using equation (4) and considering $C_v = 0.94$ and $\theta = 165°$.

7) IDEAS FOR FURTHER DISCUSSIONS

1. What causes the changes in the maximum efficiency between the two cases of needle valve openings?
2. Is there any change in the jet velocity for the two cases of needle valve openings?
3. What is the advantage of plotting $\eta_{act.}$ versus ϕ instead of $\eta_{act.}$ versus the speed of rotation N?
4. Can you use dimensional analysis to obtain an expression for $\eta_{act.}$?