ISSN (ONLINE): 2321-3051

INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING

A COMPARATIVE AND ANALYTICAL REVIEW OF IMPULSE AND REACTION TURBINE ON THE BASIS OF TYPE OF ACTION OF THE WATER ON THE BLADE

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Abstract

This paper deals with analytical review of impulse and reaction turbine on the basis of type of action of the water on the blade. The importance of mechanical drive steam turbine efficiency has been much highlighted in addition to its reliability. Some examples of modification of existing turbines for energy conservation and improving the efficiency are given.

Keywords: impulse turbine, reaction turbine, efficiency.

1. Introduction

In the case of impulse turbine all the potential energy is converted to kinetic energy in the nozzles. The impulse provided by the jets is used to turn the turbine wheel. The pressure inside the turbine is atmospheric. This type is found suitable when the available potential energy is high and the flow available is comparatively low. Some people call this type as tangential flow units. Later discussion will show under what conditions this type is chosen for operation.

In reaction turbines the available potential energy is progressively converted in the turbines rotors and the reaction of the accelerating water causes

the turning of the wheel. These are again divided into radial flow, mixed flow and axial flow machines. Radial flow machines are found suitable for moderate levels of potential energy and medium quantities of flow. The axial machines are suitable for low levels of potential energy and large flow rates. The potential energy available is generally denoted as “head available”. With this terminology plants are designated as “high head”, “medium head” and “low head” plants. Pelton wheel turbine is one of the famous turbines which are mostly used for the hydro plants. This is of impulse type and is used for high head, although the efficiency of such turbines are not so comparable with the reaction turbines, because most of the energy is wasted, due to the split-ting of water which is not being utilized and get wasted, due to this efficiency of the turbine is not so as the efficiency of other turbines like axial or reaction turbines. Related to the same context many modifications were suggested for the enhancement of the efficiency but the major change which was adopted or implemented was around 1903 after it there was no further modification were implemented.

So this work is purely intended to enhance the efficiency of the pelton wheel turbine with the modification in the blade design and some of the auxiliary attachments which ultimately lead to the enhancement of the efficiency of pelton wheel turbine.

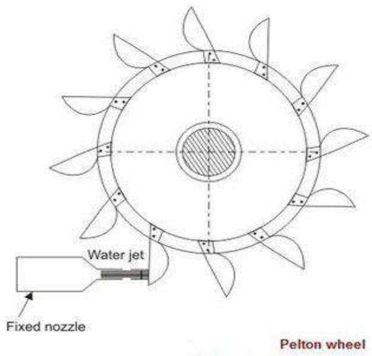

Fig. 1 Outline of pelton wheel

Figure 1 shows the outline of the pelton wheel, down the centre of each bucket, there is a splitter ridge which divides the jet into two equal streams which flow round the smooth inner surface of the bucket and leaves the bucket with a relative velocity almost opposite in direction to the original jet. For maximum change in momentum of the fluid and hence for the maximum driving force on the wheel, the deflection of the water jet should be 180. In practice, however, the deflection is limited to about 165 degree so that the water leaving a bucket may not hit the back of the following bucket. Therefore, the camber angle of the buckets is made as 165 degree. Thus we can't have the camber angle greater than 165 and up to now no consideration have been taken on the velocity of the outgoing fluid leaving on the bucket.

Working principal and hydraulic efficiency of a conventional Pelton-wheel turbine

The water from the reservoir flows through a penstock which contains a nozzle at the outlet. The nozzle increases the kinetic energy of the penstock water. At the outlet this nozzle produces a Water-jet. This Water-jet strikes on the buckets (Vanes) of the runner and transfers its kinetic energy to the bucket's wheel. The general formula of any hydro system is:

$$P = \eta \rho g Q H$$

Where:
P is the mechanical power produced at the turbine shaft (in Watts).
η is the hydraulic efficiency of the turbine, ρ is the density of water in (1000 kg/m^3).
g is the acceleration due to gravity in (9.81 m/s^2).
Q is the volume flow rate passing through the turbine in (m^3/s).
H is the effective pressure head of water across the turbine in (m).

For an impulse turbine of a Pelton-wheel type, the mechanical power can be changed by means of changing η, Q, and H inputs because ρ and g are constant.

The force exerted by the water jet on buckets (vanes) of the runner in the direction of motion is given as:

$$Fx = \rho a V_1[Vw_1 + Vw_2]$$

Here,
ρ = Density of the water.
a = Cross section area of water-jet in m^2 = $p/4d^2$,
d = diameter of jet in m.
V_1 = Velocity of the jet at the inlet of bucket splitter,
V_1 = v2gH in m/s.
H = Net head acting on the Pelton-wheel in m.
g = Acceleration due to gravitation m/s^2.
Vw_1 = Velocity of the whirl at inlet in m/s.
Vw_2 = Velocity of the whirl at outlet in m/s.

The work done by the jet on the runner per second is as:

$$W_j = F_x \times u = \rho a V_1[Vw_1 + Vw_2] \times u \text{ Nm/s}$$

Here,
u = Tangential linear velocity of the bucket wheel at pitch circle in m/s.

The energy supplied to the jet is in the form of kinetic energy which is given as ½ mv^2.

Now,
Kinetic Energy (K.E.) of the jet per second is given as:

$$K.E. = ½ \rho a V_1 \times V_1^2$$

Hydraulic Efficiency η_h = Work done by jet per second ÷ K.E. of jet per second.

$$\rho_h = \rho a V_1[Vw_1 + Vw_2] \times u \div ½ \rho a V_1 \times V_1^2$$

In terms of V_1

$$\rho_h = 2(V_1\text{-}u)[1 + \cos\beta]u \div \tfrac{1}{2}\rho\, a\, V_1 \times V_1^2$$

$$\{Vw_1 = V_1\text{-}u,\ Vw_2 = (V_1\text{-}u)\cos B - u\}$$

For maximum efficiency u is $= \tfrac{1}{2} V_1$

$$\eta_{hMax} = 1 + \cos\beta / 2$$

Here β = Angle of Vane at outlet.

Modified gravitational Pelton-wheel

The modified gravitational Pelton-wheel turbine has been designed for a low-head and heavy-discharge application. In this turbine, buckets from a conventional Pelton-wheel have been modified by additional extra bucket-cups as shown in Figure 2. The bucket-cup is designed like a cattle-pot, where water pouring is done at the top opening and water discharge occurs at another opening somewhat similar to a tea cattle-pot. The position and gap between two consecutive Pelton-wheel buckets and the additional bucket-cup on the runner is such that the water-jet axis does not interfere with conventional Pelton-wheel bucket central point, while striking on the modified bucket cup valve to store water. Similarly, it does not interfere with bucket-cup while striking on the conventional Pelton-wheel bucket. The bucket-cup's water storing action of jet takes place when the bucket cup reaches at P1 position of wheel. The bucketcup valve opens under the action of the jet strike and water gets stored in this bucket-cup. The dotted lines in insert 1 show the action of the jet on valve during filling of the bucket-cup. Inset 2 in Figure 2 exhibits a horizontal position for the bucket-cup when the valve is in a closed position and the bucket is filled with jet water. Position P4 in the figure shows the jet discharge of bucket-cup's stored water. P5 is the position of wheel when the bucket-cup is fully empty. This turbine resembles a ferry-wheel where the P1 position of the bucket-cup wheel is near the potential Headrace at the opening of the nozzle jet exit. Position P1 where the kinetic energy of jet water is stored in the bucket-cup as gravitational potential energy. P5 is the BDC position of the ferry-wheel just near P5, where the bucket-cup has become empty. Basically, the distance between potential Head-race P1 and P5 is the effective working diameter where jet water storage into the bucket-cup starts near nozzle exit and bucket-cup emptying finishes at P5. The bucket-cup pitch-circle-radius and Pelton-wheel bucket pitch-circle-radius is represented by 'x' and 'r' respectively, in Figure 2. Small 'h' is the effective average head of stored water in the bucket-cup under which jet-propulsion take place.

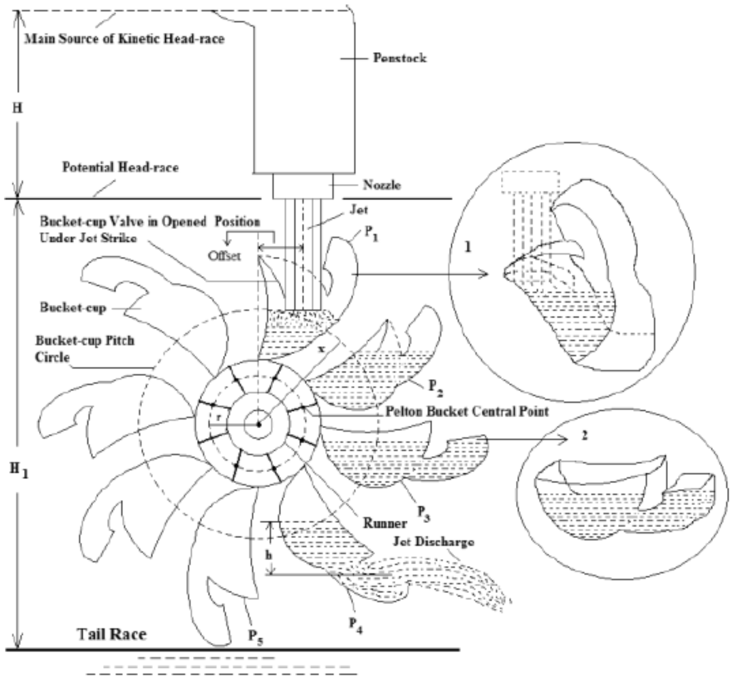

Fig. 2 A modified gravitational Pelton-wheel

Working principle of a modified gravitational Pelton-wheel

The modified gravitational Pelton-wheel is designed in such a way that it consumes the potential head H1 between P1 and P5 in the form of stored water weight in the bucket-cups. The kinetic head is consumed during the striking of the jet on bucket. The location and position of the additional bucket-cup on the conventional Pelton-wheel is such that it consumes jet water at P1. The weight of this stored water per second in the bucket-cup is the amount of potential energy per second from water jet. Further, this potential energy is converted into kinetic energy as this water filled bucket freely (assumed) falls during P1 to P5 position of the wheel. It is also possible to extract kinetic energy from the penstock nozzle discharge by means of a jet strike on the conventional Pelton-wheel vanes of the runner.

Blade and stage design

Turbine blades are of two basic types, blades and nozzles. Blades move entirely due to the impact of steam on them and their profiles do not converge. This results in a steam velocity drop and essentially no pressure drop as steam moves through the blades. A turbine composed of blades alternating with fixed nozzles is called an impulse turbine, Curtis turbine, Rateau turbine, or Brown-Curtis turbine. Nozzles appear similar to blades, but their profiles converge near the exit. This results in a steam pressure drop and velocity increase as steam moves through the nozzles. Nozzles move due to both the impact of steam on them and the reaction due to the high-velocity steam at the exit. A turbine composed of moving nozzles alternating with fixed nozzles is called a reaction turbine or Parsons turbine.

Except for low-power applications, turbine blades are arranged in multiple stages in series, called compounding, which greatly improves efficiency at low speeds. A reaction stage is a row of fixed nozzles followed by a row of moving nozzles. Multiple reaction stages divide the pressure drop between the

Vol.2 Issue.3, March 2014. Pgs: 1-13

steam inlet and exhaust into numerous small drops, resulting in a pressure-compounded turbine. Impulse stages may be either pressure-compounded, velocity-compounded, or pressure-velocity compounded. A pressure-compounded impulse stage is a row of fixed nozzles followed by a row of moving blades, with multiple stages for compounding. This is also known as a Rateau turbine, after its inventor. A velocity-compounded impulse stage (invented by Curtis and also called a "Curtis wheel") is a row of fixed nozzles followed by two or more rows of moving blades alternating with rows of fixed blades. This divides the velocity drop across the stage into several smaller drops. A series of velocity-compounded impulse stages is called a pressure-velocity compounded turbine.

By 1905, when steam turbines were coming into use on fast ships (such as HMS Dreadnought (1906)) and in land-based power applications, it had been determined that it was desirable to use one or more Curtis wheels at the beginning of a multi-stage turbine (where the steam pressure is highest), followed by reaction stages. This was more efficient with high-pressure steam due to reduced leakage between the turbine rotor and the casing. This is illustrated in the drawing of the German 1905 AEG marine steam turbine. The steam from the boilers enters from the right at high pressure through a throttle, controlled manually by an operator (in this case a sailor known as the throttle man). It passes through five Curtis wheels and numerous reaction stages (the small blades at the edges of the two large rotors in the middle) before exiting at low pressure, almost certainly to a condenser. The condenser provides a vacuum that maximizes the energy extracted from the steam, and condenses the steam into feed water to be returned to the boilers. On the left are several additional reaction stages (on two large rotors) that rotate the turbine in reverse for astern operation, with steam admitted by a separate throttle. Since ships are rarely operated in reverse, efficiency is not a priority in astern turbines, so only a few stages are used to save cost.

Velocity compounding of Impulse Turbine

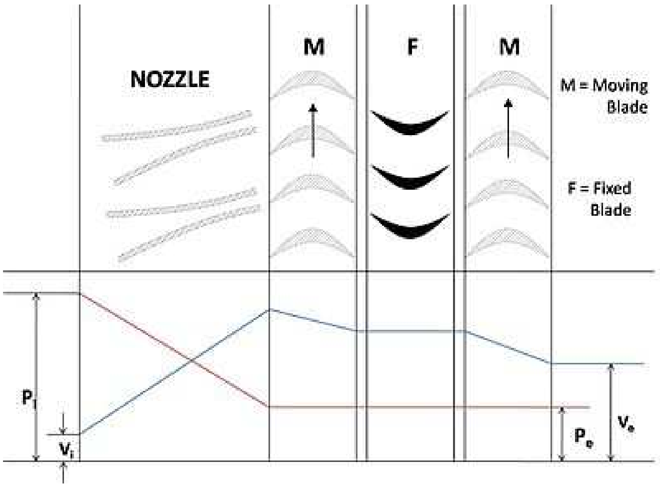

Fig.3 Schematic Diagram of Curtis Stage

Impulse Turbine

The velocity compounded Impulse turbine was first proposed by C G Curtis to solve the problem of single stage Impulse turbine for use of high pressure and temperature steam.

The rings of moving blades are separated by rings of fixed blades. The moving blades are keyed to the turbine shaft and the fixed blades are fixed to the casing. The high pressure steam coming from the boiler is expanded in the nozzle first. The Nozzle converts the pressure energy of the steam into kinetic energy. It is interesting to note that the total enthalpy drop and hence the pressure drop occurs in the nozzle. Hence, the pressure thereafter remains constant. This high velocity steam is directed on to the first set (ring) of moving blades. As the steam flows over the blades, due the shape of the blades, it imparts some of its momentum to the blades and losses some velocity. Only a part of the high kinetic energy is absorbed by these blades. The remainder is exhausted on to the next ring of fixed blade. The function of the fixed blades is to redirect the steam leaving from the first ring moving blades to the second ring of moving blades. There is no change in the velocity of the steam as it passes through the fixed blades. The steam then enters the next ring of moving blades; this process is repeated until practically all the energy of the steam has been absorbed.

A schematic diagram of the Curtis stage impulse turbine, with two rings of moving blades one ring of fixed blades is shown in figure 3. The figure also shows the changes in the pressure and the absolute steam velocity as it passes through the stages.

where,
P_i = pressure of steam at inlet
V_i = velocity of steam at inlet
P_o = pressure of steam at outlet
V_o = velocity of steam at outlet

In the above figure there are two rings of moving blades separated by a single of ring of fixed blades. As discussed earlier the entire pressure drop occurs in the nozzle, and there are no subsequent pressure losses in any of the following stages. Velocity drop occurs in the moving blades and not in fixed blades.

Disadvantages of Velocity Compounding

- Due to the high steam velocity there are high friction losses
- Work produced in the low-pressure stages is very less.
- The designing and fabrication of blades which can withstand such high velocities is difficult.

Pressure compounding of Impulse Turbine

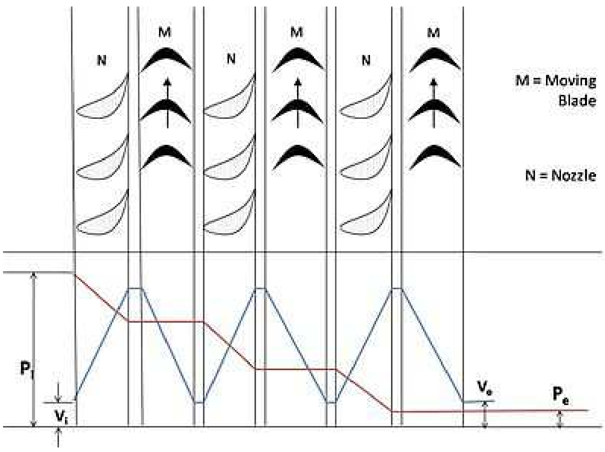

Fig. 4 Schematic Diagram of Pressure compounded Impulse Turbine

The pressure compounded Impulse turbine is also called as Rateau turbine, after its inventor. This is used to solve the problem of high blade velocity in the single-stage impulse turbine.

It consists of alternate rings of nozzles and turbine blades. The nozzles are fitted to the casing and the blades are keyed to the turbine shaft.

In this type of compounding the steam is expanded in a number of stages, instead of just one (nozzle) in the velocity compounding. It is done by the fixed blades which act as nozzles. The steam expands equally in all rows of fixed blade. The steam coming from the boiler is fed to the first set of fixed blades i.e. the nozzle ring. The steam is partially expanded in the nozzle ring. Hence, there is a partial decrease in pressure of the incoming steam. This leads to an increase in the velocity of the steam. Therefore the pressure decreases and velocity increases partially in the nozzle.

This is then passed over the set of moving blades. As the steam flows over the moving blades nearly all its velocity is absorbed. However, the pressure remains constant during this process. After this it is passed into the nozzle ring and is again partially expanded. Then it is fed into the next set of moving blades, and this process is repeated until the condenser pressure is reached.

It is a three stage pressure compounded impulse turbine. Each stage consists of one ring of fixed blades, which act as nozzles, and one ring of moving blades. As shown in the figure pressure drop takes place in the nozzles and is distributed in many stages.

An important point to note here is that the inlet steam velocities to each stage of moving blades are essentially equal. It is because the velocity corresponds to the lowering of the pressure. Since, in a pressure compounded steam turbine only a part of the steam is expanded in each nozzle, the steam velocity is lower than of the previous case. It can be explained mathematically from the following formula i.e.

$$\frac{V_1^2}{2} + h_1 = \frac{V_2^2}{2} + h_2$$

where,
V_1 = absolute exit velocity of fluid
h_1 = enthalpy of fluid at exit
V_2 = absolute entry velocity of fluid
h_2 = enthalpy of fluid at entry

Disadvantages of Pressure Compounding

- The disadvantage is that since there is pressure drop in the nozzles, it has to be made air-tight.
- They are bigger and bulkier in size.

Pressure-Velocity compounded Impulse Turbine

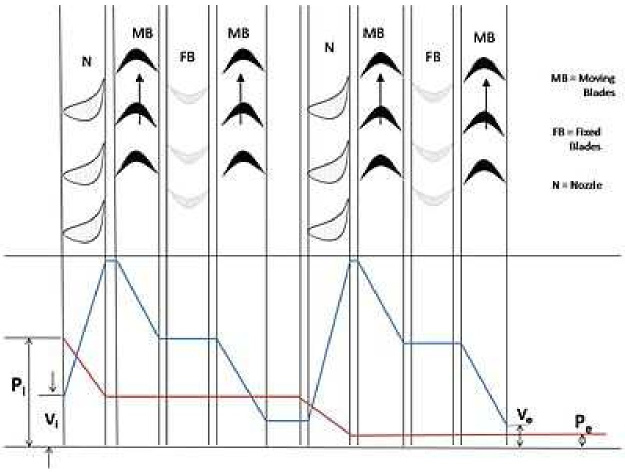

Fig. 5 Schematic Diagram of Pressure-Velocity compounded Impulse Turbine

It is a combination of the above two types of compounding. The total pressure drop of the steam is divided into a number of stages. Each stage consists of rings of fixed and moving blades. Each set of rings of moving blades is separated by a single ring of fixed blades. In each stage there is one ring of fixed blades and 3-4 rings of moving blades. Each stage acts as a velocity compounded impulse turbine.

The fixed blades act as nozzles. The steam coming from the boiler is passed to the first ring of fixed blades, where it gets partially expanded. The pressure partially decreases and the velocity rises correspondingly. The velocity is absorbed by the following rings of moving blades until it reaches the next ring of fixed blades and the whole process is repeated once again.

Pressure compounding of Reaction Turbine

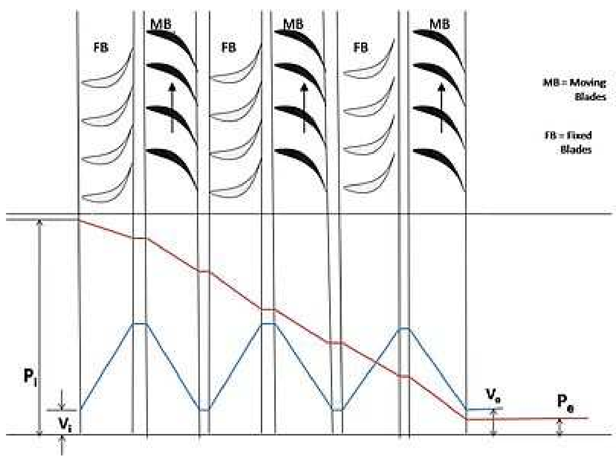

Fig.6 Schematic Diagram of Pressure compounded Reaction Turbine

As explained earlier a reaction turbine is one which there is pressure and velocity loss in the moving blades. The moving blades have a converging steam nozzle. Hence when the steam passes over the moving blades, it expands with decrease in steam pressure and increase in kinetic energy.

This type of turbine has a number of rings of moving blades attached to the rotor and an equal number of fixed blades attached to the casing. In this type of turbine the pressure drops take place in a number of stages.

The steam passes over a series of alternate fixed and moving blades. The fixed blades act as nozzles i.e. they change the direction of the steam and also expand it. Then steam is passed on the moving blades, which further expand the steam and also absorb its velocity.

Turbine efficiency

To maximize turbine efficiency the steam is expanded, doing work, in a number of stages. These stages are characterized by how the energy is extracted from them and are known as either impulse or reaction turbines. Most steam turbines use a mixture of the reaction and impulse designs: each stage behaves as either one or the other, but the overall turbine uses both. Typically, higher pressure sections are reaction type and lower pressure stages are impulse type.

An impulse turbine has fixed nozzles that orient the steam flow into high speed jets. These jets contain significant kinetic energy, which is converted into shaft rotation by the bucket-like shaped rotor blades, as the steam jet changes direction. A pressure drop occurs across only the stationary blades, with a net increase in steam velocity across the stage. As the steam flows through the nozzle its pressure falls from inlet pressure to the exit pressure (atmospheric pressure, or more usually, the condenser vacuum). Due to this high ratio of expansion of steam, the steam leaves the nozzle with a very high velocity. The steam leaving the moving blades has a large portion of the maximum velocity of the steam when leaving the nozzle. The loss of energy due to this higher exit velocity is commonly called the carry over velocity or leaving loss.

The law of moment of momentum states that the sum of the moments of external forces acting on a fluid which is temporarily occupying the control volume is equal to the net time change of angular momentum flux through the control volume.

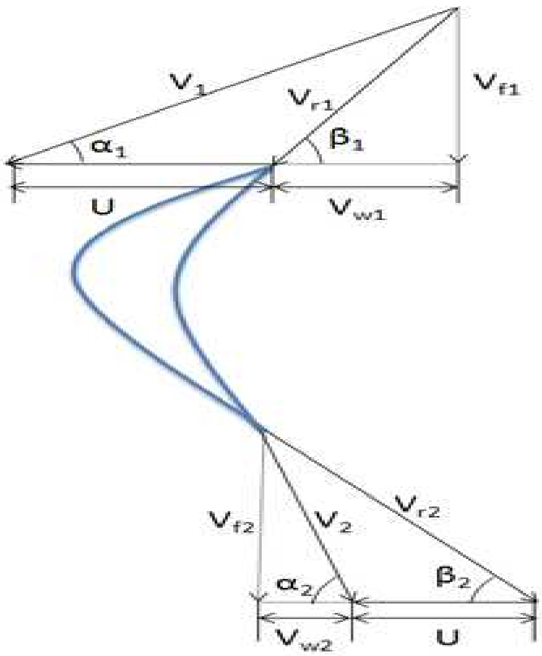

Fig.7 Velocity triangle

The swirling fluid enters the control volume at radius r_1 with tangential velocity V_{w1} and leaves at radius r_2 with tangential velocity V_{w2}.

A velocity triangle paves the way for a better understanding of the relationship between the various velocities. In the adjacent figure we have:

V_1 and V_2 are the absolute velocities at the inlet and outlet respectively.
V_{f1} and V_{f2} are the flow velocities at the inlet and outlet respectively.
$V_{w1} + U$ and V_{w2} are the swirl velocities at the inlet and outlet respectively.
V_{r1} and V_{r2} are the relative velocities at the inlet and outlet respectively.
U_1 and U_2 are the velocities of the blade at the inlet and outlet respectively.
α is the guide vane angle and β is the blade angle.

Then by the law of moment of momentum, the torque on the fluid is given by:

$$T = \dot{m}(r_2 V_{w2} - r_1 V_{w1})$$

For an impulse steam turbine: $r_2 = r_1 = r$. Therefore, the tangential force on the blades is $F_u = \dot{m}(V_{w1} - V_{w2})$. The work done per unit time or power developed:

$$W = T * \omega.$$

When ω is the angular velocity of the turbine, then the blade speed is $U = \omega * r$. The power developed is then $W = \dot{m}U(\Delta V_w)$.

Blade efficiency

Blade efficiency (η_b) can be defined as the ratio of the work done on the blades to kinetic energy supplied to the fluid, and is given by

$$\eta_b = \frac{Work\ Done}{Kinetic\ Energy\ Supplied} = \frac{2UV_w}{V_1^2}$$

Stage efficiency

A stage of an impulse turbine consists of a nozzle set and a moving wheel. The stage efficiency defines a relationship between enthalpy drop in the nozzle and work done in the stage.

$$\eta_{stage} = \frac{Work\ done\ on\ blade}{Energy\ supplied\ per\ stage} = \frac{U\Delta V_w}{\Delta h}$$

Where $\Delta h = h_2 - h_1$ is the specific enthalpy drop of steam in the nozzle.

By the first law of thermodynamics: $$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

Assuming that V_1 is appreciably less than V_2, we get $\Delta h \approx \frac{V_2^2}{2}$ Furthermore, stage efficiency is the product of blade efficiency and nozzle efficiency, or $\eta_{stage} = \eta_b * \eta_N$

Nozzle efficiency is given by $\eta_N = \frac{V_2^2}{2(h_1 - h_2)}$,

where the enthalpy (in J/Kg) of steam at the entrance of the nozzle is h_1 and the enthalpy of steam at the exit of the nozzle is h_2. $\Delta V_w = V_{w1} - (-V_{w2})$ $\Delta V_w = V_{w1} + V_{w2}$

$\Delta V_w = V_{r1}\cos\beta_1 + V_{r2}\cos\beta_2$ $\Delta V_w = V_{r1}\cos\beta_1 (1 + \frac{V_{r2}\cos\beta_2}{V_{r1}\cos\beta_1})$

The ratio of the cosines of the blade angles at the outlet and inlet can be taken and denoted $c = \frac{\cos\beta_2}{\cos\beta_1}$. The ratio of steam velocities relative to the rotor speed at the outlet to the inlet of the blade is defined by the friction coefficient $k = \frac{V_{r2}}{V_{r1}}$.

$k < 1$ and depicts the loss in the relative velocity due to friction as the steam flows around the blades ($k = 1$ for smooth blades).

$$\eta_b = \frac{2U\Delta V_w}{V_1^2} = \frac{2U(\cos\alpha_1 - U/V_1)(1 + kc)}{V_1}$$

The ratio of the blade speed to the absolute steam velocity at the inlet is termed the blade speed ratio

$$\rho = \frac{U}{V_1}$$

η_b is maximum when $\frac{d\eta_b}{d\rho} = 0$ or, $\frac{d}{d\rho}(2\cos\alpha_1 - \rho^2(1 + kc)) = 0$. That implies $\rho = \frac{\cos\alpha_1}{2}$ and therefore $\frac{U}{V_1} = \frac{\cos\alpha_1}{2}$. Now $\rho_{opt} = \frac{U}{V_1} = \frac{\cos\alpha_1}{2}$ (for a single stage impulse turbine)

Therefore the maximum value of stage efficiency is obtained by putting the value of $\frac{U}{V_1} = \frac{\cos\alpha_1}{2}$ in the expression of η_b/

We get: $(\eta_b)_{max} = 2(\rho\cos\alpha_1 - \rho^2)(1 + kc) = \frac{\cos^2\alpha_1(1 + kc)}{2}$

For equiangular blades, $\beta_1 = \beta_2$, therefore $c = 1$, and we get $(\eta_b)_{max} = \frac{cos^2\alpha_1(1+k)}{2}$. If the friction due to the blade surface is neglected then $(\eta_b)_{max} = \cos^2\alpha_1$.

Conclusions on maximum efficiency

$$(\eta_b)_{max} = \cos^2\alpha_1$$

1. For a given steam velocity work done per kg of steam would be maximum when $\cos^2\alpha_1 = 1$ or $\alpha_1 = 0$.

2. As α_1 increases, the work done on the blades reduces, but at the same time surface area of the blade reduces, therefore there are less frictional losses.

Conclusion

The issue concerned in all explanations of impulse and reaction turbine. It is concluded that reaction turbine having better performance as compared to the impulse turbine. In terms of efficiency of the turbine, the reaction turbine has higher efficiency.

From the above all the literature review and analytical study I am much more interested to analytical work on the reaction turbine.

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