AN EFFECT OF SURFACE FINISH AND SPACING BETWEEN DISCS ON THE PERFORMANCE OF DISC TURBINE

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Abstract – the turbine, invented by Nikola Tesla (1856-1943), is a bladeless turbine. Tesla disc turbine and a flexible test rig have been designed and manufactured, and experimental results are presented. An analysis of the performance and efficiency of the disc turbine is carried out. The design philosophy of the flexible test rig has been explained. Notice that there are no blades whatsoever – parallel, closely spaced discs used. Resistance to fluid flow between the plates results in energy transfer to the shaft. High velocity water enters the disk pack through inlet nozzle path tangent to the outer edge of the discs. Convergent nozzle imparts high velocity water jet tangentially on disc thickness. Lower-energy water spirals toward the central exit port, adhesion, drag and centrifugal forces continue to convert kinetic energy to shaft rotational power. The results of the study represent the step towards development boundary layer turbine. It has been determined that surface roughness and spacing affects the performance of the multiple disc turbines significantly. Efficiency may be improved at least up to 45%, which has been deemed achievable by Professor Warren Rice [2].

***Keywords** - boundary layer Turbine, disc Turbine, bladeless, Tesla.Turbine.*

I. INTRODUCTION

When an airplane flies through the air, at very high speed, there is a thin layer of air that sticks to the wing. This layer of air goes the same speed as the airplane. There is then shear action or shear plane between that boundary layer and the surrounding quiescent air around the aircraft. At that shear plane is where the drag is formed that holds the aircraft back. Aerodynamic tells us that if we could, wave a magic wand over an air craft and, eliminate or minimize boundary layer drag the aircraft could fly 40% faster or further with same amount of horse power. In aerodynamics, boundary drag layer is totally unwanted precept**.** But, Tesla was able to turn that precept around 180degrees. Tesla perceived that boundary layer drag used to do something useful.

High velocity water enters tangentially to outer periphery of the disk pack through inlet nozzle; it forms boundary layer on either side of discs. The pressure ratio is pushing it towards the center of the turbine. It forms a helical path down into the center of the device, and exit in the center, after it has transmitted all of its energy to the discs through the boundary layer drag.

In this present work, A Tesla disc turbine and a flexible test rig designed and manufactured by simple stock material. Experimental results carried out by using water medium. The overall design of the turbine is very flexible allowing parameters to be varied in order that their effect on the performance of the turbine can be measured. It is possible to change the number of discs, disc spacing.

II. LITERATURE REVIEW

H.S.Couto, J.B.Duarte and D.Bastos-Netto [1].reviews the physical principles behind the Tesla Bladeless Turbine, using basic fluid mechanics only, Considered the relative motion of rotating surfaces, it sets up the transport equations describing the flow between parallel rotating disks, estimated the boundary layer thickness under laminar and turbulent regimes, leading to expressions yielding the width between consecutive disks. H.S.Couto explained how to calculate the total number of disks required to attain the desired performance. Finally also described the device behavior acting as an air compressor or water pump and noticed that the as a unit source of rotating motion, these Tesla machines can run under a very wide spectrum of not only fuels but also fluids in general

Nikola Tesla [2] has filed a patent for a Disk Turbine which uses smooth rotating disks inside a volute casing. In his patent, Tesla described the motive forces of his machinery as being dependant on the fluid properties of viscosity and adhesion. Several successful turbines were designed and built by Tesla but they were considered not commercially feasible at the time. Lack of adequate instruments, difficulties in handling inherent speeds, and heat dissipation problems were the reasons for multiple disk devices not being further developed.

Piotr Lampart [3] presented results of the design analysis of a Tesla bladeless turbine intended for a co-generating micro-power plant of heat capacity 20 kW, which was operated in an organic Rankin cycle on a low-boiling medium. Results of investigations exhibit interesting features in the distribution of flow parameters within the turbine inter disk space. The calculated flow efficiency of the investigated Tesla turbine models show that the best obtained solutions can be competitive as compared with classical small bladed turbines.

Warren Rice [4] established most important parameters that affect the performance and efficiency of disc turbine. However, Rice constructed six disks turbine and reports some aspects of them, with the purpose of determining the feasibility of this kind of turbo machinery. Starting from the description of the Tesla´s patent, the turbine was operated with compressed air exhausting to the atmosphere, some changes as the angle of the nozzle and the use of a supersonic nozzle were made. Later, some improvements were made as reducing the gap. The comparison between analytical data and results from these experiment revealed that the geometry, flow rate, and speed combination used in the turbine where not near to those indicated by the analyses for optimum turbine efficiency. Rice observed 24% max efficiency.

K.Boyd and Warren Rice [5] have studied laminar flow of an incompressible fluid between rotating disk. K.Boyd has proposed complete solution of inlet region for various inlet conditions. A complete problem statement is formulated from the Naiver stokes equation with three parameters tangential velocity, flow rate and Reynolds number. Both the laminar and turbulent flow considered and suggested torque, Power, efficiency and Pressure drop are the function of velocity, flow and Reynolds number.

Most of the literature has considered the fluid to be laminar and incompressible. In general, it has been found that the efficiency of the rotor can be very high, at least equal to that achieved by conventional bladed rotors. Nevertheless the Tesla turbine accept all kind of fluids, most of the literature assumed a Newtonian fluid for simplifications except experimental test or those studies with two phases fluid From the literature, we establish the parameters those affects the performance of turbine.

Distance between the discs, Number of the discs, Discs surface roughness, Number of nozzles.

The following assumptions are considered for the experimental work.

i) Steady flow ii) Incompressible fluid iii) negligible body forces iv) Full admission of working fluid at the outer periphery of the discs.

III. THEROTICAL ANALYSIS

Inlet Nozzle construction fabricated as shown in fig 1 for Test Rig. Water incompressible fluid selected for performance measurement with assuming flow is steady flow.

Constant Parameters:

Spacing between Discs: 2mm Medium: Water

Disc thk: 2mm Outlet Nozzle Size: $32mm^2$
Material of Discs: SS304 Disk Dia: 152mm

Inlet Size: $21mm^2$ No of discs: 6

Surface Finish: smooth Medium: Water

Flow rate: 0.22lit/s line Pressure: $18lb/in^2$

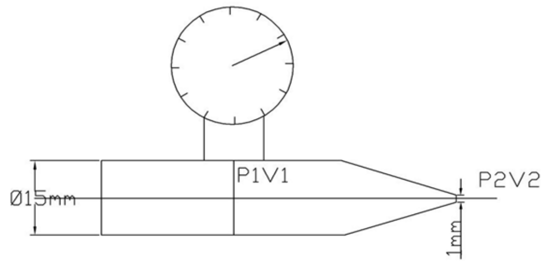

Fig. 1 : Nozzle Arrangement - line diagram for Test Rig

By applying Bernoulli’s equation, at point1 and at point2

$$P_1/\rho g + V_1^{\,2}/2g + Z_1 = P_2/\rho g + V_2^{\,2}/2g + Z_2 \qquad (1)$$

Where,

P_1 = Pressure at point 1 V_1 = velocity at point1 $Z_1 = Z_2$ Pressure head (at same elevation)

P_2 = Pressure at point 2 V_2 = velocity at point 2

ρ = water density $=1000kg/m^3$ $g = 9.81m/s^2$

We know the flow rate i.e. Q = 0.22 lit/s=$0.00021m^3/s$ and $P_1 = 18$ lb/ $in^2 = 1.15$ bar

But, by continuity equation.

$$Q = A_1V_1 = A_2V_2 \qquad (2)$$

Where A_2 = area of cross section at pt 2 = 21 mm^2 and

$\therefore A_1 = \Pi / 4 X D^2 = 3.14 / 4 X 0.0152 = 0.000225m^2$
$V_1 = Q/A1 = 0.00022 /0.000225 = 0.98$ m/s

$V_2 = Q/A_2 = 0.00022/ (22 X 10^{-6}) = 10.48$ m/s

Now put this value of V_1 and V_2 in equation (1) to get P_2. But $Z_1 = Z_2$, both the pt at same level. Now (1) becomes.

$$P_1/\rho g + V_1^{\,2}/2g = P_2/\rho g + V_2^{\,2}/2g$$

$P_2 = [(1.15X10^5 /9810+0.98^2/19.62)-(10.48^2 /19.62)]9810$ = 0.47 Bar

Now P = F/A .

We know the value of P2; Put this value in above equation to get Force acting on discs thickness.

$F = P_2 X A_2 = 69622.74 X 20 X 10^{-6} = 1.532$ N (force acting on 12mm area only)

But, we know the jet force formula from pelton turbine

$$F = \rho A(V - U)^2 \cos \varnothing \qquad (3)$$

Where, A = jet area = A2 = 21 mm^2 , V = velocity of jet = V2, U = Relative velocity of discs. & Ø = 10º angle of jet.

Now put the values in above equation to get U

$\therefore (V - U)^2 = 1.532 / (1000 \times 21 \times 10^{-6} \times 0.98) = 12.47$

$\therefore (V - U) = 8.66$

$\therefore U = 1.81$ m/s

We know the angular momentum for rotating part

$$L = mrU \qquad (4)$$

Where L = angular momentum, r= radius of disk.

$$\text{But, } L = I\omega \qquad (5)$$

Where I = moment of inertia & ω = Angular velocity

In this case,

$$I = I_{SHAFT} + I_{DISCS} \qquad (6)$$

$I_{SHAFT} = MR^2$ Where M= mass of the shaft=1.08kg, R= Radius of shaft = 15mm. (7)

$I_{SHAFT} = 1.082 \times (15/100)^2 \times 9.81 = 0.0238 N/mm^2$

$$I_{DISCS} = 1/2 \times M_d (a^2 + b^2) \qquad (8)$$

Where, M_d = Mass of disk= 0.275 kg, a = inner radius of disc = 30mm, b= outer radius of disc = 76mm.

$\therefore I_{DISC} = 1/2 \times 0.275 (0.015^2 + 0.076^2) = 0.032 Nm^2$

For 6 discs

$\therefore I_{DISCS} = 0.032 \times 6 = 0.192 Nmm^2$

Now from (6)

$I = 0.032 + 0.192 = 0.195 N/mm^2$

now, from (4) & (5) $L = I\omega$

$\therefore \omega = mrU / I = (2.73 \times 0.076 \times 1.81) / 0.195$

$\therefore \omega = 19.20$ rad /s For 6 discs

But,

$$\omega = 2\Pi N / 60 \qquad (9)$$

$N = 19.20 \times 3.75 \times 60 / 6.28 = 183.44$ rpm.

$$\text{Now Torque } T = F \times R \qquad (10)$$

From equation (3) & (10)

$T = 1.532 \times 0.076 = 0.116 Nmm$. Now,

$$P = 2\Pi NT / 60 \qquad (11)$$

By putting the values of N and T, we get

$\therefore P = (6.28 \times 183.44 \times 0.116)/60$

$\therefore P = 2.22$ watt.

IV. EXPERIMENTAL SET UP

The overall design of the turbine is very flexible as shown in fig.2, which allows varying the parameters, in order to measure their effect on the performance of the turbine. The overall features of this turbine Test Rig designed and manufactured according to the experience noted in the reviews.

The disc diameter is 152mm (6in), the thickness of each disc is 2mm, and the rotor-to-housing diametrical clearance is 1mm. An overall view of the turbine can be seen in Fig. 1(a). The discs have a 2 central outlet port, since this configuration was found to be more efficient by Rice [4].

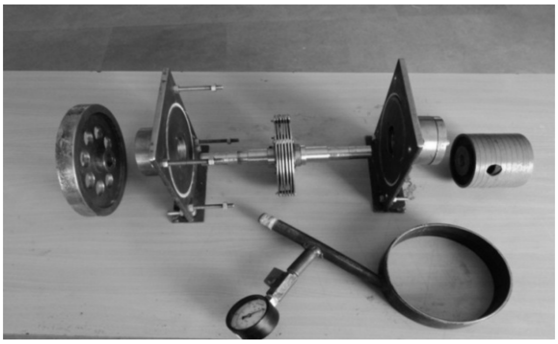

Fig. 2 : Parts of Multiple disc Turbine.

The goal behind the nozzle design is to increase the kinetic energy of the flowing medium at the expense of its pressure and internal energy and the nozzle must provide similar mass flow to each disc & space. Nozzle designed on the basis of energy conservation law and coefficient of discharge like venturimeter, it imparts equal jet on each disc and due to its convergent shape it increases kinetic energy of jet as shown in fig.3.

The diameter of this central hole is 30mm. In order to accommodate the outlet of the fluid, the shaft is supported as a boss by means of bearings inside it

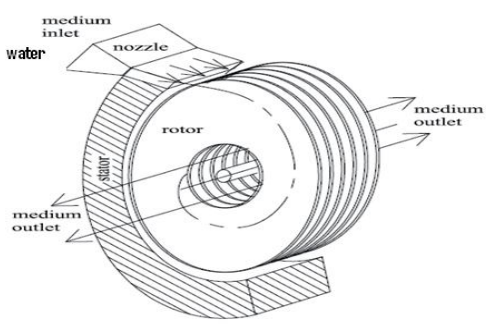

Fig. 3 : Cross Section of Test Rig

The experimental test rig is designed to investigate performance of Turbine is shown in Fig.4.The test rig used in the present research work consists of 3 cylinder piston pump. Rubber O-rings are also provided for air tight joint between casing and side plate. The shaft with disk supported on two bearings.

In previous studies air medium were used, but in this present work we used water medium. And for constant flow rate and to provide high-pressure water we used positive displacement pump as shown in fig.4. A test rig developed in such way that it measures the various parameters that are necessary to determine the performance the Turbine. Flow rate vary from the 10Lit/m to 27lit/m.

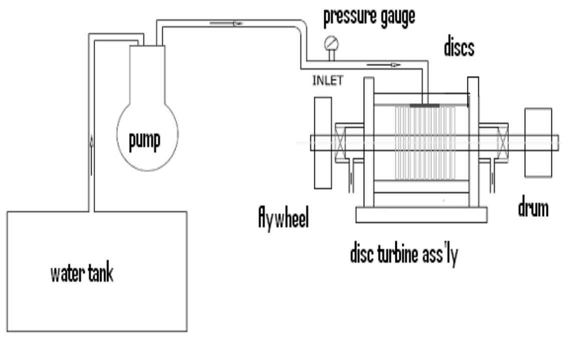

Fig. 4 : Experimental Set up.

In this study 4 variations in the number of discs used for same set up. Experiments carried out 6 discs and spacing 0.5mm, 1mm, 1.5mm, 2mm, 2.5mm. And surface finish varies from smooth to 500ra. Design of Experiments carried out in the following manner. Spiral grooves machined on discs surfaces by using lathe machine and fixture.

TABLE 1 EXPERIMENTAL DESIGN MATRIX

flow	Spacing	Surface finish
+	+	+
-	+	+
+	-	+
-	-	+
+	-	-
-	-	-
+	+	-
-	+	-

Flow + = 27Llit/m and Flow - = 10.2lit/m

Spacing + = 3mm and Spacing - = 0.5mm

Surface finish + =smooth and Surface finish - = 500ra

A. Determination of Speed of the Rotor.

In the test rig used in this research, the speed output (N) can be obtained directly by means of a Taco Meter. The methodology for the use of this device is simple. At the both end the shaft centering operation done which is provision to insert a tip of Taco Meter. This method was found to be accurate and inexpensive. Then angular velocity obtained by using formula.

$$\omega = 2\Pi N / 60 \qquad (12)$$

Using the flexibility for changing parameters that the turbine used in these investigation permits, variations in the number of discs to obtain Performance comparisons data. In this present work five variations in discs spacing and three variations in surface roughness considered to carry out experiment. Same constant parameters as considered in theoretical analysis. Variable parameters are the flow, spacing between discs and surface finish of the discs.

V. RESULT AND DISCUSSION

Nozzle size designed in such way that it imparts equal force on each disc for minimum spacing and also maximum spacing between the discs. After conducting the experiments as per matrix, we noticed that this turbine working efficiently for six discs. Due to the nozzle opening, for three discs jet imparted on $6mm^2$ areas only and for six discs jet area imparted on $12mm^2$ area means 6 disc pack use more jet forces comparatively three discs. Then all the experiments carried out for six disc and by varying spacing like 0mm .05mm, 1mm, 1.5mm, 2mm, 2.5mm

Fig 5a & 5b shows the variation of speed with respective flow. As flow increases speed increases. But spacing increases speed decreases, because for minimum spacing friction or shear force between jet and disc increases, rotating disc form a boundary layer around the wall and jet also form boundary around it. Jet high velocity boundary layer drags the discs. Disc rotating comparatively low speed hence disc trying to oppose jet velocity. Due to this shear forces developed between two boundary layers. And Kinetic energy of jet utilized for disc rotation.

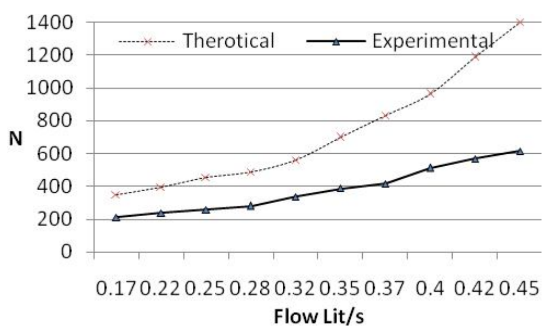

Fig.5a : A plot of Flow v/s Speed for 6 smooth discs and 2.5mm spacing

For 2.5 mm spacing, more amount of jet water passing through the gap comparatively 0.5mm spacing, boundary layer forms between discs, but jet boundary layer and disc boundary layer apart from each other. Jet uses the maximum kinetic energy to carry the fluid between discs. And flow rate difference is also observed at the outlet. For minimum spacing flow rate is comparatively less than maximum spacing.

For 0.5 mm spacing, boundary layer forms between discs, but jet boundary layer and disc boundary layer overlap with each other. Jet uses the maximum kinetic energy to drag discs.

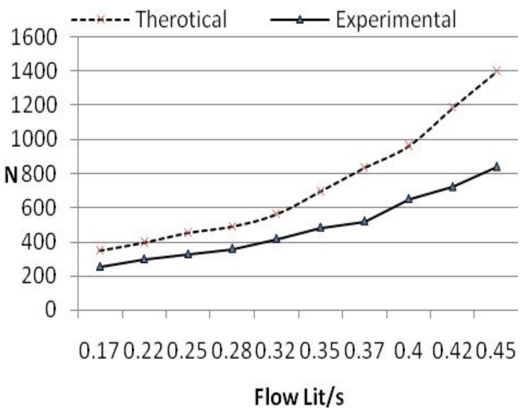

Fig. 5b : A plot of Flow v/s Speed for 6 smooth discs and 0.5mm spacing

Fig.6 shows the speed variation with respective flow, Here 6 rough (spiral grooves) discs used with 2mm spacing. From these readings plot, we observed that the experimental curve closer to the theoretical curve, actually theoretical curve plot by considering impulse force only but existence of friction force or drag force due to boundary layer effect difference found for three various conditions as shown in fig 5 & 6.

Spiral grooves on discs surfaces machined in such way that once jet enters the groove it follows the spiral path and exists thru central port. During rotation of discs centripetal force also exist which act inward to outward and that opposes to jet to come rapidly at the central port and due to this jet passes thru long spiral path and kinetic energy of jet utilized to drag the discs

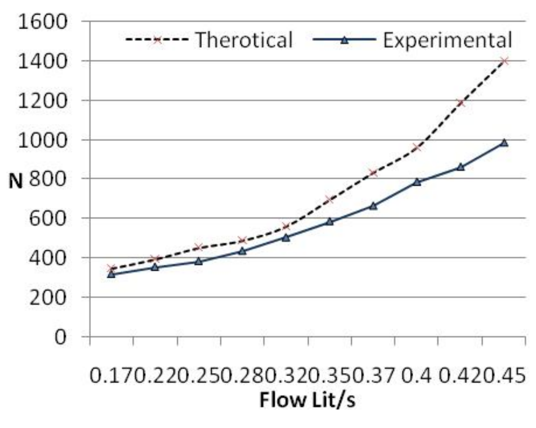

Fig. 6 : A plot of Flow v/s speed for 6 Rough discs and 0.5mm spacing

A. Output torque, power and efficiency

Torque on the shaft measured by Prony Break dynamometer. The methodology for the use of this device is simple. Essentially the measurement is made by wrapping a belt around the output shaft of the unit and measuring the force transferred to the belt through friction. The friction is increased by tightening the belt until the frequency of rotation of the shaft is reduced. In its simplest form an engine is connected to a rotating drum by means of an output shaft. A friction band is wrapped. Around half the drum's circumference and each end attached to a separate spring balance. A substantial pre-load is then applied to the ends of the band, so that each spring balance has an initial and identical reading. When the engine is starting the frictional force between the drum and the band will increase the force reading on one balance and decrease it on the other. The difference between the two readings is used to calculate torque, because the radius of the driven drum is known. Once we knew the spring balance we can determine the torque by equation.

$$T = (D + t_b) \text{ X } 9.81 \text{ X } S \qquad (13)$$

Where, D = Drum Dia., t_b = Belt thickness,

S = Spring Balance

Then the Output Power calculated by P = 2∏NT / 60

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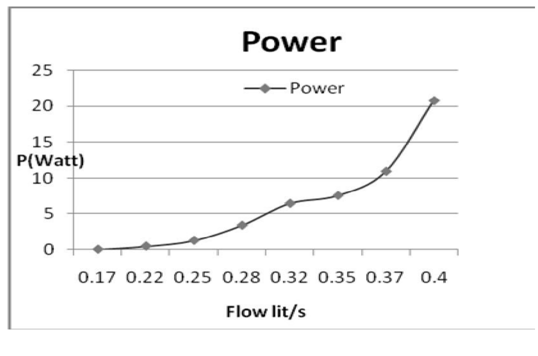

Fig.7 : A plot of flow v/s Power at various flow rates for six discs and 05mm spacing.

To determine the efficiency, we have to determine I/P power [10]. By the following formula

$$\text{Input Power} = P_{I/P} = \rho g Q h \qquad (14)$$

Where, ρ = Density of water, Q = flow of fluid m^3/s,

h = Water column height.

and $\rho g = \gamma$

$$\text{But, } P_{line} = \gamma \text{ x } h \qquad (15)$$

Where, P_{line} = static Pressure

$$P_{I/P} = P_{line} \text{ X } Q \qquad (16)$$

$$\text{Hydraulic Efficiency} = \text{Power}_{O/P} / \text{Power}_{I/P} \qquad (17)$$

By considering one case for sample calculations of efficiency.

$\therefore$ Q = 0.22lit/s = 2.2 X 10^{-4} m^3/s

P_{line} = 0.47 bar = 3238.3N/mm^2

Hydraulic Efficiency = 2.2 X 10^{-4} X 3238.3 = 7.1%

VI. CONCLUSIONS

Study and analysis of this work come to conclusion that Number of discs, spacing between discs and surface finish of discs affects the performance of turbine significantly

This work was carried out to study performance of disc turbine operating on water medium, however previous studies and experimentation carried out with air and steam medium.

Rotor speed increases with number of discs up to a certain level due to increasing area of contact of jet water and wall and it leads to the increase in friction force and boundary layer effect.

For wide spacing between the discs it works as impulse turbine only.

For appropriate spacing between discs it works with impulse force and also boundary layer effect. For minimum spacing equipment vibration also increases for high speed compare to maximum gap, Hence vibration analysis became important factor.

Present experiments showed that the losses occurring in the nozzle are large and hence this needs to be tackled for improving the overall efficiency of the Tesla disc turbine.

Experimental work shows that the efficiency of disc turbine may be increased by 5 to 6% by using spiral groove discs (Rough discs).

A. Suggestion for Future Work

Some of the following topics would be interesting for further investigation.

In a future study, it will be interesting to analyze the influence of the inlet nozzle numbers and positions .The influence of the disc holes and the outlet nozzle size must be also analyzed.

Influence of the composite material for discs on the turbine performance.

Compressible analysis of the multiple disc turbines

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