

# Effect of Disc Surface Finish and Disc Spacing On the Performance of Bladeless Turbine



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## ABSTRACT

In this investigation, instead of blades, closely packed parallel discs are used. Resistance to fluid flow between the plates results in energy transfer to the shaft. High velocity water enters the disc 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 impulse forces continue to convert kinetic energy to shaft rotational power. However, The Bladeless 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. Various complementary methods of measurement have been implemented and compared, and several operational experiences have been noted. Experimental results for a 152 mm diameter and 2 mm thick discs of turbine are presented, which shows the variation of torque, output power, and efficiency as a function of angular speed. Measurements of static pressure are also taken at the inlet; many design considerations and operational experiences are discussed. The effect of each parameter on the torque and power has been analyzed. It has been found that the spacing and surface finish has a significant effect on the power of the turbine. The maximum power obtained in this investigation was 33watts for 6discs and 0.5 mm spacing between discs with rough surface (spiral Groove). The torque and power increases with decrease in spacing up to 0.5mm and increase in surface roughness value (Ra) 500 microns. From this investigation, it is clear that the developed bladeless turbine is working efficiently at 0.5mm spacing and 500 microns roughness disc surface.

**Keywords**— disc, turbine, bladeless, boundary layer, Tesla.

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## 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; drag is formed that holds the aircraft back. Aerodynamics 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 180°. Tesla perceived that boundary layer drag used to do something useful [6].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. Experimentals results are 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 SURVEY

Nikola Tesla [1] 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 dependent 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 that 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. Warren Rice [2] 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 were not near to those indicated by the analyses for optimum turbine efficiency. Rice observed 24% max efficiency.

N.Huybrechts et.al [3] have visualized and analyzed the flow

in the disk turbine with commercial CFD software. Numerical model and mathematical model both developed. By mathematical model 43.9Watt power calculated with 32% efficiency. From the flow visualization the presence of 3 losses were detected, the windage loss, the loss of kinetic energy at the exhaust and loss due friction. Performance affecting parameters established. He also suggested that increasing the turbine power, the number of nozzles and disk must be increased and to reduce the exhaust losses, the angular velocity must be increased and the space between disks must be decreased. William Beans [4] has investigated Disk Turbine theoretically and experimentally by using the differential form of the equation of motion. A partial closed form solution obtained for the case of incompressible laminar flow. The performance of 6'' disk turbine calculated and tested over range 4000rpm with supply pressure 15-25 psi. Turbine efficiency observed in the range of 7% to 25%. He suggested that the loss of power due to the decrease in mass flow rate through the rotor is more than compensated by increase in frictional effects which is mainly due to the increase in inlet velocity. Piotr Lampart et.al.[5] have presented results of the design analysis of a Tesla bladeless turbine intended for a cogenerating micro-power plant of heat capacity 20 kW, which was operated in an organic

Rankine 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.

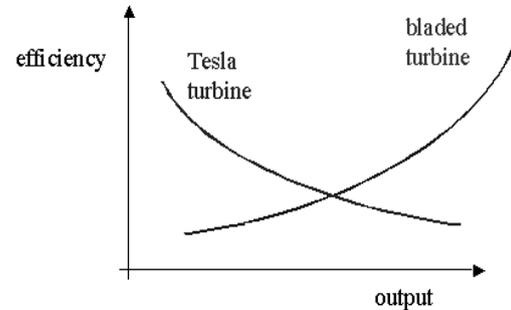


Fig. 1 Pictorial comparison of efficiency of small bladed & Tesla Turbine[3]

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.THEROCTICAL 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.

For sample calculation following data considered.

Constant Parameters:

Spacing between Discs: 2mm	Medium: Water
Disc Thickness: 2mm	Outlet Nozzle Size: 32mm <sup>2</sup>
Material of Discs: SS304	Disk Dia: 152mm
Inlet Size: 21mm <sup>2</sup>	No of discs: 6
Surface Finish: smooth	Medium: Water
Flow rate: 0.22lit/s	Line Pressure: 18lb/in <sup>2</sup>

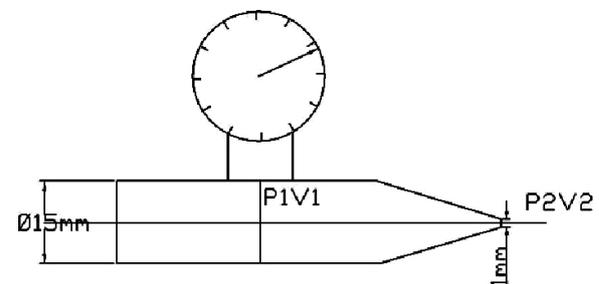


Fig. 2 Nozzle Arrangement - line diagram for Test Rig

By applying Bernoulli's equation, at point 1 and at point 2

$$P_1 / \rho g + V_1^2 / 2g + Z_1 = P_2 / \rho g + V_2^2 / 2g + Z_2 \quad (1)$$

Where,  $P_1$  = Pressure at point 1  $V_1$  = velocity at point1  
 $P_2$  = Pressure at point 2  $V_2$  = velocity at point 2  
 $Z_1=Z_2$ =Pressure head (at same elevation)  
 $\rho$  = water density =1000 kg/m<sup>3</sup>  $g$  = 9.81 m/s<sup>2</sup>  
 We know the flow rate i.e.  $Q = 0.22 \text{ lit/s} = 0.00021 \text{ m}^3/\text{s}$  and

$$P_1 = 18 \text{ lb/in}^2 = 1.15 \text{ bar}$$

By continuity equation,

$$Q = A_1 V_1 = A_2 V_2 \quad (2)$$

Where  $A_2 =$  area of cross section at pt 2 =  $21 \text{ mm}^2$

$$\text{and } \therefore A_1 = \pi / 4 \times D^2 = 3.14 / 4 \times 0.0152 = 0.000225 \text{ m}^2$$

$$V_1 = Q/A_1 = 0.00022 / 0.000225 = 0.98 \text{ m/s}$$

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

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.

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

$$P_2 = [(1.15 \times 10^5 / 9810 + 0.98^2 / 19.62) - (10.48^2 / 19.62)] \times 9810$$

$$= 0.47 \text{ Bar}$$

Now  $P = F/A$ .

We know the value of  $P_2$ ; Put this value in above equation to

get Force acting on discs thickness.

$$F = P_2 \times A_2 = 69622.74 \times 12 \times 10^{-6} = 0.8354 \text{ N (force acting on 12mm area only)}$$

We know the jet force formula from pelton turbine

$$F = \rho A (V - U)^2 \cos \theta \quad (3)$$

Where,  $A =$  jet area =  $A_2 = 21 \text{ mm}^2$ ,  $V =$  velocity of jet =  $V_2$ ,

$U =$  Relative velocity of discs. &  $\theta = 10^\circ$  angle of jet.

Now put the values in above equation to get  $U$

$$(V - U) = 0.8354 / (1000 \times 21 \times 10^{-6} \times 0.98) = 12.47$$

$$(V - U) = 6.37$$

$$U = 4.10 \text{ m/s}$$

We know the angular momentum for rotating part

$$L = m r U \quad (4)$$

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

$$L = I \omega \quad (5)$$

Where  $I =$  moment of inertia &  $\omega =$  Angular velocity

In this case,

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

$$I_{\text{SHAFT}} = M R^2 \quad (7)$$

Where  $M =$  mass of the shaft =  $1.08 \text{ kg}$ .

$R =$  Radius of shaft =  $15 \text{ mm}$ .

$$I_{\text{SHAFT}} = 1.08 \times 2 \times (15/100)^2 \times 9.81 = 0.0238 \text{ N/mm}^2$$

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

Where,  $M_d =$  Mass of disk =  $0.275 \text{ kg}$ ,

$a =$  inner radius of disc =  $30 \text{ mm}$ ,

$b =$  outer radius of disc =  $76 \text{ mm}$ .

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

For 6 discs,

$$I_{\text{DISC}} = 0.032 \times 6 = 0.192 \text{ Nm}^2$$

Now from (6)

$$I = 0.032 + 0.192 = 0.195 \text{ Nm}^2$$

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

$$\omega = m r U / I = (2.73 \times 0.076 \times 4.10) / 0.195$$

$$\omega = 4.36 \text{ rad/s For 6 discs}$$

Now,

$$\omega = 2\pi N / 60 \quad (9)$$

$$N = 4.36 \times 3.75 \times 60 / 6.28 = 156.21 \text{ rpm.}$$

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

From equation (3) & (10)

$$T = 0.8354 \times 0.076 = 0.063 \text{ Nm Now,}$$

$$P = 2\pi T N / 60 \quad (11)$$

By putting the values of  $N$  and  $T$ , we get

$$P = (6.28 \times 156.21 \times 0.063) / 60$$

$$P = 1.03 \text{ watt.}$$

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.

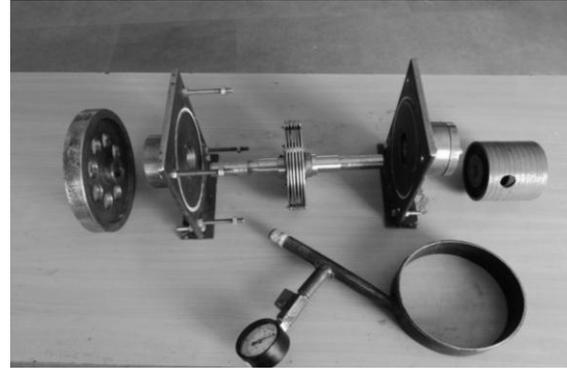


Fig. 3 Parts of Multiple disc Turbine.

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.

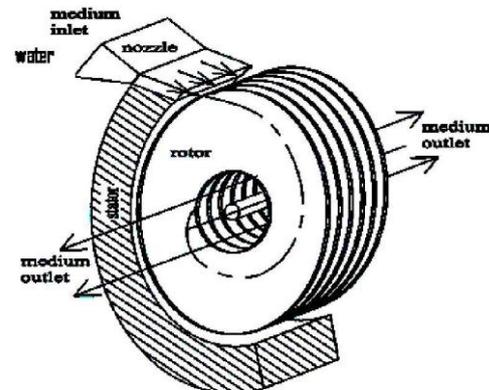


Fig. 4 Cross Section of Test Rig.

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

The diameter of this central hole is 30mm. In order to accommodate the outlet of the fluid, the shaft is supported by a boss by means of bearings inside it. 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 work we used water medium. And for constant flow rate and to provide high-pressure water at the inlet of the nozzle 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 10 lit/min to 27 lit/min.

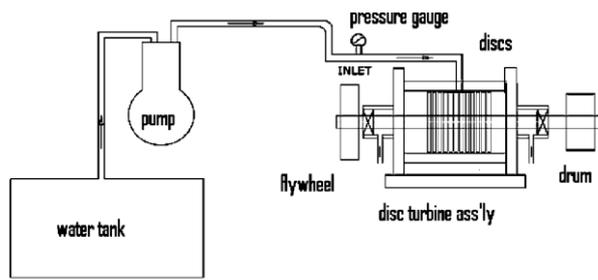


Fig. 5 Experimental Set up

Constant parameters same as selected in theoretical analysis. Variable parameters are the flow, No of discs and spacing between discs and surface finish. 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 I  
EXPERIMENTAL DESIGN MATRIX

Flow	Spacing	Surface Finish
+	+	+
-	+	+
+	-	+
-	-	+
+	-	-
-	-	-
+	+	-
-	+	-

Flow (+) = 27 lit/min and Flow (-) = 10 lit/min  
 Spacing (+) = 3mm and Spacing (-) = 0.5mm  
 Surface finish (+) = smooth and Surface finish (-) = 500Ra

**I . 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 Tachometer. 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 Tachometer. This method was found to be accurate and inexpensive. Then angular velocity obtained by using formula.

$$\omega = 2\pi N / 60 \tag{12}$$

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. Constant parameters same as considered in theoretical analysis. Variable parameters are the flow, spacing between discs and surface finish of the discs.

**IV. RESULT & DISCUSSION**

a 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 size designed in such way that it imparts equal impulse force on each discs in spite of of spacing between discs. Impulse force acts only on thickness of discs, hence spacing between discs not affecting the impulse force. But for maximum spacing i.e 2.5mm and 6 discs pack, impulse force not acting on two end discs because nozzle opening size limited to 21mm<sup>2</sup> with length 21mm and height 1mm.

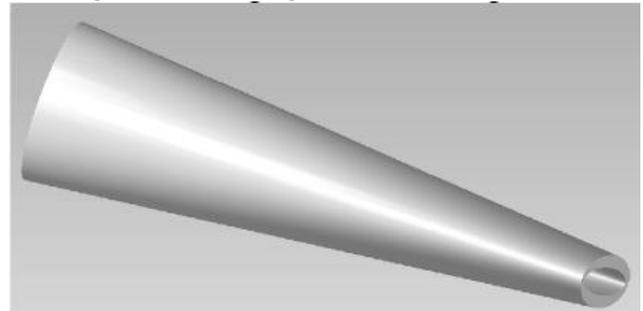


Fig. 6 Nozzle

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<sup>2</sup> areas only and for six discs jet area imparted on 12mm<sup>2</sup> 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 7 & 8 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.

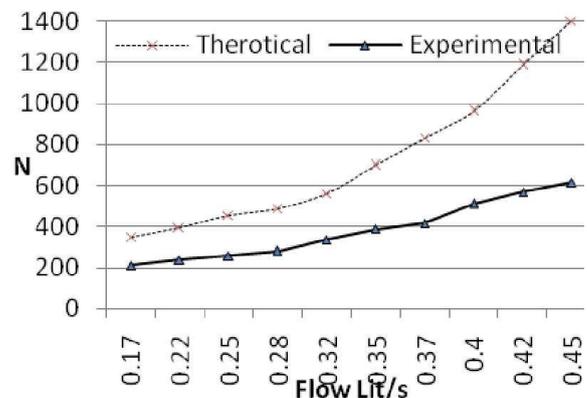


Fig. 7 Plot of Flow v/s Speed for 6 smooth discs and 2.5mm spacing.

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.

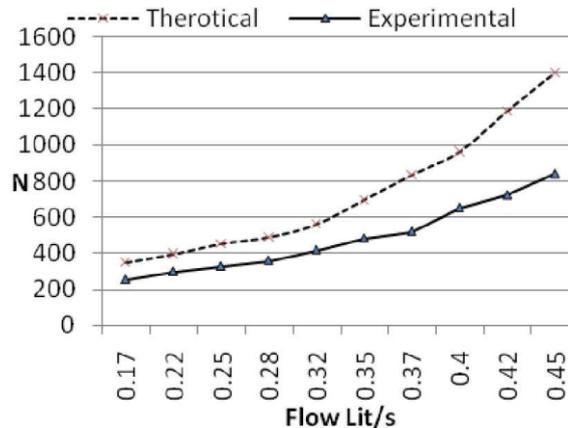


Fig. 8 plot of Flow v/s Speed for 6 smooth discs and 0.5mm spacing

Fig.9 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.

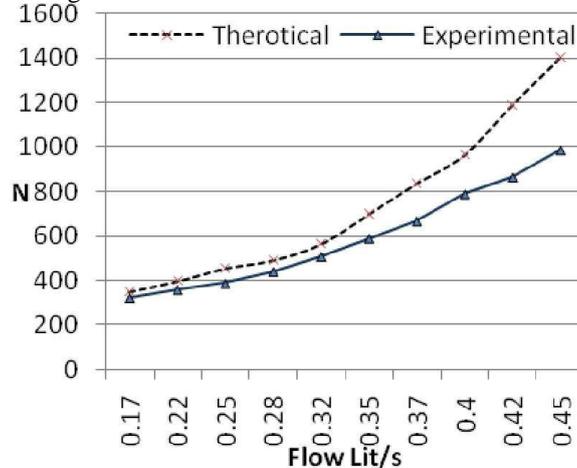


Fig.9 Plot of Flow v/s Speed for 6 Rough discs and 0.5mm spacing.

**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 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 surface roughness considered to carry out experiment. Constant parameters same as considered in theoretical analysis. Variable parameters are the flow, spacing between discs and surface finish of the discs.

$$T = (D + t_b) \times 9.81 \times S \tag{13}$$

Where, D = Drum Dia.,

t<sub>b</sub> = Belt thickness,

S = Spring Balance

Then the Output Power calculated by  $P = 2\pi INT / 60$

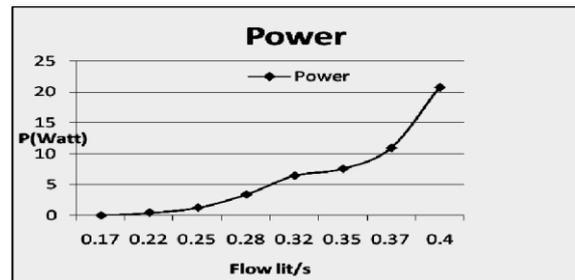


Fig. 10 A plot of flow v/s Power at various flow rates for six discs and 0.5mm spacing

To determine the efficiency, we have to determine I/P power,

by the following formula,

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

and  $\rho g = \gamma$

h = Water column height.

Where,  $\rho$  = Density of water, Q = flow of fluid m<sup>3</sup>/s

Where,  $P_{I/P} = P_{line} \times Q$

Hydraulic Efficiency = Power O/P / Power I/P

By considering one case for sample calculations of efficiency.

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

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

$$\text{Hydraulic Efficiency} = 2.2 \times 10^{-4} \times 3238.3 = 7.1\%$$

**V.CONCLUSION**

a 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.

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.

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).

## VI. 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. Compressible analysis of the multiple disc turbines. Influence of the composite material for discs on the turbine performance.

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