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Effect of Disc Number, Spacing & Surface finish on the Performance of Bladeless Turbine

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Abstract

In this Study, 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 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 manufactured and a flexible test rig has been designed 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*: Turbine, disc, bladeless, Tesla, boundary layer.*

1. Introduction

When an air plane flies through air, at speed of high, there is a thin layer of air that sticks to the wing all over. This layer of air goes the same speed as the airplane. There is then shear action between that boundary layer and the surrounding quiescent air around the aircraft. 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 unavoidable. But, Tesla was able to turn that precept around 180°means reverse. Tesla proved that boundary layer drag used to do some 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 centre of the turbine. In this present work, A Tesla disc turbine and a flexible test rig developed and manufactured by simple material. Experimental results are carried out by using water medium. The overall design of the turbine is 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.

2. Literature Review

1) Nikola Tesla

Made a patent on disc turbine in 1913 with smooth disc (Adhesion & Viscosity). 2 HP output power, 30,000 rpm, 60% efficiency

2) E William Beans

Investigated Disk Turbine theoretically and experimentally by using the differential form of the equation of motion.1.0kW output power, 12000 rpm, 24% efficiency

3)G.P.Hoya & A.Guha

Designed and manufactured test rig and results presented. An Analysis of the performance and efficiency of the disk turbine carried out. The design philosophy of the flexible test rig implemented.

4) N.Huybrechts

Suggested to increase the turbine power by increasing number of nozzle . 43.9 Watt,32% efficiency

5) Ference Lezsovits

Has investigated an advantage and disadvantages of multiple disk turbines and operation conditions.

6) Thin Engin

Have designed a multiple-disk Tesla type fan and then tested and analyzed two-dimensionally using the conservation of angular momentum principle.

7) Warren Rice Rice constructed six disks turbine and reports some aspects of them, 1.5kW output power, 12000 rpm, 23% efficiency

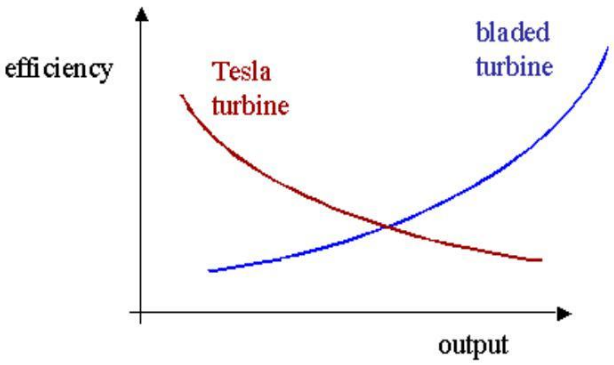

Fig. 1 Comparison of efficiency of small bladed & Tesla Turbine[3]

From the literature, we establish the parameters those affects the performance of turbine. Spacing between the discs, No. of the discs, Discs surface roughness, Number of nozzles. The following assumptions are considered for the experimental work. A) Steady flow B) Incompressible fluid C) negligible body forces D) Full admission of working fluid at outer periphery of the discs.

3. Theoretical Analysis

Inlet Nozzle fabricated as shown in fig. 2 for Test Rig. Water incompressible fluid is selected for performance checking with assuming flow is a steady flow.

For sample calculation following data are considered.

Constant Parameters:

Spacing between Discs: 2mm Medium: Water

Disc Thickness: 2mm Outlet Nozzle Size: 32mm^2 Material of Discs: Stainless Steel 304

Disk Diameter: 152mm

Inlet Size: 21mm^2 Number of discs: 6

Surface Finish: Smooth Medium: Water

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

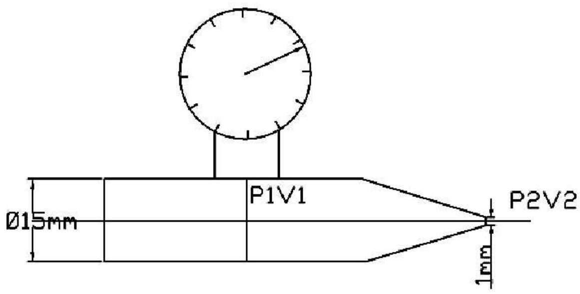

Fig. 2 Nozzle Arrangement - line diagram for Test Rig

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

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

Z1=Z2=Pressure head (At same elevation)

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

By Continuity Equation,

$Q = A_1V_1 = A_2V_2$ (2) Where A_2 = area of cross section at pt 2 = 21 mm^2

and∴$A_1 = \Pi / 4 X D^2 = 3.14 / 4 X 0.0152 = 0.000225m^2$ $V_1 = Q/A_1 = 0.00022 /0.000225 = 0.98$ 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.

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

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

= 0.47 Bar

Now P = F/A.

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

Get Force acting on discs thickness.

$F = P_2 X A_2 = 69622.74 X 12 X 10^{-6} = 0.8354$ N

(force acting on 12mm area only)

We know the jet force formula from pelton turbine

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

Where, A = jet area = A_2 = 21 mm^2 , V = velocity of jet = V_2,

U = Relative velocity of discs. & $\theta = 10^o$ angle of jet. Now put the values in above equation to get U

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

(V - U) = 6.37

U = 4.10 m/s

We know the angular momentum for rotating part

L = mrU (4)

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

L = Iω (5)

Where I = moment of inertia & ω = Angular velocity In this case,

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

$I_{SHAFT} = MR^2$ (7)

Where M= mass of the shaft=1.08 kg .

R= Radius of shaft = 15 mm .

I_{SHAFT} = 1.08 2 X (15/100)^2 X 9.81= 0.0238 N/mm^2

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

Where, M_d = Mass of disk= 0.275 kg,

a = inner radius of disc=30 mm,

b=outer radius of disc = 76 mm.

$I_{DISCS} = 1/2 X 0.275 (0.015^2 + 0.076^2) = 0.032$ Nm^2 For 6 discs,

I_{DISC}= 0.032 X 6 = 0.192 Nm2

Now from (6)

I = 0.032 + 0.192 = 0.195 Nm^2

now, from (4) & (5) L = Iω

ω = mrU / I = (2.73 X 0.076 X 4.10) / 0.195

ω = 4.36 rad /s For 6 discs

Now,

ω = 2ΠN / 60 (9)

N = 4.36 X 3.75 X 60 / 6.28 = 156.21 rpm.

Now Torque T = F X R (10)

From equation (3) & (10)

T = 0.8354 X 0.076 = 0.063 Nm Now,

P = 2 ΠNT / 60 (11)

By putting the values of N and T, we get

P = (6.28 X 156.21 X0.063)/60
P = 1.03 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.

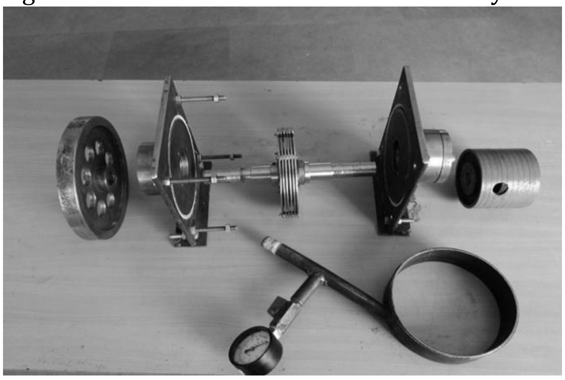

Fig. 3 Parts of Multiple disc Turbine.

Nozzle designed on the basis of energy conservation law and coefficient of discharge like venturi meter, it imparts equal jet on each disc and due to its convergent shape it increases kinetic energy of jet as shown in fig. 3.

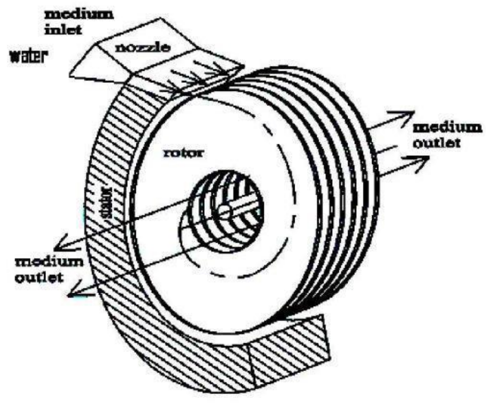

Fig. 4 Cross Section of Test Rig.

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 disc supported on two numbers of bearings. In previous studies , air medium is used, but in this study we have used water medium at for constant flow rate and to provide high-pressure water at the inlet of nozzle we used positive displacement pump as shown in Fig.4. A test rig designed in such way that it checks and measure the various parameters that are necessary to determine the performance the Turbine. Flow rate vary from the 10 lit/min to 27 lit/min.

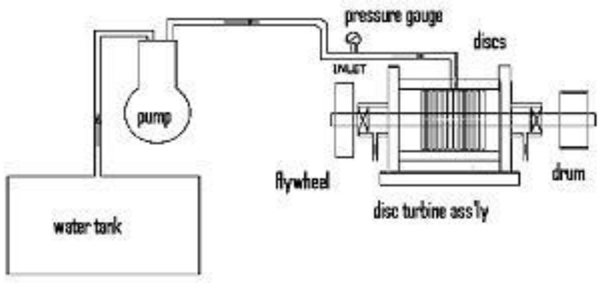

Fig. 5 Experimental Set up

Constant parameters are same as selected in theoretical analysis. Variable parameters are the flow, No of discs and spacing between discs and surface finish. In this experiment 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 Rough. Spiral grooves machined on discs surfaces by using lathe machine and fixture.

TABLE I
Experimental Design Matrix

Experiment	**Flow (Lit/Min)**	**No.of Discs**	**Spacing (mm)**	**Surface Finish (Ra)**
1	+	+	+	+
2	−	+	+	+
3	+	−	+	+
4	−	−	+	+
5	+	+	−	+
6	−	+	−	+
7	+	+	+	−
8	−	+	+	−
9	+	+	−	−
10	−	+	−	−
11	+	−	−	+
12	−	−	−	+
13	+	−	+	−
14	−	−	+	−
15	+	−	−	−
16	−	−	−	−

Flow (+) = 27 lit/min and Flow (-) = 10 lit/min
Spacing (+) = 2.5 mm and Spacing (-) =0.5mm
Surface Finish (+) =smooth and Surface Finish (-) = 500Ra Rough
No.of Discs (+) = 6 No. No. of Discs (-) = 3 No.

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 cantering operation done which is provision to insert a tip of Tachometer. This method is 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 study permits, variations in the numbers of discs to obtain Performance comparisons data. In this present work three variations in discs spacing and three variations in surface roughness considered to carry out experiment. Constant parameters are same as considered in theoretical analysis. Variable parameters are Spacing between discs and surface finish of the discs and Flow (LPM).

4. Result and Discussion

The aim 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^2 with length 21mm and height 1mm.

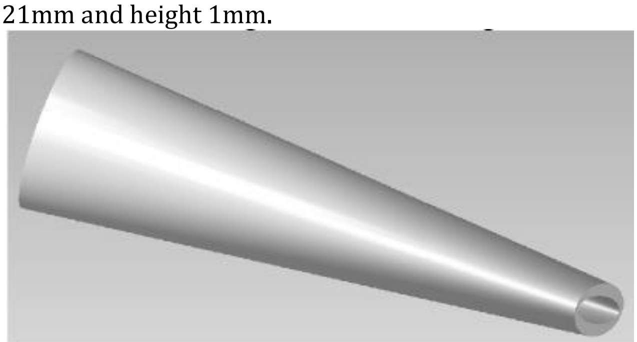

Fig. 6 Nozzle

After conducting the experiments as per matrix, we noticed that this turbine working efficiently for six discs only. Due to the nozzle opening, for three discs jet imparted on 6mm2 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 7 & 8 shows the variation of speed with respect to flow. Flow is directly proportional to speed. 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. High velocity of jet 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. Kinetic energy of jet utilized for disc rotation.

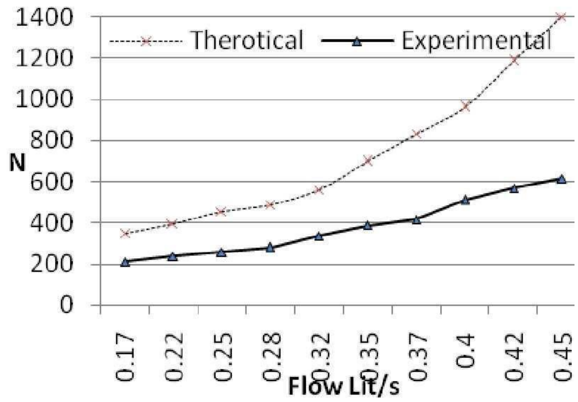

Fig. 7 Plot for Flow v/s Speed for 6 smooth discs and 2.5mm spacing.

For 2.5 mm spacing, more amount of water jet 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. Water 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 develop and overlap with each other.

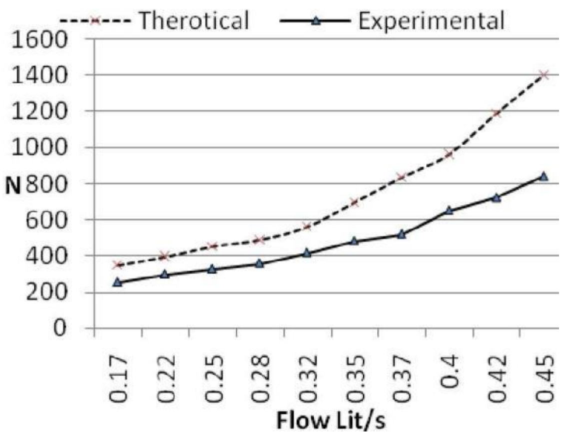

Fig. 8 plot of Flow v/s Speed for 6 smooth discs and 0.5mm spacing

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 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. while 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.

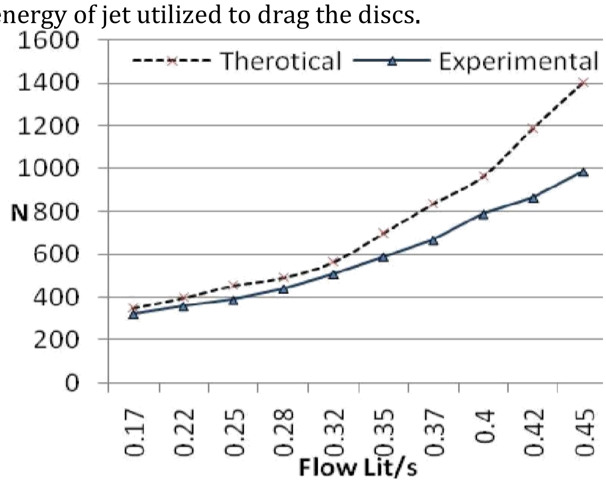

Fig.9 Plot of Flow v/s Speed for 6 Rough discs and 0.5mm spacing.

Output torque, power and efficiency:

Torque on the rotating 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 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. Approx half of 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 are same as considered in theoretical analysis. Variable parameters are the flow, spacing between discs and surface finish of the discs.

$T = (D + t_b) X 9.81X S$ (13) Where,

D = Drum Dia.,

t_b = Belt thickness,

S = Spring Balance

Then the Output Power calculated by $P = 2\Pi NT / 60$

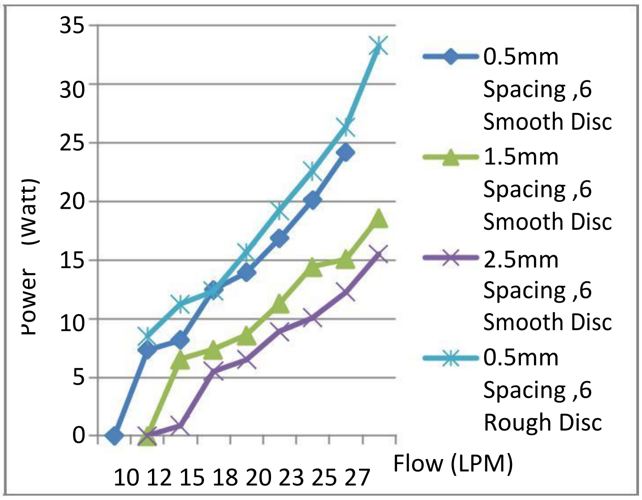

Fig.10 Flow v/s Power for different Spacing and Surface Finish

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

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

and $\rho g = \gamma$

h = Water column height.

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

Where, $P_{i/p} = P_{line} X Q$

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

by considering one case for sample calculations (efficiency)

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

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

Hydraulic efficiency $= 3238.3 X 2.2 X 10^{-4} = 7.1\%$

5. Conclusion

1. Study and analysis come to conclusion that Number of discs, spacing between discs and surface finish of discs affects the performance of turbine significantly
2. 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.
3. Rotor speed is directly proportional to number of disc .i.e. 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.
4. For wide spacing between the discs it works as impulse turbine only
5. For appropriate spacing between discs it works with impulse force and also boundary layer effect. For min. spacing equipment vibration also increases for high speed

compare to maximum gap, Hence vibration analysis became important factor.

6. 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.

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

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