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Study of Performance of Shaft output with Rotor-to-Casing Ratios versus Different Vane Angles Adopting Practical Approach on a Novel Multi-Vane Air Turbine

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STUDY OF PERFORMANCE OF SHAFT OUTPUT WITH ROTOR-TO-CASING RATIOS VERSUS DIFFERENT VANE ANGLES ADOPTING PRACTICAL APPROACH ON A NOVEL MULTI-VANE AIR TURBINE

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Study of Performance of Shaft output with Rotor-to-Casing Ratios versus Different Vane Angles Adopting Practical Approach on a Novel Multi-Vane Air Turbine

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I. Introduction

It is an established fact that the Worldwide faster consumptions of fossil fuel in transport vehicles have resulted fast depletion to energy resources and releasing huge quantities of pollutant in the atmosphere.. A US geologist Marian King Hubbert [1] in 1956 indicated that the conventional crude-oil production will attain Peak Oil within 20 years and thereafter it will start depleting which may cause serious threat to mankind within 40 years i.e. by 1995. Aleklett K. and Campbell C.J., [2] in 2003 illustrated that with current rate of consumptions, the production of oil and gas in many country will peak and begin to decline by around 2010. Such apprehension has led the search for environment friendly alternative to fossil fuel oil, or some method of conserving natural resources using non-conventional options, such as; bio-diesel, wind power, photo voltaic cells etc. and or energy conversion systems like battery storage, hydrogen cell, compressed air etc to obtain shaft work for the engines of vehicles [3-9].

Guy Negre [10] a French technologist and G. Saint Hillarie [11] an inventor of quasi turbine have carried out very important work in the area of compressed air engine. They stored highly compressed air in the energy storage systems up to 300 bar pressure within 15–20 minutes, and reused for running compressed air engines. In view of such attractive features of nearly zero pollution and air compression by using non-conventional resources, the compressed air engine became comparable in place of the other technology in vehicle markets.

In this paper author has carried out the parametric analysis of a small capacity air turbine having vane type rotor and describes the investigation of the effect of rotor to casing diameter ratios with different vanes fitted in the rotor. Results obtained by using a mathematical model are presented and analyzed here.

II. Feature of Vane Type Air Turbine

In this study a multi-vane type air turbine having casing diameter =D mm and rotor diameter =d mm is proposed as shown in Fig. 1. The considered air turbine works on the reverse working principle of vane type compressor. In this arrangement total shaft work is seen to be the cumulative effect of isobaric admission of compressed air jet on vanes and the adiabatic expansion of high pressure air. In earlier study conducted by authors a prototype of air turbine was developed [12].and its functionality was examined A storage cylinder for the compressed air having capacity of 30 minutes stored air, for the requirement of running turbine at initial stage at working pressure of 10 bar, is used as a compressed air energy source. This storage cylinder is designed to produce constant pressure for the minimum variation of torque at low volumes of compressed air and attached with filter, regulator and lubricator. The clean air then admits into air turbine

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through its inlet nozzle and vanes of air turbine are also fitted into rotor under spring loading to maintain their regular contact with the casing wall. This arrangement is proposed to minimize leakage through mating contacts and is novelty of improvement in the vane turbine. A study on highly efficient energy conversion system for liquid nitrogen [13], design and verification of airfoil and its tests, influence of tip speed ratios for small wind turbine and parabolic heat transfer and structural analysis were also carried out for conceptualizing the energy conversion system [14-17]. The study of design feasibility of vane type novel air turbine has also been carried out [18-21].

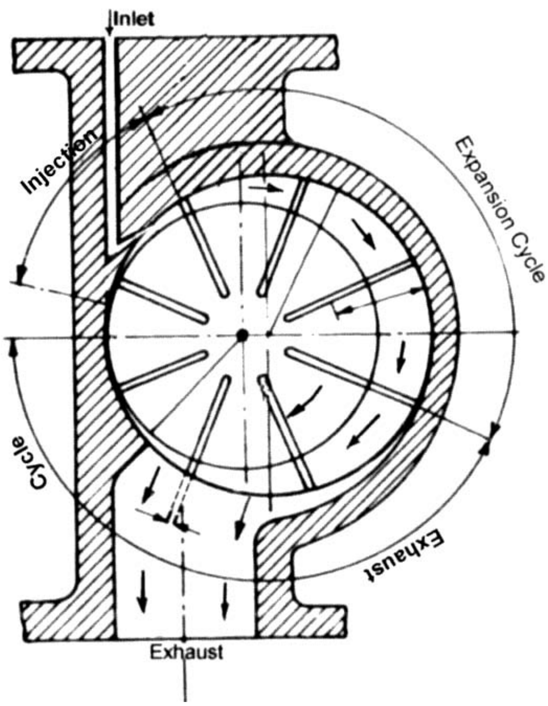

Fig.1 : Air Turbine-Model

The present objective of this study is to investigate the power output of an air turbine with different number of vanes in rotor, i.e. angle between first two consecutive vanes and the rotor/casing ratio (*d/D*). The air turbine considered has capability to yield output of 5.0 to 5.6 HP at injection pressure 4-6 bar and speed of 2000–2500 rpm and is suitable to run a motorbike.

III. Mathematical Model

The mathematical model shown here is already presented in author's earlier publications [22-34] and is reproduced here for maintaining the continuity and benefits to the readers[**1]. The high pressure jet of air at ambient temperature drives the rotor in novel air turbine due to both isobaric admission and adiabatic expansion. The high pressure air when enters through the inlet passage, pushes the vane for producing rotational movement and thereafter air so collected between two consecutive vanes of the rotor is gradually expanded up to exit passage, also contributes to the shaft out. This isobaric admission and adiabatic expansion of high pressure air both produces the total shaft power output from air turbine. Compressed air leaving the air turbine after expansion is sent out from the exit passage. It is thus noticed that the scavenging of the rotor is perfect and the work involved in recompression of the residual air is absent as seen from Fig. 1.

From Fig. 2, it is seen that work output is due to isobaric admission (E to 1), and adiabatic expansion (1 to 4) and reference 2, 3 in the figure shows the intermediate position of vanes. Thus, total work output due to thermodynamic process may be written as:

Total work output = [Thermodynamic expansion work (w_1)] + [Exit steady flow work (w_2)]

or

$$w = [(w_1) + (w_2)] \qquad (1)$$

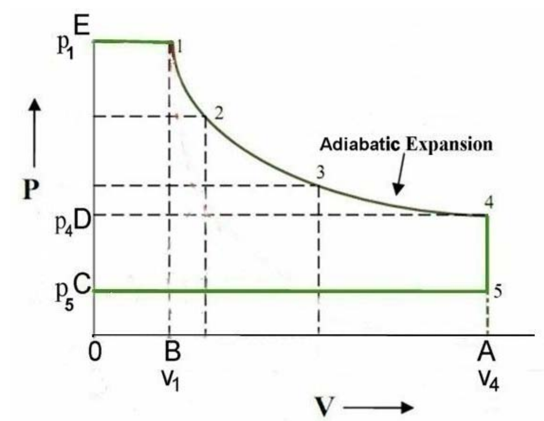

Fig.2 : Thermodynamic Processes (Isobaric, adiabatic and Isochoric Expansion)

Now thermodynamic expansion work (w_1), considering adiabatic process between state 1 and 4, it can be written as:

$$w_1 = \left(\frac{\gamma}{\gamma - 1}\right) . p_1 . v_1 . \left\{ 1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} \right\} \qquad (2)$$

The process of exit flow (4-5) takes place after the expansion process (E - 4) as shown in Fig. 2 and air is released to the atmosphere. In this process; till no over expansion takes place pressure p_4 can't fall below atmospheric pressure p_5. Thus at constant volume when pressure p_4 drops to exit pressure p_5, no physical work is seen. Since turbine is functioning as positive displacement machine and hence under steady fluid flow at the exit of the turbine, the potential work is absorbed by the rotor and flow work (w_2), can be written as:

[**1] Mathematical model is reproduced here from author's earlier publications [22-34].

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$$w_2 = \int_4^5 v.dp = v_4(p_4 - p_5) \quad (3)$$

Applying equations (2), (3) into equation (1), considering air turbine has n number of vanes, then shaft output [35] can be written as:

$$w_n = n.\left(\frac{\gamma}{\gamma-1}\right).p_1.v_1\left\{1-\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}+n.\left(p_4-p_5\right).v_4 \quad (4)$$

where w_n is work output (in Nm), for complete *one cycle.*

Therefore, the total power output or work done per unit time (W), for speed of rotation N rpm, will be:

$$W_{total} = n.(N/60).\left(\frac{\gamma}{\gamma-1}\right).p_1.v_1.\left\{1-\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}+n.(N/60).\left(p_4-p_5\right).v_4 \quad (5)$$

Figure 1 shows that if vanes are at angular spacing of θ degree, then total number of vanes will be n = (360/θ). The variation in volume during expansion from inlet to exit (i.e. v_1 to v_4) can be derived by the variable extended length of vane as shown in Fig. 3 at every point of movement between two consecutive vanes.

From Fig. 3, shows that when two consecutive vanes at OK and OL move to position OH and OB, the extended vane lengths varies from SK to IH and LM to BG, thus the variable length BG at variable α_i is assumed as $X_{at'variable'\alpha}$ can be written from the geometry:

$$BG = x_{at,variable'\alpha'} = R.\cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right).\sin\alpha\right\}\right]+(R-r).\cos\alpha - r \quad (6)$$

where 2R=D is diameter of casing and 2r=d is diameter of rotor, α is angle ∠BOF, β is angle ∠BAF and θ is angle ∠HOB or ∠H'OF or ∠KOL, between two consecutive vanes and ϕ is angle ∠KOJ at which injection pressure admits to the air turbine.

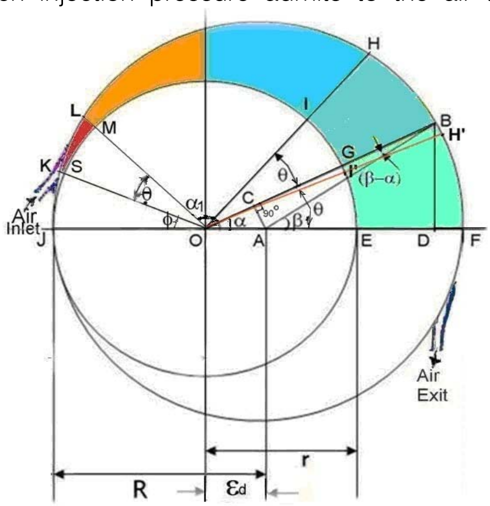

Fig.3 : Variable length BG and IH and injection angleø

Variable volume of cuboids B-G-I-H-B can be written as:

$$v_{cuboids} = L.\left\{\frac{(X_{1i}+X_{2i})(2r+X_{1i})}{4}\right\}.\sin\theta \quad (7)$$

where BG= X_{1i} and IH= X_{2i} variable length of vanes when rotate into turbine as shown in Fig. 3 and i stands for min or max length. Thus

a) The volume at inlet v_1 or v_{min} will fall between angle ∠LOF= $\alpha_{1min} = (180-\theta-\phi)$ and angle ∠KOF= $\alpha_{2min} = (\alpha_{1min}+\theta) = (180-\phi)$ as seen in Fig. 3, when air is admits into turbine at angle ϕ .

b) The Volume at exit v_4 or v_{max} will fall between angle ∠BOF = $\alpha_{1max} = \alpha = 0$ and angle ∠HOF = $\alpha_{2max} = (\alpha_{1max}+\theta) = \theta$.

Applying above conditions into equations (6), then LM=X_{1min} , SK=X_{2min}, FE=X_{1max}=Corresponding to BG at α =0 degree and I'H'=X_{2max}=Corresponding IH at $(\alpha+\theta) = \theta$ degree

Applying values of X_{1min} and X_{2min} to equation (7),

$$v_{min} = v_1 = L.\left\{\frac{(X_{1min}+X_{2min})(2r+X_{1min})}{4}\right\}.\sin\theta \quad (8)$$

Applying values of X_{1max} and X_{2max} to equation (7),

$$v_{max} = v_4 = L.\left\{\frac{(X_{1max}+X_{2max})(2r+X_{1max})}{4}\right\}.\sin\theta \quad (9)$$

Applying values of v_1 and v_4 from equations (8) and (9) to equation (5), the total power output available W_{total} can be written as :

$$W_{total} = n.(N/60).\left(\frac{\gamma}{\gamma-1}\right).\left\{1-\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}p_1.\left[L.\left\{\frac{(X_{1min}+X_{2min}).(2r+X_{1min})}{4}\right\}.\sin\theta\right]$$
$$+n.(N/60).\left(p_4-p_5\right).\left[L.\left\{\frac{(X_{1max}+X_{2max}).(2r+X_{1max})}{4}\right\}.\sin\theta\right] \quad (10)$$

IV. Parametric Considerations and Analysis

In this study various input parameters are listed in Table 1 for investigation of performance of vane turbine at different rotor to casing diameter ratios (d/D) with respect to different vane angle when D=100 mm, injection pressure 6 bar (90 psi) and its optimization for larger shaft output.

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Table 1 : Input Parameters

Symbols	Parameters
d /D Ratio	0.95, 0.9, 0.85, 0.80, 0.75 and 0.70 when casing diameter is kept D=100 mm
p_1	6 bar(≈ 90 psi) – inlet pressures
p_5	$(p_4 / 1.2)$ >1.0132 bar (atmospheric pressure)
p_4	$(v_1 / v_4)^{\gamma} \cdot p_1$ > p_5 assuming adiabatic expansion
N	2500 rpm (as total power is directly proportion to rpm)
L	45 mm length of rotor (assumed minimum)
γ	1.4 for air
n	Number of vanes in the rotor=(360 / θ)
θ	Vane angle =30°(12 vanes), 36°(10 vanes), 45° (8 vanes), 60°(6 vanes), and 90° (4 vanes)
ϕ	60° angle at which compressed air enters through nozzle into rotor

While actual fabrication of air turbine is carried out, it is noticed that the rotor / casing (d/D) ratio can possibly be varied from 0.95 to 0.70 only.

V. Results and Discussion

Based on the various input parameters listed in Table-1 and mathematical model, the effects of different rotor to casing diameters ratios, at different vane angles, 2500 rpm of speed of rotation and 6 bar of injection pressure on the expansion work, exit flow work and total work output from air turbine are studied. Here the injection angle (ϕ) of the air turbine is considered to be constant at 60° for whole study.

The results obtained have been plotted in Figs. 4 to 9, for the rotor to casing diameter ratio (d/D), varied as 0.95, 0.90, 0.85, 0.80, 0.75 and 0.70 at different vanes of rotor 12, 10, 8, 6, 4 (i.e. corresponding vanes angles of 30°, 36°, 45°, 60°, 90°) and injection angle of 60° at injection pressures of 90 psi and at the speed of rotation 2500 rpm.

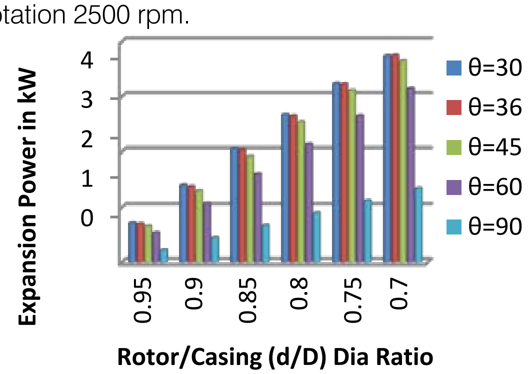

Fig.4 : Expansion power versus rotor / casing diameter (d/D) ratio at different rotor vanes when D= 100 mm, injection pressure = 6 bar and speed of rotation =2500 rpm.

Figure 4 shows the variation of expansion power for the rotor vanes =12 nos. (θ=30°), 10 nos. (θ= 36°), 8 nos. (θ= 45°), are increasing linearly at rotor / casing ratios =0.95, 0.90, 0.85, 0.80, 0.75 to 0.70 and it almost varies in similar values. Thus optimal performance of expansion power is seen at rotor / casing ratios =0.70 for vanes number 12 to 8. But variation of expansion power for the rotor vanes =6 nos. (θ=60°), and 4 nos. (θ= 90°), are increasing almost linearly at rotor / casing ratios =0.95, 0.90, 0.85, 0.80, 0.75 & 0.70 and developed power values are smaller.

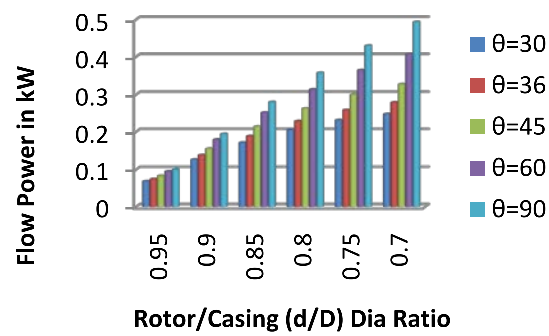

Fig.5 : Exit flow power versus rotor / casing diameter (d/D) ratio at different vane number (vane angles) when D=100 mm, injection pressure = 6 bar and speed of rotation =2500 rpm.

Figure 5 shows the variation of flow power for the rotor vanes =12 nos. (θ=30°), 10 nos. (θ= 36°), and 8 nos. (θ= 45°), 6 nos. (θ=60°), and 4 nos. (θ= 90°), are increasing parabolically at rotor / casing ratios =0.95, 0.90, 0.85, 0.80, 0.75 and 0.70 and it almost varies at different power values. It is also noticed that for the rotor vanes 6 nos. (θ=60°), and 4 nos. (θ= 90°) the values of power are found large.

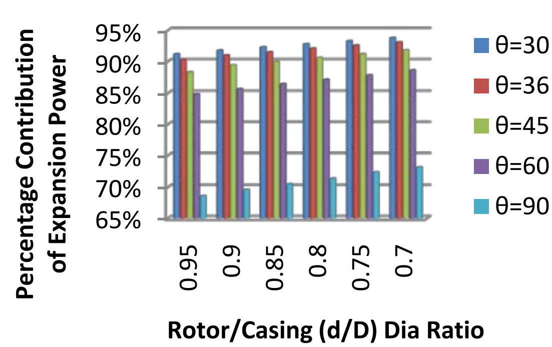

Fig.6 : Percentage contribution of expansion power versus rotor / casing diameter (d/D) ratio at different vane number (vane angles) when D=100 mm, injection pressure = 6 bar and speed of rotation =2500 rpm

Figure 6 shows the percentage contribution of expansion power against total power output for different rotor / casing ratios =0.95, 0.90, 0.85, 0.80, 0.75 and 0.70 at the rotor vanes =12 nos. (θ=30°), 10 nos. (θ= 36°), and 8 nos. (θ= 45°), are increasing from 91-94%, 90-93% and 88-92% respectively. But at the rotor 6 nos., (θ=60°), and 4 nos. (θ= 90°), it varies from 85% to 87% and from 69% to 73% respectively.

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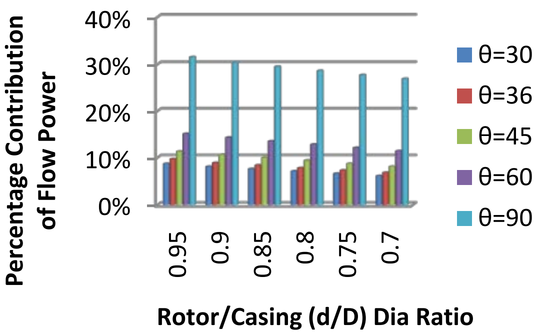

Fig.7 : Percentage contribution of exit flow power versus rotor / casing diameter (d/D) ratio at different vane number (vane angles) when D=100 mm, injection pressure = 6 bar and speed of rotation =2500 rpm

Figure 7 shows the contribution of flow power against the total power output for different rotor / casing ratios =0.95, 0.90, 0.85, 0.80, 0.75 and 0.70 at the rotor vanes =12 nos. (θ=30°), 10 nos. (θ= 36°), and 8 nos. (θ= 45°) are decreasing from 11% to 6% and at the rotor 6 nos. (θ=60°), and 4 nos. (θ= 90°), it varies from 15% to 11% and from 31% to 27% respectively.

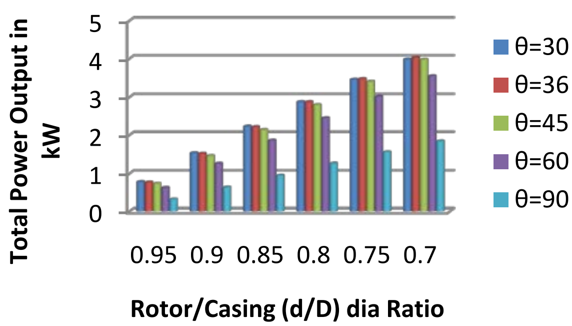

Fig.8 : Total power output versus rotor/casing diameter (d/D) ratio at different vane angle when injection pressure = 6 bar and speed of rotation =2500 rpm

Variation of total power output is seen from Fig. 8, with respect to different vane angle =30° (12 vanes), 36° (10 vanes), 45° (8 vanes), 60° (6 vanes) and 90° (4 vanes) are increasing linearly at rotor / casing ratios =0.95, 0.90, 0.85, 0.80, 0.75 to 0.70 and it almost varies in similar values. Thus optimal performance of total power is seen to be 4.02 kW at rotor / casing ratios =0.70 for vanes number 10 nos. (θ= 36°). But variation of total power for the rotor vanes =6 nos. (θ=60°), and 4 nos. (θ= 90°), are increasing linearly from rotor / casing ratios =0.95 to 0.75 and found smaller in its value than (*d/D*) =0.70.

From Fig. 8, a critical study is carried out and it is observed that the total power bar lines are almost equals at vane angle 30°-36° (vanes= 12-10 nos.) for *d/D* ratios 0.90, 0.85, 0.80 and 0.75. The power output appears to be maximum when vane angle is 36° (vanes= 10 nos.) at *d/D* ratio=0.70. Accordingly a graph is drawn between "Total output power versus different vane angles" e.g., = 30° (12 vanes), 36° (10 vanes), 45° (8 vanes), 60° (6 vanes) and 90° (4 vanes) as shown in Fig. 9. It is thus observed that in the vane turbine total shaft power output is although the combined effect of the component of expansion power and exit flow power, but contribution of expansion power is predominant. The contribution of exit flow power due to steady flow in respect to total power output varies from 6.2% to 31.4% for injection pressure of 6 bar and speed of rotation at 2500 rpm.

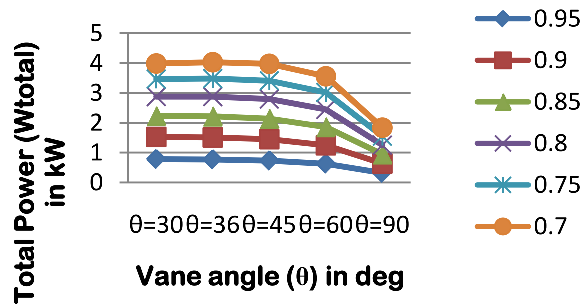

Fig.9 : Total power output versus vane angles (vane numbers) at different rotor / casing (d/D) ratio when injection pressure = 6 bar and speed of rotation =2500 rpm

From Fig. 9, it is obvious that the expansion power output as well as total power output is found optimum when vane angle ranges from 30°-45° (12-8 vane nos.), injection angle at 60° or above, at rotor/casing diameter ratio 0.70, speed of rotation at 2500 rpm and injection pressure at 6 bar and will be a deciding factor for desired shaft power output.

VI. Conclusions

From the above investigations based on input parameters, such as; injection pressure, injection angle and speed of rotation when kept 6 bar, 60° and 2500 rpm respectively, following conclusions are drawn:

Total output power from the air turbine is seen to be larger at injection pressure 6 bar, speed of rotation 2500 rpm and different rotor/casing diameter ratios at particular vane angles and total power ranges as shown below :

- 3.98 kW to 4.02 kW, when rotor to casing diameter ratio is 0.70 and vane nos. 12-8 (vane angle 30° -45°),
- 3.46 kW to 3.48 kW, when rotor to casing diameter ratio is kept 0.75 and vanes nos. 12-10 (vane angle 30° to 36°),
- 2.79 kW to 2.87 kW, when rotor to casing diameter ratio is kept 0.80 and vanes nos. 12-10 (vane angle 30° to 36°).

Thus optimum shaft power output of a novel vaned type air turbine is obtained when the design parameters for rotor diameter to casing diameter (*d/D*) ratios are kept between 0.75 to 0.70 and vanes nos. 12-10 (vane angle is of 30° to 36°) and plays an important role in designing the air turbine.

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VII. Nomenclature

d	diameter of rotor (2r) in meter
D	diameter of outer (2R) cylinder in meter
L	length of rotor having vanes in meter
n	no. of vanes=(360/θ)
N	no. of revolution per minute
p	pressure in bar
p_1, v_1	pressure and volume respectively at which air strike the Turbine,
p_4, v_4	pressure and volume respectively at which maximum expansion of air takes place,
p_5	pressure at which turbine releases the air to atmosphere.
v	volume in cu-m
w	theoretical work output in Nm
W	theoretical power output (Nm/s)
X_{1i}	variable extended lengths of vane at point 1
X_{2i}	variable extended lengths of vane at point 2
bar	(1 / 1.0132) atmospheric pressure

Subscripts

1, 2...4, 5	subscripts – indicates the positions of vanes in casing
e, exp	expansion
f, flow	flow
t, total	total
min	minimum
max	maximum

Greek symbols

α	angle BOF
α_1	angle LOF(=180-ϕ)
α_2	angle KOF (=180- θ-ϕ)
β	angle BAF
γ	1.4 for air
θ	angle between 2-vanes (BOH)
ϕ	angle at which compressed air enters into rotor through nozzle
ξ_d	eccentricity *(R-r)*

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