Analytical Evaluations of Shaft Output on Different Rotor to Casing Diameter Ratios at Optimal Value of Vane and Injection Angles for a Multi-Vane Air Turbine

Bharat Raj Singh[1*] and Onkar Singh[2]

ABSTRACT

The atmospheric air is freely available but once it is compressed, it develops the potential power source for running any prime mover as a zero pollutant. Thus air turbine proposed to be installed here on motorbike transforms the energy of the compressed air into shaft work in place of an internal combustion engine. This paper details a mathematical modeling and performance evaluation of an air driven vane type rotary novel air turbine / engine. The effect of isobaric admission and adiabatic expansion of high pressure air for different rotor to casing diameter ratios at optimum vane and injection angles as 45^0, 30^0 respectively have been considered and analyzed. The optimum work output is seen at some typical values of rotor / casing diameter ratios at moderate air consumptions. In this study, the power obtained along linear expansion (without excessive air consumption) as 6.0-7.6 kW (8-10 HP) between rotor to casing diameter ratios 0.85 to 0.80, at casing diameter of 200 mm, injection pressure of 90 psi (6 bar) and speed of rotation 2500 rpm which is enough to run any motorbike or light vehicle.

Keywords : *Zero pollution, compressed air, air turbine, injection angle, rotor / casing diameter ratios, motorbike.*

I. INTRODUCTION

WORLDWIDE increasing demand of transport vehicles has resulted heavy consumptions of fossil fuel, thereby causing threat to fast depletion to energy resources. Marion King Hubbert [1] a noted US based geophysicist was the first man who predicted in 1956 that conventional crude-oil production would attain Peak Oil in 1970 and thereafter start depleting. This would cause serious threat to mankind within 40 years (i.e. by 1995). He also indicated that the environment may get badly affected due to release of huge quantities of pollutant in the atmosphere. Aleklett K. and Campbell C.J., [2] indicated in 2003 that the world is depleting its resources of oil and gas at such a rate that oil production is set to peak and begin to decline by around 2010. This apprehension necessitates 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 some energy conversion systems like battery storage, hydrogen cell, compressed air etc to obtain shaft work for the engines of vehicles [3-9].

Compressed air has enormous potential as an alternative to these issues due to its zero pollutant capability and for running prime mover like air turbine. The French technologist Guy Negre [10] and quasi turbine inventor G. Saint Hilaire [11] did pioneering work in the field of compressed air engine. The use of compressed air as working fluid offers a prime mover which does not involve combustion process for producing shaft work. Thus, the great advantages in terms of free availability of air as fuel and the emissions free from carbon dioxide, carbon monoxide and nitrous oxides is apparent from such air engine. Compressed air driven prime movers are also found to be cost effective compared to fossil fuel driven engines. In view of these attractive features, the compressed air engine may become the dominant technology in

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place of the electric and hydrogen cell vehicle markets. Some studies [12-21] for optimizing the efficiency of various types of turbines have also been done.

In this paper the study of influence of rotor / casing dimension on the performance of air turbine is evaluated. Such air engine is proposed to be equipped on the motorbike in place of an internal combustion engine, which transforms the energy of the compressed air into shaft work for running the vehicle. The mathematical modeling and performance evaluation of a small capacity compressed air driven vaned type novel air turbine is presented here. The effect of isobaric admission and adiabatic expansion of high pressure air for different rotor diameters, casing diameters and rotor / casing diameter ratios (d/D) of the turbine have been considered and analyzed for linear expansion to optimize performance at moderate air consumptions rather than excessive one.

II. AIR TURBINE MODEL

The proposed rotary vane type novel air turbine as shown in Figures 1a and 1b has been considered to work on the reverse of working principle of vane type compressor. In this arrangement total shaft work is cumulative effect of isobaric admission of compressed air jet on vanes and the adiabatic expansion of high pressure air. Vanes of novel air turbine were placed under spring loading to maintain their regular contact with the casing wall to minimize leakage.

The present objective is to investigate the performance of an air turbine with the variation of rotor / casing dimensions. The air turbine considered has capability to yield output of 8.1 to 9.8 HP at 4-6 bar air pressure and for speed of 2500-3000 rpm, which is enough power to run any motorbik, which is suitable for a motorbike.

A cylinder for the storage of compressed air with a minimum capacity of storing air for the requirement of 30 min running at initial stage and maximum pressure of 20 bar is used as a source of compressed air.

III. MATHEMATICAL MODELING

The mathematical model shown here is already presented in the author's earlier publications [22-34]. But for the benefits of readers it is again reproduced here. The high pressure jet of air at ambient temperature drives the rotor in novel air turbine due to both

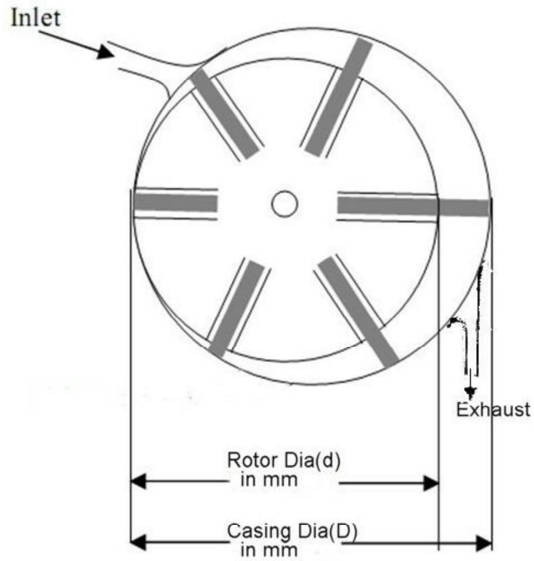

Fig.1a Air Turbine-Schematic Drawing

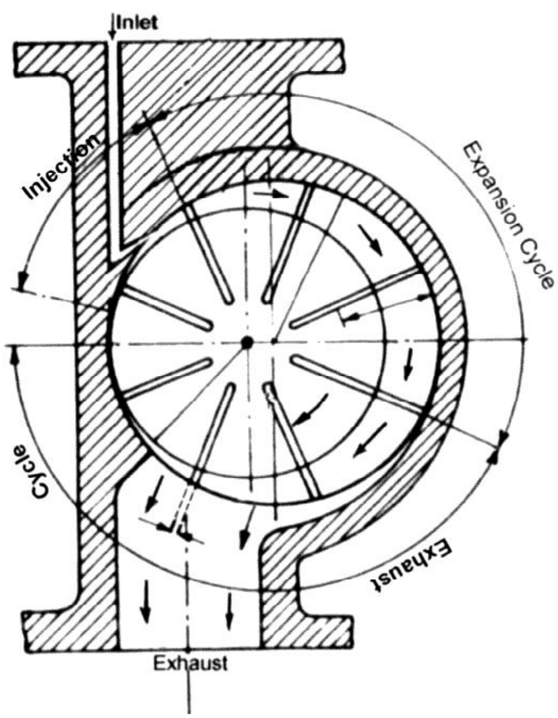

Fig.1b Air Turbine- Model

isobaric admission and adiabatic expansion. Such high pressure air when enters through the inlet passage, pushes the vane for producing rotational movement through this vane and thereafter air so collected between two consecutive vanes of the rotor is gradually expanded up to exit passage. This isobaric admission and adiabatic expansion of high pressure air both contribute in producing the shaft work from air tur-

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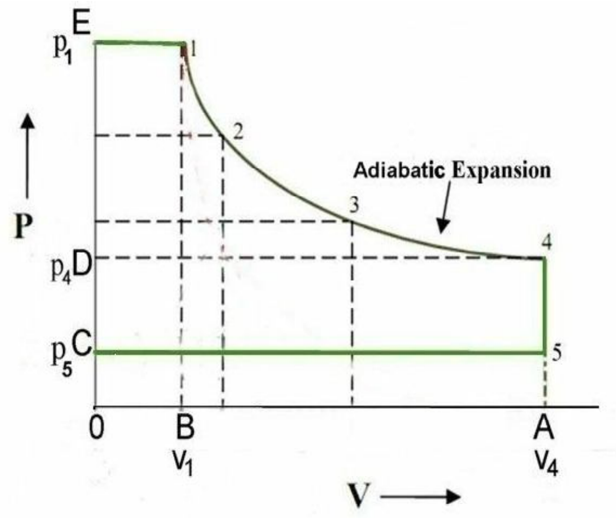

Fig. 2 Thermodynamic Processes (Isobaric, adiabatic and Isochoric Expansion)

bine. Compressed air leaving the air turbine after expansion is sent out from the exit passage. It is assumed that the scavenging of the rotor is perfect and the work involved in recompression of the residual air is absent.

From Figure 2, it is seen that work output is due to isobaric admission (E to 1), adiabatic expansion (1 to 4) and steady exit flow work (4 to 5). Thus, total work done due to thermodynamic process may be written as:

[Area under (E145CE)] = [Area under (E1BOE) +Area under (14AB1) - Area under (4AOD4) + Steady Flow (45CD4)]

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

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

From the above eq. (1) thermodynamic expansion work can be written as

$$w_1 = p_1.v_1 + \left(\frac{p_1.v_1 - p_4.v_4}{\gamma - 1}\right) - p_4.v_4 \text{ or}$$

$$w_1\left[\left(\frac{\gamma}{\gamma-1}\right).(p_{1.}v_1 - p_{4.}v_4)\right]$$

For adiabatic process, $p.v^\gamma = p_1.v_1^\gamma = p_4.v_4^\gamma$ or

$$v_4 = \left(\frac{p_1}{p_4}\right)^{\frac{1}{\gamma}}.v_1$$

Thus thermodynamic expansion work output can be written as

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

From the above eq. (1) steady flow work can be written as

$$w_2 = \int_4^5 v.dp = (p.v_4 - p_5.v_5) \qquad (3)$$

After the expansion process during exit flow the pressure cannot fall below atmospheric pressure . Thus, from eq. (1) the net work output will be:

$$w = (w_1 + w_2) = \left(\frac{\gamma}{\gamma-1}\right).p_1.v_1.\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\} + (p_4 - p_5).v_4 \qquad (4)$$

When air turbine is having n number of vanes, then shaft output [34] can be written as,

$$w_n = n.\left(\frac{\gamma}{\gamma-1}\right).p_1.v_1\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\} + n.(p_4 - p_5).v_4 \qquad (5)$$

Where is work output (in Nm), for complete one cycle.

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

$$W_{total} = n.(N/60).\left(\frac{\gamma}{\gamma-1}\right).p_1.v_1.\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\} + n.(N/60).(p_4 - p_5).v_4 \qquad (6)$$

Where $W_{exp} = n.(N/60).\left(\frac{\gamma}{\gamma-1}\right).p_1.v_1.\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}$

and $W_{flow} = n.(N/60).(p_4 - p_5)v_4$

Figure 1a and 1b 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. v1 to v4) can be derived by the variable extended length of vane as shown in Figure 3 at every point of movement between two consecutive vanes.

From Figure 3, it is seen that when two consecutive vanes at OK and OL moves 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 is assumed as can be written from the geometry:

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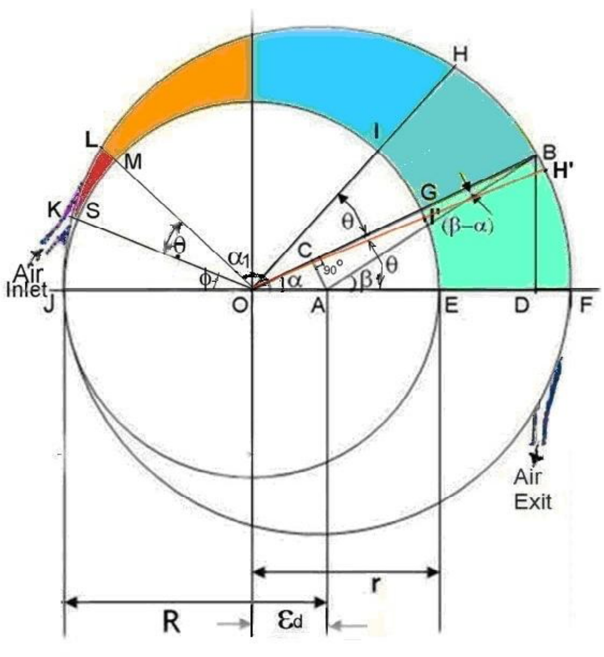

Fig. 3 Variable length BG and IH and injection angle ϕ

$$BG = x_{\alpha t,\text{variable}''\alpha''} = (1/2).D.cos\left[\sin^{-1}\left\{\left(\frac{D-d}{D}\right).\sin\alpha\right\}\right] + (1/2).(D-d).\cos\alpha - d/2 \quad (7)$$

Where D is diameter of casing and d is diameter of rotor, α is angle ∠BOF, β is angle ∠BAF and θ is angle ∠HOB or ∠KOL, between two consecutive vanes and ϕ is angle ∠KOJ at which injection pressure enters the air turbine.

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

$$v_{cuboids} = L.\left\{\frac{(X_{1i} + X_{2i})(d + X_{1i})}{4}\right\}.\sin\theta \quad (8)$$

where BG= X_{1i} and IH=X_{2i} variable length of vanes as shown in Figure 3.

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

Applying above conditions into equations (7), then LM = X_{1min} and SK = X_{2min} can be written as:

$$X_{1min} = \frac{D}{2}Cos\left[\sin^{-1}\left\{\left(\frac{D-d}{D}\right) - \sin(180 - \theta - \phi)\right\}\right] + \left[\left(\frac{D-d}{2}\right) - Cos\,(180 - \theta - \phi)\,\frac{d}{2}\right] \quad (9)$$

$$X_{2\,min} = \frac{D}{2}Cos\left[\sin^{-1}\left\{\left(\frac{D-d}{D}\right).\sin(180 - \phi)\right\}\right] + \left[\left(\frac{D-d}{2}\right).\cos(180 - \phi) - \frac{d}{2}\right] \quad (10)$$

$$v_1 = v_{min} = L.\left\{\frac{(X_{1min} + X_{2min})(d + X_{1min})}{4}\right\}.\sin\theta \quad (11)$$

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 (7), then FE =X_{1max} = Corresponding to BG at =0 degree and I'H' = X_{2max} = Corresponding IH at $(\alpha + \theta) = \theta$ degree can be written as:

$$X_{1max} = (D - d) \quad (12)$$

$$X_{2max} = \left(\frac{D}{2}\right)Cos\left[\sin^{-1}\left\{\left(\frac{D-d}{D}\right) - \sin\theta\right\}\right] + \left\{\left(\frac{D-d}{2}\right).Cos\,\theta\,\frac{d}{2}\right\} \quad (13)$$

Applying values of X_{1max} and $X_{2\,max}$ to equation (8),

$$v_4 = v_{max} = L.\left\{\frac{(X_{1max} + X_{2\,max})(d + X_{1max})}{4}\right\}.\sin\theta \quad (14)$$

Applying values of and from equations (11) and (14) to equation (6), the total power output available W_{total}, can be written as:

$$W_{total} = n(N/60).\left(\frac{\gamma}{\gamma-1}\right).\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}.p_1\left[L.\left\{\frac{(X_{1min+}X_{2min})(d + X_{1min})}{4}\right\}.\sin\theta\right] + n(N/60).(p_4 - p_5).\left[L.\left\{\frac{(X_{1max} + X_{2max}).(d + X_{1max})}{4}\right\}.\sin\theta\right] \quad (15)$$

IV. ASSUMPTION AND INVESTIGATION PARAMETERS

Various input parameters considered for study are listed in Table 1. Using the mathematical model the effect of speed of rotation, rotor/casing diameter ratio and injection pressure on the expansion power output, flow work output and total power output from air turbine is studied. Here the vane angle, injection angle and speed of rotation of the air turbine are considered to be constant for whole study. The results obtained have been plotted in Figures 4 to 8, for the rotor/casing diameter ratio (d/D), varied as 0.95,

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0.90, 0.85, 0.80, 0.75, 0.65, 0.60 and 0.55 at vane angle of 45^0, injection angle of 30^0, at different injection pressures of 2, 3, 4, 5 and 6 bar (i.e., 30, 45, 60, 75 and 90 psi), speed of rotation of 2500 rpm, and casing diameter 200 mm.

Figure 4 shows the variation of expansion power at different rotor/casing diameter ratios with respect to different injection pressure. It is evident that the shaft power due to expansion at 2 bar is lower at higher rotor/casing diameter ratio of 0.95, thereafter gradually increases linearly upto 0.85 to 0.80 and, largest when rotor/casing diameter ratio is kept 0.55. For higher injection pressure 4 to 6 bar, this is attributed to the large work output per time unit in similar pattern. It is evident that there exists maximum rotor/casing diameter for every injection pressure which offers the linear expansion power at moderate air consumption and beyond 0.75 to 0.55 rotor / casing (d/D) ratios, the value of maximum expansion power is more but expansion is parabolic which shows the higher air consumption for higher shaft output. The higher injection pressures produces higher shaft power in similar manner as compared to lower injection pressures.

Table 1
Input Parameters

Symbols	Parameters
Ratio of Rotor to Casing diameter (d/D)	0.95, 0.9, 0.85, 0.80, 0.75 , 0.70, 0.65, 0.60 and 0.55 when casing diameters are kept D=200 mm
p_1	2 bar (?30 psi), 3 bar (?45psi), 4bar (?60psi), 5bar (?75psi), 6bar (?90psi) – inlet pressures
p_4	$(v_1/v_4)^{\gamma}.p_1 > p_5$ assuming adiabatic expansion
p_5	(p_4/1.1)= 1.0132 bar- exit pressure
N	2500 rpm
L	45 mm length of rotor (assumed minimum)
n	($360/\theta$) number of vanes in rotor
γ	1.4 for air
θ	45^o angle between 2-vanes, (i.e. rotor contains correspondingly 8 number of vanes)
ϕ	30^o angle at which compressed air through nozzle enters into rotor

V. RESULTS AND DISCUSSION

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the rotor/casing diameter ratio (d/D), varied as 0.95, 0.90, 0.85, 0.80, 0.75, 0.65, 0.60 and 0.55 at vane angle of 45^0, injection angle of 30^0, at different injection pressures of 2, 3, 4, 5 and 6 bar (i.e., 30, 45, 60, 75 and 90 psi), speed of rotation of 2500 rpm, and casing diameter 200 mm.

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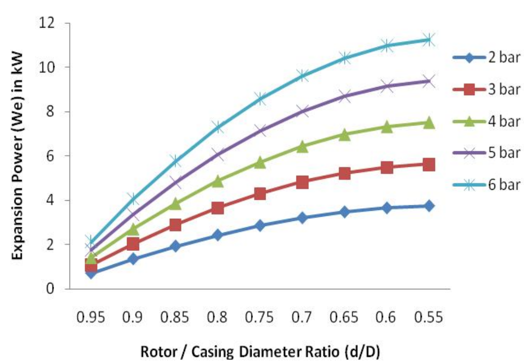

Fig. 4 Expansion Power vs. Rotor / Casing Diameter (d/D) Ratio when D= 200 mm

Figure 5 shows the variation of exit flow power at different rotor/casing diameter ratios with respect to different injection pressure. It is evident that the shaft power due to exit flow work is lowest at 2 bar and parabolically increases up to rotor/casing diameter ratio of 0.55. It is quite evident that the shaft power due to exit flow work gradually increases with reducing value of rotor/casing diameter ratio in view of the gap between the rotor and casing as increases gradually. On these ground, the exit flow power is nearly insignificant for rotor/casing diameter ratio of 0.95 and would be absent when this ratio value is unity.

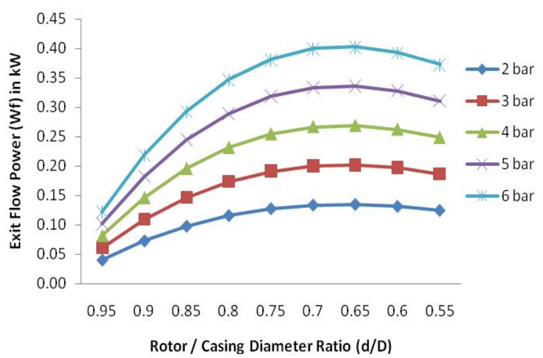

Fig. 5 Exit Flow Power vs. Rotor / Casing Diameter (d/D) Ratio when D= 200 mm

Figure 6 shows the percentage contribution of expansion power against total work output at different rotor/casing diameter ratios with respect to different injection pressure. It is evident that percentage contribution of expansion power is low at (d/D) ratio =0.95 and highest at (d/D) =0.55 for all injection pressure 2- 6 bar. At rotor / casing ratio 0.95 the contribution of expansion power against total power is lowest and gradually increases from 94.58% to 96.80% as rotor/casing diameter ratio decreases from 0.95 to 0.55.

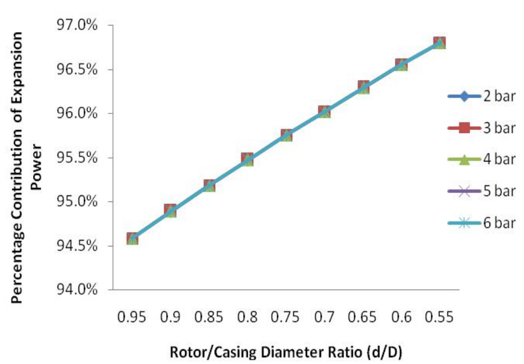

Fig. 6 Percentage Contribution of Expansion power vs. Rotor / Casing Diameter (d/D) Ratio when D= 200 mm

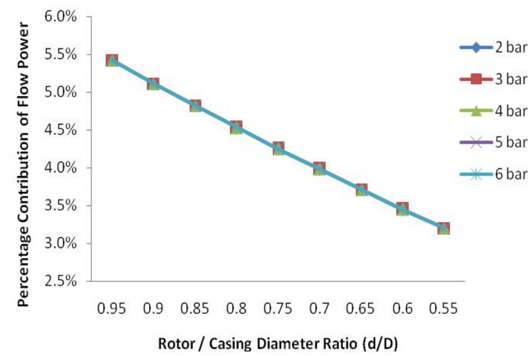

Fig. 7 Percentage Contribution of Exit Flow Power vs. Rotor / Casing Diameter (d/D) Ratio when D= 200mm

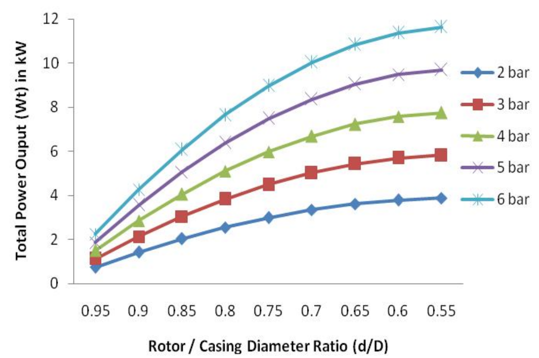

Fig. 8 Total Power output vs. Rotor / Casing Diameter (d/D) Ratio when D= 200 mm

Figure 7 shows the percentage contribution of exit flow power in total power output at different rotor/casing diameter ratios with respect to different injection pressure. It is evident that percentage contribution of exit flow power is higher, when rotor/casing diameter ratio is 0.95 and gradually decreases from 5.42% to 3.20% as this diameter ratio drops up to 0.55 when casing diameter is kept 200 mm at all injection pressure from 2-6 bar.

Variation of total power output with respect to different rotor/casing diameter ratios with respect to different injection pressure2-6 bar is shown in Figure 8. Total power at 2-6 bar is seen increasing linearly from rotor/casing diameter ratio 0.95 to 0.80 and gradually further increases parabolically to highest when rotor/casing diameter ratio reaches to 0.55. This shows the behavior of higher air consumption. Thus for moderate air consumption maximum value of shaft power output is obtained as 6.08 -7.65 kW at rotor/casing diameter ratio 0.85 to 0.80.

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Thus it is observed that in the rotary vane type air turbine total shaft power output is combined effect of the component of expansion power and exit flow power. The contribution of exit flow power due to steady flow in respect to total power output varies from 3.20% to 5.42% only for all injection pressure 2-6 bar at constant injection angle 30^0, constant vane angle 45^0, at speed of rotation 2500 rpm. Thus it is obvious that the expansion power output as well as total power output is found optimum as 7.30 and 7.65 kW respectively for moderate air consumption when rotor/casing diameter ratio lies between 0.85 to 0.80 at casing diameter 200 mm and is a deciding factor for desired shaft power output.

VI. CONCLUSIONS

The results obtained from above investigations based on input parameters such as injection angle, vane angle and speed of rotation are kept 30^0, 45^0 and 2500 rpm respectively, following conclusions are drawn:

- There exists an linear value of shaft power output at rotor/casing diameter ratio (approx. 0.85 to 0.80) and between 0.75 to 0.55, though the shaft output increases in parabolic form that indicates higher air consumption for the considered air turbine for all air injection pressures. Thus the rotor /casing diameter ratio 0.80 offers the optimum expansion power to 7.299 kW at injection air pressures 6 bar for the moderate air consumption.
- The exit flow power due to steady flow is seen to increase parabolically for the rotor/casing diameter ratio varying from 0.95 to 0.55 and is found maximum 0.12 to 0.37 at 6 bar injection pressure.
- Total output power from the air turbine is seen to be optimum for the higher injection air pressure and there exists an optimum value of rotor/casing diameter ratio for all injection pressure 2-6 bar at linear increase (moderate air consumptions). The optimum power output is seen to be 7.645 kW at 0.80 rotor/ casing ratio for injection pressure of 6 bar.

Thus the optimum shaft power output of a rotary novel vane type air turbine having casing diameter 200 mm and other design parameters such as: rotor to casing diameter (d/D) ratios between 0.85 to 0.80 and at optimum value of vane angle 45^0 (8 vanes) and pressure injection angle 30^0, as already investigated in the earlier studies, offers 7.645 kW power output. Thus air consumptions have also an important role for optimizing the power output.

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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

Subscripts

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

Greek symbols

α	angle BOF
α_1	angle LOF (=180-θ-ϕ)
α_2	angle KOF (=180-θ-ϕ)
β	angle BAF
γ	1.4 for air
θ	angle between 2-vanes(BOH)

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ϕ angle at which compressed air enters into rotor through nozzle

ξ_d eccentricity (R-r)

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