MIT International Journal of Mechanical Engineering Vol. 1, No. 1, Jan 2011, pp 9-16
ISSN No. 2230-7699 © MIT Publications

Performance Investigations for Power Output of a Vaned Type Novel Air Turbine

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Abstract- **Greater utilization of hydrocarbon fuel in the transport sector is causing serious challenges to depletion of oil and thereby environmental and ecological imbalances. Thus the major thrust has now been given towards search of alternative energy source. Atmospheric air can also be one of the cost effective energy conversion system. The atmospheric air could be got compressed by alternative natural energy sources such as: solar energy, wind energy etc. and can be stored in the energy tank at every domestic house. Such compressed air can be utilized as potential zero pollution working fluid for producing shaft work in the air turbine. This paper details the mathematical modeling and performance evaluation of a small capacity compressed air driven vaned type novel air turbine. The air turbine parameters are considered as; vane angle 45^0, injection angle 60^0, and rotor / casing diameters ratio (*d/D*= 0.75). The effects on shaft power output due to different air pressure from 2-6 bar, have been analyzed. The study shows that the flow power to the total power output has significant contributions and is found 8.8% and contribution of expansion power is found 91.2% at all injection pressures from 2 to 6 bar and speed of rotation from 500-3000 rpm for the said novel air turbine. It is also observed that maximum power output of 4.1 kW(5.5 HP) is obtained at 6 bar injection pressure and 3000 rpm speed of rotation, which is presently sufficient to drive any motorbike or domestic appliances in near future.**

Keywords: zero pollution, compressed air, air turbine, energy conversion, impingement action, injection angle.

I. INTRODUCTION

About 100 years ago, the major thrust of energy shifted from recent solar to fossil fuel (hydrocarbons). Technological advances have led to a greater use of hydrocarbon fuels [**1**], making civilization vulnerable to depletion in supply. The made study by Aleklett and Campbell [**2**] in the year 2004, predicts that if the oil is consumed at the current rates, then by 2020, we will be consuming 80% of the entire available resources. This necessitates the search for alternative of oil as energy source or preserving it by tapping some other alternatives such as non-conventional energy like battery operated vehicles, wind mills, photocells etc. and to convert their output into mechanical energy.

Presently because of better developments and availability of facilities, urban population is raising use of vehicles rapidly, causing air pollution and greenhouse gases that come from vehicle emissions. This is the primary motivation behind developing alternative transportation technologies that do not rely on combustion of fossil fuels. Consumer acceptance of a replacement transportation technology, however, is highly dependent on the new vehicle sticker price, operating expenses, reliability, and convenience of use. The worldwide researches are also going on for other alternatives such as use of Hydrogen Fuel Cell, which is presently very costly, use of Bio-Diesel or use of compressed air for vehicle engines [**3-9**].

Thus, it is advantageous for the energy storage system of a Zero Pollution Vehicle (ZPV) to have low initial cost, be quickly and economically recharged, and to provide driving performance comparable to that of conventional automobiles. In addition, the most desirable technological solutions will eliminate the release of automotive combustion products in areas of poor air quality, while also reducing the net amount of pollutants released to the environment as a consequence of their implementation. Careful consideration of the overall environmental impact of a particular ZPV technology and the corresponding costs of necessary infrastructure developments are required to evaluate the ability of any new transportation system to meet the goals of society. Currently, the battery-powered electric vehicles and hydrogen cell vehicles are the only commercially available technology that can meet ZPV standards; however, these vehicles have not captured market and sold well due to their limited range, no proper facilities for recharge, and high initial cost. All of these issues have given the birth to the technology of compressed air energy storage and its utilization in transport vehicles and other domestic utilities.

The *Guy Negre*, a French technologist and inventor has developed 4- cylinder compressed air engine, which can run the vehicle at 60-80 miles per hour speed without tail pipe emission [**10**]. They registered about 52- patents during 1998 to 2006. In January 2007, MDI and Tata Motors entered into an agreement to develop such vehicles in commercial use. **'**G. Saint Hilaire' an inventor of quasi turbine has also developed hybrid car running on compressed air and gasoline [**11**]. These highly compressed air energy storage systems with 300

MIT International Journal of Mechanical Engineering Vol. 1, No. 1, Jan 2011, pp 9-16
ISSN No. 2230-7699 © MIT Publications

psi, which can be filled within 15-20 minutes, may remain the dominant technology in the electric and hydrogen cell vehicle market.

This paper presents a mathematical model for the analysis of shaft output of a vaned type novel air turbine running on compressed air energy. The different total shaft power output is obtained by applying the various parametric conditions such as: injection angle, vane angle, rotor and casing diameter ratio.

II. Vaned Type NOVEL AIR TURBINE

A schematic vaned type air turbine as shown in Fig.1 (a) and air turbine model as shown in Fig.1 (b) have been considered here. The present objective is to develop an air engine using air turbines with the output of 5.50 to 6.80 HP at 500–750 r/min for meeting starting torque requirements at 4–6 bar air pressure and the required torque at a normal speed of 2000–2200 r/min at 2–3 bar air pressure, which is suitable for a motorbike. Salient features of the development of air turbine are given ahead. A cylinder for the storage of compressed air with a minimum capacity of storing air for the requirement of 30min running at initial stage and maximum pressure of 200–300 psi (13–20 bar) is used as a source of compressed air.

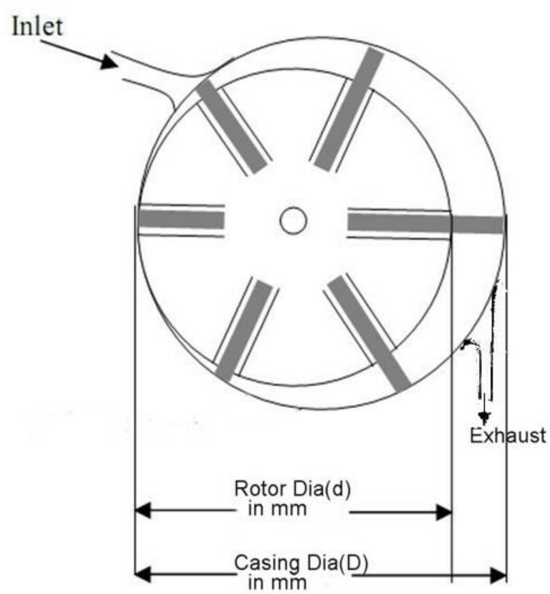

Fig.1 (a) Air Turbine Schematic Model

The compressed air storage cylinder is also designed to produce constant pressure without the much variation in the torque at low air pressure. For this, a spring-loaded baffle is installed into the cylinder and filter, regulator and regulator are attached to maintain constant air pressure. The various design aspects for optimizing shaft output were also studied [**12 - 20**].

The vanes of novel air turbine are also placed under spring loading to maintain regular contact with the casing /cylinder wall to minimize leakage which is proposed as improvisation over the currently available vane turbine.

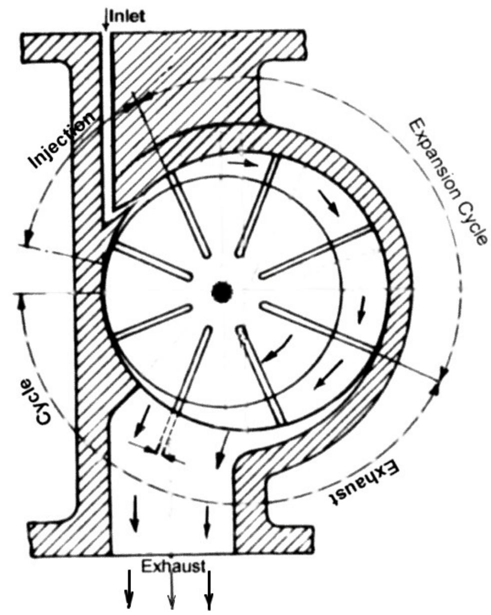

Fig.1 (b) Air Turbine Model

Thus such novel vaned type air turbine is proposed to be installed on the motorbike as shown in Fig.1(c), with a twin compressed air cylinder and attachments of filter, regulator and lubricator, which could be capable to develop desired output.

Fig.1(c) Computerized Sketch of Motorbike installed with Air Tank and Air Turbine

III. Mathematical Modeling

The high pressure jet of air at ambient temperature drives the rotor in novel air turbine due to both 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 contribute in producing the shaft work from air turbine. 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 as seen from Figure 1b. Similar type mathematical modeling is considered in earlier publications by authors and it is being reproduced here for maintaining continuity and benefits to the readers [**21-34**].

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

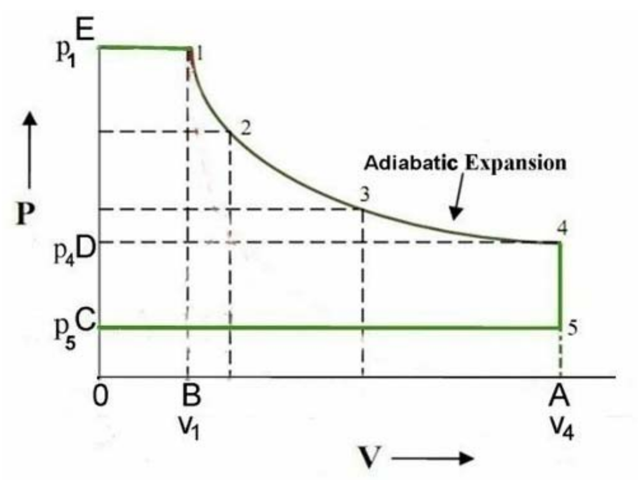

Fig. 2 Thermodynamic Processes (Isobaric, adiabatic and Isochoric Expansion)

[Area under (E145CE)] = [Area under (E1BOE) +Area under (14AB1) – Area under (4AOD4) + Exit steady flow (45CD4)] or

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

or $$w = [(w_1) + (w_2)] \tag{1}$$

Now thermodynamic expansion work (w_1), can be written as:

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

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

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

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

Thus thermodynamic expansion work output would be:

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

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:

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

Applying equations (2), (3) into equation (1), therefore net work output will be:

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

when air turbine is having *n* number of vanes, then shaft output [**35**] can be written as:

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

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 mentioned as:

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

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

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.

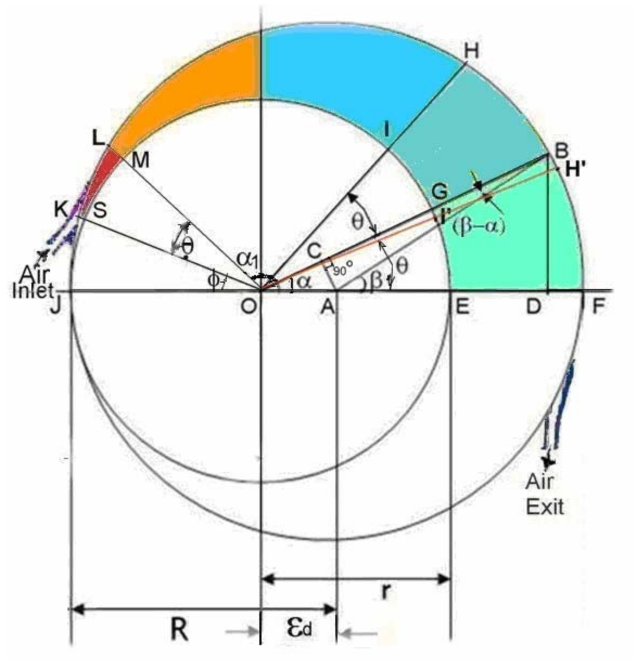

Fig. 3 Variable length BG and IH and injection angle ø

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 α_i is assumed as $X_{at'\mathrm{variable}'\alpha}$ can be written from the geometry [23-25]:

$$BG=X_{at,\mathrm{variable}'\alpha'}=R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right).\sin\alpha\right\}\right]+(R-r).\cos\alpha-r \quad (7)$$

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

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

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

Where BG= X_{1i} and IH= X_{2i} variable length of vanes when rotate into turbine as shown in Figure 3. *The lengths (IG, HB and LK, SM.), are considered linear whereas all are chords of circles. This approximation is done in mathematical model which has very least impact on the overall values.*

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

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

$$X_{1\min}=R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right).\sin(180-\theta-\phi)\right\}\right]+\left[(R-r).\cos(180-\theta-\phi)-r\right] \quad (9)$$

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

Applying values of X_{1min} and X_{2min} to equation (8),

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

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

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

$$X_{1\max}=(D-d)=2(R-r) \quad (12)$$

$$X_{2\max}=R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right).\sin\theta\right\}\right]+\left\{(R-r).\cos\theta\right\}-r \quad (13)$$

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

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

Applying values of v_1 and v_4 from equations (11) and (14) to equation (6), the total power output available W_{total} can be written as:

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

IV. INPUT PARAMETERS AND ASSUMPTIONS

Detailed analysis to derive relations between injection angles to vane angle for optimizing power output has been carried out in earlier studies by authors in respect to variation in expansion and flow work, percentage contribution of expansion and flow work, and total work output at admission pressure of 2 - 6 bar and speed of rotation of 2500 rpm [**24-28**]. The optimal vane angle (θ) and injection angle (ϕ) of air turbine were found 45° (i.e. 8 vanes) and 60° respectively at rotor/casing (*d/D*) ratio of 0.75. Here other input parameters are listed in Table 1 for investigation of optimum shaft power output at different injection pressure and speed of rotation.

TABLE I
INPUT PARAMETERS

Symbols	Parameters
$D=2R$	100 mm (outer)
$d=2r$	75 mm (inner) corresponding
p_1	2 bar (=30 psi), 3 bar (=45 psi), 4 bar (=60 psi), 5 bar (=75 psi), 6 bar (=90 psi)
p_4	$=(v_1/v_4)^{\gamma}.p_1$ assuming adiabatic expansion
p_5	1 atm = 1.0132 bar
θ	45° angle between 2- consecutive vanes (i.e. rotor contains 8 vanes)
N	500, 1000, 1500, 2000, 2500 and 3000 rpm
L	45 mm length of rotor
γ	1.4 for air
n	$(360 / \theta)$ = number of vanes
$Ø$	60° Injection angles at which compressed air enters through nozzle into rotor

V. RESULTS AND DISCUSSIONS

Various input parameters considered here are listed in Table-1. Using the mathematical model, the effect of speed of rotation at different air pressure (2-6 bar) on the expansion power, flow power and total power output is studied. Here the vane angle (θ), injection angle ($Ø$) and rotor/casing diameter ratio (*d/D*) of the air turbine is considered to be constant for whole study. The results obtained are shown in Figs. (4) to (8) for the speed of rotation (*N*) varying as 500, 1000, 1500, 2000, 2500 and 3000 rpm, at vane angle of 45°, injection angle 60°, at different injection pressures 2-6 bar (30, 45, 60, 75 and 90 psi) and at the rotor/casing diameter ratio (*d/D*)=0.75 where *D*=100 mm.

Figure 4 shows the variation of expansion power at different the speed of rotation (*N*) varying from 500, 1000, 1500, 2000, 2500 to 3000 rpm, at constant vane angle 45°, air injection angle 60°, different air injection pressure from 2 to 6 bar and rotor/casing diameter ratio (*d/D*)=0.75 when casing diameter of 100 mm. It is evident that the shaft power output due to expansion at 2 bar (30 psi) and 500 rpm is lower and thereafter gradually it increases when injection pressure is increased from 3-6 bar (45-90 psi). It is also seen that the power becomes large at higher speed of rotation and higher injection pressure.

Figure 5 shows the variation of flow power output at different the speed of rotation (*N*), varying from 500, 1000, 1500, 2000, 2500 to 3000 rpm, at constant vane angle 45°, air injection angle 60°, different air injection pressure from 2 to 6 bar and rotor/casing diameter ratio (*d/D*)=0.75, when casing diameter is kept 100 mm. It is evident that the shaft power output due to flow power at 2 bar (30 psi) and 500 rpm is lower but significant and thereafter gradually it increases from 3-6 bar (45-90 psi) and is found large at higher speed of rotation and higher injection pressures.

Figure 6 shows the percentage contribution of expansion power to the total power output for different speed of rotation (*N)* varying from 500, 1000, 1500, 2000, 2500 to 3000 rpm, at constant vane angle 45°, air injection angle 60°, different air injection pressure from 2 to 6 bar and rotor/casing diameter ratio (*d/D*) =0.75 when casing diameter is 100 mm. It is evident that percentage contribution of expansion work is constant at injection pressure 2-6 bar. The contribution of expansion power to the total power is 91.2% at all speed of rotation and all pressure from 2- 6 bar. The contribution of flow power to the total power at all injection pressure 2-6 bar and speed of rotation from 500 – 3000 rpm as evident from Fig. 7 is 8.8 % though it is significant but the major contribution is due to expansion power only.

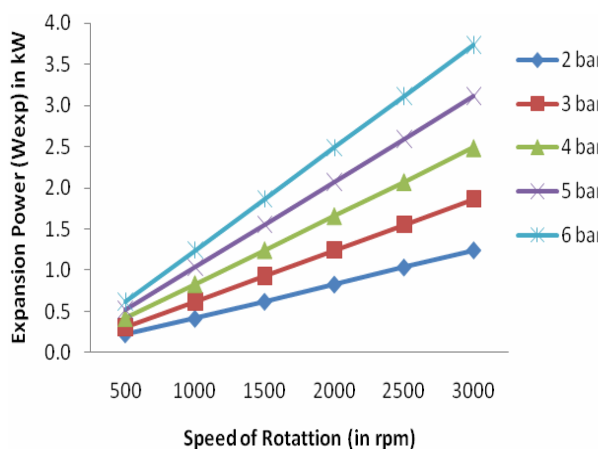

Fig. 4 Expansion power (*Wexp*) versus speed of rotation at vane angle 45°, injection angle 60° and *d/D* ratio 0.75 when D=100 mm

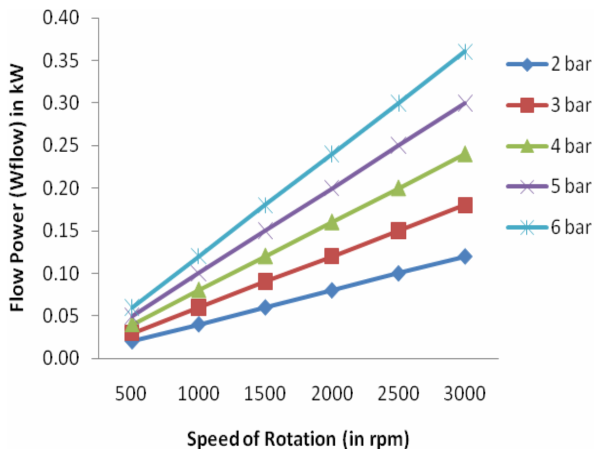

Fig. 5 Flow power (*Wflow)* versus speed of rotation at vane angle 45°, injection angle 60° and *d/D* ratio 0.75 when D=100 mm

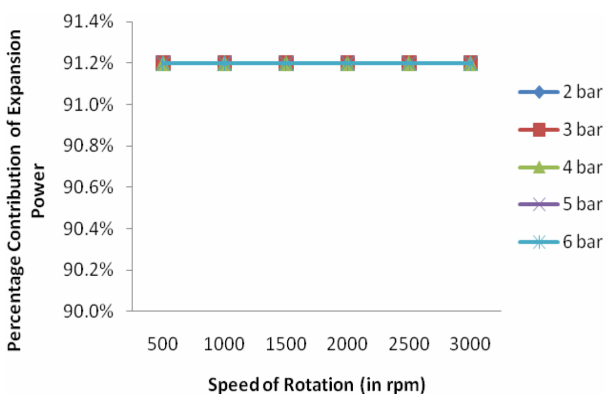

Fig. 6 Percentage contribution of expansion power versus speed of rotation at vane angle 45°, injection angle 60° and *d/D* ratio 0.75 when D=100 mm

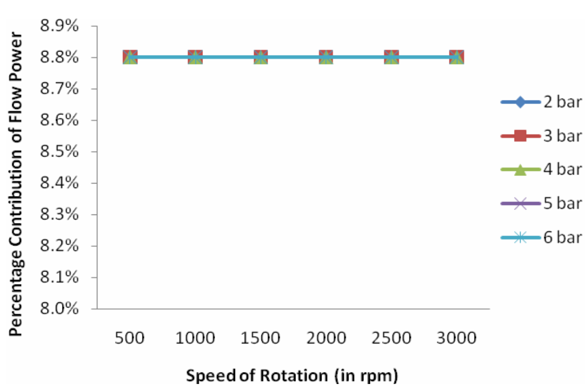

Fig. 7 Percentage contribution of flow power versus speed of rotation at vane angle 45°, injection angle 60° and *d/D* ratio 0.75 when D=100 mm

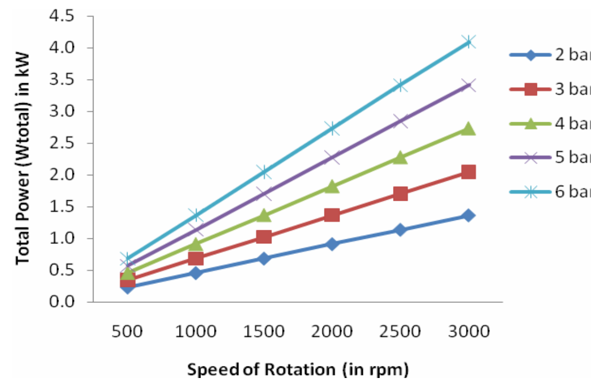

Fig. 8 Total power (*Wtotal*) versus speed of rotation at vane angle 45°, injection angle 60° and *d/D* ratio 0.75 when D=100 mm

Variation of total power output with respect to the different speed of rotation (*N*), varying from 500, 1000, 1500, 2000, 2500 to 3000 rpm, at constant vane angle 45°, air injection angle 60°, different air injection pressure from 2 to 6 bar and rotor/casing diameter ratio (*d/D*) =0.75 when casing diameter is kept 100 mm. From Fig. 8 it is seen that at different air injection pressure and total power output at 2 bar (30 psi) is seen lower at speed of rotation of 500 rpm and highest when speed of rotation becomes 3000 rpm.

It is thus, observed that total shaft power output is combined effect of the component of expansion and flow power. The contribution of flow power in respect to total work output at 2 - 6 bar (30 - 90 psi) is constant as 8.8 % but it is significant at all speed of rotations from 500 to 3000 rpm, constant vane angle 45° and constant injection angle 60°. It is also concluded that the expansion power as well as total power outputs are optimum when speed of rotation are on higher side 3000 rpm and at higher injection pressure 6 bar (90 psi).

VI. Conclusions

Based on optimal input parameters as (such as; vane angle 45°, injection angle 60° and *d/D* = 0.75, when casing diameter is 100 mm) and results obtained at higher injection pressure 6 bar and speed of rotation 3000 rpm, following conclusions are drawn;

- There exists a higher value of expansion power.
- The flow power is seen to be large and significant.
- Total power output from the air turbine is seen to be large at the higher injection air pressure and speed of rotation.

Thus total power output is seen to be 4.1kW (5.5 HP) at 6 bar and 3000 rpm, which is sufficient to run any motorbike as commercial one also runs at 80 km /hr speed (at 2200- 2500 rpm).

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 cum
w : theoretical work output in (J) Joules
W : theoretical power output (W) Watts
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
exp : expansion
min : minimum
max : maximum

Greek symbols

α : angle BOF (see Fig.3)
α_1 : angle LOF (=180-ϕ)(see Fig.3)
α_2 : angle KOF (=180-θ-ϕ)(see Fig.3)
β : angle BAF (see Fig.3)
γ : 1.4 for air
θ : angle between 2-vanes (BOH) (see Fig.3)
ϕ : angle at which compressed air enters into rotor through nozzle
ξ_d : eccentricity *(R-r)*

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