Critical Effect of Rotor Vanes with Different Injection Angles on Performance of a Vaned Type Novel Air Turbine

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***Abstract*—Globally faster consumption of hydrocarbon fuel in the transport sector is posing threat to environmental and ecological imbalances and thereby depletion of hydro-carbon fuel is causing another challenge to oil reserves. In view of these issues extensive researches are being carried out to explore the alternative energy source and or to find out appropriate energy conversion system. The atmospheric air once compressed, it is found as potential working fluid to produce shaft work for an air turbine and releases almost zero pollution in the environment. This paper details the mathematical modeling of a small capacity compressed air driven novel multi-vanes type air turbine. Effect of expansion of high pressure air collected between two consecutive vanes at different vane angles and varying inlet pressure have been analyzed here. The study shows that the total shaft power is found optimum at injection angle 60^o when vane angle θ=36^o (10 vanes) and it reduces at injection angle 45^o when vane angle θ=51.4^o (7 vanes) and further goes down at injection angle 30^o when vane angle θ=60^o-72^o (6-5 vanes), for injection pressure 6 bar and speed of rotation 2500 rpm.**

***Keywords*—zero pollution, compressed air, air turbine, flow power, vane angle.**

I. INTRODUCTION

The global fast consumption scenario of hydrocarbon fuel [**1, 2**] in transport sector alone is contributing about 70% of the total air pollution and its implications upon the environment and ecology are compelling factors to search for an environment friendly alternative to oil [**3-7**]. The important factor for such an alternative should have a zero or near zero pollution level, low initial cost, low running expenses, high degree of reliability, convenience and versatility of use. Use of compressed air for running prime mover like air turbine offers a potential solution to these issues, which does not involve combustion process for producing shaft work. Thus, the great advantages of such air motors are ; free of cost availability of air as fuel and free from emissions such as carbon dioxide, carbon monoxide and nitrous oxides. Compressed air driven prime movers are also found to be cost effective compared to fossil fuel driven engines. It only has perennial compressed air requirement which needs some source of energy for running compressor and overall analysis shows that the compressed air system is quite comparable and attractive option for light vehicle applications.

Worldwide researches are going on to make a sustainable alternate for transport vehicles and other domestic utilities [**8**]. The energy conversion technology for compressed air is in state of infancy. A French technologist Guy Negre [**10**] and an inventor of quasi turbine called G. Saint Hilaire [**11**] have carried out the pioneering work in the area of compressed air engine. The highly compressed air energy storage systems can be filled up to 20 bar pressure within 15–20 minutes, and reused for running compressed air engines. In view of these attractive features, the compressed air engine technology is found a cost effective alternative for vehicle markets rather than the electric and hydrogen cell vehicle.

This paper describes the study and analysis of a very small capacity air turbine with vane type rotor has been carried out for studying the effect of vane angle and inlet air pressure variation on air turbine performance. Results obtained using the mathematical modeling of proposed novel air turbine have been presented graphically and analyzed.

II. SCOPE OF COMPRESSED AIR ENGINE

The per capita income of a person in India, as a developing country, is very low to meet livelihood requirements. On the basis of the recent data 80% of the population of the country still lives in rural and suburban areas where the means of transport is either bicycle or motorbike. The continuous hikes of fossil fuel prices at the rate of around 20–30 per cent every year are making the situation miserable. Extrapolation shows that at this rate, by 2010-12, prices may be double as what they were in 2005, and by 2030–40, may touch Rs. 1000 per litre. A time will come when the common person will not be able to purchase fuel to run motorcycles. This is not only due to the high demand for vehicles or its increasing numbers worldwide, but also due to the cost of fossil fuel going high as 80 per cent of the available fossil fuel is presently being consumed in transport. Thus, it is imperative to explore the possibility of alternatives to fossil fuel to make the environment free from emission for keeping the present and future generations healthy.

During last two decades, major work has been done to tap the air freely available in the atmosphere and to compress it for storage in cylinders for further use. Apart from other uses of compressed air, this can also be used to run combustion engines with the mixture of gas and air getting fired after the compression stroke at top dead centre. The use of compressed air will eliminate the need of having a separate compression stroke. Compressed air helps in the attainment of the expansion stroke after ignition takes place. Thus, the efficiency of the internal combustion engine is improved, and without running all four stroke cycles, it runs on two stroke cycles. The air engines developed so far are basically running on hybrid systems [**10, 11**] such as compressed air and gases, and are not 100 per cent pollution free.

III. Concept of multi vanes Air Turbine model

This study proposes a multi-vanes type air turbine as shown in Figs. 1a and 1b. Such air turbine is considered to work on the reverse 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.

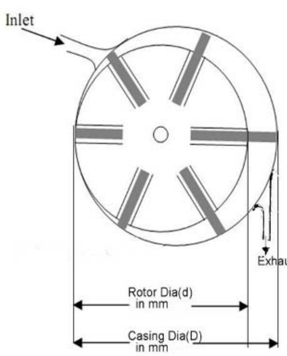

Fig. 1a Air Turbine-Diagram

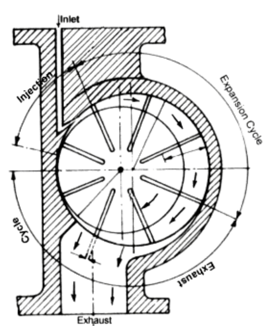

Fig. 1b Air Turbine- Model

A prototype of air turbine was developed in an earlier study [**12**]. At initial stage a cylinder for minimum capacity of storing compressed air for the requirement of 30 minutes running and maximum pressure of 20 bar is used as a source of storage energy. The compressed air 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 which regulate and maintain the constant pressure. The clean air then admits into air turbine through inlet passage / nozzle. Vanes of novel air turbine are placed under spring loading to maintain their regular contact with the casing wall to minimize leakage which is proposed as improvement over the currently available vane turbine. A study on high efficiency 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 and design of the air turbine [**14-17**], A detailed method of energy conversion processes, development concept of utilization of air turbine and optimization of its shaft work were made and presented in various symposium, seminars and conferences of international levels [**18-23**].

The objective of this study is to investigate the performance of an air turbine by varying vane angles with a particular injection angle, i.e. at which angle air should admit into the turbine between first two consecutive vanes. The air turbine considered has capability to yield output of 4.0 to 5.5 kW at 4-6 bar air pressure and for speed of 2000–2500 rpm, which is suitable for a motorcycle.

IV. 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 [**24-32**].

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:

[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 $$w = [(w_1) + (w_2)] \quad \textbf{(1)}$$

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.

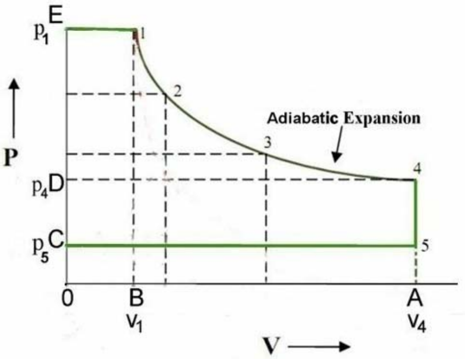

Fig. 2 Thermodynamic Processes (Isobaric, adiabatic and Isochoric Expansion)

Thus, the total power output or work done per unit time (W_{total}), for *speed of rotation* N rpm, will be mentioned as [**33**]:

$$W_{total}=n.(N/60).\left(\frac{\gamma}{\gamma-1}\right).p_1.v_1.\left\{1-\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}+n.(N/60).\left(p_4-p_5\right).v_4 \quad \textbf{(2)}$$

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 1a 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.

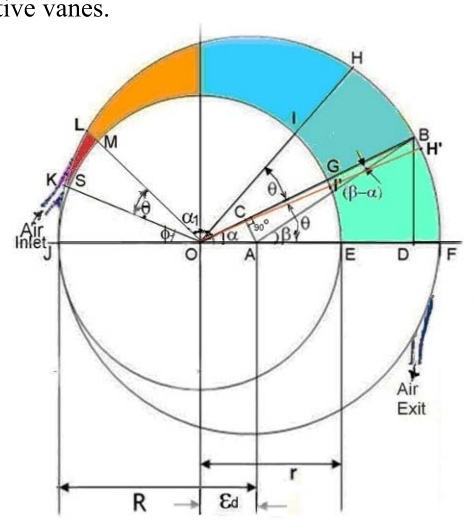

Fig. 3 Variable length BG **a**nd IH and injection angle ø

From Fig. 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 $X_{at'\text{var}iable'\alpha}$ can be written from the geometry

$$BG=X_{at,\text{var}iable'\alpha'}=R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right).\sin\alpha\right\}\right]+(R-r).\cos\alpha-r \quad \textbf{(3)}$$

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.

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

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

where BG= X_{1i} and IH= X_{2i} variable length of vanes when rotate into turbine as shown in Fig. 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 least effect on the overall values.*

The volume at inlet v_1 or $v_{\min}$ will fall between angle ∠LOF= $\alpha_{1\min}=\left(180-\theta-\phi\right)$ and 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.

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

Applying values of v_1 and v_4 to equation (2), the total power output available W_{total} can be written as:

$$W_{total}=n.(N/60).\left(\frac{\gamma}{\gamma-1}\right).\left\{1-\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}p_1.\left[L.\left\{\frac{\left(X_{1\min}+X_{2\min}\right).\left(2r+X_{1\min}\right)}{4}\right\}.\sin\theta\right] \quad \textbf{(5)}$$
$$+n.(N/60).\left(p_4-p_5\right).\left[L.\left\{\frac{\left(X_{1\max}+X_{2\max}\right).\left(2r+X_{1\max}\right)}{4}\right\}.\sin\theta\right]$$

where,

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

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

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

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

$$X_{1\max}=(D-d)=2\left(R-r\right) \text{ and}$$

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

V. INPUT PARAMETERS AND ASSUMPTIONS

Detailed analysis to derive relation between injection angle 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**].

In this study the vane angle (θ) of air turbine is considered 30°, 45°, 51.4°, 60°, 72°, 90° and 120° (i.e. 12, 8, 7, 6, 5,4 and 3 vanes respectively) and injection angle (ϕ) is kept 30°, 45° and 60°. Other various input parameters are listed in Table 1 for investigation of optimum shaft power output at different vane angle.

TABLE I
Input Parameters

Symbols	**Parameters**
D=2R	150mm (outer)
d =2r	100mm (inner) corresponding
p_1	2bar (=30psi), 3bar (=45 psi), 4bar (=60 psi), 5bar (=75 psi), 6bar (=90 psi)
p_4	$= (v_1 / v_4)^\gamma . p_1$ assuming adiabatic expansion
p_5	1 atm = 1.0132 bar
θ	30 °, 36 °, 45°, 51.4°, 60 °, 72 °, 90 ° and 120 ° angles between 2- consecutive vanes (i.e. rotor contains correspondingly 12, 10, 8, 7, 6, 5, 4, 3 vanes)
N	2500 rpm
L	35mm length of rotor
γ	1.4 for air
N	(360 / θ) Number of vanes
Ø	30 °, 45 ° and 60 ° Injection angles at which compressed air enters through nozzle into rotor

VI. RESULTS AND DISCUSSION

Various input parameters considered for study are listed in Table-1. Using the mathematical model and input parameters in Table-1 the effect of vane angles on the expansion work, flow work and total work output from air turbine is studied with speed of rotation 2500 rpm and different injection pressures of 2 bar, 3 bar, 4 bar, 5 bar, and 6 bar at fixed injection angles Ø = 30°, 45° and 60°.

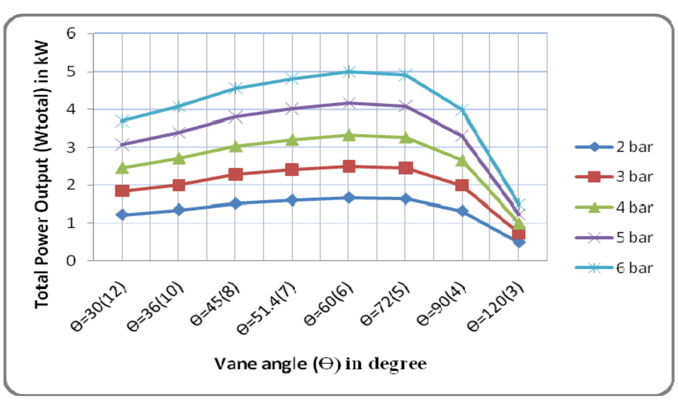

Fig. 4: Total turbine Power output vs Vane angles at injection angle (Ø) =30° and speed of rotation 2500 rpm

Fig. 4 shows the variation of total power output of the air turbine at injection angle (Ø) = 30°, is seen to increase with increasing vane angles from 36° to 60° or with the decrease in number of rotor vanes from 12 to 6 and thereafter it declines from vane angles (θ) = 72° to 120°. With increase in injection pressure the work output increases gradually as shown in graphical patterns. *Thus the* optimal total shaft output is found at vane angle (θ) = 60° (i.e. 6- vanes).

Fig. 5 shows the variation of total power output of the air turbine at injection angle (Ø) = 45°, is seen to increase with increasing vane angle up to 36° to 51.4 ° or with the decrease in number of rotor vanes from 12 to 7 and thereafter it declines from vane angles (θ) = 60° to 120°. With increase in injection pressure the work output increases gradually as shown in graphical patterns. *Thus the optimal total shaft output is found at vane angle (θ) = 45° and 51.4° (i.e. approx. 8-7 vanes).*

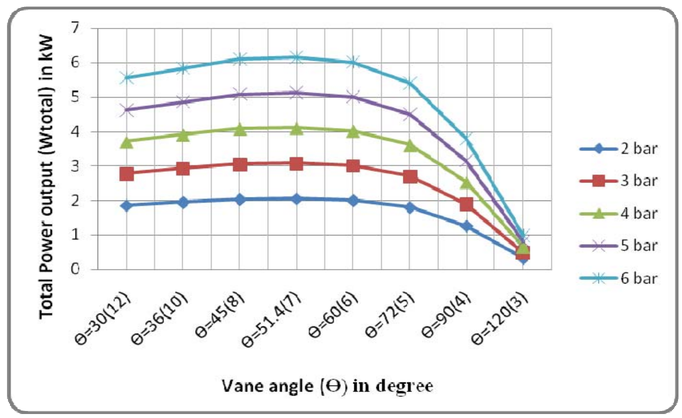

Fig. 5: Total turbine Power output vs Vane angles at injection angle (Ø) =45° and speed of rotation 2500 rpm

Fig. 6 shows the variation of total power output of the air turbine at injection angle (Ø) = 60°, is seen to increase with increasing vane angle up to 36° or with the decrease in number of rotor vanes from 12 to 10 and thereafter it declines from vane angles (θ) = 45° to 120°. With increase in injection pressure the work output increases gradually as shown in graphical patterns. *Thus the optimal total shaft output is found at vane angle (θ) = 36° (i.e. 10 vanes).*

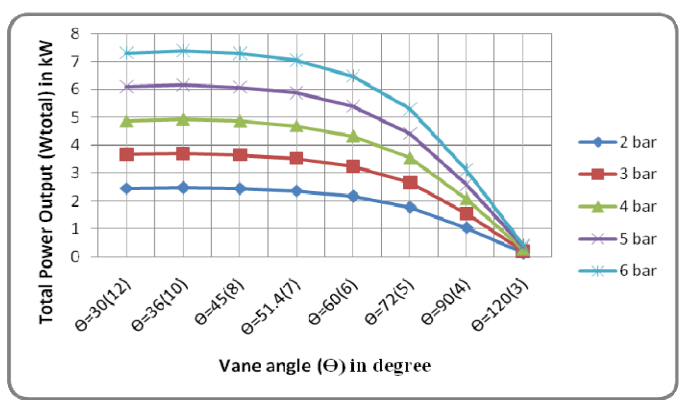

Fig. 6: Total turbine Power output vs Vane angles at injection angle (Ø) =60° and speed of rotation 2500 rpm

From Figs.4, 5 and 6, it is critically observed that at 6 bar injection pressure, 2500 rpm speed of rotation and at different injection angles; (Ø) =30°, 45° and 60°, the optimum total power output are obtained as:

- *5.00 kW when vane numbers are. 6-5 and injection angle (Ø) = 30°,*
- *6.10 kW when vane number are 7 and injection angle (Ø) = 45° and*
- *7.31 kW when vane numbers are 10 and injection angle (Ø) = 60°.*

VII. CONCLUSIONS

Based on input parameters and investigations carried out, following conclusions are drawn:

- The total power is found optimum for a particular dimension of multi vane turbine at a particular injection angle and at specific rotor vanes.
- The optimum power output is obtained at 30° injection angle when number of rotor vanes is 6-5 (vane angle is from 60°-72°). It is larger at injection angle 45°, when number of rotor vanes is 7 (vane angle is from 51.4°) and further largest at injection angle 60° when number of rotor vanes is 10 (vane angle is 36°).

Thus the maximum shaft power output is seen at 60° injection angle when number of rotor vanes is 10 (vane angle is from 36°) for the vane turbine of 150 mm casing diameter and 100 mm rotor diameter.

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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