Numerical Investigations on Performance of Vaned Type Novel Air Turbine

Bharat Raj Singh[a] and Onkar Singh[b]

[a]Professor & Head, Department of Mechanical Engineering,
Sagar Institute of Technology & Management, Barabanki-225001, UP, India
E-mail: brsinghlko@yahoo.com
[b]Professor & Head, Department of Mechanical Engineering,
Harcourt Butler Technological Institute, Nawabganj, Kanpur-208002, UP, India
E-Mail: onkpar@rediffmail.com

ABSTRACT

Greater utilization of hydrocarbon fuel in the transport sector is causing serious challenges to depletion of oil and thereby environmental & 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 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. Effect of impingement and expansion action of high pressure air for optimum shaft work (output) have been analyzed and parameters such as injection angle (22.5 deg), Vane angle (36 deg) and rotor / casing diameters ratio (d/D= 0.7) were found out earlier. With these parameters the turbine have been considered and analyzed here. Study shows that the impingement work has significant contribution in total work output and varies from 39.20% to 13.50% for D=200 mm and d= 140 mm and contribution of expansion work varies from 60.80% to 86.50%, at varying injection pressures 2 to 6 bar, constant injection angle (22.5^0) & constant vane angle (36^0) for low pressure turbine. It is also resulted a **maximum work output** *of 4.376 Kw at 6 bar injection pressure, which is sufficient to drive motorcycle.*

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

1. 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 & 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 & 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 & sold well due to their limited range, no proper facilities for recharge, and high initial cost. All of these issues have given birth to the technology of compressed air energy storage and its utilization in transport vehicles and other domestic utilities.

The "Guy Negre" [10], 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. So far about 52- patents were made during 1998 to 2006 and recently MDI and Tata Motors entered into an agreement to develop such vehicles in commercial use. 'G. Saint Hilaire' [11] an inventor of quasi turbine has also developed hybrid car running on Compressed air and gasoline. These highly compressed air energy storage systems with 300 psi, which can be filled within 15-20 minutes, may remain the dominant technology in the electric and hydrogen cell vehicle market.

In this paper, we are going to explore the use of compressed air storage energy for running vaned type novel air turbine and its total shaft work output obtained using the mathematical modeling are presented and analyzed in light of various conditions of injection angle, vane angle, rotor& casing diameter ratio.

2. VANED TYPE NOVEL AIR TURBINE

A vaned type air turbine as shown in Figure (1) has been considered. 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. A compressed air storage cylinder is designed to produce constant pressure for the minimum variation of torque at low volumes of compressed air. For this, a spring-loaded baffle is installed into the cylinder to regulate the constant air pressure and various aspects for optimum shaft output were 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. Thus such novel vane turbine is capable to develop desired output required to run the motor bike at 500–750 rpm for meeting starting torque at 4-6 bar air pressure and running torque at a normal speed of 2000–2200 rpm at 2-3 bar air pressure.

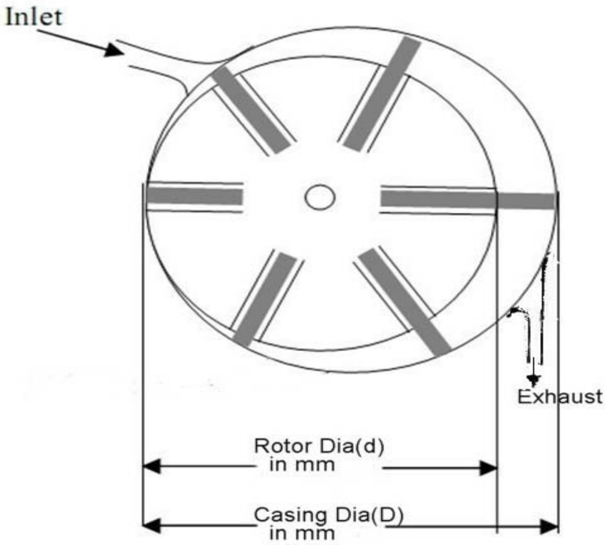

Fig. (1) Air Turbine

3. MATHEMATICAL MODELING

3.1 Thermodynamic Analysis

The high pressure of air at ambient temperature drives the rotor in novel air turbine Fig. (1). When high pressure air enters through the inlet passage and impinges upon the vanes it producing impulse. Also the high pressure air entering the rotor in consecutive vanes is gradually expanded up to exit passage. This impingement action and the expansion of high pressure air both contribute in producing the shaft work from air turbine. Assuming isentropic expansion as shown in Fig.(2), the theoretical expansion work output from air turbine having "n" number of vanes is given as under: -

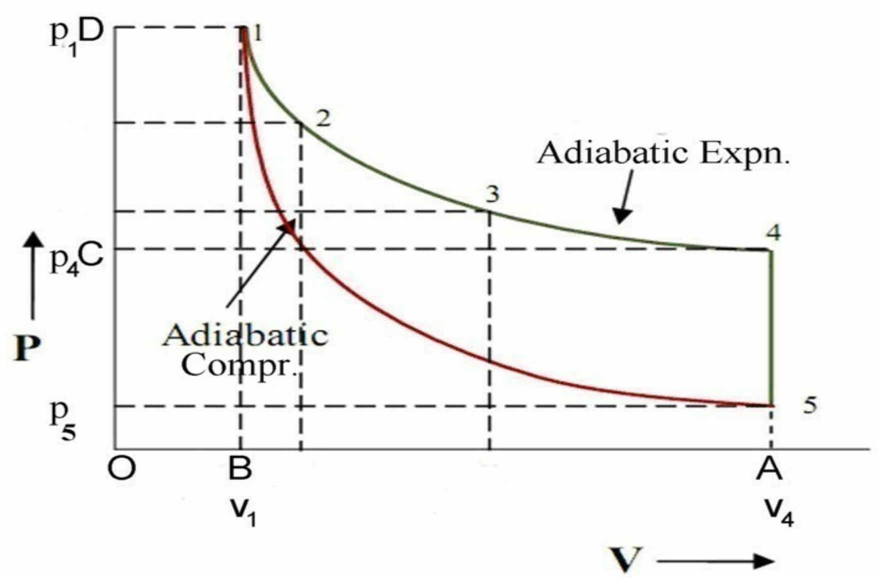

Fig. (2) PV Diagram for Isentropic Expansion in Air Turbine

Work done due to Expansion Cycle = Area under (D14CD) = Area under (14AB1) + Area under (1BOD1) – Area under (OA4CO).

$$w = \left(p_1.v_1 - 0\right) + \left(\frac{p_1.v_1 - p_4.v_4}{\gamma - 1}\right) - \left(p_4.v_4 - 0\right)$$

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

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

Considering isentropic (reversible adiabatic) expansion,

$$p.v^{\gamma} = constant = p_1.v_1^{\gamma} = p_4.v_4^{\gamma}$$

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

Applying equations (2) in (1);

$$\boxed{w = \left(\frac{\gamma}{\gamma - 1}\right).p_1.v_1.\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}\right\}} \qquad \textbf{(1.0)}$$

During isentropic expansion process; pressure $'p_4'$ can not fall below atmospheric pressure $'p_5'$. Under steady flow conditions in any positive displacement device the work done will get affected and here a negative (-) work done will be developed as follows,

$$-\Delta w = \int_4^5 v.dp = \left(p_4 - p_5\right).v_4 \qquad \textbf{(2.0)}$$

Thus total effective work done would be,

$$w_e = w - \Delta w$$

or $\boxed{w_e = \left(\frac{\gamma}{\gamma - 1}\right).p_1.v_1.\left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}\right\} - \left(p_4 - p_5\right).v_4}$

When there is n vanes, work done will be,

$$W = n\left(\frac{\gamma}{\gamma - 1}\right) p_1 v_1 \left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\} - n\left(p_4 - p_5\right) v_4 \tag{3.0}$$

It is obvious from Fig. (1) that if vanes are at angular spacing of θ degree, then total number of vanes would be, n = (360/θ).

3.2 Numerical Approach to find out Work done

The volumetric 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.

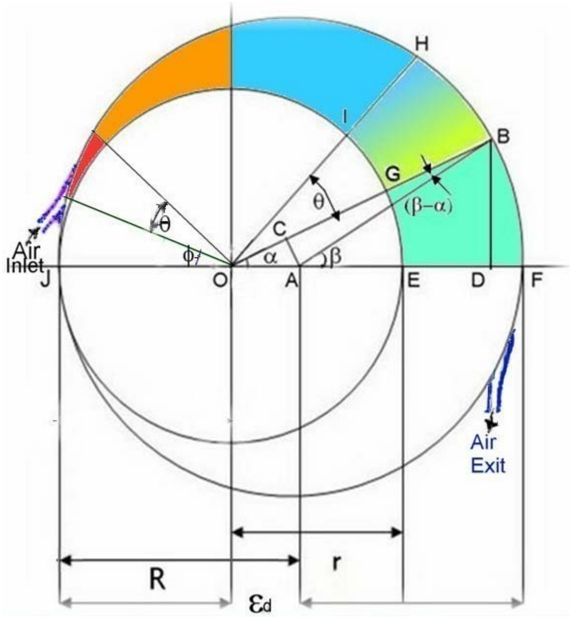

Fig. (3): Variable length BG & IH and injection angle" Ø"

From Fig.(3) variable length BG,

$$BG = X_{at,\text{variable}"\alpha"} = R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right)\sin\alpha\right\}\right] + \left(R-r\right).\cos\alpha - r \tag{3.1}$$

& Variable Volume of Cuboid "B-G-I-H-B",

$$V_{cuboids} = L.\left\{\frac{\left(X_1 + X_2\right)\left(2r + X_1\right)}{4}\right\}.\sin\theta \tag{3.2}$$

The Volume at inlet "v_1" or "v_{min}" will fall between angles $\alpha_1 = \left(180 - \theta - \phi\right)$ & $\alpha_2 = \left(\alpha_1 + \theta\right) = \left(180 - \phi\right)$ when air is injected at angle" Ø" into Turbine

Applying above conditions into equations (3.1), then

$$X_{1\min} = R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right)\sin\left(180-\theta-\phi\right)\right\}\right] + \left[\left(R-r\right).\cos\left(180-\theta-\phi\right) - r\right] \tag{3.3}$$

$$X_{2\min} = R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right)\sin\left(180-\phi\right)\right\}\right]+\left[\left(R-r\right).\cos\left(180-\phi\right)-r\right] \quad \textbf{(3.4)}$$

Applying values of X_{1min} & X_{2min} to equation **(3.2),**

$$V_{1\min} = L.\left\{\frac{\left(X_{1\min}+X_{2\min}\right)\left(2r+X_{1\min}\right)}{4}\right\}.\sin\theta \quad \textbf{(3.5)}$$

The Volume at exit $"V_2"or"V_{\max}"$ will fall between angles $\alpha_1 = 0$ & $\alpha_2 = \left(\alpha_1+\theta\right) = \theta$

Applying above conditions into equations (3.1), then

$$\boxed{X_{1\max} = 2\left(R-r\right)} \Rightarrow \left(D-d\right) \quad \textbf{(3.6)}$$

$$X_{2\max} = R.cos\left[\sin^{-1}\left\{\left(\frac{R-r}{R}\right)\sin\theta\right\}\right]+\left\{\left(R-r\right).\cos\theta\right\}-r \quad \textbf{(3.7)}$$

Applying values of X_{1max} & $X_{2\ max}$ to equation **(3.2),**

$$V_{2\max} = L.\left\{\frac{\left(X_{1\max}+X_{2\max}\right)\left(2r+X_{1\max}\right)}{4}\right\}.\sin\theta \quad \textbf{(3.8)}$$

Work output from novel air turbine will be sum of both impingement action and expansion work

$$W_{total} = W_{Expansion} + W_{impingement} \quad \textbf{(3.9)}$$

The work available due to expansion **W** $_{\textbf{expansion}}$ can be written as

$$W_{Expansion} = n\left(\frac{\gamma}{\gamma-1}\right)*\left\{1-\left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\}p_1*\left[L*\left\{\frac{\left(X_{1\min}+X_{2\min}\right)\left(2r+X_{1\min}\right)}{4}\right\}*\sin\theta\right]$$
$$-n\left(p_4-p_5\right)*\left[L*\left\{\frac{\left(X_{1\max}+X_{2\max}\right)\left(2r+X_{1\max}\right)}{4}\right\}*\sin\theta\right] \quad \textbf{(3.10)}$$

Due to the impact by nozzle air the work done can be given

$$W_{impingement} = Torque \times AngularVelocity = \left(T\times\omega\right) \quad \textbf{(3.11)}$$

where torque $T = Force \times Mean\Pr ojectedDis\tan ce = F_{impingement} \times (r + \Delta r_{mean})$ **(3.12)**

and $F_{impingement} = p_1 \times a$ **(3.13)**

Assuming p_1 = Pressure at which it strikes the vane

a = Area on which nozzle air pressure impinging

$$\Delta r_{mean} = \frac{X_{1\min} + X_{2\min}}{2}$$

$$\boxed{W_{impingement} = 2\pi n * N * p_1 * L * \left(\frac{X_{1\min} + X_{2\min}}{2}\right) * \left[r + \left(\frac{X_{1\min} + X_{2\min}}{2}\right)\right]} \quad \textbf{(3.14)}$$

Substituting the values of $W_{Expansion}$ and $W_{impingement}$ into equation (3.9),
The total work **(**W_{Toatal} **)** by turbine would be,

$$\boxed{\begin{aligned} W_{Total} &= n * N * \left(\frac{\gamma}{\gamma - 1}\right) * \left\{1 - \left(\frac{p_4}{p_1}\right)^{\frac{\gamma-1}{\gamma}}\right\} * p_1 * \left[L * \left\{\frac{(X_{1\min} + X_{2\min})(2r + X_{1\min})}{4}\right\} * \sin\theta\right] \\ &- n * N * (p_4 - p_5) * \left[L * \left\{\frac{(X_{1\max} + X_{2\max})(2r + X_{1\max})}{4}\right\} * \sin\theta\right] \\ &+ 2\pi n N * p_1 * L * \left(\frac{X_{1\min} + X_{2\min}}{2}\right) * \left[r + \left(\frac{X_{1\min} + X_{2\min}}{2}\right)\right] \end{aligned}} \quad \textbf{(3.15)}$$

Table-1
Input Parameters

Symbols	**Parameters**
D=2R, d=2r	(200,140) i.e. d/D=0.7
p_1	30psi(≃2bar), 45 psi(≃3bar), 60 psi(≃4bar), 75 psi(≃5bar), 90 psi(≃6bar)
p_5	1 atm = 1.0132 bar
p_4	1.1≃1.2 p5 =1.1 bar
n	Number of vanes (360 / θ)
N	500 rpm, 1500 rpm, 2500 rpm
L	35mm length of rotor
γ	1.4 for air
θ	36^0 angle between 2-vanes, (i.e. rotor contains correspondingly 10 number of vanes)
Ø	22.5^0 angle at which compressed air through nozzle enters into rotor

4. RESULTS AND DISCUSSION

Various input parameters considered for study are listed in Table-1. Using the mathematical model the effect of speed of rotation at constant vane angle, injection pressure and rotor/casing diameter ratio on the expansion work, impingement work and total work output from air turbine 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 have been plotted in Figs. (4) to (6) for the speed of rotation "N", varied as 500, 1000, 1500, 2000, 2500 and 3000 rpm at vane angle of 36^o, injection angle of 22.5^o at different injection pressures of 30,45,60,75 & 90 psi and at the rotor/casing diameter ratio (d/D)=0.7.

Figure (4) shows the variation of expansion work at different the speed of rotation "N", varied as 500, 1000, 1500, 2000, 2500 and 3000 rpm at constant vane angle 36^o, air injection angle 22.5^o, different air injection pressure of 2 to 6 bar and rotor/casing diameter ratio (d/D)=0.7, at casing diameter of 200mm. It is evident that the shaft work due to expansion at 2 bar (30 psi) is lower at 500 rpm and thereafter gradually increases at 3-6 bar (45-90 psi) and is higher at higher speed of rotation, which is attributed to the large work capacity at higher injection pressures.

Figure (5) shows the variation of impingement work at different the speed of rotation "N", varied as 500, 1000, 1500, 2000, 2500 and 3000 rpm at constant vane angle 36^o, air injection angle 22.5^o, different air injection pressure of 2 to 6 bar and rotor/casing diameter ratio (d/D)=0.7, at casing diameter of 200mm. It is evident that the shaft work due to impingement at 2 bar (30 psi) is lower but significant at 500 rpm and thereafter gradually increases at 3-6 bar (45-90 psi) and is higher at higher speed of rotation, which is attributed to the large work capacity at higher injection pressures.

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Figure (6) shows the percentage contribution of expansion work in total work output for different the speed of rotation "N", varied as 500, 1000, 1500, 2000, 2500 and 3000 rpm at constant vane angle 36^o, air injection angle 22.5^o, different air injection pressure of 2 to 6 bar and rotor/casing diameter ratio (d/D) =0.7, at casing diameter of 200mm. It is evident that percentage contribution of expansion work is highest at 90 psi (6 bar). At 2bar (30 psi) injection pressure the contribution of expansion work in total work is lowest at all speed of rotation and almost constant as 60.80%. Higher contribution of expansion work at 3-6 bar injection pressures are attributed 78.40% to 86.50%, in caparison to the smaller contribution of impingement work at these injection pressure as evident from Fig.(7) that is from 39.2 % to 13.5 % and so the majority of work is due to expansion work only.

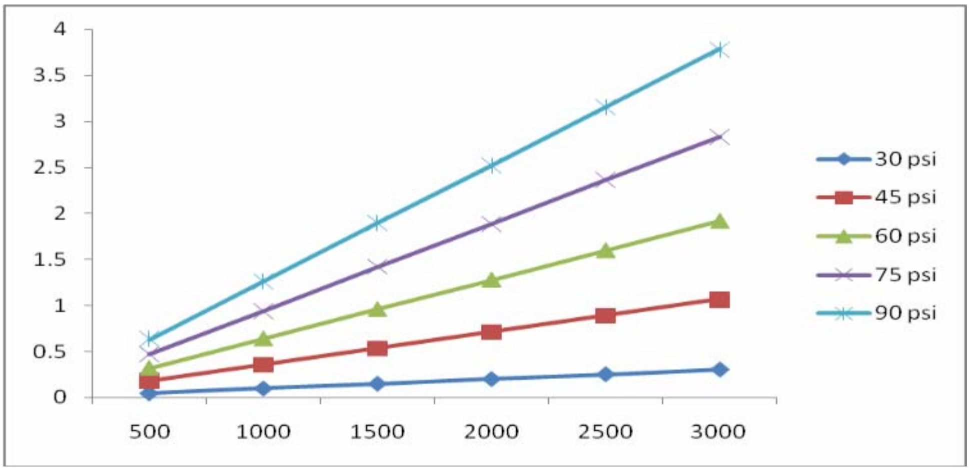

Fig. 4 Expansion work versus speed

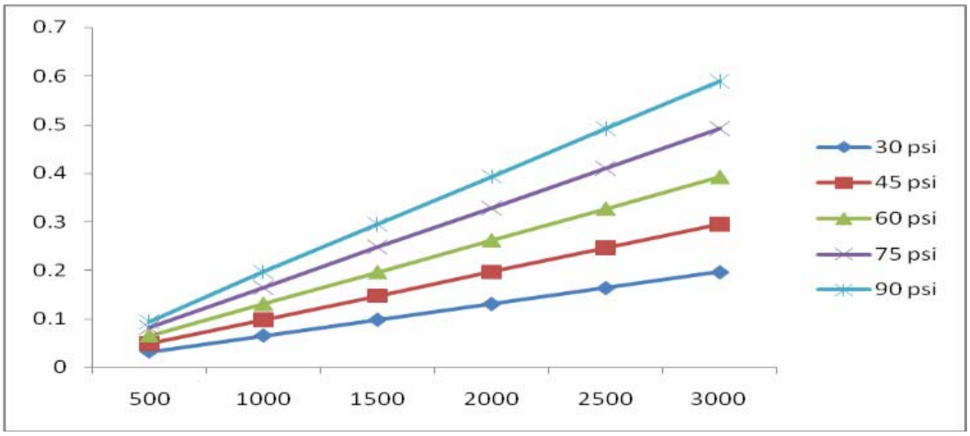

Fig. 5 Impingement work versus speed

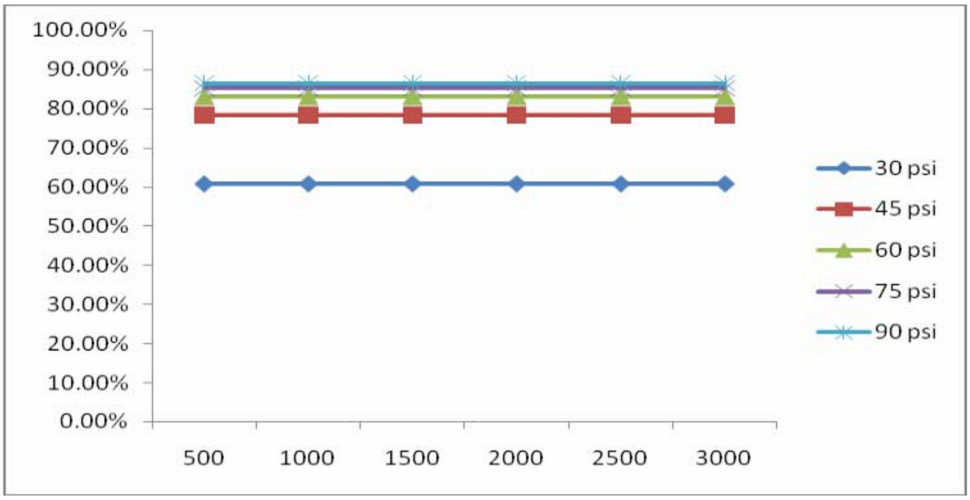

Fig. 6 Percentage contribution of Expansion Work (We) vs. Speed

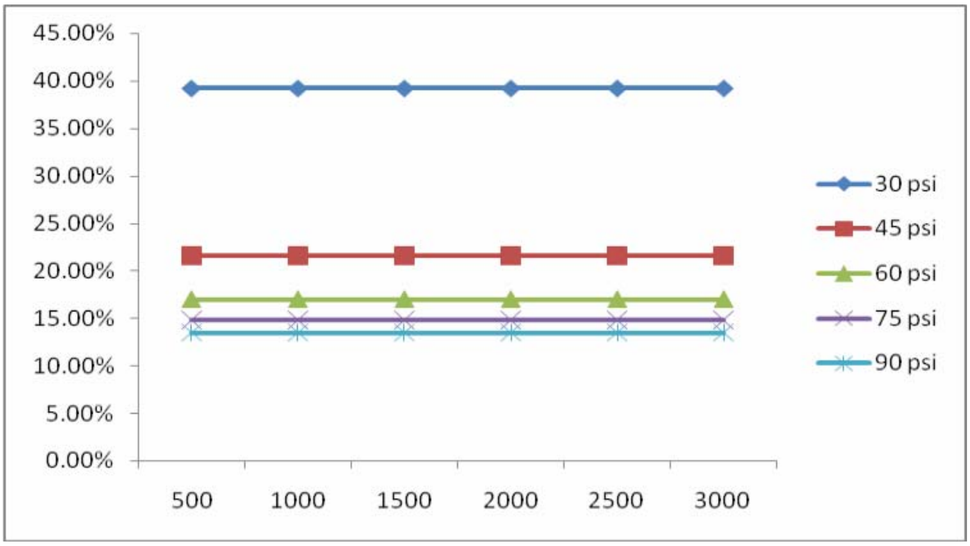

Fig. 7 Percentage contribution of Impingement Work (Wimp) vs. Speed

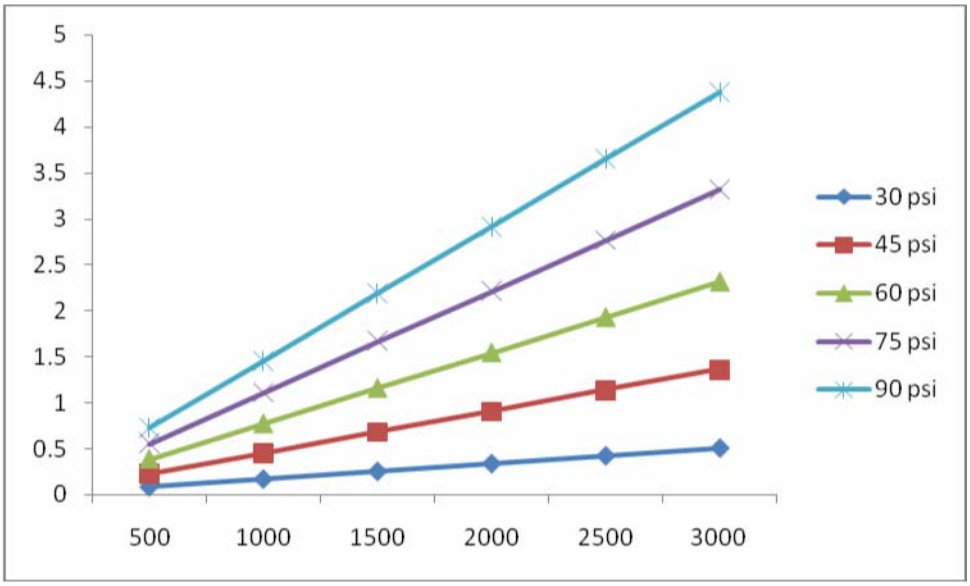

Fig. 8 Total Work (Wt) vs. Speed

Variation of total work output work with respect to different the speed of rotation "N", varied as 500, 1000, 1500, 2000, 2500 and 3000 rpm at constant vane angle 36^o, air injection angle 22.5^o, different air injection pressure of 2 to 6 bar and rotor/casing diameter ratio (d/D) =0.7, at casing diameter of 200mm is shown for different air injection pressure in Fig.(8) at 500 rpm. Total work at 2 bars (30 psi) is seen lower at 500 rpm and highest when speed of rotation becomes 3000 rpm.

It is thus, observed that in the vane turbine total shaft work is combined effect of the component of expansion and impingement work. The significant contribution of impingement work in respect to total work output at 2 bar (30 psi) is 39.2 % and 21.6% to 13.5 % as speed of rotation gradually increases for 3-6 bar injection pressure at constant injection angle (22.5^o) & constant vane angle (36^o). Thus it is also concluded that the expansion work output as well as total work output is optimum when speed of rotation are higher side 2000- 3000 rpm and at higher injection pressure 4-6 bar (60-90 psi)

5. CONCLUSIONS

On the basis of input parameters (injection angle 22.5^o and vane angle 36^o and d/D = 0.70 at 200mm casing diameter) considered and results obtained, following conclusions are drawn;

- There exists an optimal value of expansion work 3.785 Kw at 3000 rpm and 6 bar (90 psi).
- The impingement work is seen to increase 0.589 KW at 3000 rpm and 6 bar (90psi).
- Total work output from the air turbine is seen to be maximum for the higher injection air pressure and is seen to be 4.375 Kw for injection pressure of 6 bar at rpm of 3000 rpm.
- In view of different values speed of rotation and higher injection pressure the impingement work is low but significant to the order of 0.589 Kw at the desired work output of 4.375 Kw (5.5 HP) output which is sufficient to run motorcycle.

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