Global Scenario of Two Wheeled Vehicle's Emission and its Emerging Future Technology

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ABSTRACT

As civilization is growing, uses of transport vehicles become essential part of life and such fast utilization of transport vehicles is causing serious threat to the depletion of oil reserves and releasing heavy carbon dioxide (CO_2) in the atmosphere globally. The five major contributors are China, USA, Russia, India and Japan. Thus environmental contributors are not only developed countries but developing countries also are also responsible to release the highest carbon dioxide and other un-burnt Green House Gases through transport vehicle's tail pipe emissions. This leads to the current burning problem of Global Warming and Climate change.

In developing countries, as per recent data, about 77.8 % contribution of Green House Gases is from transport only and the remaining through Thermal Power Stations, Industrial Exhausts and other sources. Out of which it is noticed that, population of two-wheeled vehicles in India, China, Bulgaria, Taiwan, Korea; Thailand etc. is around 80 % and hence 50-60% contribution of emission of carbon dioxide in the atmosphere is due to motor-bikes, scooters etc.

This paper focuses on a recent design of an air engine which proposes to replace the conventional internal combustion engine on motorbike. The proposed air engine runs only on the compressed air and release nearly zero emission and could contribute to curb emission of vehicles up to 50-60% if implemented widely on motorbike globally.

Keywords: Transport vehicles, Global warming, Compressed air engine,

1. INTRODUCTION

Worldwide huge demand of vehicles being added every year, are resulting into a rapid consumption of fossil fuels and causing fast depletion of the energy resources. A noted geophysicist Marion King Hubbert [1] applied the Principles of Geology, Physics and Mathematics in 1956 for the future projection of oil production from the US reserve base. Hubbert indicated that, conventional crude-oil production would attain Peak Oil in 1970 and thereafter start depleting. This may cause serious threat to mankind within 40 years i.e. by 1995. This will also affect environment due to release of huge amount of pollutants 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 this oil production is set to a peak and begins to decline by around 2010.

A study was carried about vehicles populations in the developing countries like: India, China, Taiwan etc. In India the data for transport vehicles registered from 1951 to 2004 is shown in Table 1.

Table No. 1

Total Number of Registered Motor Vehicles in India - 1951-2004 (In thousands)

Year (As on 31st March)	All Vehicles	Two Wheelers	Cars, Jeeps and Taxis	Buses	Goods Vehicles	Others*
1	**2**	**3**	**4**	**5**	**6**	**7**
1951	306	27	159	34	82	4
1956	426	41	203	47	119	16
1961	665	88	310	57	168	42
1966	1099	226	456	73	259	85
1971	1865	576	682	94	343	170
1976	2700	1057	779	115	351	398
1981	5391	2618	1160	162	554	897
1986	10577	6245	1780	227	863	1462
1991	21374	14200	2954	331	1356	2533
1996	33786	23252	4204	449	2031	3850
1997	37332	25729	4672	484	2343	4104
1998	41368	28642	5138	538	@ 2536	4514
1999	44875	31328	5556	540	@ 2554	4897
2000	48857	34118	6143	562	@ 2715	5319
2001	54991	38556	7058	634	@ 2948	5795
2002	58924	41581	7613	635	@ 2974	6121
2003(R)	67007	47519	8599	721	@ 3492	6676
2004 (P)	72718	51922	9451	768	@ 3749	6828

Note:

* : Others include tractors, trailors, three wheelers (passenger vehicles) and other miscellaneous vehicles which are not separately classified.

@ : Includes Omni buses.

(P): Provisional

(R) : Revised

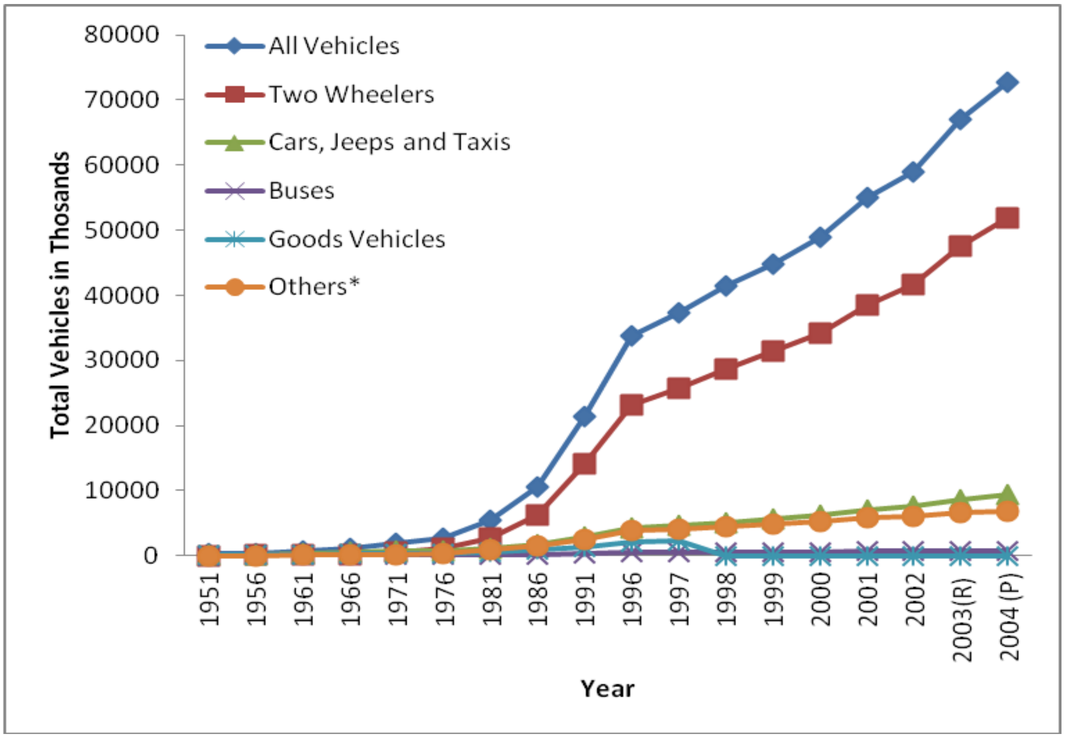

Fig.1 Yearwise Total Registered Vehicles in India

This data shows that in 2004, the percentage of population of two wheelers as shown in Fig. 1, in respect to the total vehicles was around 70-75%, whereas from the recent report, total vehicle's population in Uttar Pradesh, that largest State of India is around 10.5 million, out of which 8.2 million is only two wheeled (Source: Dainik Jagran- Jan' 2011). Thus the percentage of two wheelers has increased from 70% to 82% within 6 years. On other hand, globally transport sector alone is consuming huge quantity of hydrocarbon fuel and releasing about 77.8 percentage of air pollutants in the atmosphere. Recent study also indicates that in the developing countries like India, China, Taiwan etc., 80 percentage pollutants are generated by the motorbikes/two wheelers driven by internal combustion (IC) engines. As transport vehicles are major contributors to tail- pipe emission, generating around 77.8% air pollutants such as: Carbon Monoxide (CO), Carbon Dioxide (CO2) and unburned Hydrocarbon (HC). Thus the motorbikes/ two wheelers are contributing 50-60% air pollutants in the atmosphere and is a major player for global warming. This study also confirms that two wheelers / motorbike's IC engines are generating more than double the pollutants as compared to the remaining automobiles / transport vehicles.

In order to reduce the emission and eliminate 50- 60% of the exhausting pollutants, this paper presents a new concept of an air engine using compressed air as the potential power source to run motorbikes in place of using IC engines. Such motorbikes are proposed to be equipped with compressed air engines that transform the compressed air stored energy into mechanical work.

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. Pioneering work in the area of compressed air engine has been done by French technologist Guy Negre [3] and also by an inventor of quasi turbine G. Saint Hilaire [4]. Use of compressed air as working fluid offers a prime mover which does not involve combustion process for producing shaft work. Thus, 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 motors. 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 whose overall analysis shows that the compressed air system is quite attractive option for light vehicle applications [5].

From the author's earlier study [6,7], the compressed air driven prime movers are cost effective as compared to fossil fuel driven engines and may become a dominant technology in place of the electric, hydrogen cell and other alternative fueled vehicles available in the market [8-9]. Some of studies for performance optimization of the low capacity of air turbines have also been carried out in the author's earlier publications [10-12] and from other authors [13-33].

2. AIR TURBINE MODEL

A vaned type air turbine as shown in Fig. 1 has been considered. Proposed air turbine is 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. In earlier study conducted by authors, a prototype of air turbine was developed and its functionality was ensured [34-50]. 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 5.50 to 6.80 HP at 4-6 bar air pressure and for speed of 2000–2500 rpm, 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.

3. 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 as shown in Fig. 2. 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 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.

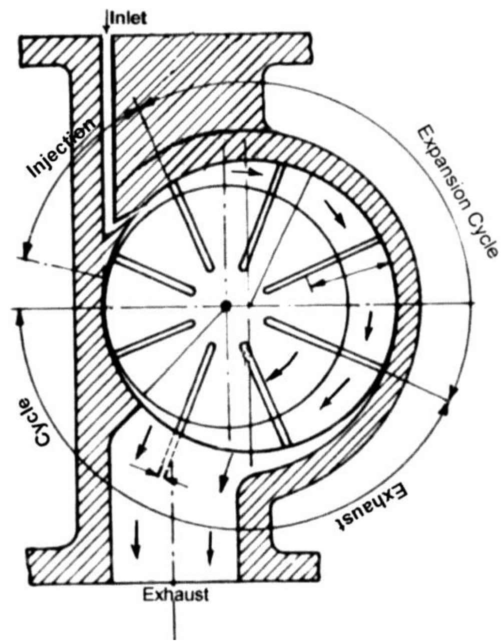

Fig.2: Air turbine- model

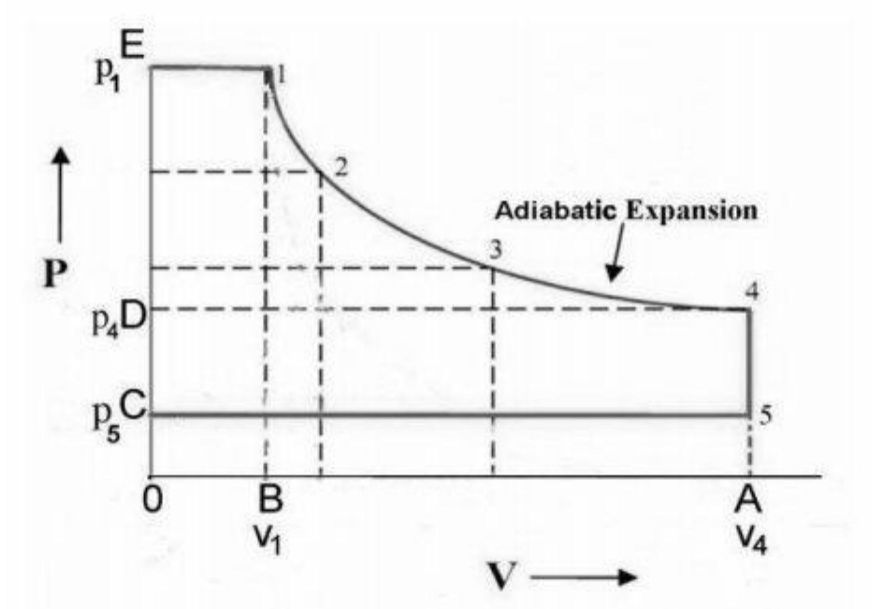

Fig. 3: Thermodynamic processes (isobaric, adiabatic and isochoric expansion)

From Fig. 3, 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)]

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

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

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

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

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

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

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

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

After the expansion process during exit flow the pressure p_4 cannot fall below atmospheric pressure p_5. Thus, from eq. (1) the net work output will be:

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

When air turbine is having *n* number of vanes, then shaft output [51] 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 (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 $W_{flow} = n.(N/60).\left(p_4 - p_5\right)v_4$

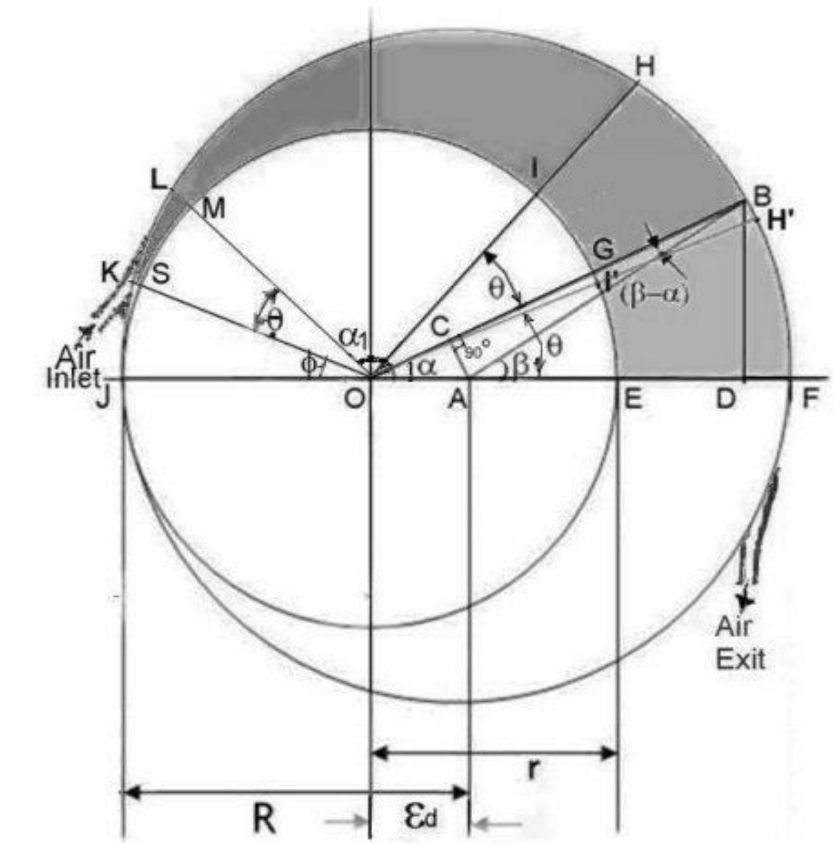

Fig. 4: Variable length BG and IH and injection angle ϕ

From Fig. 4, 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 $X_{at'\text{variable}'\alpha}$ can be written from the geometry:

$$BG = x_{at,\text{variable}"\alpha"} = (1/2).D.cos\left[\sin^{-1}\left\{\left(\frac{D-d}{D}\right).\sin\alpha\right\}\right] + (1/2).(D-d).\cos\alpha - d/2 \quad (7)$$

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

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

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

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= $\alpha_{1\min} = (180-\theta-\phi)$ and angle $\angle$KOF $= \alpha_{2\min} = (\alpha_{1\min}+\theta) = (180-\phi)$ 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:

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

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

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

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

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

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

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

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

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

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

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:

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

4. ASSUMPTIONS AND INVESTIGATION PARAMETERS

Following assumptions and investigation parameters are taken while analyzing the theoretical and experimental setup:

- The temperature of compressed air entering through inlet into rotor and casing space is at ambient temperature.
- Vanes are spring loaded and hence leakages across vane tip and casing liner are ignored.
- Friction between vane tip and casing liner is ignored.

Various input parameters are considered as shown in Table-2 for investigation. 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 θ, injection angle ϕ and speed of rotation N of the air turbine are considered to be constant for whole study. The results obtained have been plotted in Figs. 5 to 8, for the rotor/casing diameter ratio (*d/D*), as 0.75 at vane angle of 45^{o}, injection angle of 60^{o} at different injection pressures of 2-6 bar (30, 45, 60, 75 and 90 psi) and at the speed of rotations 500 rpm, 1000 rpm, 1500 rpm, 2000 rpm and 2500 rpm, at casing diameter 100 mm and rotor diameter 75 mm.

Table- 2 Input Parameters

Symbols	Parameters
Rotor to Casing (d/D) ratio	0.75, when casing diameter is kept D=100 mm and rotor diameter d=75 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	500 rpm, 1000 rpm, 1500 rpm, 2000 rpm, 2500 rpm and 3000 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	60^o, injection angle at which air enters into turbine.

5.0 RESULTS AND DISCUSSION

5.1 Theoretical Investigation

From the author's earlier study for investigation of optimum input parameters, Fig. 5 shows that the theoretical power at different speed of rotation is increasing with increase of injection pressure. The rate of increase of power is higher at higher injection pressure compared to lower injection pressure. This can be attributed to the fact that at higher injection pressures the flow power and the expansion power is more. Due to higher admission pressure total amount of air admitted is more and it offers the overall increase in total power output.

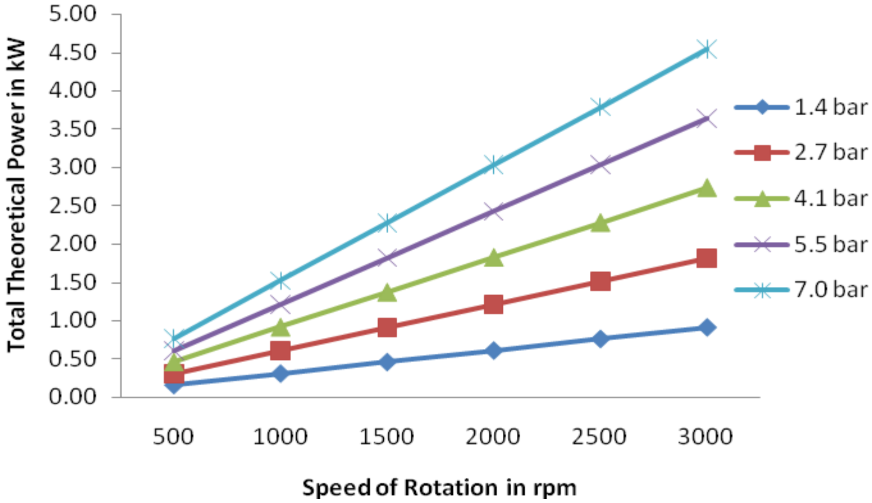

Fig 5: Total theretical power output (*Wttheo*) vs speed of rotation

5.2 Experimental Investigation

5.2.1 Experimental Test setup

The complete schematic of test setup is shown in Fig. 6. It consists of compressor, compressed air storage cylinder, supply of compressed air through air filter, regulator and lubricator to air turbine. The dynamometer consisting of load pulley, weight load and load dial gauge are also shown in the set up.

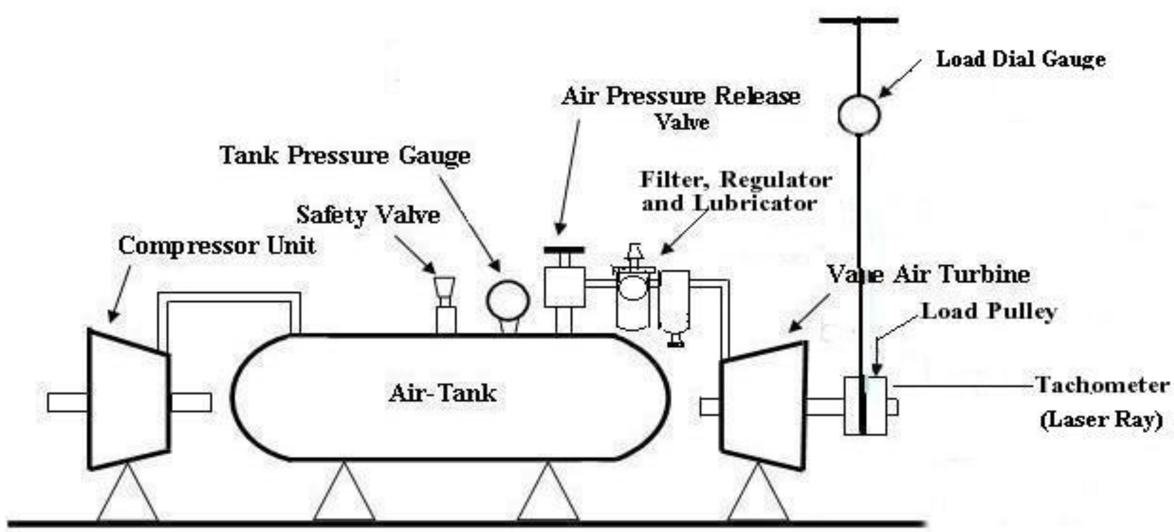

Fig. 6: Schematic Test Setup

The experimental setup consisting of a heavy duty two stage compressor with suitable air storage tank, air filter, regulator and lubricator, novel air turbine, rope dynamometer has been created for validation of theoretical results as shown in Figs.7 (a) and (b).

Fig. 7: (a) Actual test rig

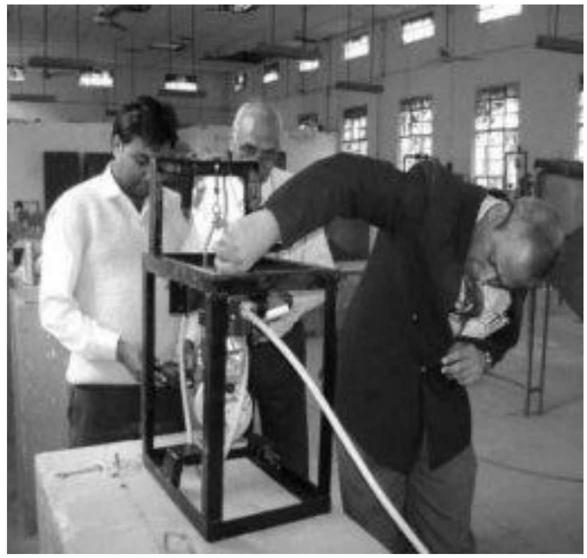

Fig.7: (b) Actual air turbine under test

The actual setup of test rig of air engine / turbine was fabricated and air turbine was tested in the laboratory. The compressed air is produced by a heavy duty two stage compressor and stored in a suitable capacity of air tank to maintain nearly constant supply pressure of 300 psi. The compressed air is connected to air filter, regulator and lubricator to produce desired air pressure for testing. The data is recorded with various parametric conditions and performance evaluation of the prototype air turbine is carried out.

5.2.2 Experimental Results

Performance evaluation is conducted on a compressed air driven vaned type novel air turbine. The comparison of theoretical total shaft outputs with respect to experimental values are carried out on following optimum input parameters such as high pressure air 1.4 bar (20 psi), 2.8 bar (40 psi), 4.2 bar (60 psi), 5.6 bar (80 psi) and 7 bar (100 psi), at different input parameters (injection angle 60^o, vane angle 45^o, L=45 mm, and d= 75 mm rotor diameter and D= 100 mm casing diameter (or d/D=0.75).

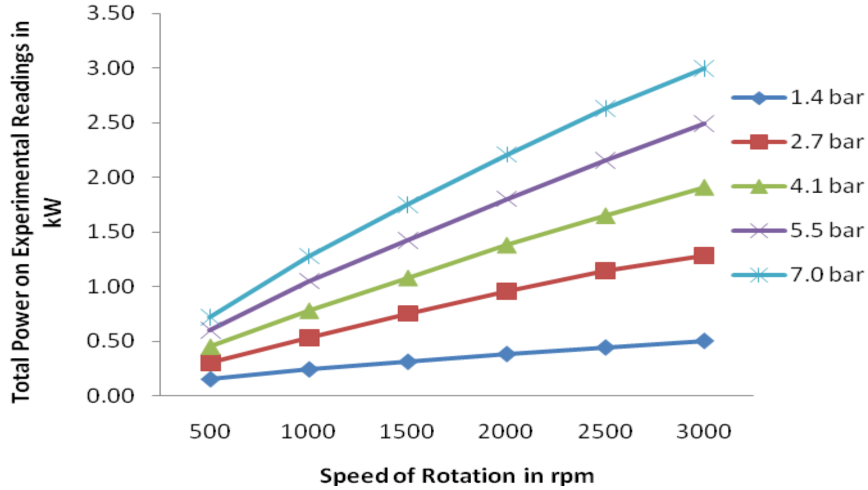

Fig. 8: Total experimental power (W_{texper}) versus speed of rotation

Figure 8 shows that the experimental values of power output increases with higher injection pressure and at different speeds of rotation. Comparison of power output for theoretical and experimental conditions shows that for a given injection pressure the experimental power output is less than theoretical value at same operating condition. This is because of leakage at interface of vane and casing, throttling of air at admission, and friction losses. From Figs. 5 and 8, the theoretical performance of the air turbine can be compared with the experimental performance. It is seen that the results obtained experimentally match significantly with the theoretical results to the extent of around 70% to 98% for different operating parameters.

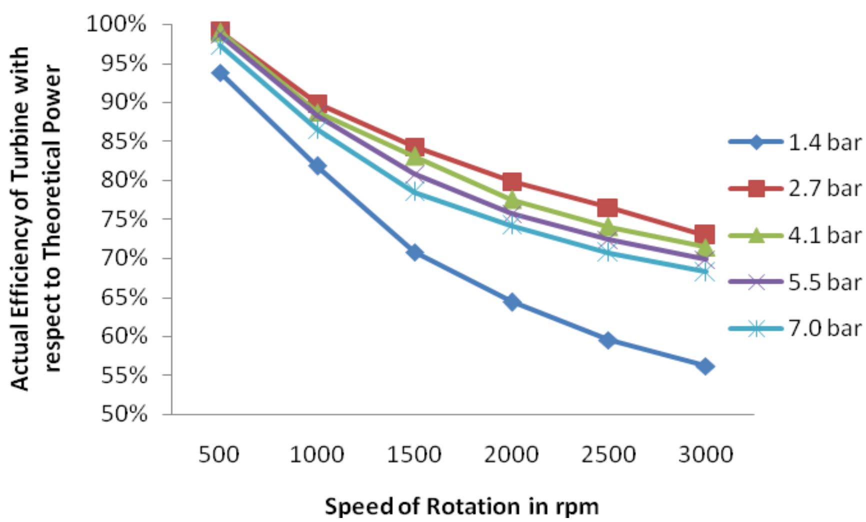

Fig. 9: Actual performance of vane turbine with respect to theoretical power

Figure 9 depicts the variation of performance efficiency of air turbine for different injection pressure at different speeds of rotation such as: 99%, 89.8%, 84.3%, 79.8%, 76.5% and 72.5% at speed of rotation 500 rpm, 1000 rpm, 1500 rpm, 2000 rpm, 2500 rpm and 3000 rpm respectively when injection pressure varies from 2.8 - 4.2 bar. But the performance efficiency for injection pressure 1.4 bar is not in parity with higher pressure. This indicates that turbine power output is utilized in overcoming the friction losses at injection pressure 1.4 bar and centrifugal forces on vanes are also not effective at speed of rotation 500-3000 rpm. Thus air turbine offers best performance at injection pressure 2.8 to 4.2 bar (40-60 psi).

5.2.3 Reasons for large Deviation Between Theoretical and Experimental Results

The above study shows that there is large difference between the theoretical results and experimental observations ranging from 72.5% to 99%. This is attributed due to the following reasons:

- Theoretically expansion is considered to be adiabatic but the same will not be possible in this case as there is no isolation of engine from the surroundings. In actual case the expansion will not be adiabatic and the index of expansion will be different from 1.4 (i.e. theoretically considered value).
- The leakage at interface of vane and casing cannot be completely eliminated in view of running clearance required between the mating surfaces. Although vanes are of spring loaded type but the too high stiffness of vane spring will lead to increase in friction resistance loss. This leakage can be experimentally observed and suitable leakage model may be defined in future studies.
- The throttling of air occurs at the time of admission due to restricted passage available for the injection of air into air turbine. This throttling effectively reduces the initial pressure at the beginning of expansion of air inside air turbine. Adverse influence of throttling at different injection pressures will be different and the output varies accordingly.
- The friction losses which are there at all rotating parts and mating surfaces eventually reduce the power output from the engine. These losses are there at the mating surface of vanes / casing and at the shaft bearing.
- Air lubricator adds some trace of lubricants in the air injected into the air turbine. These traces of lubricant also expand with the expanding air and the work output is different from theoretically predicted value.
- The experimental observation errors may be there in various measurements.

Thus an empirical relation between tip leakage and throttling losses due to effect of size of air nozzles for different injection pressure and speed of rotation which ultimately varies the air consumption, are taken into account to reduce this large deviations between the theoretical and experimental results.

6 CONCLUSIONS

Based on the considered input parameters and above investigations, following conclusions are drawn:

- The expansion power output is found maximum as 3.65 kW for moderate / minimum air consumption.
- The significant contribution of exit flow power with respect to total power output varies from 6.68% to 11.33%.
- The total optimal power output is obtained as 3.98 kW for minimum air consumption, at rotor/casing diameter ratio 0.70, injection pressure 6 bar and speed of rotation 2500 rpm.
- The theoretical optimum shaft power output 3.98 kW match significantly from the results obtained experimentally and performance efficiency of the novel air turbine ranges from 72.5% to 99% for injection pressure 2.8- 4.2 bar.

Thus the investigation would be very useful for designing the air engine for light vehicles such as motorbikes which can run as zero pollution vehicles.

From these results it is emerged out that the air turbine engine having casing diameter = 100 mm and rotor diameter = 75 mm, can be easily equipped on two wheelers as a replacement to IC Engines. The above investigation also shows that such data could be useful for designing the air engine for light vehicles / motorbikes. Further, if this technology is implemented widely in developing countries where major contributors of CO_2 are two wheelers in 70-80 % population of vehicles, this can curb the emission 50-60%. Thus the technology can be used as alternative to fossil fuel, contribute to energy sustainability and can also check the global warming and climate change largely.

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