Study of Performance of Shaft output with Rotor-to-Casing Ratios versus Different Vane Angles Adopting Practical Approach on a Novel Multi-Vane Air Turbine

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Abstract

A concept of using compressed atmospheric air as an alternative to fossil fuel and zero pollution power sources for running light vehicle such as: motorbikes etc.. Here considered vehicle is equipped with an air turbine in place of an internal combustion engine, and transforms the compressed air energy into shaft work. The mathematical modeling shown here is reproduced from author?s earlier publications on a small capacity compressed air driven vaned type novel air turbine. The effect of different rotor to casing diameter ratios with respect to different vane angles (number of vanes) have been considered and analyzed under specific parametric conditions. The shaft work output is found optimum adopting practical conditions of rotor / casing diameter ratios on a particular value of vane angle (no. of vanes). In this study, the maximum power is obtained as 4.02 kW (5.6 HP) when casing diameter is taken 100 mm, and rotor to casing diameter ratio is kept from 0.70, as the construction of turbine can be fabricated between rotor to casing (d/D) ratio from 0.95 to 0.70 only. It is learnt that the generated power output of 4.02 kW (5.6 HP) is sufficient to run any motorbike.

Index terms— zero pollution, compressed air, air turbine, vane angle, rotor / casing diameter ratio, air-o-cycle.

1 INTRODUCTION

t is an established fact that the Worldwide faster consumptions of fossil fuel in transport vehicles have resulted fast depletion to energy resources and releasing huge quantities of pollutant in the atmosphere.. A US geologist Marian King Hubbert [1] in 1956 indicated that the conventional crude-oil production will attain Peak Oil within 20 years and thereafter it will start depleting which may cause serious threat to mankind within 40 years i.e. by 1995. Aleklett K. and Campbell C.J., ??2] in 2003 illustrated that with current rate of consumptions, the production of oil and gas in many country will peak and begin to decline by around 2010. Such apprehension has led the search for environment friendly alternative to fossil fuel oil, or some Author ? : Department of Mechanical Engineering, SMS Institute of Technology, Gosainganj, Lucknow-227125, UP, India. (Telephone: +91-9415025825; E-mail : brsinghlko@yahoo.com) Author ? : Department of Mechanical Engineering, Harcourt Butler Technological Institute, Kanpur-208002, UP, India. (Telephone: +91-9415114011; E-mail: onkpar@rediffmail.com) method of conserving natural resources using nonconventional options, such as; bio-diesel, wind power, photo voltaic cells etc. and or energy conversion systems like battery storage, hydrogen cell, compressed air etc to obtain shaft work for the engines of vehicles ??3] ??4] ??5] ??6][7][8][9].

Guy Negre [10] a French technologist and G. Saint Hillarie [11] an inventor of quasi turbine have carried out very important work in the area of compressed air engine. They stored highly compressed air in the energy storage systems up to 300 bar pressure within 15-20 minutes, and reused for running compressed air engines. In view of such attractive using non-conventional resources, the compressed air engine became comparable in place of the other technology in vehicle markets.

In this paper author has carried out the parametric analysis of a small capacity air turbine having vane type rotor and describes the investigation of the effect of rotor to casing diameter ratios with different vanes fitted in the rotor. Results obtained by using a mathematical model are presented and analyzed here.

2 II.

3 FEATURE OF VANE TYPE AIR TURBINE

In this study a multi-vane type air turbine having casing diameter =D mm and rotor diameter =d mm is proposed as shown in Fig. 1. The considered air turbine works on the reverse working principle of vane type compressor. In this arrangement total shaft work is seen to be the 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 [12].and its functionality was examined A storage cylinder for the compressed air having capacity of 30 minutes stored air, for the requirement of running turbine at initial stage at working pressure of 10 bar, is used as a compressed air energy source. This 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. The clean air then admits into air turbine This paper describes a concept of using compressed atmospheric air as an alternative to fossil fuel and zero pollution power sources for running light vehicle such as: motorbikes etc.. Here considered vehicle is equipped with an air turbine in place of an internal combustion engine, and transforms the compressed air energy into shaft work. The mathematical modeling shown here is reproduced from author's earlier publications on a small capacity compressed air driven vaned type novel air turbine. The effect of different rotor to casing diameter ratios with respect to different vane angles (number of vanes) have been considered and analyzed under specific parametric conditions. The shaft work output is found optimum adopting practical conditions of rotor / casing diameter ratios on a particular value of vane angle (no. of vanes). In this study, the maximum power is obtained as 4.02 kW (5.6 HP) when casing diameter is taken 100 mm, and rotor to casing diameter ratio is kept from 0.70, as the construction of turbine can be fabricated between rotor to casing (d/D) ratio from 0.95 to 0.70 only. It is learnt that the generated power output of 4.02 kW (5.6 HP) is sufficient to run any motorbike.

features of nearly zero pollution and air compression by through its inlet nozzle and vanes of air turbine are also fitted into rotor under spring loading to maintain their regular contact with the casing wall. This arrangement is proposed to minimize leakage through mating contacts and is novelty of improvement in the vane turbine. A study on highly efficient 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 [14][15][16][17]. The study of design feasibility of vane type novel air turbine has also been carried out [18][19][20][21]. The present objective of this study is to investigate the power output of an air turbine with different number of vanes in rotor, i.e. angle between first two consecutive vanes and the rotor/casing ratio (d/D). The air turbine considered has capability to yield output of 5.0 to 5.6 HP at injection pressure 4-6 bar and speed of 2000-2500 rpm and is suitable to run a motorbike.

III.

4 MATHEMATICAL MODEL

The mathematical model shown here is already presented in author's earlier publications [22][23][24][25][26][27][28][29][30][31][32][33] **??**34] and is reproduced here for maintaining the continuity and benefits to the readers **1 ** 1 Mathematical model is reproduced here from author's earlier publications [22][23][24][25][26][27][28][29][30][31][32][33] **??**34].

. The high pressure jet of air at ambient temperature drives the rotor in novel air turbine due to both isobaric admission and adiabatic expansion. The high pressure air when enters through the inlet passage, pushes the vane for producing rotational movement and thereafter air so collected between two consecutive vanes of the rotor is gradually expanded up to exit passage, also contributes to the shaft out. This isobaric admission and adiabatic expansion of high pressure air both produces the total shaft power output from air turbine. Compressed air leaving the air turbine after expansion is sent out from the exit passage. It is thus noticed that the scavenging of the rotor is perfect and the work involved in recompression of the residual air is absent as seen from), considering adiabatic process between state 1 and 4, it can be written as:1 4 1 1 1 1 . . . 1 1 p w p v p ? ? ? ? ? ? ? ? ? ? ? ? ? = ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? (2)

The process of exit flow (4-5) takes place after the expansion process (E -4) as shown in Fig. **??** and air is released to the atmosphere. In this process; till no over expansion takes place pressure can't fall below atmospheric pressure . Thus at constant volume when pressure drops to exit pressure , 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 ( ), can be written as: considering air turbine has n number of vanes, then shaft output [35] can be written as:

( )4 1 1 4 5 4 1 1 . . . 1 . . 1 n p w n p v n p p v p ? ? ? ? ? ? ? ? ? ? ? ? ? = ? + ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? **(4)**

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

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

(5)

Figure 1 shows that if vanes are at angular spacing of ? degree, then total number of vanes will be n = (360/?). 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.

From Fig. 3, shows that when two consecutive vanes at OK and OL move to position OH and OB, the extended vane lengths varies from SK to IH and LM to BG, thus the variable length BG at variable is assumed as can be written from the geometry:

(6)

where 2R=D is diameter of casing and 2r=d is diameter of rotor, is angle BOF, is angle BAF and is angle HOB or H'OF or KOL, between two consecutive vanes and 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: (7) where BG= and IH= variable length of vanes when rotate into turbine as shown in Fig. 3 and i stands for min or max length. Thus a)

The volume at inlet or will fall between angle LOF= and angle KOF= as seen in Fig. 3, when air is admits into turbine at angle .

5 b)

The Volume at exit or will fall between angle BOF = and angle HOF =

.

Applying above conditions into equations (6), then LM=X1min , SK=X2min, FE=X1max=Corresponding to BG at =0 degree and I'H'=X 2max =Corresponding IH at = degree Applying values of X 1min and X 2min to equation (7),

Applying values of X 1max and X 2 max to equation (**??**),

Applying values of and from equations (8) and (9) to equation (5) , the total power output available can be written as :n w W N ( ) 4 1 1 4 5 4 1 1 .(/ 60). . . . 1 .(/ 60). . 1 total p W n N p v n N p p v p ? ? ? ? ? ? ? ? ? ? ? ? ? ? = ? + ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? ? i ? 'var ' at iable X ? . sin .sin ( ).cos at iable R r BG x R cos R r r R ? ? ? ? ? ? ? ? ? ? ? ? = = + ? ( )( ) 1 2 1 2 . .sin 4 i i i cuboids X X r X v L ? + + ? ? = ? ? ? ? 1i X 2i X 1 v min v ? ( ) 1min 180 ? ? ? = ? ? ? ( ) ( ) 2min 1min 180 ? ? ? ? = + = ? ? 4 v max v ? 1max 0 ? ? = = ? ( ) 2max 1max ? ? ? ? = + = ? ( ) ? ? + ? ( )( ) 1min 2min 1min min 1 2 .

.sin 4X X r X v v L ? + + ? ? = = ? ? ? ? ( )( ) 1max 2max 1max max 4 2 .

.sin 4 X X r X v v L ? + + ? ? = = ? ? ? ? 1 v 4 v total WX X r X p W n N p L p X X r X n N p p L ? ? ? ? ? ? ? ? ? ? ? + + ? ? ? ? ? ? ?? ? = ? + + ? ? + ? ? ? ? ? ? ? ? ? ? ?**(10)**

IV.

6 PARAMETRIC CONSIDERATIONS AND ANALYSIS

In this study various input parameters are listed in Table 1 for investigation of performance of vane turbine at different rotor to casing diameter ratios (d/D) with respect to different vane angle when D=100 mm, injection pressure 6 bar (90 psi) and its optimization for larger shaft output. While actual fabrication of air turbine is carried out, it is noticed that the rotor / casing (d/D) ratio can possibly be varied from 0.95 to 0.70 only.

V.

7 RESULTS AND DISCUSSION

Based on the various input parameters listed in Table -1 Figure 4 shows the variation of expansion power for the rotor vanes =12 nos. (?=30 o), 10 nos. 9. It is thus observed that in the vane turbine total shaft power output is although the combined effect of the component of expansion power and exit flow power, but contribution of expansion power is predominant. The contribution of exit flow power due to steady flow in respect to total power output varies from 6.2% to 31.4% for injection pressure of 6 bar and speed of rotation at 2500 rpm. 9, it is obvious that the expansion power output as well as total power output is found optimum when vane angle ranges from 30 o -45 o (12-8 vane nos.), injection angle at 60 o or above, at rotor/casing diameter ratio 0.70, speed of rotation at 2500 rpm and injection pressure at 6 bar and will be a deciding factor for desired shaft power output.

8 VI.

9 CONCLUSIONS

From the above investigations based on input Total output power from the air turbine is seen to be larger at injection pressure 6 bar, speed of rotation 2500 rpm and different rotor/casing diameter ratios at particular vane angles and total power ranges as shown below :

? 3.98 kW to 4.02 kW, when rotor to casing diameter ratio is 0.70 and vane nos. 12-8 (vane angle 30 o -45 o), ? 3.46 kW to 3.48 kW, when rotor to casing diameter ratio is kept 0.75 and vanes nos. 12-10 (vane angle 30 o to 36 o),

? 2.79 kW to 2.87 kW, when rotor to casing diameter ratio is kept 0.80 and vanes nos. 12-10 (vane angle 30 o to 36 o).

Thus optimum shaft power output of a novel vaned type air turbine is obtained when the design parameters for rotor diameter to casing diameter (d/D) ratios are kept between 0.75 to 0.70 and vanes nos. 12-10 (vane angle is of 30 o to 36 o) and plays an important role in designing the air turbine. [1] [2] [3] [4]

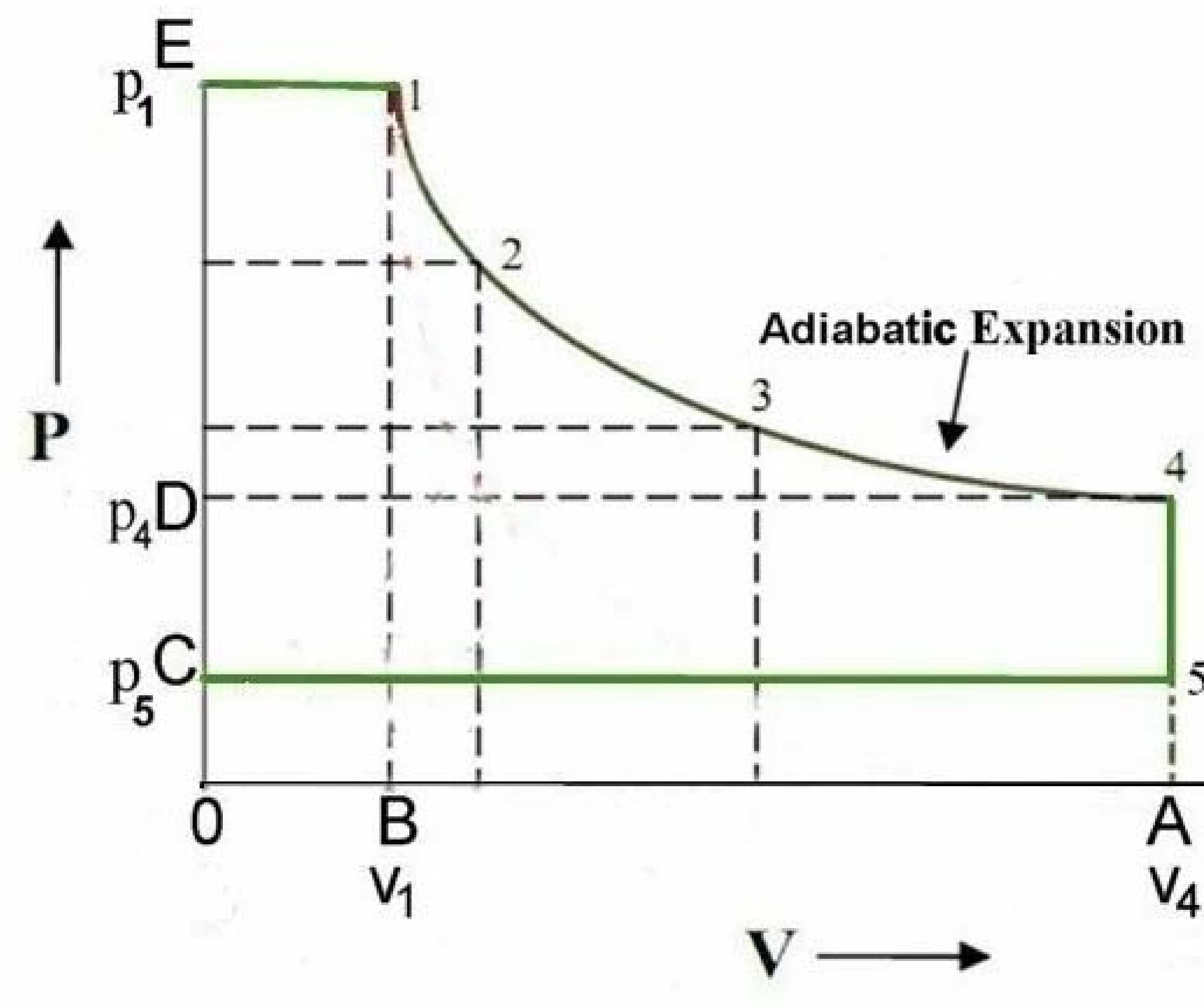
1

Figure 1: Fig. 1 :

[2],var' '

[3]Practical Approach on a Novel Multi -Vane Air Turbine

[4]International Journal of Mathematical, Physical and Study of Performance of Shaft output with Rotor-to-Casing Ratios versus Different Vane Angles Adopting © 2011 Global Journals Inc. (US)



1

Figure 2: Fig. 1 .

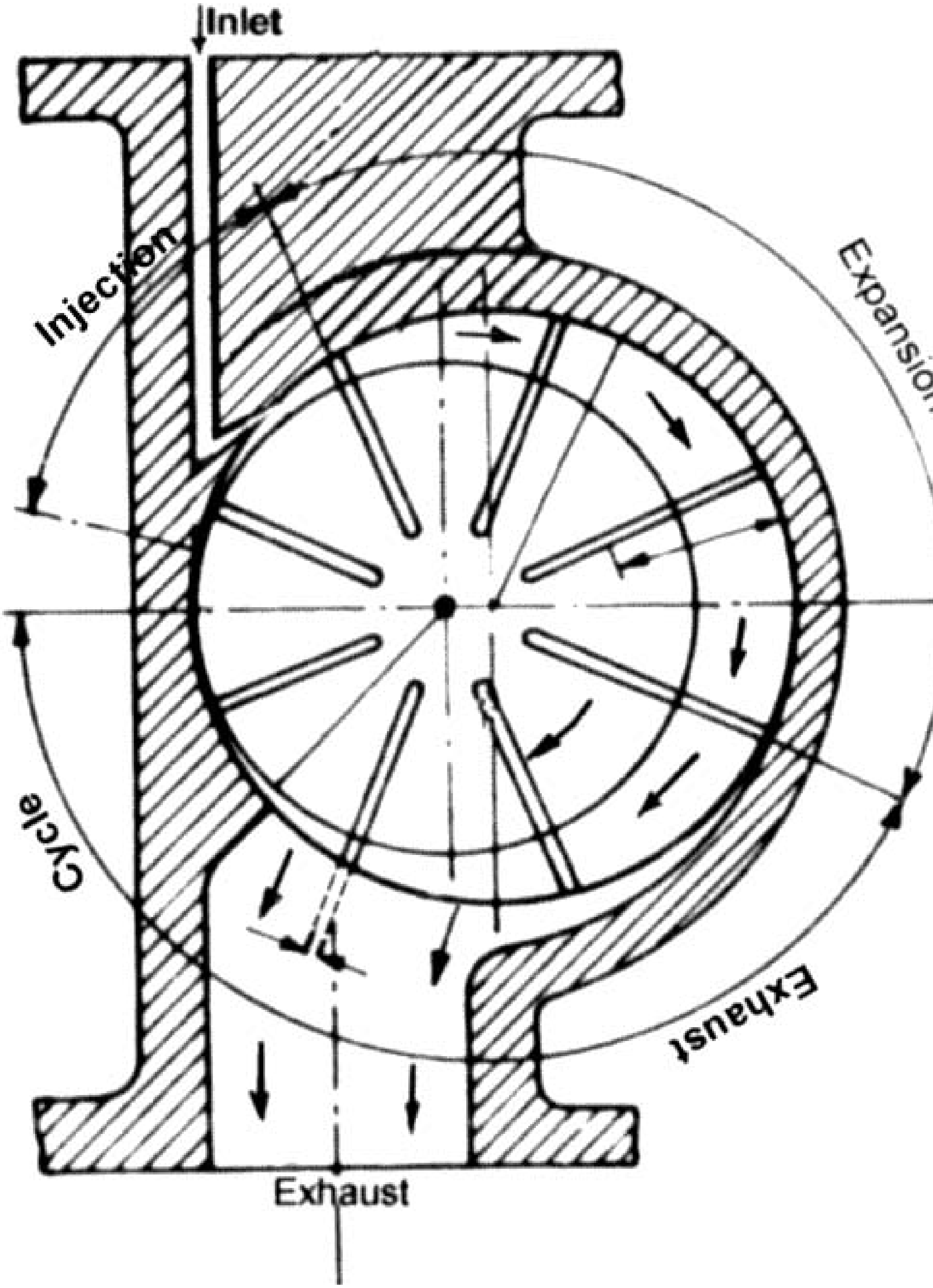


Figure 3:

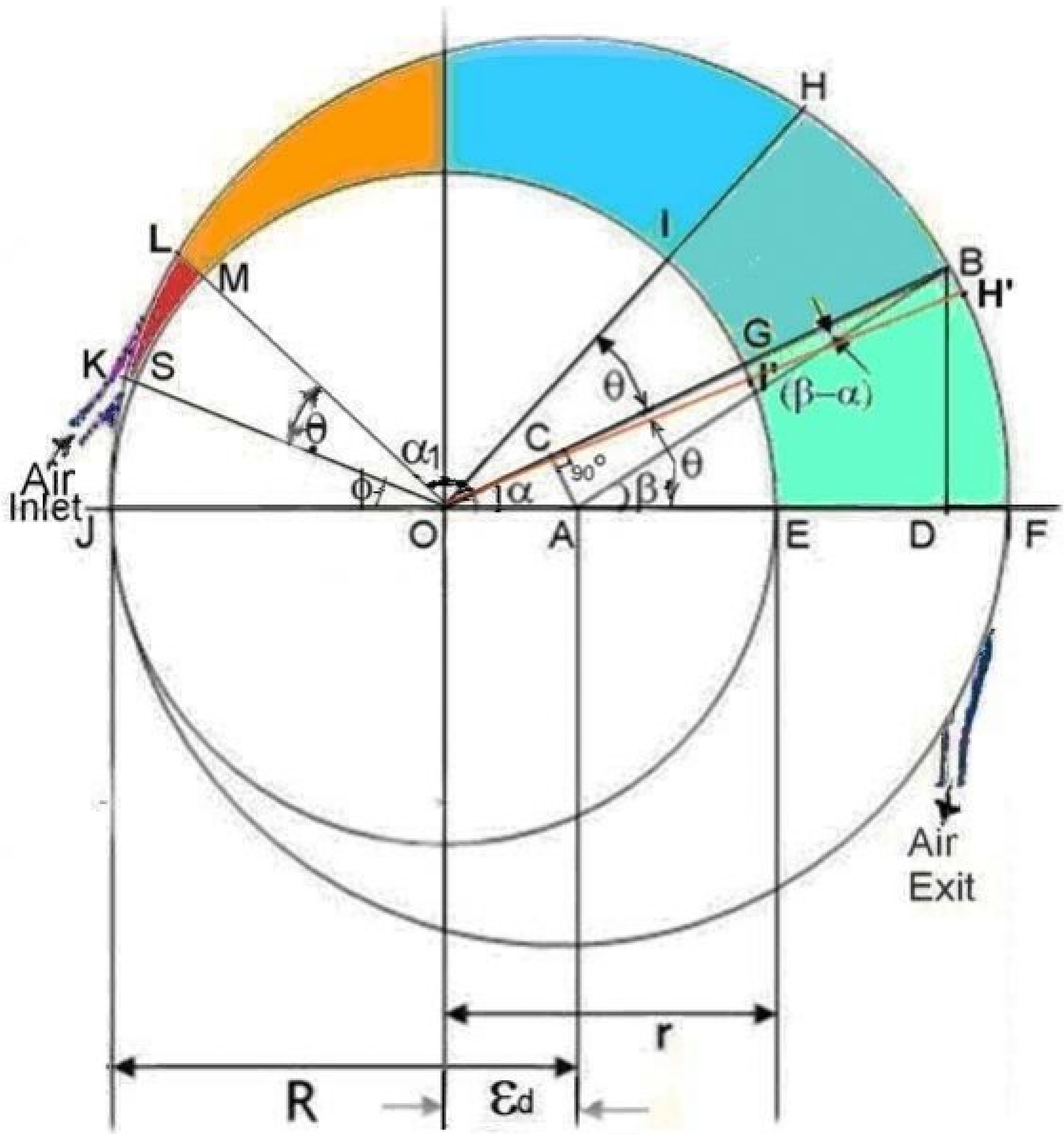


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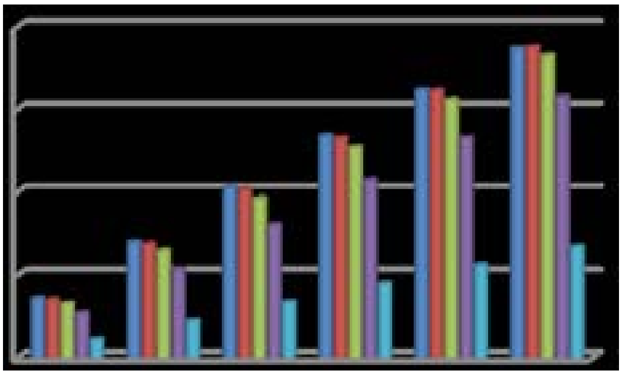
Figure 4: Fig. 3 :



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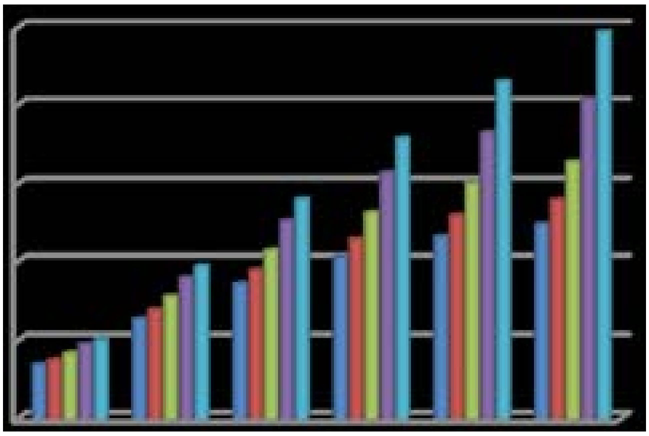

Figure 6:

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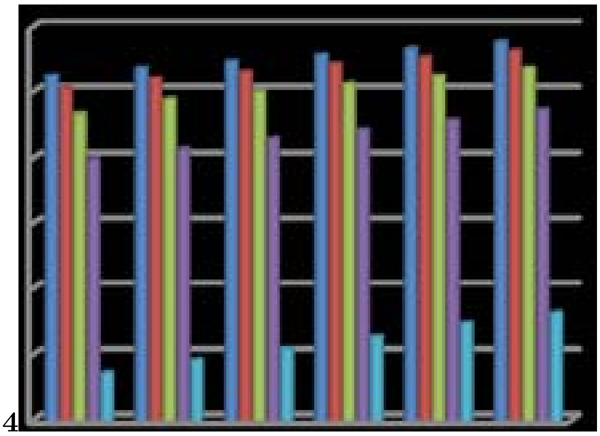


Figure 7: Fig. 4 :

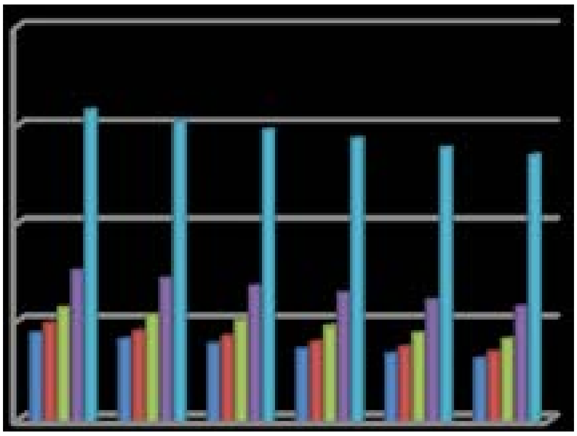


Figure 8: (

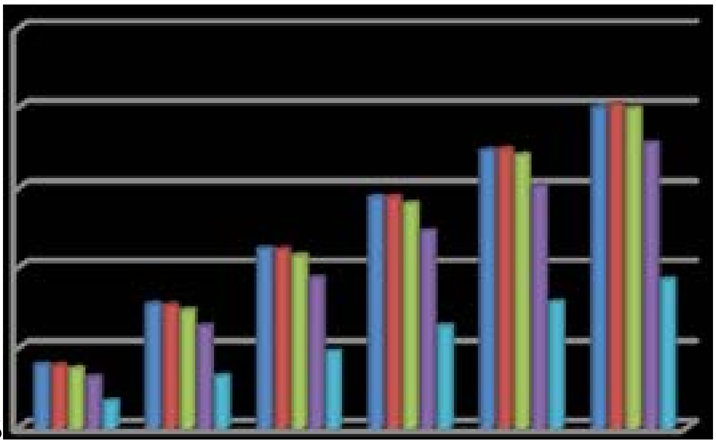
5 

Figure 9: Fig. 5 :

2011
August
1
total .(/ 60). 1 . 1 4 1 1 . . (1min) (. 2 4
2min
.(/ 60). (4 5) . . (1max) (. 2 4 2max 1max)

[Note: V (a)]

Figure 10:

1

Figure 11: Table 1 :

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