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# IMPROVED DESIGN OF AIR FLOW FOR A TWO STROKE INTERNAL COMBUSTION ENGINE TO PROMOTE CLEANER COMBUSTION

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## ABSTRACT

In the present work, an attempt is made to reduce scavenging problems by developing a new model of two-stroke spark-ignition engine. This model allows a flow of fresh air through intake valves positioned at the bottom of the cylinder and exit of burnt gases through exhaust ports situated at the top of cylinder. The exhaust ports are closed by the piston as it moves towards bottom of the cylinder following which gasoline is injected minimizing the possibility of mixing of fuel with outgoing exhaust gases. During expansion the piston unravels the exhaust ports and the burnt gases escape to the atmosphere due to pressure difference. Due to low density at high temperatures, the exhaust gases naturally rise against the direction of gravity reducing the possibility of mixing with incoming fresh air. Further, a comparative study of fuel distribution inside the cylinder has shown greater distribution when injection takes place against gravity than along gravity, which promotes cleaner combustion. The combustion analysis is done using Diesel-RK software and flow analysis is done using ANSYS FLUENT.

**Keywords:** two stroke engine, SI engine, scavenging, intake valves and exhaust ports, conversion of 4 stroke to 2 stroke.

## INTRODUCTION

The air flow system in two stroke engines is governed by the scavenging process. The process of introducing air and expelling burnt gases is called scavenging. The incoming air is used to clean out or scavenge the exhaust gases and then to fill or charge the space with fresh air [1]. Scavenging arrangements are classified into: (a) cross-scavenged, (b) loop- scavenged, and (c) unifow-scavenged configurations [2]. In case of crankcase scavenging method, the air is drawn into the crankcase due to the suction created by the upward stroke of the piston. On the down stroke of the piston it is compressed in the crankcase that increases its pressure enabling the air to flow into the cylinder through the transfer port. The piston compresses the air as it ascends, till the piston is nearly at the top of its stroke. The compression pressure is increased sufficiently and the cycle continues. The uniflow system may use inlet ports with exhaust valves in the cylinder head [2]. This is because of flow of scavenging air in one (uni) direction. Loop scavenged engines are fitted with exhaust ports which are located just above the scavenge ports. As the piston uncovers the exhaust ports on the power stroke, the exhaust gas starts leaving the cylinder. When the exhaust ports are uncovered, scavenge air, loops around the cylinder and pushes the remaining exhaust gas out of the cylinder.

The real scavenging process is characterized by two problems common to two-stroke engines: short-circuiting losses and mixing [1]. In short circuiting process, there is a mixing of fresh air fuel mixture with exhaust gases and nearly 35% of fresh charges are lost through exhaust valve. This is dead loss and it should be avoided [4]. Mixing consists on the fact that there is a small amount of residual gases which remain trapped without being expelled are being mixed with some of the new air charge [1]. This leads to an incomplete combustion emitting significant amount of particulate matter, un-burnt hydrocarbons, carbon mono-oxide (CO) and Nitrous oxides (Nox). Conventional existing two stroke engines are well known to pollute badly and their future will be limited due to excessive pollution [3].

During the recent decades, new factors such as the need to control the automotive air pollution and, the need to achieve improvements in fuel consumption have become important. With depletion of oil resources, the cost of fuel is rising steadily which demanded the need for fuel efficient engines. They concern mainly new combustion processes to improve light load operation, and new fuel introduction technologies to reduce fuel short circuiting to the atmosphere [3]. To design a new engine we need to know the engine performance in earlier cases and then finding the remedy [5].

The performance of 2-cycle internal combustion engines is calculated very largely by the process of charging [6]. The combustion products resulting from the previous explosion have to be exhausted as far as possible by the fresh charge of air in the course of less than one quarter of a revolution. It is blown in through the inlet valves and drives before it through the open exhaust ports the exhaust gases. some of the fresh charge or of the scavenging air passes away to the exhaust and is wasted and Inevitably some mixing can occur. The performance of the engine mainly depends on the amount of this waste. Its economy affected by the quantity of waste of fuel and also the classic mixing equation for scavenging is derived.

A spatial stratification of the incoming fresh charge is utilized in many low-emission concepts [7]. With temporal stratification, the so called airhead stratified scavenging, an air buffer is loaded upstream of the mixture

in the transfer ports. This air buffer enters the cylinder first and separates the fresh fuel charge from the exhaust. A specific embodiment of the stratified charging technique utilizes burned gas for the "air" head stratified scavenging approach. The burned gas is taken directly from the muffler. By guiding the gas from the exhaust to the transfer ports by means of a circumferential groove in the piston, this concept is very compact and does not need additional part. engine with diesel and blends of bio-diesel and diesel The innovative two-stroke car diesel engine concept from Aumet Oy is based on a very rapid gas exchange through poppet valves in the cylinder head when the piston approaches top dead centre [8]. This rapid gas exchange is achieved by high- pressure scavenging air that is produced externally.The mathematical model and numerical schemes are developed for solving the flow through a two-stroke engine with moving parts [9]. First, the computation of the convective flux function is carried out by the AUSMDV Riemann solver, which has been proven to be very efficient in comparison to other schemes. Then the introduction of the temperature dependency of the material properties of the fluid has augmented the realistic setting within the compression and expansion of the hot gas within the cylinder. A further reduction in computational time has been obtained by the parallelization of the numerical scheme and the mesh routines. A simulation of flow through real-world geometry of an existing two- stroke engine has been performed, the results have been validated with measured pressure data for this engine, and the flow has been qualitatively and quantitatively studied.

This work deals with air flow design of a fuel injected, two stroke cycle engine. The focus of this design is to avoid scavenging problems of short circuiting and mixing. Figure-1 gives a pictorial representation of the proposed model.

The working cycle of the above model is explained below:

a) Initially, the piston is at the end of the compression stroke. Spark initiates combustion and the piston starts moving towards the bottom dead Centre.
b) As the piston uncovers the exhaust ports, due to high pressure difference between the cylinder and the exhaust manifold, the burnt gases inside the cylinder start moving out.
c) When the pressure inside the cylinder drops below the pressure at inlet, the inlet air valves are opened. Due to the pressure difference the inlet air aids in pushing out the exhaust gases.
d) As the piston covers the ports, the inlet air valves are closed followed by fuel injection and the cycle gets repeated.

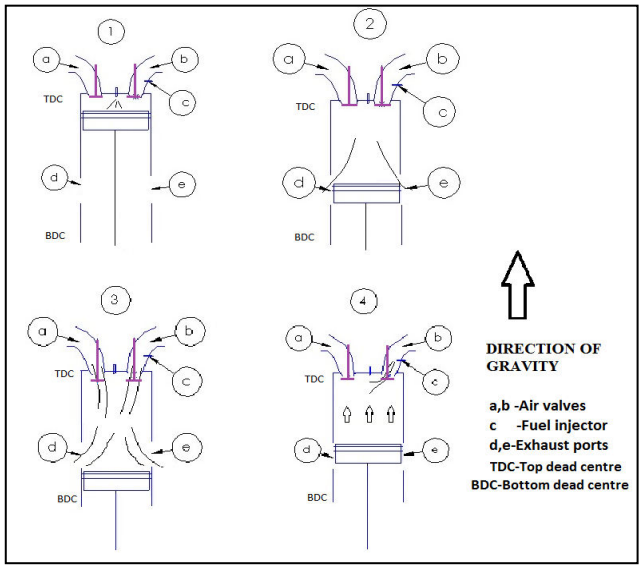


**Figure-1.** Working model of the proposed engine.

## METHODOLOGY

Four stroke IC engines use valves for intake and exhaust strokes. Changing their valve timing by cam adjustment followed by the introduction of ports in the same cylinder would satisfy the structural requirements of the proposed model. Also, a majority of real world two stroke engines do not have ports and valves configuration. Hence, modifying them according to the proposed model would be more complex than that of a four stroke engine.

The air flow system of a two stroke gasoline fuel injected engine is developed based on a four stroke gasoline engine. The valve timing of a four stroke Royal Enfield Super Meteor engine as in Figure-2 of was initially altered for a two stroke engine. Table-1 represents the initial conversion.

**Table-1.** Initial conversion of valve timing.

| **Operating stroke** | **Royal enfield super meteor engine** (degrees) | **Proposed model** (degrees) |
|---|---|---|
| Suction | 270 | 115 |
| Compression | 120 | 120 |
| Expansion | 105 | 120 |
| Exhaust | 290 | 120 |

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"Super Meteor" Engine

| | |
|---|---|
| Cubic Capacity ... ... ... | 692 c.c. |
| Stroke ... ... Nominal | 90 m.m. |
| Bore ... ... ... Nominal | 70 m.m. |
| Actual | 69·874 m.m./2·751 in. |
| (Rebore to ·020 in. when wear exceeds ·0065 in. and again to ·040 in. after further ·0065 in. wear.) | |
| Compression Ratio ... ... ... | 7½ to 1 |
| Piston Diameter— | |
| Bottom of Skirt—Fore and Aft. | 69·811 m.m. |
| Top Lands ... | 69·40/69·35 m.m. |
| Skirt is tapered and oval-turned. | |
| Piston Rings— | |
| Width—Plain Rings ... ... | ·0625/·0635 in. |
| Scraper Ring ... ... | ·1550/·1560 in. |
| Radial Thickness ... ... | 2·883/3·085 m.m. |
| Gap when in unworn Cylinder ... | ·011/·015 in. |
| Clearance in grooves ... ... | ·001/·003 in. |
| Renew Piston Rings when gap exceeds 1/16 in. | |
| Oversize Pistons and Rings available ·020 and ·040 in. | |
| Piston Boss Internal Diameter ... | ·7499/·7501 in. |
| Gudgeon Pin Diameter ... ... | ·7499/·7501 in. |
| Con. Rod Small End Internal Diameter | ·7507/·7505 in. |
| Big End Internal Diameter ... | 1·8760/1·8755 in. |
| Crank Pin Diameter ... ... | 1·875/1·8795 in. |
| Driving Side Main Ball Bearing— | |
| Type ... ... ... ... | Hoffman—145 or R and M—LJ 45 |
| Outside Diameter ... ... | 85 m.m. |
| Inside Diameter ... ... | 45 m.m. |
| Width ... ... ... ... | 19 m.m. |
| Timing Side Main Roller Bearing— | |
| Type ... ... ... ... | Hoffman—R145 or R and M—LRJ54 |
| Outside Diameter ... ... | 85 m.m. |
| Inside Diameter... ... ... | 45 m.m. |
| Width ... ... ... ... | 19 m.m. |
| Rocker Inside Diameter ... ... | ·5627/·5622 in. |
| Rocker Bearing Inside Diameter ... | ·5622/·5617 in. |
| Rocker Spindle Diameter ... ... | ·5617/·5615 in. |
| Inlet Valve Stem Diameter... ... | ·3430/·3425 in. |
| Exhaust Valve Stem Diameter ... | ·3410/·3405 in. |
| Valve Guide Internal Diameter ... | ·3437/·3447 in. |
| Valve Guide External Diameter ... | ·6275/·6270 in. |
| Valve Guide Hole in Cylinder Head Dia. | ·625/·626 in. |
| Tappet Stem Diameter ... ... | ·3743/·3740 in. |
| Tappet Guide Internal Diameter | ·3755/·3745 in. |
| Tappet Guide External Diameter... | 1·0125/1·0130 in. |
| Tappet Guide Hole in Crankcase Dia. | 1·011/1·010 in. |
| Tappet Clearance with cold engine— | |
| Normal Running: | |
| Inlet ... ... ... ... | Nil |
| Exhaust ... ... ... ... | Nil |
| Continuous High Speed Running: | |
| Inlet ... ... ... ... | Nil |
| Exhaust ... ... ... ... | ·005 in. |
| Valve Spring Free Length— | |
| Inner ... ... ... ... | 2 1/32 in. |
| Outer ... ... ... ... | 2 3/32 in. |
| (Renew when reduced by 1/16 in.) | |
| Valve Timing with ·012 in. clearance— | |
| Exhaust Opens ... ... ... | 75° before B.D.C. |
| Exhaust Closes ... ... | 35° after T.D.C. |
| Inlet Opens ... ... ... | 30° before T.D.C. |
| Inlet Closes ... ... ... | 60° after B.D.C. |
| Camshaft Bearing External Diameter | ·9095/·9085 in. |
| Camshaft Bearing Internal Diameter (Bored in position in crankcase) | ·7505/·7495 in. |
| Cam Lift ... ... ... ... | ·3125 in. |
| Valve Lift (approx.) ... ... | ·3125 in. |
| Timing Sprocket ... ... ... | 12 Teeth |
| Camshaft Sprocket ... ... ... | 24 Teeth |
| Magneto Sprocket ... ... ... | 19 Teeth |
| Timing Chain—Type ... | Single No. 110038 endless |
| Length ... ... | 66 pitches |
| Width ... ... | ·225 in. |
| Pitch ... ... | ·375 in. |
| Roller ... ... | ·250 in. |
| Magneto Chain—Type ... | Duplex No. 114500 endless |
| Length ... ... | 44 pitches |
| Width ... ... | 8·64 m.m. |
| Pitch ... ... | 8 m.m. |
| Roller ... ... | 5 m.m. |
| Magneto Speed ... ... ... | Half Engine Speed |
| Points ... ... ... ... | ·015 in. |
| Timing Retarded ... ... | 1/32 in. before T.D.C. |
| Timing Advanced ... | 1/8 in.—1/16 in. before T.D.C. |
| Distributor Chain—Type ... | Single No. 110500 endless |
| Length ... ... | 40 pitches |
| Width ... ... | 8·64 m.m. |
| Pitch ... ... | 8 m.m. |
| Roller Diameter | 5 m.m. |
| Distributor Speed ... ... ... | Half Engine Speed |
| Points Gap ... ... ... | ·015 in. |
| Timing Retarded ... ... | 1/32 in. before T.D.C. |
| Timing Advanced ... | 1/8 in.—1/16 in. before T.D.C. |
| Engine Sprocket ... ... ... | 33 Teeth |
| Clutch Sprocket ... ... | 56 Teeth |
| Final Drive Sprocket (Solo) ... ... | 18 Teeth |
| Final Drive Sprocket (Sidecar) ... | 16 Teeth |
| Primary Chain Type ... | Duplex No. 114038 endless |
| Length ... ... ... ... | 94 pitches |
| Width ... ... ... ... | ·628 in. |
| Pitch ... ... ... ... | ·375 in. |
| Roller ... ... ... ... | ·250 in. |
| Feed Oil Pump—Speed ... ... | 1/6 Engine Speed. |
| Piston Diameter ... | ·24975/·24950 |
| Stroke ... ... | ·5 in. |
| Return Oil Pump—Speed ... ... | 1/6 Engine Speed. |
| Piston Diameter | ·375/·3755 in. |
| Stroke ... ... | ·5 in. |
| Sparking Plug. Type (i) Engines Nos. up to SMQA 4687 (Short Reach): Lodge H14, KLG F70 or Champion L7 (formerly L10S). (ii) Engines Nos. SMQA. 4688 onwards (Long Reach): Lodge HLN, KLG FE70 or Champion N5 (formerly NA8). | |
| Diameter ... | 14 m.m. |

**Figure-2.** Technical details of royal enfield super meteor engine.

To obtain the air flow and thermodynamic parameters of a four stroke engine, geometrical parameters (stroke, bore, compression ratio, valve size and timing) of Royal Enfield Super Meteor Engine are used as inputs to Diesel R K software to form the required reference engine parameters as shown below.

Parameters of Efficiency and Power
5000.0 - Engine Speed, rev/min
0.02346 - Mass of Fuel Supplied per cycle, g
Environmental Parameters
1.0000 - Total Ambient Pressure, bar
288.00 - Total Ambient Temperature, K
Turbo-charging and Gas Exchange
1.0001 - Total Air Fuel Equivalence Ratio
Combustion
50.487 - Maximum Cylinder Pressure, bar
15.000 - Angle of Max. Cylinder Pressure, degree After TDC
25.000 - Ignition Timing, degree before TDC Main Engine Construction Parameters
7.5000 - Compression Ratio
75.000 - Exhaust Valve Opening, degree before BDC
35.000 - Exhaust Valve Closing, degree after TDC
30.000 - Intake Valve Opening, degree before BDC
60.000 - Intake Valve Closing, degree after BDC

The compression and expansion stroke duration of the proposed model are fixed almost the same as that of the reference engine by altering the intake and exhaust stroke duration. This leads to the formulation of the objectives to ensure the feasibility of the proposed model.

- Why is fuel injector mandatory according to this idea?
- Will sufficient amount of air be injected through the valves in the stipulated time?
- Will forced induction be necessary to achieve the working of the cycle?
- Has the cylinder pressure dropped below the inlet pressure during inlet valve opening?
- Is it feasible to send the burnt gases out of the cylinder within the stipulated time?
- Will there be a mixing of exhaust gases with inlet fresh charge?

## DETERMINATION OF MASS AND VELOCITY OF AIR DURING INTAKE

The chemical formula for pure octane fuel is $C_8H_{18}$.

$$C_8H_{18} + 12.5\ O_2 = 8\ CO_2 + 9\ H_2O \quad (1)$$

The stoichiometric air to fuel ratio is 15.1:1.

However, chemical analysis for perfect gasoline/air is

$$C_8H_{15.4} + 11.85\ O_2 = 8\ CO_2 + 7.7\ H_2O \quad (2)$$

The stoichiometric air fuel ratio is 14.7:1. However, under wide open throttle conditions it is considered as 14.6:1 [Heywood]. In the present work, the engine is run at a constant 5000 revolutions per minute. As throttling is not done wide open, throttle conditions are considered for calculations. The total mass of air is found from the mass of fuel supplied and air fuel equivalence ratio as in Figure-3.

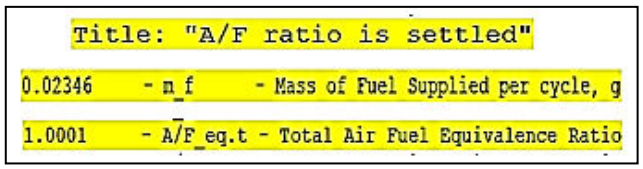


**Figure-3.** Air fuel ratio and mass of fuel supplied in reference engine.

Mass of Air determination:

$$1.001 = (A/F\ actual) / A/F\ stoichiometric) \quad (3)$$

$14.6 \times 1.001 = A/F$ actual

Mass of fuel supplied = $0.02346 \times 10^{-3}$ kg

Actual air supplied = $3.425 \times 10^{-4}$ kg

Total flow through intake = mass of air + mass of fuel

$= 3.425 \times 10^{-4}$ kg $+ 0.02346 \times 10^{-3}$ kg

$= 3.65 \times 10^{-4}$ kg

Velocity of air flow through valve orifice:

By applying flow rate equation,

$\Pi/4 \times d^2 \times \rho \times$ Velocity$\times$ intake duration = $3.65\times10^{-4}$ kg
Intake duration = $9\times 10^{-3}$ seconds (270 degrees)
d = diameter of valve orifice (2.66 cm)
Velocity = 60.513 m/s.

## EXHAUST CALCULATIONS

The focus of exhaust calculations was to narrow down to the dimensions of exhaust port that is sufficient enough to send the mass of burnt gases out of the cylinder within 120 degrees of crank angle and simultaneously attain a cylinder pressure lesser than the inlet pressure. The pressure at inlet was initially assumed to be atmospheric.

Figure-4 represents the cylinder dimensions of the proposed model.

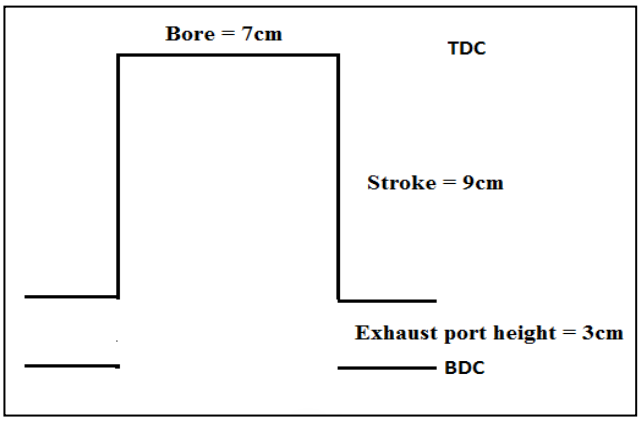


**Figure-4.** Exhaust port location and dimension in model engine.

The existing crankcase scavenging system engine was modified in such a way that the exhaust port opens at 6 cm from the top dead Centre. The height of transfer port measured from the bottom dead Centre was adjusted to vary the height of exhaust port. Initially, the transfer port extended 2 cm above BDC. Almost the same amount of fuel (0.02350 g of gasoline) as in the reference engine was supplied. The air fuel equivalence ratio was maintained at 1.001. The pressure drop at the end of 1 cm and 7 cm wide exhaust port was 6 bar which would not permit the flow of charge into the cylinder during intake. So the transfer port was reduced to 1cm above BDC .This increased the exhaust port height to 2cm and the width of the port was extended to 20 cm. The amount of fuel supplied was 0.02336 g per cycle (slightly lesser than that of the reference engine).

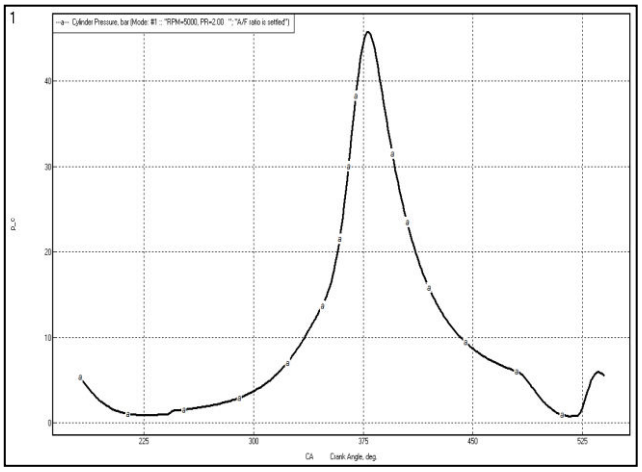

**Figure-5.** Pressure vs crank angle diagram of crankcase scavenged model (0.02336 g of gasoline).

The pressure at the end of 2 cm was found to be 0.83 bar which was less than atmospheric pressure.

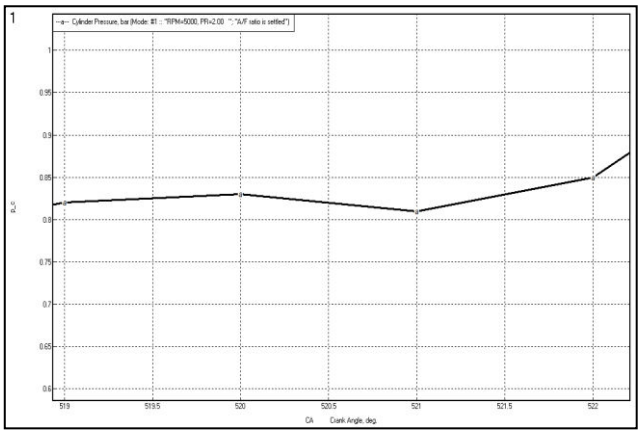

**Figure-6.** Pressure at the end of 2 cm height of exhaust port (0.02336 g of gasoline).

In the next iteration, 0.02361 g of gasoline was supplied (slightly higher than that of the reference engine).

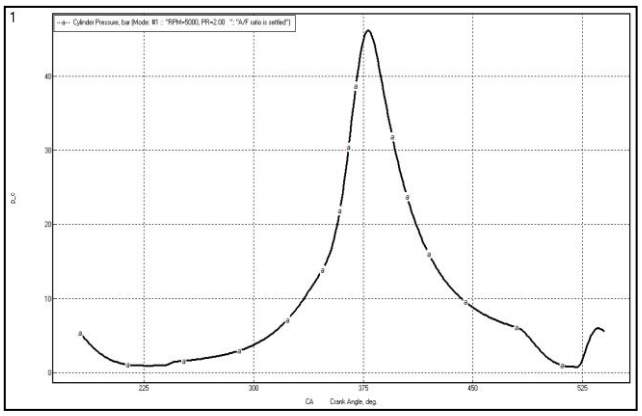

**Figure-7.** Pressure vs crank diagram of crankcase scavenged model (0.02361 g fuel).

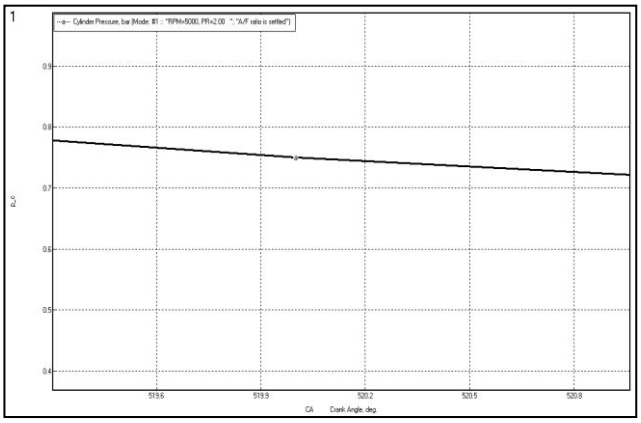

**Figure-8.** Pressure at the end of 2 cm height of Exhaust Port (0.02361 g fuel).

The port of dimension 2 cm height and 20 cm width lowered the cylinder pressure, below the atmospheric pressure at the end of expansion. To check whether the exhaust port is sufficient enough to send out 3.65 $\times 10^{-4}$ kg of burnt gases, two separate integrals were done: one for velocity (Figure-9) and another for area (Figure-10).

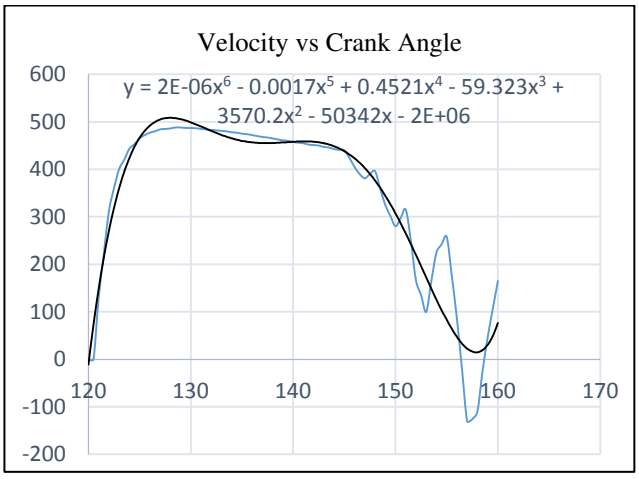


**Figure-9.** Velocity curve (m/s) of exhaust gases.

The curves from 0.02336 g fuel injection was chosen for calculation as it is closest to the amount of fuel injected in reference engine (0.02346 g).

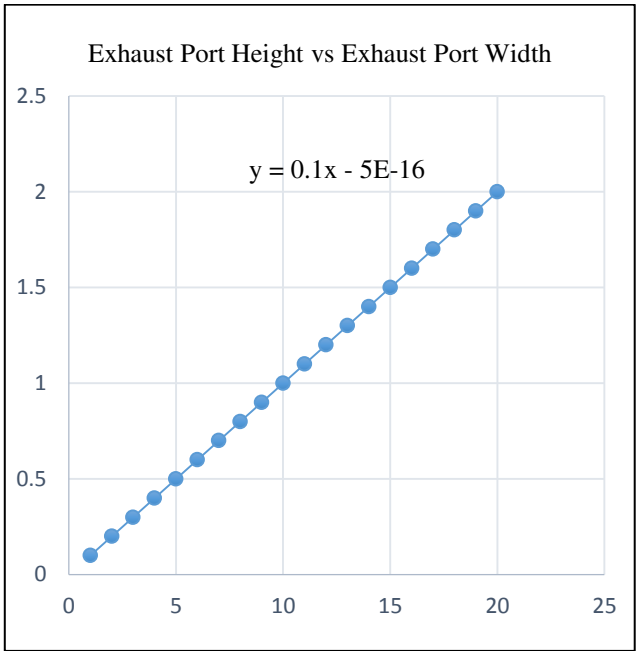


**Figure-10.** Area (cm$^2$) curve of exhaust ports.

The velocity is integrated for the time interval of 40 degrees.

$$\int_0^{0.00133} (2\times10^{-6}x^6 - 0.0017x^5 + 0.4521x^4 - 59.323x^3$$

$$+3570.2x^2 - 50342x - 2\times10^6)dx$$

$$\begin{bmatrix} 2\times10^{-6}(x^7/7) - 0.0017(x^6/6) + 0.4521(x^5/5) \\ -59.323(x^4/4) + 3570.2(x^3/3) - 50342(x^2/2) - 2\times10^6(x) \end{bmatrix}_0^{0.00133}$$

$$\begin{bmatrix} 2\times10^{-6}(1.05\times10^{-21}) - 0.0017(9.22\times10^{-19}) + 0.4521(2.66\times10^{-16}) \\ -59.323(7.842\times10^{-10}) + 2.799\times10^{-6} - 2660 \end{bmatrix}$$

$= -2660.04$m

The equation of area is integrated for the length of the port,

$$\int_0^2 10^{-2}(0.1x - 5\times10^{-16})dx$$

$$= \left[0.1(x^2/2) - 5\times10^{-16}(x)\right]_0^2$$

$$= \left[10^{-2}(0.2) - 10^{-15}(10^{-2})\right]$$

$=0.002$m$^2$

The peak temperature is 2545 degree celsius and the corresponding pressure is 46.8 bar. The corresponding density of air is 0.5780 kg/m$^3$.

Mass =density x- 2660.04m x 0.002m$^2$ =3.075 kg (4.1) which was greater than the mass of incoming charge (3.65$\times10^{-4}$ kg).

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## INTAKE CALCULATIONS

The objective of intake design was to permit 3.65 $\times 10^{-4}$ kg of fresh air within 80 degree of crank angle as the desirable cylinder pressure occurs only after 40 degree of exhaust process. However, the compression in cylinder starts when the first piston ring reaches point A. Figure-11 illustrates the scenario of sealing of combustion chamber.

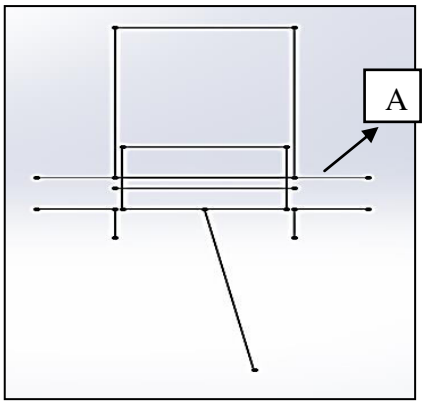


**Figure-11.** Effect of piston rings on intake duration.

So the distance between the piston face and compression ring provides additional time for intake before the compression process actually starts. Figure-12 represents the cross section of a piston.

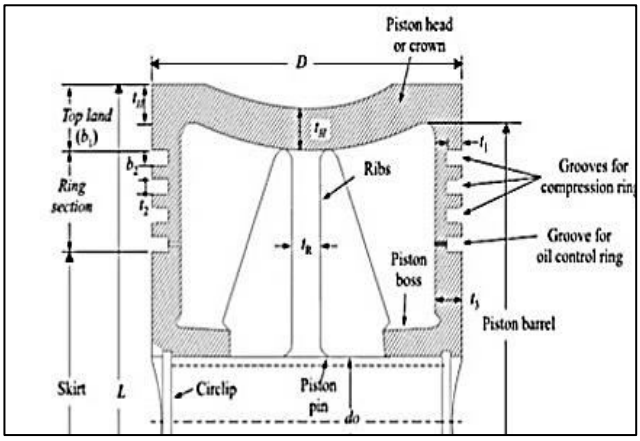


**Figure-12.** Cross sectional view of a piston.

Width of top land ($b_1$) = $t_h$ or 1.2 $t_h$

$$t_h = \sqrt{3PD^2 / 16\sigma_t} \quad (4)$$

P = maximum gas pressure
D = Bore diameter (mm)
$\sigma_t$ = Permissible stress (90 MPa)

For maximum gas pressure of 4.68 MPa

$t_h$ = 6.91 mm.

The thickness can also be determined using formula in Figure-13.

The thickness of the piston $t_H = \frac{H}{12.56k(T_c - T_e)}$ (in mm)
H = Heat flowing through the piston Head
k = Heat conductivity factor in W/m/°C
= 46.6 W/m/°C for Grey Cast Iron
= 51.25 W/m/°C for Steel
= 174.75 W/m/°C for Aluminum alloys
Tc = Temperture at Center, Te = Temperature at Edge

Heat flowing through the head H= C x HCV x m x B.P (kW)
C = Constant = 0.05
HCV = Higher Calorific Value of the Fuel in kJ/kg
= 45x103 kJ/kg for diesel & 47x103 kJ/kg for Petrol
m = Mass of the fuel used (kg/B.P/sec)
B.P = Brake Power of the engine per cylinder

**Figure-13.** Determination of top land height.

$T_c - T_e = 75^o C$ Thus, the thickness of the piston is found out by the following calculations.

$$B_g = (B_{MEP} \times L \times A \times N \times K) / 60$$
$$B_g = 47.851 kw$$
$$B_g = 64.169 BHP \quad (5)$$

$$H = C \times HCV \times m \times BP(kw)$$
$$H = 0.05 \times 47 \times 103 \times m \times 47.851 \quad (6)$$
$$360^o \times 5000 = 60 \sec$$
$$360^o = x$$
$$x = 0.012 \sec$$

Here, x denotes time for one complete revolution

$$m = (1.95 \times 10^{-3}) / \mathrm{BP} = 4.085 \times 10^{-5}$$
$$H = 0.4732 kw$$
$$t_h = H / (12.56 \times k \times (T_c - T_e))$$
$$t_h = 2.87 mm$$

Thus, an intermediate value of 5mm (between 2.87 mm and 6.91 mm) was chosen $b_1$. This 5 mm gave an additional 10 degree to the intake stroke. The diameter required for the valves are found out by the following calculation. The number of valves is taken as 3 considering geometrical limitations.
Time = 90 degree = 0.0003 sec
Mass of the air inducted =

$$(\pi / 4) \times d^2 \times \rho \times V \times time \times n = 3.65 \times 10^{-4}\ Kg \qquad (7)$$

d = 2.66 cm. The diameter of the valve orifice was fixed and the intake pressure was modified to achieve the suitable design. Figure-14 shows the variation of valve opening area with crank angle.

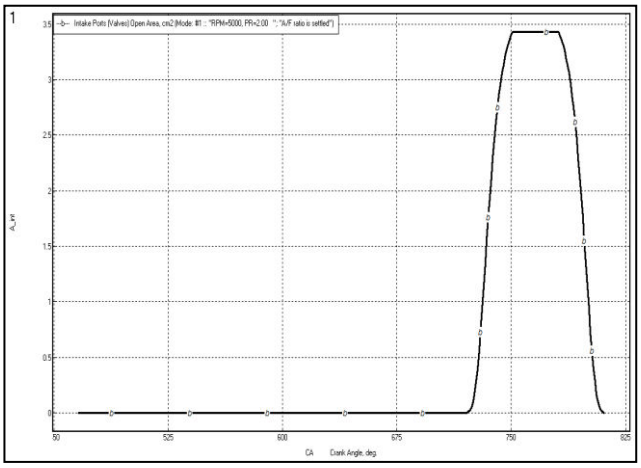

**Figure-14.** Intake valve open area.

Forced induction at 1.3 bar inlet pressure was carried out in the first iteration. 3 valves with the diameter of 26.6 mm pump in $4.7 \times 10^{-4}\ Kg$ of air was higher than the required amount of air during intake (3.65 X 10-4 kg.)

The next iteration was carried out at a pressure of 1.1 bar. Accounting in density variations with pressure

$$P = \rho \times R \times T \qquad (8)$$

$1.1\times10^{-5}/ (288 \times297) =1.303$ kg/m$^3$

The mass of air through the inlet was 3.9783 X 10-4 kg which was still more than required.

Figure-15 and Figure-16 shows variation of intake pressure and velocity with crank angle for forced induction at 1.1 bar.

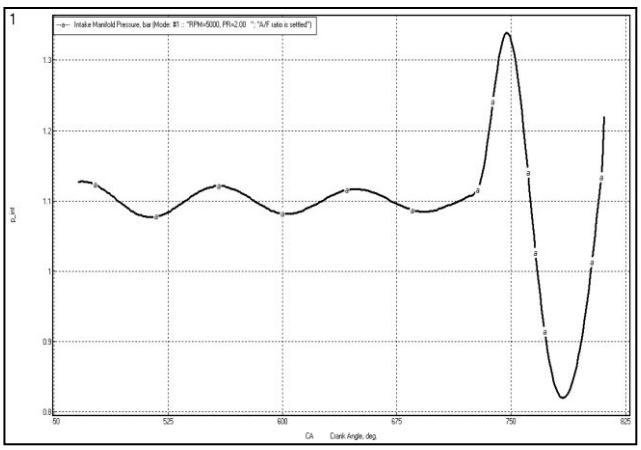

**Figure-15.** Intake pressure variation vs crank angle at 1.1 bar inlet pressure.

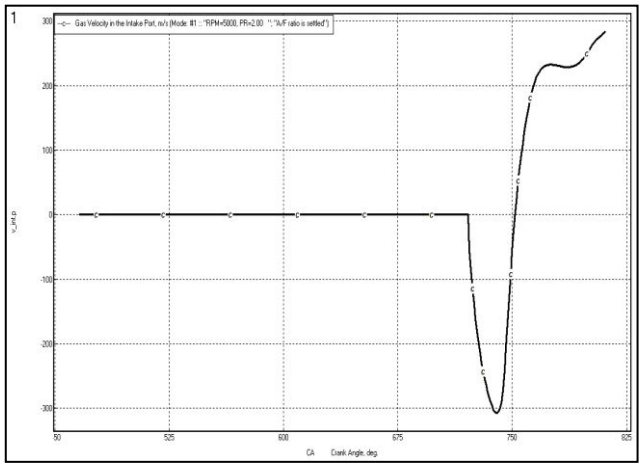

**Figure-16.** Intake velocity variation vs crank angle at 1.1 bar inlet pressure.

The next iteration was carried out for natural aspiration. Mass of air through inlet was 3.6016 X $10^{-4}$ kg, which was slightly less than the required. Figure-17 and Figure-18 show the variation of intake pressure and velocity with crank angle.

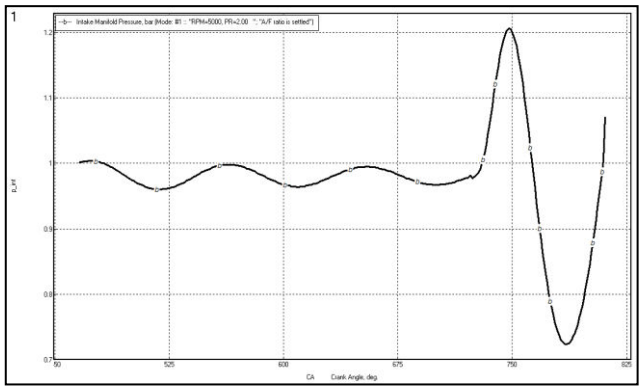

**Figure-17.** Intake pressure variation vs crank aangle for natural aspiration.

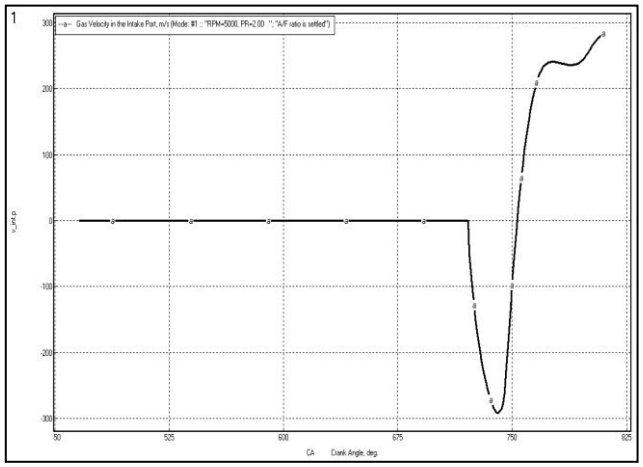

**Figure-18.** Intake velocity variation vs crank angle for natural aspiration.

The cylinder pressure during the opening of inlet valve is 0.75 bar which would permit the flow of air from intake at 1 bar into the cylinder. Thus the natural aspiration would serve for the intake purpose. However, forced induction at 1.1 bar would be the most suitable intake design.

## EFFECT OF THE DISTANCE BETWEEN THE PISTON FACE AND COMPRESSION RING ON COMPRESSION RATIO

The clearance between piston and bore is between 0.1 mm to 0.3 mm. In case of the 4 stroke engine, the combustion chamber is completely sealed. Thus, the compression ratio is not altered by the position of piston rings. In the case of the proposed model, the combustion chamber gets sealed only when the first piston ring reaches the end of exhaust port, which causes a small reduction in the compression ratio.

Bore=7 cm  Piston diameter = 7 cm- 0.2 mm =69.8 mm.
Difference in area = 3.14/4× (bore$^2$ – piston diameter$^2$)
= 21.9 mm$^2$
Clearance volume = (height from piston ring × 21.9 mm$^2$)
= 5 mm × 21.9 mm$^2$ = 0.1097 cm$^2$
Maximum volume of the cylinder =
= (3.14/4) ×D$^2$×L = 436.18 cm$^3$
Compression ratio theoretical =
7.5 = (346.18 cm$^3$ / Minimum volume of the cylinder)
Minimum volume of the cylinder = 46.158 cm$^3$
Actual minimum volume= 46.15 cm$^3$+clearance volume
= 46.267 cm$^3$.
Actual compression ratio =7.482.

## GRAVITY EFFECTS ON CRANK AND SLIDER

The piston opens the exhaust ports as it moves against gravity during expansion stroke. This is analogous to inverting the cylinder of conventional IC engine. The changes in acceleration and velocity of the mechanism due to inversion was examined.

**Table-2.** Specifications of crank slider components.

| Part name | Height /Length (cm) | Width/ Diameter (cm) | Thickness (cm) |
|---|---|---|---|
| Piston | 3 | 7 | 2 |
| Connecting rod | 16.5 | 2 | 2 |
| Crank rod | 4.488 | 2 | 2 |

Material for piston was chosen as aluminum, steel for connecting rod and crank rod. The mechanism was simulated at 5000 rpm and a force of 18000.62 N (46.8 bar peak pressure) was applied on the cylinder head for every cycle. Figure-19 represents the ADAMS simulation setup.

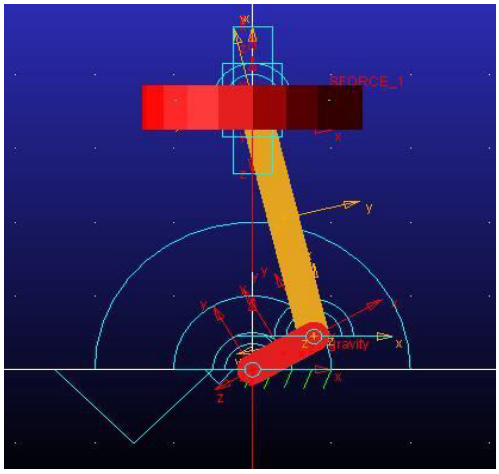


**Figure-19.** Crank slider mechanism simulation setup.

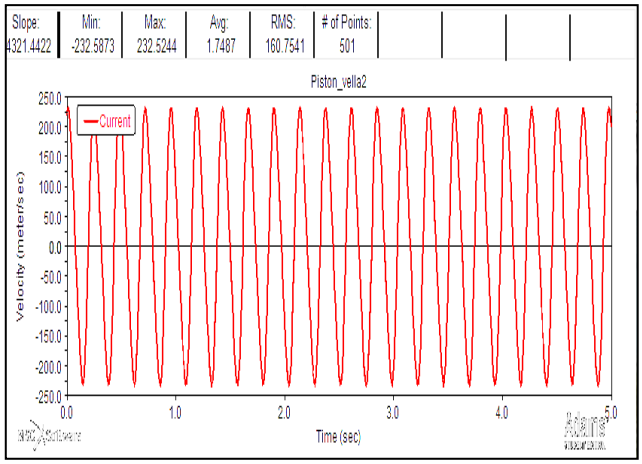


**Figure-20.** Velocity graph for inverted crank slider mechanism.

The plots of velocity and acceleration versus time were unaffected by the change in gravity direction. Figure-20 and Figure-21 represent the velocity versus time graph and acceleration versus time graph respectively.

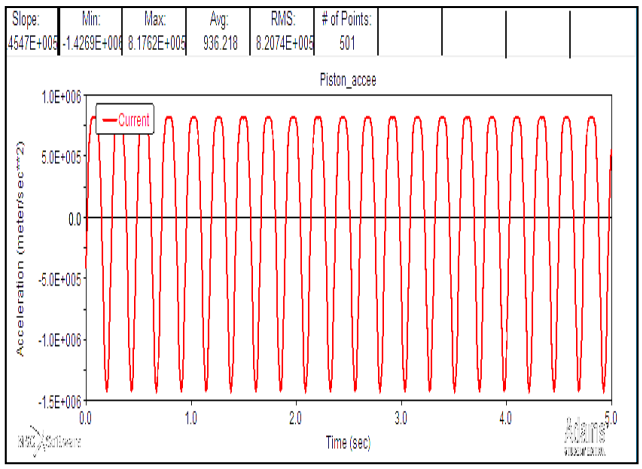


**Figure-21.** Acceleration graph inverted crank slider mechanism.

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## GRAVITY EFFECTS ON FUEL INJECTION AND AIR FLOW

Bottom induction of fresh air is chosen for the following reasons. The intake manifold heats up the incoming air which is at a lower temperature. The temperature of exhaust gases is also very high. As a natural phenomenon of hot gases moves towards higher region of atmosphere where the density is lesser. To avoid any possibility backflow of incoming fresh air and mixing of fresh air with burnt gases in the combustion chamber, the gases are allowed to flow in the direction that is facilitated naturally.

Figure-22 shows temperature distribution inside the cylinder.

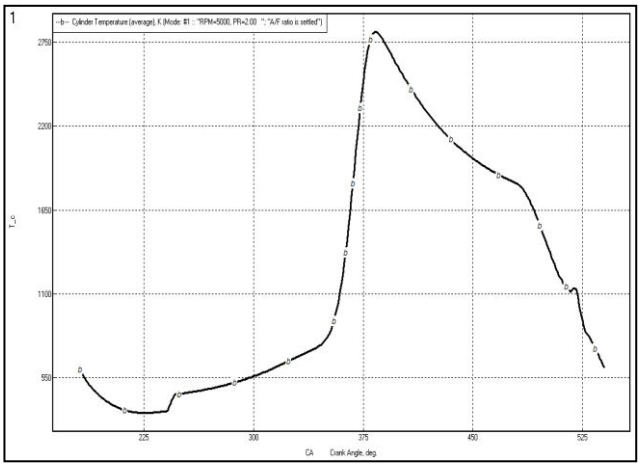


**Figure-22.** Temperature distribution inside the cylinder.

The peak temperature of the cylinder is 2545 (Figure-23) degree celsius and 46.8 bar. The density of air inside the cylinder will be lesser than 0.5780 kg/m$^3$.The average temperature at crankcase wall was 60 degree celsius. Assuming this temperature and ambient pressure of 1 bar to exist in the intake manifold, the density of fresh air will be higher than 1.060 kg/m$^3$ during the intake process, the incoming charge is usually cooler than the walls and the flow velocities are high. [2].This gradient will force the fresh charge into the cylinder to push out the burnt gases.

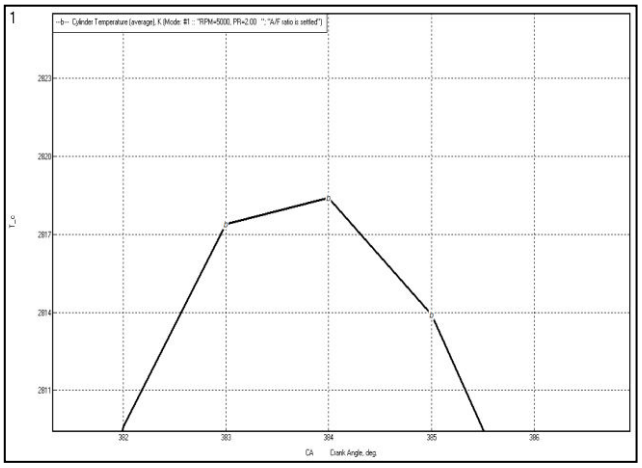


**Figure-23.** Peak temperature occurrence.

The temperature of exhaust manifold is 727 degree Celsius and corresponding pressure of air is 0.6 bar at the time of opening of exhaust ports. As the gases of reduced density from combustion chamber start moving out through the exhaust port, the cylinder pressure at 0.85 bar prevents the backflow into the combustion chamber from the exhaust manifold. This process is supported opening of inlet valves that permits flow of the fresh air which is relatively at a higher density from the intake manifold of pressure 1 bar. This will avoid all possibilities of backflow of burnt gases back into the combustion chamber.

The changes in velocity due to effect of gravity was examined for entry of fresh air into the cylinder and injection of fuel. Rate at which air was supplied was 0.114166 kg/sec for 0.003 seconds (5000 RPM and 90 degree inlet).This ensures supply of required 3.425x10$^{-4}$ kg of air. Atmospheric pressure was maintained at exhaust port.

Figure-24 and Figure-25 show the velocity contour for air flow along and against the direction of gravity respectively. The contours show that there is a small change in inlet velocity. Air having very less density (1.20 Kg/m$^3$ at 15 degree Celsius and 1 bar pressure) and gravity do not have a major impact.

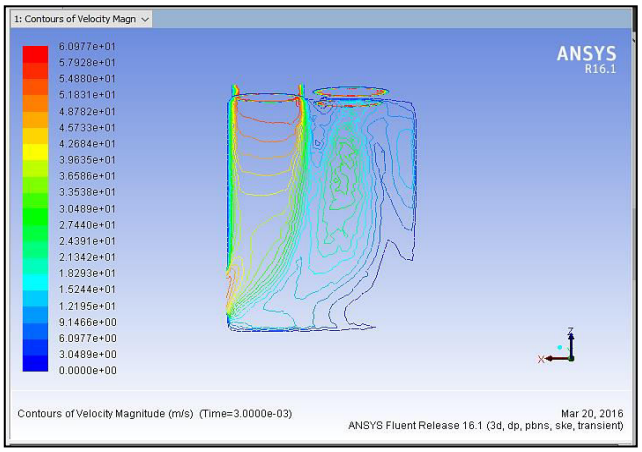


**Figure-24.** Velocity contour for air flow along gravity.

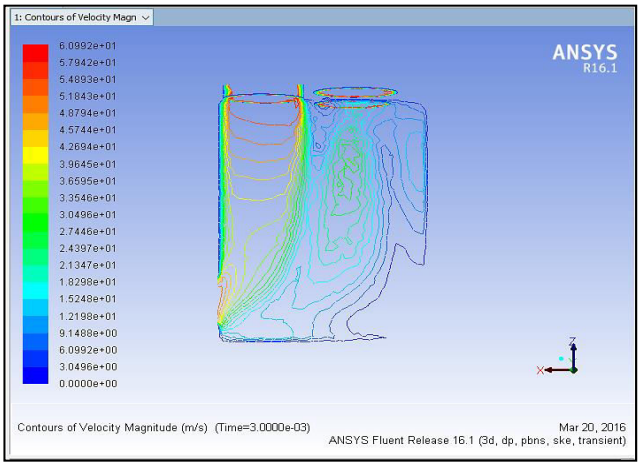


**Figure-25.** Velocity contour for engine placed against gravity.

The direction of fuel injection was decided from studies related to fuel injection and its effect on engine operation. There is an urgent need for the reduction of liquid fuel impingement on the piston top, because a liquid fuel film vaporizes more slowly than airborne droplets, resulting in high hydrocarbon emissions [10]. For stratified cold operation, the spray impingement on piston may yield to film formation, large hydrocarbons and soot exhaust emissions. Moreover, liquid film may get enough heat from the piston-wall to evaporate, and then decreases the air cooling effect and the compression ratio [11]. Mie scattering images show the liquid exiting the injector probe as a stream and directly impacting the piston top. Schlieren imaging was used to show the fuel vaporizing off the piston top late in the expansion stroke and during the exhaust stroke. Emissions tests showed that the presence of liquid fuel on in-cylinder surfaces increase the engine-out hydrocarbon emissions [12]. Spray/wall interaction has a significantly influence on the mixture formation process in gasoline direct injection (GDI) engines. Moreover, the fuel wall film and the resulting delayed evaporation of the liquid fuel are the main sources of soot formation in the internal combustion engines [13]. Thus, if the fuel is not fully vaporized and properly mixed with the air in the engine's cylinders during the combustion process, a part of this fuel may go out of the cylinders as unburned hydrocarbons. The carbon deposits can also cause cold start and drivability problems as the engine warms up because they can actually act as a sponge by momentarily absorbing some of the fuel that is need for proper combustion. Hence, in order to improve vaporization, the fuel stream has to be broken into finer droplets. This is achieved by distributing the fuel over a greater area inside the cylinder.

The direction of fuel injection was chosen with an objective to enhance distribution inside the cylinder by enhancing vaporization and reducing the magnitude of deposition over piston head. Injection against the direction of gravity showed the characteristics satisfying the objectives. The fuel has to be injected precisely at 3.5 cm from the BDC to ensure sealed combustion chamber by avoiding scavenging. The flow of fuel into the cylinder, without the presence of air was analyzed for $6.6 \times 10^{-4}$ seconds at a rate of 0.035 kg/sec which equals 0.02346 kg of fuel getting injected in 20 degrees at constant 200 bar pressure in both the cases. Figure-26 and Figure-27 show the velocity contour for fuel flow along and against the direction of gravity respectively.

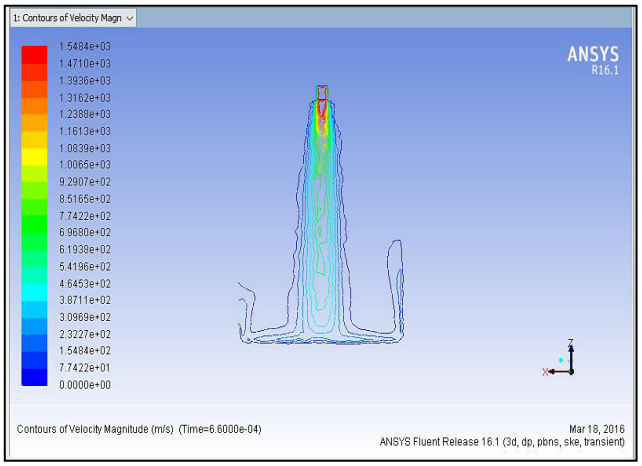


**Figure-26.** Velocity contour for fuel injection along gravity.

There was 12 m/s decrease in velocity (density of gasoline being 720 $Kg/m^3$) due to injection against the direction of gravity and the contours show that by injecting fuel against gravity enhances distribution inside the cylinder.

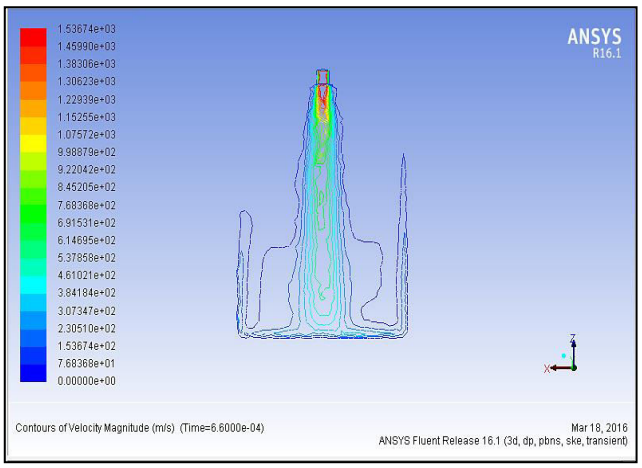


**Figure-27.** Velocity contour for fuel injection against gravity.

## RESULTS AND DISCUSSIONS

Following are the inferences to the mentioned objectives.

- Is fuel injector mandatory according to this idea?

Yes, Short circuiting is avoided only when the fuel is injected in a sealed combustion chamber to provide injection at the exact timing in which fuel injector is mandatory.

- Will sufficient amount of air be injected through the valves in the stipulated time?
- Will forced induction be necessary to achieve the working of the cycle?

Yes, with the manifold design same as that of reference engine 90 degree of intake stroke and forced induction at 1.1 bar with 3 intake valves of diameter 2.66cm is sufficient to send in the required amount of air. However, forced induction need not be mandatorily as the mass of charge inducted case of natural aspiration 3.60 x $10^{-4}$ kg (Required is 3.65 x $10^{-4}$ kg).Modification of intake can achieve the required parameters with natural aspiration.

- Has the cylinder pressure dropped below the inlet pressure during inlet valve opening?
- Is it feasible to send the burnt gases out of the cylinder within the stipulated time?

Yes, the 120 degree of exhaust timing with exhaust port dimensions of 2 cm height and 20 cm width causes the cylinder pressure to drop (Figure-5.3 and Figure-5.5) below the inlet pressure (Figure-6.5 and Figure-6.7). Also, it is capable of sending out mass greater than incoming mass within the stipulated duration.

- Will there be a mixing of exhaust gases with inlet fresh charge?

Through bottom induction for inlet and injection of fuel into sealed combustion chamber the possibilities of mixing of exhaust gases with inlet fresh charge has been avoided.

**ACKNOWLEDGEMENTS**

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