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# MODELING AND PARAMETRIC STUDIES OF HYDROGEN FUELLED MULTICYLINDER S. I. ENGINE CONSIDERING WITH THE EFFECT OF EQUIVALENCE RATIO USING ORDINARY DIFFERENTIAL EQUATIONS

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## ABSTRACT

In more than 100 years of evolution of the automobile, the times we live in today - at the start of a new century - may ultimately take the form of a true “sequel to the beginning” in a larger story of transportation history. After all, even after so many dramatic technological breakthroughs in the 20th century, much of what took place underneath the sheet metal of global automobiles was relatively incremental in nature. The benefits of using hydrogen are thus simple and clear - used in power generation or as a vehicle fuel, it contains nothing that pollutes and so all emissions are dependent on the way in which it is combusted. Widespread use of hydrogen could therefore have a significant impact on urban pollution. Fossil and nuclear fuel reserves are becoming increasingly limited, and the world's energy future will have to include several renewable alternatives to these failing resources. A promising possibility is to exploit the energy potential of the most plentiful element in the known universe — **HYDROGEN**.

The two main motivating reasons for development of hydrogen fuelled I.C. engine and to build a necessary infrastructure are

- Hydrogen, insight of its unlimited supply potential, will be key fuel in future sustainable energy systems that will rely on renewable energy resources.
- The extraordinarily clean combustion properties along with zero emissions have minimal environmental impacts.

In this paper, all the four basic processes taking place in an S.I. engine are analyzed and the values of pressure and temperature at every 2° of crank rotation are found out with the aid of certain assumptions. The model involves good deal of calculations and iterations and hence, it is coded in ‘c’.

It also explains the results obtained from simulation and discussion related with the results. It compares the predicted results of simulation with ideal Otto cycle. It is manifested that the ideal Otto cycle is ineffective in simulating combustion in a S.I. engine. The mass fraction burned commencing at the end of ignition lag calculated using Vibe function, gives quite good correlation with analytical results. The volumetric efficiency is 89%, which shows the role of friction losses in the intake system of the engine.

**KEYWORDS:** Computer Simulation, Mathematical Model, Delayed Entry Technique, Hydrogen Fuel

## INTRODUCTION

Internal Combustion Engines are those engines in which combustion of fuels takes place inside the engine and hence the chemical energy is converted in to thermal energy, which is further converted into mechanical work.

The present acute shortage of conventional fuels has necessitated the need for alternate fuel research. Hydrogen, which can be produced from natural gas or water, is proved to be a practical and potential alternate fuel for the I.C. Engine. The replacement of hydrocarbons by Hydrogen in automotive vehicles is expected to results in a considerable reduction in environmental pollution, since the harmful emission of unburned hydrocarbons and oxides of nitrogen are either avoided or minimized. With Hydrogen as a fuel, the engine exhaust is free from carbon monoxide and hydrocarbon emission, except very small quantities, which may be due to the combustion of lubricating oil. Further it does not contain sulfur, lead compounds or smoke and is virtually odorless. When Hydrogen-air combustion takes place in an I. C. engine cylinder, the only product of combustion are water vapor and oxides of nitrogen and the engine will be pollution free.

It has been proved that the higher thermal efficiency of Hydrogen engine can offset the higher production cost. With only minor modifications, the conventional diesel cycle engine can be operated efficiently using Hydrogen as fuel with atmospheric air supplying the necessary oxygen.

## PROPERTIES OF HYDROGEN

Table 1 shows that main combustion properties of Hydrogen provide its use as an IC engine fuel. A low fuel conversion rate is problem with gaseous-fueled engines run with high amounts of excess air. The low quenching distance of Hydrogen offers improvement in this matter. Hydrogen flames can easily penetrate into difficult chamber zones and reach the un burnt mixtures than that of fossil fuels. Optimized Hydrogen engines can be run at higher compression ratio than that with unleaded gasoline. It makes Hydrogen powered engines 15-25 % more efficient than gasoline engines.

**Table 1: Properties of Hydrogen**

| Description | Hydrogen |
|---|---|
| Laminar flame speed | 1.96 m/sec |
| Theoretical flame Temperature | 2140 $^oC$ |
| Minimum ignition energy | 0.02 MJ |
| Quenching distance | 0.6 mm |
| Normalized flame emmisivity | 1 |
| Normal Boiling Point | 20.27 K |
| Auto ignition temperature | 858 K |
| Burning velocity | 265 to 325 cm/sec |

## LITERATURE SHOWCASE

Beauties of Hydrogen were recognized as early as in 1820. In 1820, W. Cecil [1] read a paper before Cambridge philosophical society on “The Application of Hydrogen gas to produce a motive power in Machinery”.

Then after an elapse of century,. Ricardo [1] published in the “Report of the Empire Motor Fuel Committee” a very instructive paper on experiments carried out with Hydrogen and air used as a promoter with Petrol and Kerosene. He noticed that with a rich mixture pained by backfire, Ennen [2] in Germany, in 1933 dealt successfully with the backfire problem by injecting Hydrogen directly in to the cylinder, but the knocking persisted. King[3] made valuable contribution on the subject of pre-ignition and combustion knock in Hydrogen engine. He found that any particulate matter provides hot spot for pre-ignition and the combustion knock is an inherent property of near stoichiometric Hydrogen-air mixture due to the extremely high flame velocity.

The major conclusions derived from the available literature are as follows:

- Any existing engine can be converted to Hydrogen fuelled engine with minor modifications.
- The part load & thermal efficiencies of $H_2$ fuelled engine are higher than gasoline air engine.
- Hydrogen induction technique is easier to adopt as compared to Hydrogen injection technique.
- Emission levels of $H_2$ - air engine are far less than that of gasoline – air engine if equivalence ratio is not exceeded 0.6 in $H_2$ - air engine (i.e. Lean operation)
- Equivalence ratio more than 0.6 results in back fire problems. If $H_2$ – air engine has to be operated in the range of 0.6 to 1.0-equivalence ratio, we have to go for EGR or water induction or delay entry technique to achieve backfire free operation and lower NOx emission.
- The reported optimum spark advance for $H_2$ – air engine lies in between $7^o$ to $12^o$ BDC.
- The optimum compression ratio lies in between 8 to 12 for $H_2$ – air engine.

## AIM OF THE PRESENT WORK

The aim of the present work is to model All Processes in Hydrogen fueled Engine and by that improve fuel economy and govern power capacity of the engine. And also to describe the safe and backfire free $H_2$ fuelled engine using Delayed Entry Technique.

## DEVELOPMENT OF MATHAMATICAL MODEL

Internal combustion engines are the main power plants of the transportation systems and are responsible for a substantial fraction of fuel consumption. The scarcity of oil resources and the ever increasing standards on air pollution and emissions have dictated a need for improved, more efficient and less polluting internal combustion engine. Improvements on engine design have been achieved by traditional methods based on extensive experience. The advent of computers and the possibilities of performing numerical experiments may provide a new way of designing I. C. Engines. In fact, a stronger interaction between engine modelers, Designers and experimenters may result in improved engine designs in the not-to-distant future.

The modeling of reciprocating or rotary engine is a multidisciplinary subject that involves chemical thermodynamics, fluid mechanics, turbulence, heat transfer, combustion and numerical methods.

## STARTING OF SUCTION PROCESS

On the starting of suction valve will be closed, so there will not be any flow of air. Thus pressure drop will not occur due to the friction. Hence at the starting of suction process the pressure in the pipeline will be equal to atmospheric pressure. At the starting of suction process we will assume that at the end of exhaust process, pressure will be 1.05 bars and the temperature will be 553 Kelvin. Also the error involved in these assumptions will be nullified due to the iterations of whole cycle. Thus, we know the condition outside and inside the cylinder. So we are able to start the simulation.

First Valve lift of inlet valves are measured at every 5° of interval, with the help of Dial gauge and angle measurement device(pro-circle). These data are interpolated at every 2° crank interval using MATLAB for simulation.

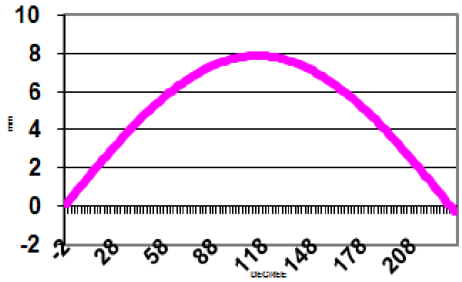


**Figure 1**

Now Injecting Hydrogen into the engine cylinder is an inherently difficult task and considerable engine modification is required to convert existing l engine to use Hydrogen. The present method consists of inducting of Hydrogen along with air in to engine cylinder to use Hydrogen with the help of delayed entry valve. This method has the virtue of simplicity and flexibility since exiting engine is easily converted to work on this principle.

## SIMULATION OF THE SUCTION PROCESS

The simulation of the process starts by assuming for the small rotation (dθ) of the crank. The volume inside the cylinder after the small rotation (dθ) can be calculated as follows:

$$V_{new} = V_c\left[1+\tfrac{1}{2}(r_c-1)\left(n+1-\cos\theta-\sqrt{n^2-\sin^2\theta}\right)\right]$$

Flow area $A_f$ is calculated from valve lift $L_v$.

The value of valve lift for different crank angle is given in appendix-1

$$L_{\lim} = \frac{(d_{os} - d_{is})}{2\sin\phi\cos\phi}$$

For the first stage of poppet valve lift where,

$$L_v \le L_{\lim} \Rightarrow$$

$$A_f = \pi L_v \times \cos\phi\left(d_{is} + L_v\sin\phi\cos\phi\right)$$

For the second stage of poppet valve lift where,

$$L_v > L_{\lim} \Rightarrow$$

$$A_f = \pi\left(\frac{d_{os}+d_{is}}{2}\right)\sqrt{\left(L_v - \frac{d_{os}-d_{is}}{2}\tan\phi\right)^2 + \left(\frac{d_{os}-d_{is}}{2}\right)^2}$$

The mass dm that is coming inside the cylinder can be calculated by formula given by Heywood

$$dm = \frac{C_d \times A_{\theta+d\theta}\times P_{atm}}{\sqrt{R_{charge}\times T_{atm}}}\times\left[\frac{P_{\theta+d\theta}}{P_{atm}}\right]^{1/\gamma}\times\left[\frac{2\gamma}{\gamma-1}\left\{1-\left(\frac{P_{\theta+d\theta}}{P_{atm}}\right)^{\gamma-1/\gamma}\right\}\right]^{1/2}\times dt$$

Mass of the exhaust gases inside the cylinder can be calculated as follows

$$mexh = \frac{Patm \times Vc}{Rxh \times Texh}$$

Hence total mass at (θ + dθ)° can be obtained by

$$M_{new} = M_{exh} + dm$$

$Cp_{new}$ can be calculated as

$$C_{P_{new}} = \left(\frac{dm}{M_{new}}\right) \times C_{P_{ch}} + \left(\frac{M_{exh}}{M_{new}}\right) \times C_{P_{exh}}$$

$R_{new}$ can be calculated as

$$R_{new} = \left(\frac{dm}{M_{new}}\right) \times R_{ch} + \left(\frac{M_{exh}}{M_{new}}\right) \times R_{xh}$$

Applying energy balance between the mass that was present at θ°, the mass 'dm' that is coming from outside into the cylinder and the new mass at (θ + dθ)°, we get new temperature

$$T_{new} = \frac{\left[\left(M_{exh} \times Cp_{exh} \times T_{exh}\right) + \left(dm \times Cp_{ch} \times T_{atm}\right)\right]}{\left[M_{new} \times Cp_{new}\right]} \quad (3.8)$$

Also, new pressure at θ + dθ

$$P_{new} = \frac{R_{new} \times m_{new} \times T_{new}}{V_{new}}$$

Considering heat transfer losses using Woschni's heat transfer formula

$$h = 0.82 \times B^{-0.2} \times p^{0.8} \times Wmv^{0.8} \times T^{-0.53}$$

where $W_{mv} = C_1 \times C_m$

($C_1$ =6.18 for gas exchange process)

p is pressure in MPa

The surface area at which heat transfer takes place can be obtained by

$$A_{surface} = \frac{\pi}{2}B^2 + \frac{\pi BL}{2}\left[n + 1 - \cos\theta - \left(n^2 - \sin^2\theta\right)^{1/2}\right]$$

The temperature of the cylinder wall is obtained by

$$T_{wall} = (426 - 0.388 \times \theta)$$

Heat transfer through walls can be obtained by

$$Q_{conv} = h \times A_{surface} \times (T_{wall} - T_{new}) \times dt$$

Corresponding temperature with heat transfer consideration

$$T_{new\_h} = T_{new} + \left[ \frac{Q}{(M_{new} \times Cp_{new})} \right]$$

The temperature so obtained above is put back in equation for heat transfer coefficient and is proceeded further until it falls under the desired accuracy. The value of temperature ($T_{new\_h}$) is used to calculate the corrected value of pressure by taking into account of Cp, R and heat transfer.

Thus, the corrected pressure is obtained as follows:

$$P_{new\_h} = \frac{T_{new\_h} \times R_{new} \times M_{new}}{V_{new}}$$

## SIMULATION OF COMPRESSION PROCESS

In four- stroke combustion engine compression process is of fundamental importance and requires great understanding of the micro processes taking place. An effort is made to analyze the compression process and evaluated the properties of the mixture so as to compare the variations of properties at each stage. In compression process, the mass inducted during the previous process of suction is enclosed in the volume of the cylinders. This air-fuel charge mixture is to be compressed by action of the piston moving from the outer dead center to the inner dead center. Work is supplied to the system in the compression but recovered in later process of combustion and expansion

As a result of work done on the mixture the internal energy of the mixture is increased. The pressure and temperature of the mixture increase slowly at first, than steadily due to the progressive work of compressing the mixture. Consequently the specific heat capacity of the mixture also increases due to the temperature change. At the end of compression at 356º crank angle the introduction of electric spark inside the cylinder takes place. Compression is continued up to that. Simulation of the compression process is treated in this work up to 344º of crank movement.

- **Homogeneity:** It is assumed that the charged is mixed homogenously with the residual gases like water vapor and other constituents.
- **Range:** The process of effective compression starts at 234º of crank rotation and completes at 356º of crank angle. The beginning of compression is governed by the establishment of the pressure in the cylinder, which occurs at 234º of crank angle. The end of compression is governed by the initiation of effective combustion.

## STARTING OF COMPRESSION PROCESS

The simulation of the process starts by assuming for the small rotation (dθ) of the crank.

The volume inside the cylinder after the small rotation (dθ) can be calculated using equation.

We find out the corresponding temperature using the relation

$$\frac{T_{\theta+d\theta}}{T_\theta} = \left(\frac{V_{\theta+d\theta}}{V_\theta}\right)^{1-\gamma}$$

We also find out the corresponding pressure using the relation

$$\frac{P_{\theta+d\theta}}{P_\theta} = \left(\frac{V_{\theta+d\theta}}{V_\theta}\right)^{-\gamma}$$

Calculate the total fuel trapped inside the cylinder during suction process, Mass of fuel vaporized and loss of heat from the cylinder.

Calculate the density of the gas

$\rho_{\theta + d\theta} = P_{\theta + d\theta} / R_{charge} * T_{\theta + d\theta}$

Using the value of $T_\theta$ and $T_{\theta + d\theta}$ the average value of $T_{mean}$ can be calculated

$T_{mean} = (T_\theta + T_{\theta + d\theta}) / 2$

Find out the viscosity of gas, thermal conductivity of the gas, Reynold no. and Nusselt no. using the relation given as below,

$\mu_{cylinder} = 7.457 * 10^{-6} + 4.1547 * 10^{-8} * T_\theta - 7.4793 * 10^{-12} * T_\theta^{2}$

$C_k = 6.1944 * 10^{-3} + 7.3814 * 10^{-5} * T_\theta - 1.2491 * 10^{-8} * T_\theta^{2}$

Reynold no. (Re) = $\rho_{\theta + d\theta} * V_p * D_{cylinder} / \mu_{cylinder}$

Nusselt no. (Nu) = $0.49 * Re^{0.7}$

Find out the convective heat transfer coefficient

$C_h = C_k * Nu / D_{cylinder}$

Determine the convectional heat transfer loss

$Q_{conv} = C_h * A_{surface} * (T_{wall} - T_{\theta + d\theta}) * dt$

heat loss

$Q = Q_{conv}$

Corrected temperature

$T_{\theta + d\theta\_h} = Q / m_c * C_v + T_{\theta + d\theta}$

At this corrected temperature, calculate $C_p$, $C_v$ and $\gamma$.

Substitute the above value of $\gamma$ in equation and repeat the whole cycle till the required accuracy is achieved.

## SIMULATION OF COMBUSTION PROCESS

Mass fraction burned, $B_\theta$

$$B_\theta = 1 - e^{-a\left[\frac{\theta-\theta_s}{\Delta\theta}\right]^{m+1}} \quad (3.24$$

Where, a = 7.62, m= 1.69

$dB = B_{\theta+d\theta} - B_\theta$ Mass burned at θ+dθ,

$mb_{\theta+d\theta} = mb_1 + dB*m_c$

Mass unburned at θ+dθ,

$mu_{\theta+d\theta} = mu_1 - dB*m_c$ Gas constant R at θ,

$R_\theta = B_\theta R_b + (1-B_\theta)R_u$ Specific heat at constant volume at θ,

$Cv_\theta = B_\theta Cv_b + (1-B_\theta)Cv_u$ Specific heat at constant pressure at θ,

$Cp_\theta = R_\theta + Cv_\theta$ Ratio of specific heat at constant pressure to Specific heat at constant volume at θ,

$\gamma_\theta = Cp_\theta / Cv_\theta$ Pressure rise due to change in cylinder volume,

$\Delta P_v = P_\theta (v_\theta / v_{\theta+d\theta})^\gamma$

Pressure rise due to combustion,

$P_c = (m_c*R*\Delta T)/\Delta v$

Where,

$\Delta T = m_c*C_v/m_f*C_{vf}$

Δv is a differential volume during combustion process.

Total pressure rise during combustion = Pressure rise due to change in cylinder volume + Pressure rise due to combustion

$P_{\theta+d\theta} = \Delta P_v + \Delta P_c$ $P_{\theta+d\theta} = P_\theta (v_\theta / v_{\theta+d\theta})^\gamma + dB*P_c$

Net heat transfer,

$\delta Q_R - \delta Q_L = [(P_{\theta+d\theta} V_{\theta+d\theta} - P_\theta V_\theta)/\gamma - 1] + [(P_\theta + P_{\theta+d\theta})/2*(V_{\theta+d\theta} - V_\theta)]$

Using mass fraction burned approach, the overall behavior of the entire cylinder space may be found simply as below, but for that the time interval should be sufficiently short. So the combustion process in the present simulation is simulated for every 1° crank interval.

$T_{\theta+d\theta} = [(\delta Q_R - \delta Q_L) + m_c* Cv_\theta*T_\theta - P_\theta (V_{\theta+d\theta} - V_\theta)] / m_c*Cv_\theta$

Corrected Pressure,

$P_{corr} = m_c*R_\theta*T_{\theta+d\theta} / V_{\theta+d\theta}$

Now, a simple solution for the properties within the two zones is only possible if some assumption is made regarding inter-zone heat transfer. €Without such an assumption, it is not possible to determine the individual volumes within each zone, and hence the individual zone temperature cannot be determined.

It is reasonable to assume that the process in the unburned zone is adiabatic, because the unburned zone is gaining as heat from the burned zone as it is losing to the surface of the cylinder and the piston crown.

$P_{\theta+d\theta}/P_{\theta} = [(mu_{\theta+d\theta}/mu_{\theta})(Vu_{\theta}/Vu_{\theta+d\theta})]^{\gamma u}$

In the above formula only volume of the unburned zone $Vu_{\theta+d\theta}$ is unknown.

Volume of the burned zone at $\theta+d\theta$,

$Vb_{\theta+d\theta} = V_{\theta+d\theta} - Vu_{\theta+d\theta}$

As thermodynamic equation of state must be satisfied for each of them, burn zone Temperature at $\theta+d\theta$,

$Tb_{\theta+d\theta} = P_{\theta+d\theta} * Vb_{\theta+d\theta} / mb_{\theta+d\theta} * Rb$

unburn charge Temperature at $\theta+d\theta$

$Tu_{\theta+d\theta} = P_{\theta+d\theta} * Vu_{\theta+d\theta} / mu_{\theta+d\theta} * Ru$

At this temperature, i. e at $Tu_{\theta+d\theta}$ and $Tb_{\theta+d\theta}$ calculate $C_pu$ and $C_pb$.

$C_vb = C_pb - Rb$

$C_vu = C_pu - Ru$

$\gamma u = C_pu / C_vu$.

The value of $\gamma u$ is substituted in equation (which gives the value of pressure at $(\theta + d\theta)$. That value of $P_{\theta+d\theta}$ is resubstituted in equation to calculate the net heat loss and the cycle is repeated till the desired accuracy is achieved.

## SIMULATION OF EXANSION & EXHAUST PROCESSES

The expansion and exhaust processes are merely the reverse of the compression and suction process respectively

## RESULTS & DISCUSSIONS OF THE MODEL

### Results of Suction Process

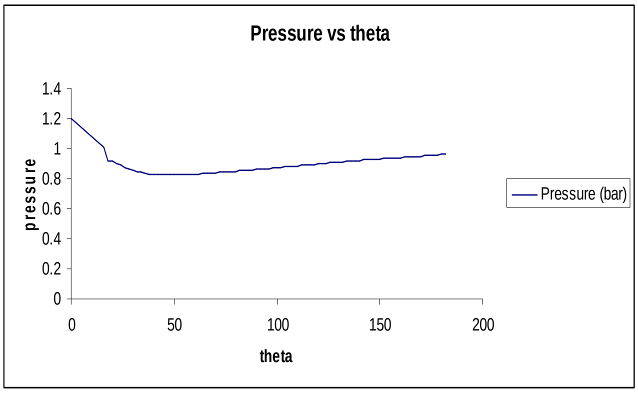


**Figure 2: Pressure v/s Theta for Suction Process**

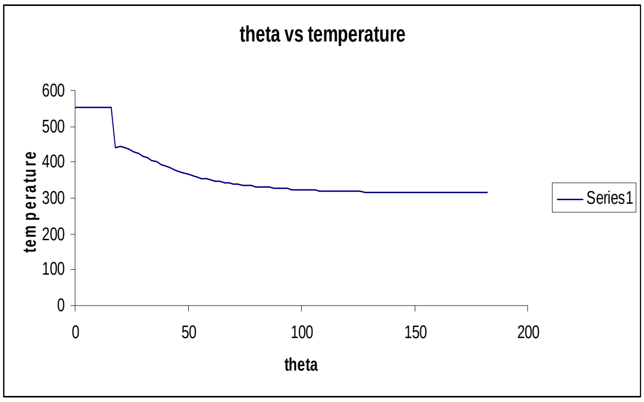


**Figure 3: Temperature v/s Theta for Suction Process**

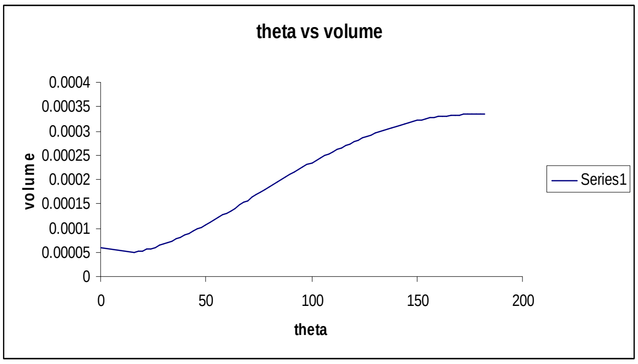


**Figure 4: Pressure v/s Volume for Suction Process**

The nature of P-theta curve is quite interesting. Initially a strong decrease in cylinder pressure is observed. his happens due to starvation of mass flow due to restricted valve intake area during the initially stage of valve lift. However, thereafter a gradual pressure building is observed due to increased availability of mass flow with higher range.

The temperature obviously will reduce with increased availability of mass flow with increasing crank angle. The m-θ curve clearly shows gradual rise in mass flow during initial valve lift and thereby explain the trend of P-θ curve too.

Thus, the basic results of suction process are as per logical trend observed in actual I.C. engines and this validates the model used in present case

**Results of the Compression Process**

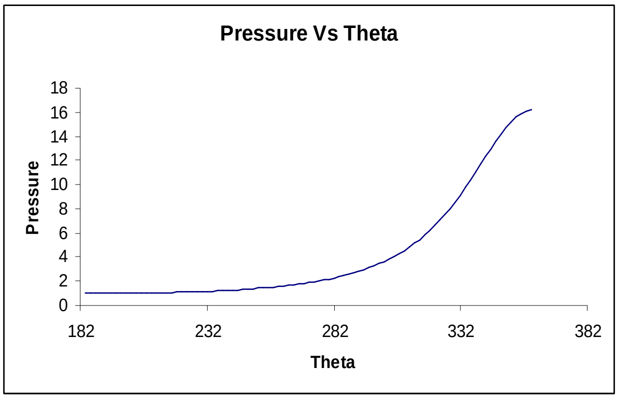


**Figure 5: Pressure v/s Theta for Compression Process**

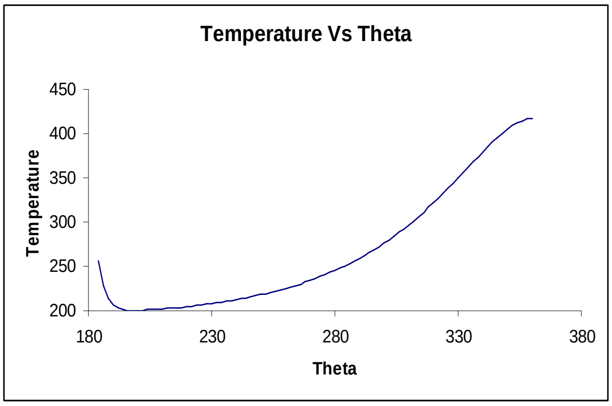


**Figure 6: Temperature v/s Theta for Compression Process**

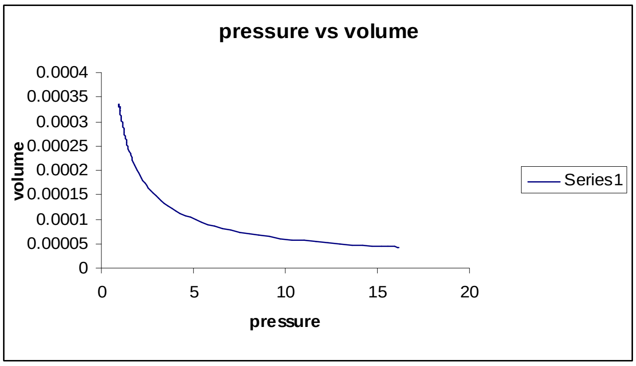


**Figure 7: Pressure v/s Theta for Compression Process**

It is seen from P-θ curve that initially there is a gradual rise in pressure, this happens because the charge gets trapped within the sealed cylinder, experiences compression. Compression process starts at 236° crank angle when the cylinder pressure equals to the atmospheric pressure.

As the pressure inside the sealed cylinder increases, the temperature will also increase in the same manner. The nature of T-θ curve is similar to P-θ curve. The temperature soars to 396.875 K at the end of compression process.

During compression process, heat loss due to fuel vaporization is also considered. It is seen from the calculation that it is worth to employ vaporization loss to the convective heat loss.

**Results of the Combustion Process**

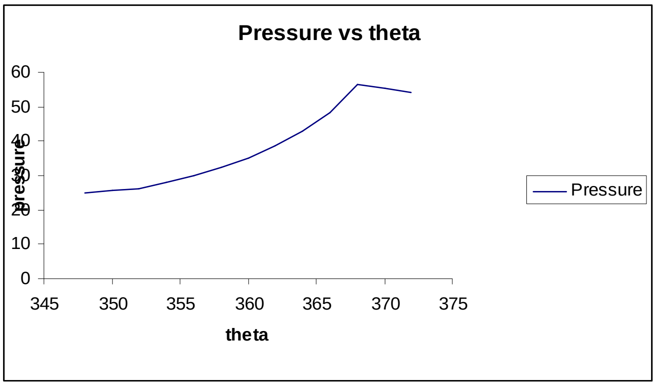


**Figure 8: Pressure v/s Theta for Combustion Process**

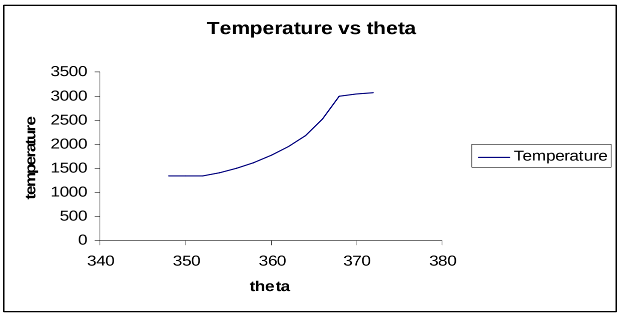


**Figure 9: Temperature v/s Theta for Combustion Process**

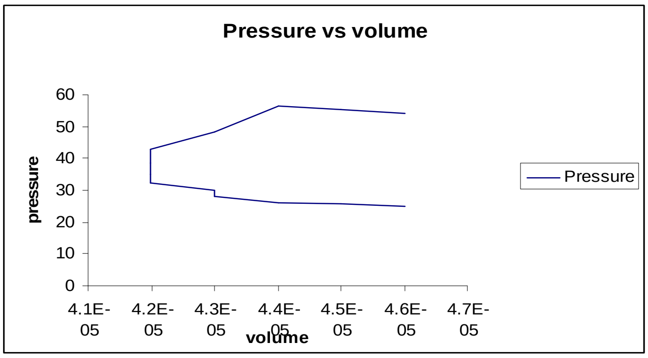


**Figure 10: Pressure v/s Volume for Combustion Process**

The p-θ curve shows that there is rapid rise in pressure. The pressure reaches to its peak value of 47.50 bar at 14 °atdc. This is because, at the conclusion of compression stroke combustion takes place. It is observed from the mass fraction burned curve that nearly 75% charge gets burned at 14 °atdc. After that the pressure begins to fall due to increase in cylinder volume.

The nature of T- θ curve is quite interesting. The temperature increases initially with increase in pressure. It reaches to its peak value of 2704.069 K at 21 °atdc. The heat release continues as the charge keeps on burning even after 14 °atdc when the peak pressure is reached. The rate of heat addition under these circumstances is more than the heat losses. As a result the temperature continues to rise and reaches to its peak value 21 °atdc. the mass fraction burned curve, evaluated using vibe function. Graph shows the mass fraction burned characteristic determined from the analysis of cylinder pressure diagram from a conventional automobile naturally aspirated spark ignition engine. Both graph give comparable trend and thereby validates the combustion model used in present case.

**Results of the Expansion Process**

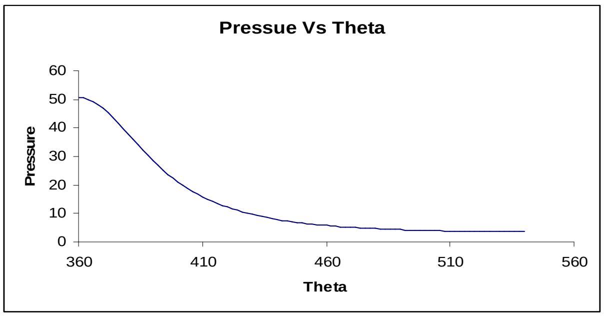


**Figure 11: Pressure v/s Theta for Expansion Process**

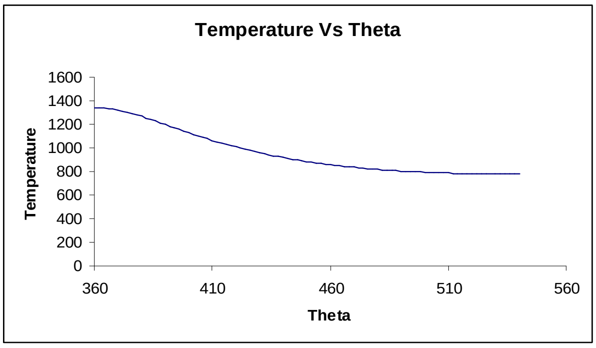


**Figure 12: Temperature v/s Theta for Expansion Process**

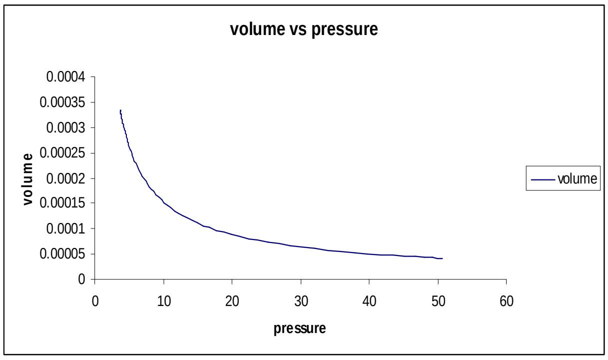


**Figure 13: Pressure v/s Volume for Expansion Process**

Above Graphs show the trend of cylinder pressure variation with increasing crank angle. Combustion products with very high temperature get expanded due to increase in cylinder volume, which in turn reduce the pressure inside the cylinder drastically. At the end of combustion process the pressure inside the cylinder is 35.023 bar. Pressure falls to 6.22 bar. This gives higher energy extraction.

The T-θ curve shows that the temperature continues to decrease with increasing crank angle. Temperature reaches to 1883.2382 K at the end of expansion process from 2616.5024 K at the conclusion of effective combustion process.

It is observed from the calculation that the value of convective heat transfer coefficient is 1710.58 W/m$^2$K, which is much higher compare to the value of 374.2725 W/m$^2$K of compression process. This is obviously because of the effect of heat addition, which enhances the temperature level of gas which is responsible for such a high value of heat transfer coefficient

**Results of the Exhaust Process**

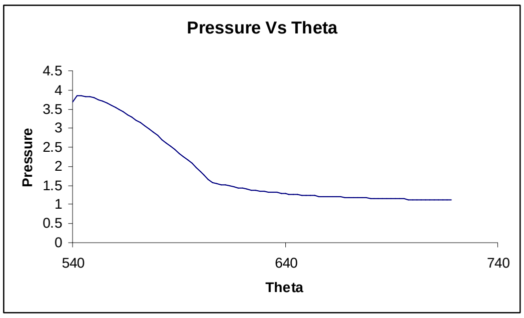


**Figure 14: Pressure v/s Theta for Exhaust Process**

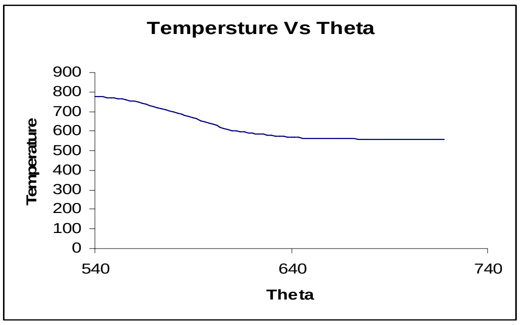


**Figure 15: Temperature v/s Theta for Exhaust Process**

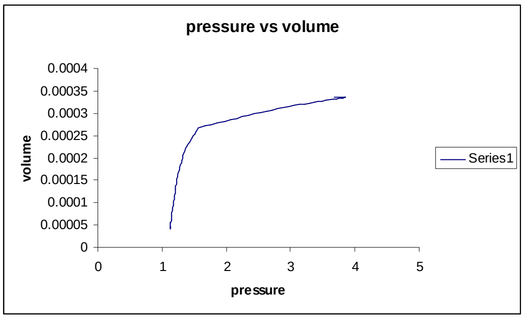


**Figure 16: Pressure v/s Volume for Exhaust Process**

Above Graphs give the experimentally measured value of valve lift at different crank angle. By comparing, it is clear that the maximum valve lift of exhaust valve is less compared to maximum valve lift of inlet valve. Higher valve lift of inlet valve gives higher volumetric efficiency especially at high speed. Figure represents the valve flow areas obtained by Heywood [22] and Gordan's[2] approach. As Gordan's [2] approach does not take stem area in the consideration, at higher valve lifts, effective flow area is equal to the port area.

The P-θ curve shows that there is rapid fall in cylinder pressure. It falls to 1.0762 bar at 640° crank angle. This happens because burned mass is forced to leave the cylinder space due to very high pressure differential. After 660° crank angle slow pressure building inside the cylinder is observed due to throttling effect.

The temperature obviously will reduce with high temperature burned mass leaving to the atmosphere with increasing crank angle. It is observed from the T-θ curve that, like pressure, there is not any rise of temperature during the later stage of exhaust process. At the end of exhaust process, i.e at 720° crank angle temperature reaches to 406.4757 K. The value of pressure and temperature obtained at the end of exhaust process is to be substituted again in the initial assumption of the analysis of the suction process and the whole calculation needs to be repeated till the required accuracy is achieved.

**Overall Performance of Hydrogen Fuel Engine**

**Table 2: Overall Performance Parameters**

| Parameters | Discrete Approach |
|---|---|
| $P_0$ | 1.01325 bar |
| $T_0$ | 553 K |
| $P_1$ | 0.96158 bar |
| $T_1$ | 313.6781 K |
| $P_2$ | 16.1652 bar |

**Table 2: Contd.,**

| | |
|---|---|
| $T_2$ | 417.6876 K |
| $P_3$ | 50.5909bar |
| $T_3$ | 1341.17 K |
| $P_4$ | 3.6814 bar |
| $T_4$ | 775.8594 K |
| Efficiency (η) | 48.60 % |
| Indicated Power kw | 96.3274 kW |
| Brake Power kw | 77.87 kW |

**Effect of Compression Ratio on Engine Parameters**

**Comparison of Overall Performance Parameters at Different Compression Ratio**

**Table 3**

| Compression Ratio | 8:1 | 9:1 | 10:1 | 12:1 |
|---|---|---|---|---|
| Pmep (bar) | 4.16 | 3.166 | 2.248 | 0.6543 |
| I.P (KW) | 127.28 | 96.69 | 69.95 | 19.88 |
| B.P (KW) | 101.82 | 77.35 | 54.92 | 15.25 |
| Efficiency (ή) | 47.75 % | 48.91 % | 46.11 % | 47.234 % |

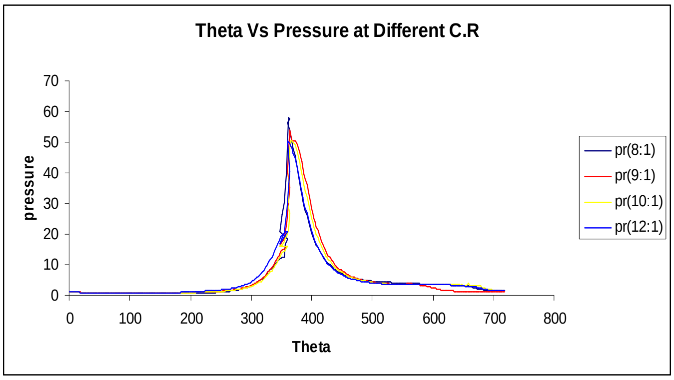


**Figure 17: Pressure v/s Theta for Different Compression Ratio**

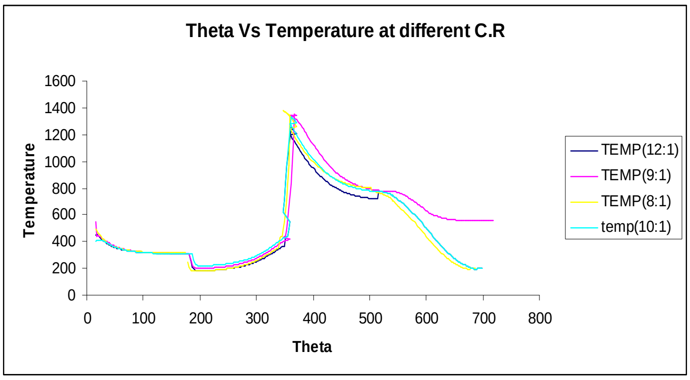


**Figure 18: Temperature v/s Theta for Different Compression Ratio**

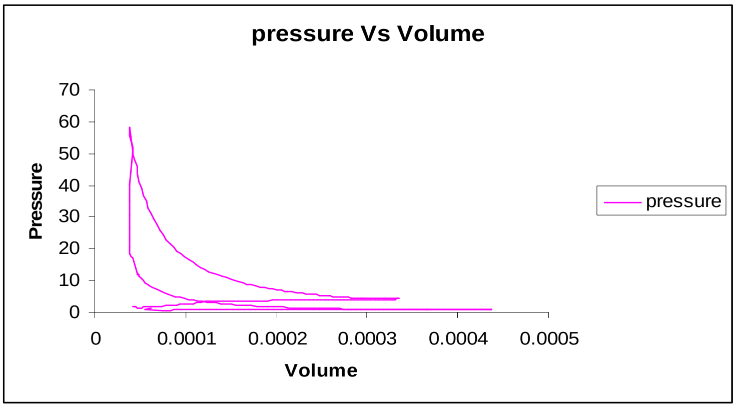


**Figure 19: Pressure v/s Volume for Compression Ratio 8:1**

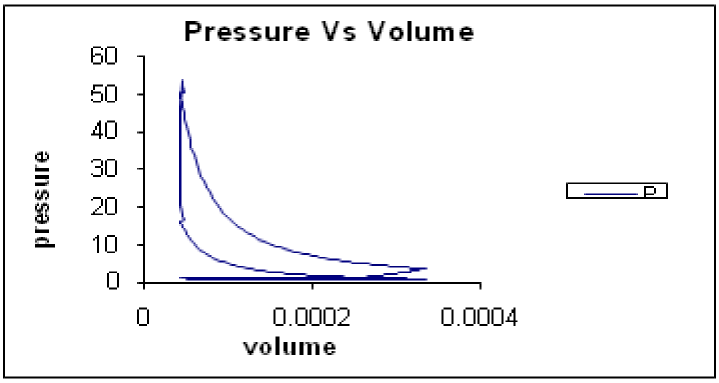


**Figure 20: Pressure v/s Volume for Compression Ratio 9:1**

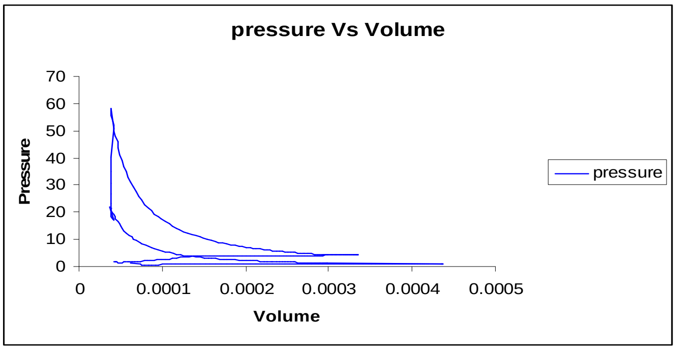


**Figure 21: Pressure v/s Volume for Compression Ratio 10:1**

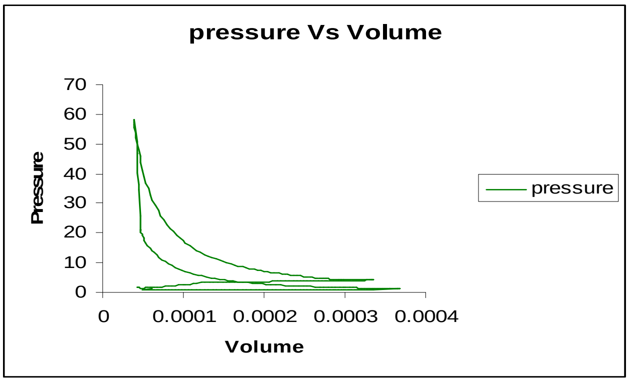


**Figure 22: Pressure v/s Volumes for Compression Ratio 12:1**

**Effects of Equivalent Ratio**

**Mass Fraction of Reactant at Different Equivalence Ratios**

**Table 4**

| Φ | $H_{2r}$ | $O_{2r}$ | $N_{2r}$ | Total Reactant[Kg/Mole] |
|---|---|---|---|---|
| 0.2 | 0.005792 | 0.2316 | 0.7625 | 345.33 |
| 0.4 | 0.01151 | 0.2303 | 0.7581 | 173.66 |
| 0.6 | 0.0171 | 0.229 | 0.7538 | 116.43 |
| 0.8 | 0.0227 | 0.2277 | 0.7459 | 87.8333 |
| 1 | 0.0283 | 0.2264 | 0.7452 | 70.6666 |
| 1.2 | 0.0337 | 0.2251 | 0.741 | 71.0666 |
| 1.4 | 0.03917 | 0.2238 | 0.7369 | 71.466 |

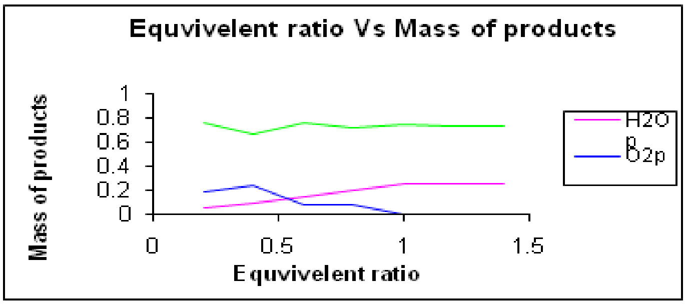


**Figure 23: Equivalence Ratio vs Mass of Reactant**

**Mass Fraction of Product at Different Equivalence Ratios**

**Table 5**

| Φ | $H_2O_p$ | $O_{2p}$ | $N_{2p}$ | Total Product[Kg/mole] |
|---|---|---|---|---|
| 0.2 | 0.05212 | 0.1853 | 0.7625 | 345.33 |
| 0.4 | 0.091 | 0.2429 | 0.6659 | 173.66 |
| 0.6 | 0.1481 | 0.0877 | 0.7641 | 116.43 |
| 0.8 | 0.196 | 0.0871 | 0.7168 | 87.8333 |
| 1 | 0.2547 | 0 | 0.7452 | 70.6666 |
| 1.2 | 0.2532 | 0 | 0.741 | 71.0666 |
| 1.4 | 0.2518 | 0 | 0.7369 | 71.466 |

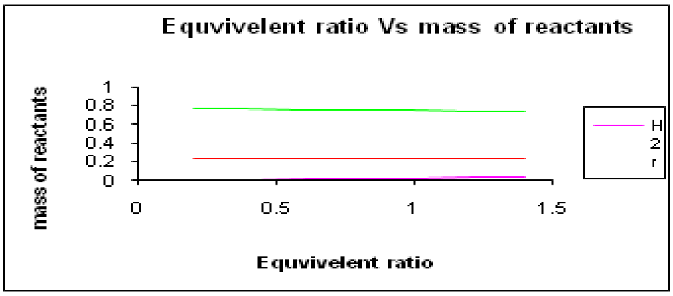


**Figure 24: Equivalence Ratio Vs Mass of Products**

## CONCLUSIONS

Hydrogen's potential as a vehicular fuel is a subject that has recently aroused great attention. Hydrogen is a unique fuel with unmatched properties, which makes it a deal fuel. Based on the extensive literature review on Hydrogen fuelled engine, simulation exercise carried out during the course of this work and developmental work pertaining to delayed entry valve

- The simulation of suction pressure using discrete approach suggests that the pressure 30º after TDC and 30 º before BDC are of the order of 0.856 bar and 0.931 bar, respectively. This clearly indicates that the entry of hydrogen during this period will certainly offer back fire free operation of the engine
- The pressure and temperature at the end of the compression process with discrete approach is 16.165 bar and 417.68 K respectively as against the pressure and temperature of 22.180 bar and 729.69 K with ideal Otto cycle analysis
- The peak pressure and temperature obtained with discrete approach is of the order of 50.590 bar and 1341.17 K respectively.

Thus, the present work offers new approaches for simulation of H2-air engine with delayed entry technique and advocates for the use of rotary type delayed entry valve for backfire free operation of multi-cylinder engine

## SCOPE OF THE FUTURE WORK

The simulation code written in $C^{++}$ should be written in a software code such as Microsoft™ Quick Basic for the Macintosh, Microsoft Visual Basic for the PC, or True Basic- a cross platform language for either the PC or Macintosh. These three software code permits a highly visual data input procedure, with the cylinders or the valves or the ducting of the engine appearing as moving entities on the computer screen. It shows the variations of pressure, Volume & Gas flow rate that takes place during the engine cycle in a pictorial form. It allows the viewers imagine the unimaginable. It forms such pictorial information that a designer conceives of future improvements.

Wave action model should be adopted to calculate the unsteady flow in the manifold pipes.

The coefficient of discharge, that is assumed constant, should be evaluated.

Other than two zone combustion model, one may look for multi-zone model which can give more accurate results.

Future work will involve validation against experiment and comparison.

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