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Simulation of Suction & Compression Process with Delayed Entry Technique Using Discrete Approach for Hydrogen Fuelled Engine

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ABSTRACT -- The rapidly increasing worldwide demand for energy and the progressive depletion of fossil fuels has led to an intensive research for alternative fuels which can be produced on a renewable basis. Hydrogen in the form of energy will almost certainly be one of the most important energy components of the early next century. Hydrogen is a clean burning and easily transportable fuel. Most of the pollution problems posed by fossil fuels at present would practically disappear with Hydrogen since steam is the main product of its combustion. This Paper deals with the modeling of Suction and Compression Processes for Hydrogen Fuelled S.I.Engine and also describes the safe and backfire free Delayed entry Technique. A four stroke, Multicylinder, Naturally aspirated, Spark ignition engine, water cooled engine has been used to carrying out of investigations of Suction Process. The Hydrogen is entered in the cylinder with the help of Delayed Entry Valve. This work discusses the insight of suction process because during this process only air and Hydrogen enters in to cylinder, which after combustion provides power. Simulation is the process of designing a model of a real system and conduction experiment with it, for the purpose of understanding the behavior of the design. The advent of computers and the possibilities of performing numerical experiments may provide new way of designing S.I.Engine. In fact stronger interaction between Engine Modelers, Designers and Experimenters may results in improved engine design in the not-to-distant future. A computer Programme is developed for analysis of suction and Compression processes. The parameter considered in computation includes engine speed, compression ratio, ignition timing, fuel-air ratio and heat transfer. The results of computational exercise are discussed in the paper.

KEYWORDS: Computer simulation, Mathematical model, Suction Process, 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 vapour 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 unburnt 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 ^{o}C
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:

(i). Any existing engine can be converted to Hydrogen fuelled engine with minor modifications.
(ii). The part load & thermal efficiencies of H_2 fuelled engine are higher than gasoline air engine.
(iii). Hydrogen induction technique is easier to adopt as compared to Hydrogen injection technique.
(iv). 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)
(v). 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.
(vi). The reported optimum spark advance for H_2 – air engine lies in between 7^o to 12^o BDC.
(vii). 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 suction and Compression 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

During this stroke, charge (Air+Hydrogen fuel) enters into the cylinder. It mixes with exhaust gases that are left out after exhaust stroke. The entry of fresh charge takes place only when the pressure inside the cylinder falls below atmospheric even when the inlet valve remains opened. The variation of pressure and temperature in this suction process is obtained by considering heat transfer and friction at each crank degree rotation in discrete manner. The heat transfer coefficient and frictional losses are evaluated based on average gas velocity inside the cylinder which is obtained using momentum balance and not through available procedures [62, 63].

The modeling of the process starts by assuming that for the small rotation (dθ) of the crank, the temperature inside the cylinder does not vary, which is corrected later on by iteration.

Considering the equation of state ,

$$\frac{P_\theta \times V_\theta}{T_\theta} = \frac{P_{\theta+d\theta} \times V_{\theta+d\theta}}{T_{\theta+d\theta}} \text{ [For Ideal Gas]} \qquad (4.1)$$

Where, volume at any crank angle is given by [64]:

$$V_\theta = Fc \times r[(n+1) - \sqrt{(n^2 - \sin^2\theta)n} - \cos\theta + \frac{1}{n}] \quad (4.2)$$

From the above relation, the values of V_θ and $V_{\theta+d\theta}$ can be calculated. Hence, $P_{\theta+d\theta}$ can be calculated.

Therefore, $d_p = P_{atm} - P_{\theta+d\theta}$ (4.3)

Hence, from Bernoulli's equation [64], the velocity at which the charge is coming inside the cylinder can be calculated [64] as:

$$U = C_d \times \sqrt{(2 \times \frac{d_p}{D_{charge}})} \quad (4.4)$$

Where, D_{charge} is the density of the charge that can be calculated by relation.

$$D_{charge} = \frac{P_{atm}}{(R_{charge} \times T_{atm})} \quad (4.5)$$

Therefore, the mass 'd_m' that is coming inside can be calculated by:

$$d_m = D_{charge} \times U \times A_{planer} \times C_d \times d_t \quad (4.6)$$

A_{planer} is calculated as per the lift 'h' of the inlet valve, which is given by Blair [64]. The mass of exhaust that is present at θ^0 of the crank can be calculated by

$$m_\theta = P_{\exp} \times V_0 \, (R_{\exp} \times T_{\exp}) \quad (4.7)$$

Hence the total mass at $(\theta + d\theta)^o$ can be obtained by

$$m_{\theta+d\theta} = m_\theta + d_m \quad (4.8)$$

Applying the Energy balance between the mass that was present at $\theta^{o,}$ the mass 'd_m' that is coming from outside into the cylinder and the new mass at $(\theta + d\theta)^o$.

$$T_{\theta+d\theta} = \frac{\left[\left(m_\theta \times C_{p_\theta} \times T_\theta\right) + \left(d_m \times C_{p_{charge}} \times T_{atm}\right)\right]}{\left[\left(m_{\theta+d\theta} \times C_{p_{\theta+d\theta}}\right)\right]} \quad (4.9)$$

The surface area at which heat transfer takes place can be obtained by [64]

$$A_{surface} = \left(\frac{\pi \times \beta^2}{2} + \left(\frac{B \times stroke\ Length}{2}\right) \times \theta_{rad}\right) \quad (4.10)$$

The temperature of the cylinder walls is varying along the length. Assuming this variation to be linear based on experimental measurements [64],the instantaneous wall temperature at any crank angle θ is evaluated as [6]:

$$T_{wall} = (0.388 \times \theta) + 423\ K \quad (4.11)$$

Initially we store the approximation of pressure obtained from equation 4.1 to a new variable 'P' for taking heat transfer losses into account. Taking Momentum balance,

$$V_{mean} = \frac{\left[\left(m_\theta \times V_{piston}\right) + \left(d_m \times U\right)\right]}{\left(m_{\theta+d\theta}\right)} \quad (4.12)$$

Using this mean velocity, we calculate the heat transfer coefficient using the relation [63]

$$H = 0.13 \times B^{0.12} \times P_{\theta+d\theta}^{-0.5} \times T_{\theta+d\theta}^{-0.5} \times V_{mean}^{0.8} \times 10^{-3} \qquad (4.13)$$

Frictional losses can be calculated by [64]

$$Fq = \frac{\left[2 \times 8.5 \times 10^{-5} \times V_{mean}^{2} \times \pi \times B/2 \times Stroke\ Length \times d_T\right]}{Clearance} \qquad (4.14)$$

Heat transfer through the walls can be obtained by:

$$q = H \times A_{surface} \times \left(T_{wall} - T_{\theta+d\theta}\right) \times dt \qquad (4.15)$$

Net heat transfer losses,

$$Q = \left(q - Fq\right) \qquad (4.16)$$

Corresponding temperature,

$$T_{\theta+d\theta,h} = \left[\frac{Q}{\left(m_{\theta+d\theta} \times C_{v_\theta} \times 10^{3}\right)}\right] + T_{\theta+d\theta} \qquad (4.17)$$

The temperature so obtained above is substituted back in equation 4.14 and is proceeded further until it falls under the desired accuracy. Hence, the so corrected value of temperature ($T_{\theta+d\theta\_h}$) is used to calculate the corrected value of pressure for the obtained approximate pressure from equation (4.1) by taking into account of Cp, R, Heat transfer & Frictional losses.

$$P_{\theta+d\theta,h} = \frac{\left[T_{\theta+d\theta,h} \times R_{new} \times m_{\theta+d\theta}\right]}{V_{\theta+d\theta} \times 10^{-5}} \qquad (4.18)$$

This so obtained corrected value of pressure is substituted back into equation (4.5) and the process continues until the desired accuracy is achieved and this is used for next step. The entire calculations are carried out for 1^{o} crank rotation till the closing of suction valve or pressure approaches just near to atmosphere.

COMPRESSION PROCESS

During this stroke both the inlet valve and exhaust valves are in the closed position. The mixture, which fills the entire cylinder volume, is now compressed into the clearance volume at the end of the compression stroke. The analysis of the process is carried out by finding the values of pressure and temperature for every 1° of crank rotation. Initially, it is assumed that the compression process is isentropic i.e. γ = 1.40. Hence, for this value of γ, the corresponding temperature will be,

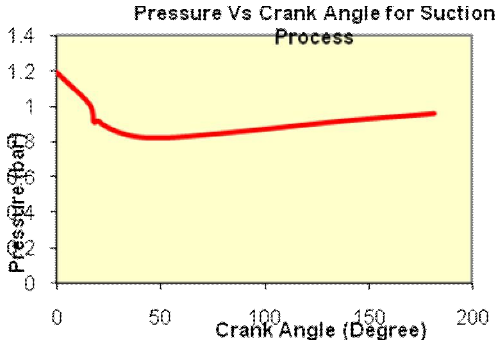
$$T_\theta \times V_\theta^{\gamma-1} = T_{\theta+d\theta} \times V_{\theta+d\theta}^{\gamma-1} \qquad (4.19)$$

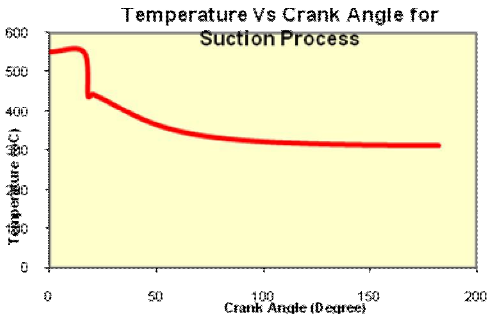
Where, V_θ and $V_{\theta+d\theta}$ are obtained from equation and $T_{\theta+d\theta}$ is evaluated. This temperature is further corrected by considering wall heat transfer and fluid friction through equation 4.10 to 4.16 and the $T_{\theta+d\theta}$ is evaluated as:

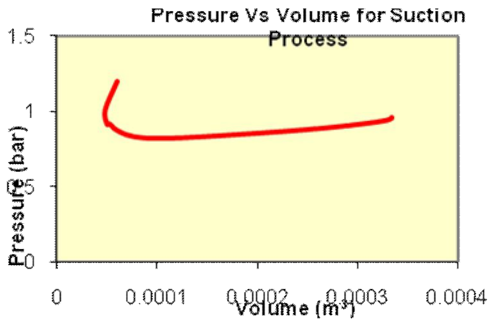
$$T_{\theta+d\theta_{new\,h}} = \left[\frac{Q}{m_{new} \times C_{v\exp} \times 10^{3}}\right] + T_{\theta+d\theta} \qquad (4.20)$$

Using the values of T_θ and $T_{\theta+d\theta\ new,h}$ the average value of temperature T_{mean} can be calculated. Further using the value of T_{mean} the value of C_{vmea} and revised value of γ may be evaluated as per the procedure outlined. This new value of γ is again used in eq. to calculate $T_{\theta+d\theta}$ and this is continued until the final and the pre assumed values do not vary significantly. Now using the final value of $T_{\theta+d\theta}$ the value of $P_{\theta+d\theta}$ can be calculated through eq.

RESULTS & DISCUSSION OF THE MODEL **<u>RESULTS OF SUCTION ROCESS</u>**





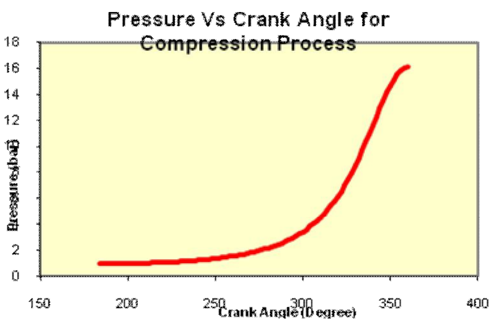


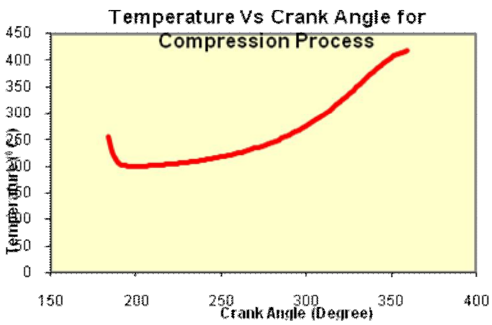
The nature of P-theta curve is quite interesting. Initially a strong decrease in cylinder pressure is observed. his happens due to starvation of mass flow due to restricted valve intake area during the initially stage of valve lift. However, thereafter a gradual pressure building is observed due to increased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

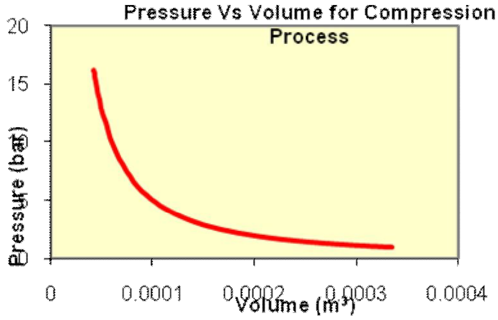
The temperature obviously will reduce with increased availability of mass flow with increasing crank angle. The m-θ curve clearly shows gradual rise in mass flow during initial valve lift and thereby explain the trend of P-θ curve too.

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







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.

CONCLUSION

- From the results of simulation, it concludes that the trend of pressure and temperature with increasing crank angle is quite logical to the actual S.I. engine.
- During suction process, pressure falls to 0.67 bar at 58 ˚atdc. Initial fall of pressure in the suction stroke is also observed by Blair and Lumley. According to Winterbone and pearson[13,14], if the wave action theory is included in the intake system, it utilizes the ramming effect of the traveling compression wave in the inlet system, that will improve the trend of pressure curve.
- The calculation of pressure losses in the intake system of an engine carried out as per ASHRE, gives 89% volumetric efficiency, comparable to the real engine. According to Zhao and Winterbone [13,14] the use of loss coefficients improves the evaluation of the pressures and mass flow in the manifold.
- The authors feel that the Delayed Entry Technique is designed for the backfire free operation will become an essential feature of future H_2 fuelled engines. It is also felt that this valve can be used on any gas engine for utmost safe operation.

SCOPE FOR 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..

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