A Numerical Study of Expansion and Exhaust Process Variability in Hydrogen Fuelled Engine with the new concept of Differential Technique using Real Gas Equations

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ABSTRACT -- The rapidly increasing world wide 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 Expansion and Exhaust Processes for Hydrogen Fuelled S.I.Engine. A four stroke, Multicylinder, Naturally aspirated, Spark ignition engine, water cooled engine has been used to carry out investigations of Expansion and Exhaust Processes. The Hydrogen is entered in the cylinder with the help of Delayed Entry Valve. This work discusses the insight of Exhaust process. Simulation is the process of designing a model of a real system and conducting 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 a 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 Expansion and Exhaust 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, Exhaust Process, 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 into 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

<table>
<thead>
<tr>
<th>Description</th>
<th>Hydrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar flame speed</td>
<td>1.96 m/sec</td>
</tr>
<tr>
<td>Theoretical flame Temperature</td>
<td>2140 °C</td>
</tr>
<tr>
<td>Minimum ignition energy</td>
<td>0.02 MJ</td>
</tr>
<tr>
<td>Quenching distance</td>
<td>0.6 mm</td>
</tr>
<tr>
<td>Normalized flame emissivity</td>
<td>1</td>
</tr>
<tr>
<td>Normal Boiling Point</td>
<td>20.27 K</td>
</tr>
<tr>
<td>Auto ignition temperature</td>
<td>858 K</td>
</tr>
<tr>
<td>Burning velocity</td>
<td>265 to 325 cm/sec</td>
</tr>
</tbody>
</table>

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 H2 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 H2 - air engine are far less than that of gasoline – air engine if equivalence ratio is not exceeded 0.6 in H2 - air engine (i.e. Lean operation)
(v). Equivalence ratio more than 0.6 results in back fire problems. If H2 – 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 H2 – air engine lies in between 7° to 12° BDC.
(vii). The optimum compression ratio lies in between 8 to 12 for H2 – air engine.

AIM OF THE PRESENT WORK

The aim of the present work is to model Expansion and Exhaust Processes in Hydrogen fueled Engine and by that improve fuel economy and govern power capacity of the engine.

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.

LITERATURE RIVIEW

Recent development of S.I. engines, aiming to the higher power, better fuel economy, lower air pollution and better drivability have much increased the importance of computer simulation in engine research and development.
In the various papers reviewed below, the following major areas should be noted:

1. Model for gas exhaust systems.
2. Model for flow through the valves.
3. Model for heat transfer

**Model for the gas flow in the Exhaust system**

The application of one-dimensional non-steady compressible flow equations to the intake and exhaust system of a reciprocating internal combustion engine has been well known. There are several methods which enable the equations to be solved. The characteristics theory reduces the set of equations into a set of simultaneous ordinary differential equations and the solution of these equations gives the wave and gas path line characteristics. Riemann originally formulated this theory in 1885. De Haller\[17\] used a graphical method to solve the set of simultaneous differential equations for the gas flow in an engine exhaust system. Later, Jenny\[18\] extended the graphical solution to include pipe wall heat transfer, friction, entropy gradient and gradual pipe cross-sectional area change. This type of graphical solutions has been extensively used in the calculation of gas flow in the reciprocating I.C. engines. With the availability of high speed digital computer, Benson, Garg and Woollatt[13] proposed a numerical scheme to solve the non-steady flow equations with the method of characteristics (MOC). The pressure losses have been calculated considering differential approach with the help of pressure wave theory. The pressure drop can be calculated by the equation given below.

\[
\frac{dP_f}{dt} = \frac{2C_f \rho_s C_s^3 dt}{d}
\]

Where \(C_f = \frac{0.0791}{R_t^{0.25}}\)

\(R_t = \frac{\rho_s C_s d}{\mu}\)

\(\mu = 7.457 \times 10^{-6} + 4.1547 \times 10^{-8}T - \cdot 7.4793 \times 10^{-12}\)

\(dt = 0.99 \frac{L}{\alpha_s}\)

\(\alpha_s = a_0 \sqrt{\gamma + 1 \frac{P}{P_0} \frac{\gamma - 1}{\gamma}}\)

\(a_0 = \sqrt{\gamma RT}\)

**Model for the flow through the valve**

Blair presents a unique empirical approach for the design and dimensioning of the valving and the ducting of a high performance, naturally aspirated, spark ignition automotive engine so as to attain the required performance levels at a given engine speed.

**The fundamental geometry of the cylinder head**

The cylinder head of a four stroke engine contains intake and exhaust poppet valves which link the cylinder to its manifold. This is shown in fig.A. The number of intake and exhaust valves is \(n_i\) and \(n_e\), respectively. The port area at each of these valves is \(A_{ip}\) and \(A_{ep}\) respectively. The valves are connected to intake and exhaust manifold apertures of area \(A_{in}\) and \(A_{em}\), respectively. This gives rise to the concept of pipe (manifold) to port area ratio \(C_{mp}\), which are defined as follows:

\(C_{imp} = \frac{A_{im}}{n_i} \times \frac{A_{ip}}{A_{ip}} = \frac{A_{im}}{A_{ep}}\)

\(C_{emp} = \frac{A_{em}}{n_e} \times \frac{A_{ep}}{A_{ep}} = \frac{A_{em}}{A_{ep}}\)
According to Blair[10] these manifolds to port area ratios are critical for the performance of an engine. For this area ratio, as either an expansion or a contraction, controls the amplitude of any pressure wave created within the ducting by the cylinder state conditions, as the strength of any such unsteady gas flow tuning is a function of its pressure.

The throat or minimum port aperture area at any valving, $A_{pt}$, is shown in fig.B. Inflow or outflow at any valve passes through the valve curtain areas which are side areas of a frustum of a cone. However, at the highest valve lifts the minimum flow area may become that at the inner port where the diameter is dip.

The valve curtain area, $A_v$, for this particular geometry is often simplistically, and quite incorrectly, expressed as the side surface area of a cylinder of diameter $d_a$ and height $L$ as,

$$A_v = \pi d_a L$$

It is vital to calculate correctly the geometrical throat area of the restriction $A_s$. In fig.B, the valve curtain area at the throat, when the valve lift is $L$, is that which is represented by the frustum of a cone defined by the side length dimension $x$, the valve seat angle, $\Phi$, the inner or outer seat diameters, i.e. $d_{in}$ and $d_{out}$, and the radius, $r$, which depends on the amount of valve lift $L$. The side surface area of a frustum of a cone, i.e. $A_s$, is,

$$A_s = \pi[(d_{major} + d_{minor})/2] * x$$

Where $x$ is the sloping side and $d_{major}$ and $d_{minor}$ are its top and bottom diameters. This area $A_s$ is the maximum geometrical gas flow area through the seat of any one valve for flow to, or from, the port where that area is $A_{pt}$ or $A_{ept}$.

The dimension $x$ through which the gas flows has two values which are shown in fig.B. On the left, the lift is sufficiently small that the value $x$ is at right angle to the valve seat and, on the right, the valve lift has lifted beyond a lift limit $L_{lim}$ where the value $x$ is no longer normal to the valve seat at angle $\Phi$. By simple geometry, this limiting value of lift is given by,
\[ L_{\text{lim}} = \frac{(d_m - d_o)}{2\sin\Phi \cos\Phi} \]
\[ = \frac{(d_m - d_o)}{\sin 2\Phi} \]

For the first stage of poppet valve lift

Where, \( L \leq L_{\text{lim}} \)

Then the valve curtain area \( A_t \) is given from the value of \( x \) as,

\[ X = L \cos\Phi \]
\[ r = \frac{d_m}{2} + x \sin\Phi \]

in which case,

\[ A_t = \pi L \cos\Phi (d_m + L \sin\Phi \cos\Phi) \]

For the second stage of poppet valve lift where, \( L > L_{\text{lim}} \)

Then the valve curtain area \( A_t \) is given from the value of \( x \) as,

\[ X = \left\{ \frac{L - [d_m - d_o]/2 \tan\Phi}{1 + [d_m - d_o/2]^2} \right\}^{1/2} \]

Whence,

\[ A_t = \pi \left( \frac{d_m + d_o}{2} \right) \left( \frac{d_m - d_o}{2} \tan\Phi \right)^{1/2} \left( \frac{d_m - d_o}{2} \right)^{1/2} \]

If the seat angle is 45°, which is conventional, then \( \tan\Phi \) is unity and the above equation simplifies somewhat.

**Model for heat transfer**

For considering heat transfer, Woschni’s[20] equation was used by Shashikantha[24], which is based on the similarity law of steady turbulent heat transfer.

The formula is,

\[ h_c = 0.820 D^{0.2} P^{0.8} w_{\text{av}}^{0.8} T^{-0.53} \text{(kw/m}^2\text{K).} \]

According to Patterson [12], of the several expressions in the literature, two are of special interest. Nusselt [25] was among the first to formulate an expression for the heat transfer coefficient in an I.C.Engine. His expression based on experimental observation, included radiation as well as convective effects, & it is;

\[ h_c = 0.99 (P^4 T)^{1/2} (1+1.24 w) + 0.362 \left\{ \frac{(T/100)^4 - (T_w/100)^4}{T - T_w} \right\} \]

Where \( h \) in kcal/ m² hr K.

According to Anand[22] the expression for the heat transfer coefficient in an I. C. Engine is expressed as

\[ h_c = \frac{d \left( \frac{\lambda}{B} \right) \text{Re}^{0.7} + \left[ \frac{C}{T - T_c} \left( \frac{T}{100} \right)^4 - \left( \frac{T_w}{100} \right)^4 \right] }{1} \]

This expression is widely used for I. C. Engines. Many books on I. C. engines have referred this expression and we have also selected the same reference.

**Model for Specific properties**

According to Blair, enthalpy, internal energy and specific heat at constant pressure are the function of temperature. They are controlled by the simple formula.

\[ h = K_0 + K_1 T + 2K_3 T^2 + 3K_5 T^3 \text{ KJ/kgK.} \]
\[ U = K_0 + (K_1 - R)T + 2K_2 T^2 + 3K_3 T^3 \]
\[ C_p = K_1 + 2K_2 T + 3K_3 T^2 \frac{KJ}{Kg - k} \]
The differential form can be derived by simply differentiation of the equation with respect to temperature. The differential form is given below.

\[ \frac{dU_c}{dT} = K_1 - R + 4K_2T + 9K_3T^2 \]

**Model for Coefficient of discharge**

A fundamental experimental study was conducted by Blair[10] to visualize the effect of size of the engine ducting on the discharge coefficient of the cylinder porting aperture. It was found to have no significant influence. The general applicability of this conclusion in design and simulation study is debated. A study of the discharge coefficients of restrictions or throttles within engine ducting was carried out and the ensuing map was determined to be significantly dissimilar in profile to that of all published data on engine porting or valving.

\[ Cd = -23.543 + 60.686P - 51.04P^2 + 14.387P^3 \quad (1 < P < 1.4) \]
\[ Cd = 0.838 \quad (P \geq 1.4) \]

Where P = pressure ratio.

With all the properties of hydrogen, we must know the practical aspect while dealing with the hydrogen. This aspect has been described in detail further in this chapter.

For using hydrogen, we must know the different criteria for dealing with hydrogen. Further in this chapter, we will see the production, storage and transportation of hydrogen.

**Model for cylinder volume**

The figure shows the schematic diagram of the piston-cylinder arrangement of I. C. engines. The volume of the cylinder is changing continuously with the crank rotation. The volume inside the cylinder can be calculated from the geometry and the generalized equation derived by Blair is given below. The differential of volume and the piston speed can be calculated as below.

\[ V_\theta = V_\circ + \frac{\pi B^2}{4} (l + r) \left( \sqrt{l^2 - r^2 \sin^2 \theta} \right) \]
\[ \frac{dV}{d\theta} = \frac{\pi B^2}{4} \frac{r \sin \theta - r^2 \sin^2 \theta}{\sqrt{l^2 - r^2 \sin^2 \theta}} \]

**Model of Wall temperature**

The temperature of wall can be found from the empirical formulas suggested by authors As given below.

\[ T_{\text{wall}} = (423 - 0.388 * \theta) \]

**STARTING OF EXPANSION PROCESS**

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

As a result of work done by the mixture the internal energy of the mixture is decreased. The pressure and temperature of the mixture decrease slowly at first, than steadily due to the progressive work of expansion the mixture. Consequently the specific heat capacity of the mixture also decreases due to the temperature change.
**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 expansion starts at 364º of crank rotation and completes at 492º of crank angle. The beginning of expansion is governed by the establishment of the pressure in the cylinder, which occurs at 362º of crank angle. The end of expansion is governed by the completion of effective combustion.

### SIMULATION OF EXPANSION PROCESS

The approach is very simple suggested by Richard stone. It is merely the some modification of first law of thermodynamics. As we all know that first law of thermodynamics is

\[
dQ = dU + dW
\]

Now in the compression process, there will be no mass change nor do the heat generation involve. Thus in the compression process, the heat transfer will be only the heat loss from the walls. The heat loss from the wall can be found from the simple equation given below.

\[
dQ = hc * A * (T^4 - Tw^4)
\]

Here the ‘A’ i.e. the area term can be found from the simple geometry. The heat transfer coefficient can be found from the approaches suggested in the literature review. We have in our case have taken the equation given by Anand. The reason behind the selection is that many books on simulation on I. C. Engines have suggested it to use.

The temperature of the wall can be found from the equation given by Dr. S. A. Channiwala. The equations are given in the literature review. It is the equation of crank rotation. Thus the dQ term is only the function of temperature.

Now the second term is dU i.e. the internal energy difference. Now as it is mentioned earlier, the internal energy is the function of temperature only. Now the the last term is the work done term expressed as below

\[
dW = \left( \frac{P_2 + P_1}{2} \right) (V_2 - V_1)
\]

Now as we all know that the pressure is the function of temperature and volume. we know the volume of the cylinder at the each degree of crank rotations. Thus the only unknown variable is the temperature. Thus the all term in the first law of thermodynamic can be expressed in the two basic variable i.e. temperature and volume. And as we know the volume at each degree of the crank rotation, we left with the only variable and that is temperature. Now to find the temperature from the modified equation it is almost impossible to express in single side variable. Thus to find the temperature we used bisection method to find the root of the equation.

Now we have to consider real gas equation. There are many real gas equations but one which fits in our temperature range is Redlich Kwong real gas equation. This is given by,

\[
P = \frac{RT}{V_m - b} - \frac{a}{\sqrt[4]{TV_m(V_m + b)}}
\]

But as we have to consider the differential approach, the differential form of the same equation is,

\[
\frac{1}{p} \frac{dp}{d\theta} = \frac{1}{m} \frac{dm}{d\theta} (C 1) + \frac{1}{V} \frac{dV}{d\theta} (C 1) + \frac{1}{2T} \frac{dT}{d\theta} (C 2)
\]

\[
C1 = \frac{\left( \frac{2RT}{V_m^2} VM (VM + mb) \right) - \frac{am(VM - mb)^2(2VM + mb)}{m^2b^2}}{\left( \frac{2RT}{V_m^2} (VM + mb) - ma(VM + mb) \right) V^2 M^2 - m^2b^2}
\]

\[
C2 = \frac{\left( \frac{2RT}{V_m^2} VM(VM - mb)(VM + mb) + m^2d(VM + mb)(VM - mb) \right) \left( V^2 M^2 - m^2b^2 \right)}{\left( \frac{2RT}{V_m^2} VM(VM + mb) - m^2d(VM - mb) \right) V^2 M^2 - m^2b^2}
\]

Thus we are able to find pressure and temperature. These values will be again put in to the starting of the loop and after the iterative method we will get the pressure and temperature. The whole loop is to work over whole cycle.
STARTING OF EXHAUST PROCESS

At the starting of exhaust process we will assume that at the end of expansion process. These assumptions are commonly used in the most of the automobile books. 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 Exhaust 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.

SIMULATION OF THE EXHAUST PROCESS

During the flow due to friction the pressure drop occurs. The pressure drop due to friction can be calculated from the method described in previous title. So gas will flow from intake valve to cylinder. First the exhaust gas will discharge from the cylinder than the fresh charge will enter the cylinder. The volume of cylinder can be calculated by formula given below. The $dV/d\theta$ can also be getting from below equation.

$$V_0 = V_i + \frac{\pi}{4} B (l+r - \sqrt{l^2 - r^2 \sin^2 \theta - r \cos \theta}) \frac{dV}{d\theta} = \frac{\pi B^2}{4} \frac{r \sin \theta - r^2 \sin 2\theta}{\sqrt{l^2 - r^2 \sin^2 \theta}}$$

Mass calculation

Now In general, $m = \rho AV$

Now we can calculate the density and the speed from the pressure wave theory given by Blair. According to him,

$$\rho = \rho_0 Xs_2 \frac{G_5}{G}$$

$$V = G_5 a_{02} (X_{i2} - X_{i2})$$

So we have,

$$\frac{dm}{d\theta} = \rho A_{valve} V \frac{dt}{\omega}$$

The different terms are given below.

Pi1 = pressure at inlet.
Pi2 = pressure at outlet.
A1 = inlet area.
A2 = outlet area.
Ar = A2 / A1.
T1 = inlet temperature.
T2 = outlet temperature.

$$G17 = \frac{\gamma - 1}{2\gamma}, \quad G5 = \frac{2}{\gamma - 1}$$
\[ Xi_1 = \left( \frac{P_{i1}}{P_0} \right)^{G17}, \quad Xi_2 = \left( \frac{P_{i2}}{P_0} \right)^{G17} \]

\[ X_{i1} = \frac{[1 - Ar]X_i + 2Xi_1Ar}{1 + Ar} \]

\[ X_{i2} = \frac{2Xi_1 - Xi_1[1 - Ar]}{1 + Ar} \]

\[ X_s = Xi + Xr - 1 \]

\[ T_s = \frac{T_1}{Xs^2}, \quad T_{s2} = \frac{T_2}{Xs^2} \]

\[ \alpha_{s2} = \sqrt{\frac{RT}{T_{s2}}} \cdot \rho_{s2} = \frac{P_0}{RT_{s2}} \]

### Temperature and Pressure Variations

So, now we know the mass entered and left from the cylinder. Now we have to consider real gas equations. There are many real gas equations but one which fits in our temperature range is Redlich Kwong real gas equation. This is given by,

\[ P = \frac{RT}{V_m - b} - \frac{a}{\sqrt{TV_m(V_m + b)}} \]

But as we have to consider the differential approach, the differential form of the same equation is,

\[ \frac{1}{P} \frac{dp}{d\theta} = \frac{1}{m} \frac{dm}{d\theta} (C1) - \frac{1}{V} \frac{dV}{d\theta} (C1) + \frac{1}{2T} \frac{dT}{d\theta} (C2) \]

Where,

\[ C1 = \left( \frac{R^{3/2}V^2M^2(VM + mb)^2}{m^{3/2}V^2M^2(VM + mb)^2} \right) \]

\[ C2 = \left( \frac{2R^{3/2}V^2M^2(VM + mb)^2 + n^2 dV}{m^{3/2}V^2M^2(VM + mb)^2} \right) \]

Temperature can be calculated by energy conservation method, i.e. by

\[ T_{a-c} = \frac{(T_{m} \times M_{m} \times C_{p_m}) + (T_{cyl} \times M_{cyl} \times C_{p_{cyl}})}{(M_{m} + M_{cyl}) \times C_{p_{new}}} \]

\[ dp = 128.6 \times \frac{P_{in}}{BS} \times \frac{C_m}{1000}, \quad dp = 10 \times 0.377Sn_{pt} \]

The new value of \( C_p \) will be the function of new \( T \). Thus this equation will be in iteration. Thus we are able to calculate new temperature so from the differential equation we will get the new pressure. Now the gases have to move the piston and the pressure on rear side of the piston is atmospheric. So the movement of the piston will be on the cost of pressure. The pressure drop according to Ganeshan is given by,

\[ dP = 0.42P_{atm} - P_{eo} \times \left( \frac{L}{B^2} \times \frac{0.0888 + 0.182e^{0.3099 \times 100}}{100} \right) \]

This temperature and pressure will be again put into the equation to do the iteration. The whole loop will continue up to the whole exhaust process. Thus we will get the pressure and temperature at the end of the exhaust process. Also mass of burnt gases leaving will be known. The result that we got has been plotted on graph.
RESULTS & DISCUSSION OF THE MODEL

Results of the Expansion Process

Above Graphs show the trend of cylinder pressure variation with increasing crank angle. Combustion products with very high temperature gets 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/m²K, which is much higher compare to the value of 374.2725 W/m²K 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
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. Fig.4.15 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.
CONCLUSION

A graph shows 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 at 484° crank angle. 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 much higher compare to the value 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.

Graph gives 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.

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.

REFERENCES