International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 2 Issue 9, September - 2013

Determination of Improvement in Characteristics of a Spark-Ignition (SI) Engine with Spark Plug Embedded in Piston

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Abstract

This technical paper attempts to determine the improvements in the I. C. Engine performance parameters once the spark plug has been placed at the centre of the piston with the entire spark plug assembly embedded inside the piston. A basic theoretical analysis of the combustion has been carried out in this paper as a comparison with existing conditions to point out a striking differencein the performance parameters of the engine shown herewith.

Key Words:*performance parameters, theoretical, analysis, combustion.*

1. Introduction

$$\text{Flame velocity} = \frac{dm_b/dt}{A_f\,\rho}$$

where m_b is the mass burning in the combustion, A_f is the flame area and ρ is the density of the fuel. Thus the differentiation term defines the mass burning rate in the combustion chamber [1].

By simple deduction, we can see that flame velocity is directly proportional to the mass burning rate at a constant A_f and inversely proportional to A_f at a constant mass burning rate. When we consider the first case, i.e. the conventional spark position, the spark firing is in such a position that the flame front progresses only in one direction. The flame velocity is enough for low engine speeds to achieve complete combustion. However, for example, when engine speeds increase to the order of 5000 to 6000 rpm,
Time allotted for complete combustion is around 5 ms. The flame front in the conventional case cannot develop enough torque to drive the engine load at high speedsat stoichiometric air-fuel ratio due to which rich mixtures are required. There have been some advances in engine technology to accommodate for such changes such as the advancement in spark timing, addition of supercharger, etc. However, if the spark is advanced, the maximum temperature and pressure achieved during the cycle will not be as high as that at lower speeds. Hence the amount of useful work decreases. On the other hand if the A/F ratio is changed by carburetion or intelligent fuel injection systems, the charge burnt is less for a given cycle. The cycle in both cases, does not achieve optimum efficiency because of high load condition.

However, when the spark plug is embedded in the piston, as the combustion progresses, the fuel occupies space below the combusted fuelso that the flame front may acquire a greater A_f. The flame front becomes more spherical than hemispherical, thus achieving better combustion quality. Also, by achieving a greater A_f, the requirement of flame velocity to achieve complete combustion drastically reduces. Hence, the combustion quality and performance parameters of the engine should undergo substantial improvement.

2. Experiment Setup

The theoretical engine used here is the General motors' 60^o V-6 Engine[3] with a displacement of 2800 cc, bore 89 mm, stroke 76 mm, compression ratio 8.5, maximum power 86 kW at 4800rpm.

The setup mainly consists of two hypotheticalGM V6 engines, same in all respects except that one of them has the spark plug in the conventional position, whereas the other has the spark plug embedded inside the piston. The valve timings, the ignition delay, the shape of the piston and other such parameters have been considered as ideal and identical for both engines. This assumption is valid in this case because this is a comparative study. Once significant results are achieved, it is possible to increase the level of accuracy in the simulation by considering non-ideal conditions.

There are some assumptions regarding the model of combustion. We assume that the combustion is laminar. Also, the fuel-air mixture is perfectly atomized and homogenous and present in equal density throughout the combustion chamber, i.e. ρ is constant.
Thus by defining these assumptions we can assume that the flame front will be perfectly spherical in all directions. Also it has been assumed for these calculations that the spark is fired at such a time that

$$\text{Ignition delay} = \text{Spark advance}$$

Thus it can be assumed that appreciable combustion starts at TDC. Also, it must be assumed that the flame velocity is constant at its mean.

3. Basic Calculations

Assume the maximum power output condition at 4800 rpm. In this case, the time available to the engine to complete combustion equals that for half of a complete revolution of the crankshaft. Any further time taken for combustion will cause a reverse effect on the power output of the engine and hence is counter-productive. For optimum usage of fuel heat, the combustion must take place in the expansion stroke only.Now, the flame must propagate through a maximum portion of the combustion chamber so that a large portion of the charge is burnt in every cycle. Thus the flame must travel the length of the piston and to the deepest corner of the piston or at least to a reasonable distance of it so that charge is not burnt during the exhaust stroke.

Hence time available for combustion $= (\frac{4800x2}{60})^{-1}$

$= 6.25ms$

This is the maximum time available for combustion. Hence this is limit on any calculation of combustion taking place henceforth.

Swept volume $V_s = \frac{\pi}{4} D^2 L$

$$= \frac{\pi}{4} x89^2 x76$$

$$= 4.728x10^5 mm^3$$

$$= V_1 - V_2$$

Now,

Compression ratio $= \frac{V_1}{V_2} = 8.5$

Solving,

$V_1 = 5.3585x10^5 mm^3$

$V_2 = 6.3041x10^4 mm^3 = V_c$ where V_c is the clearance volume. Hence, when appreciable combustion begins, the volume of the combustion chamber is $6.3041x10^4$ mm^3.Clearance height is the height of the combustion chamber at full compression, i.e. at V_2.

Clearance height (h_2):

$$6.3041x10^4 = \frac{\pi}{4} x89^2 xh_2$$

$$h_2 = 10.133mm$$

To get a good view of the change in performance, some parameters must be checked. In this paper, the parameter under consideration is the burned mass fraction (x_b).

$x_b = \frac{m_b}{m}$, where, m_b is the amount of charge burnt and m is the total mass of charge present in the combustion chamber. The burned mass fraction as a function of time after combustion begins gives a clear indication of the combustion quality in the combustion chamber. Here it must be stipulated that if there is an appreciable increase in the characteristics of burned mass fraction over the time period of the expansion stroke, then the combustion quality will improve as will the power and performance parameters of the engine. A higher burned mass fraction corresponds to more power, hence better thermal efficiency and torque capacity.

Now,

$$x_b = \left[1 + \frac{\rho_u}{\rho_b}\left(\frac{1}{y_b} - 1\right)\right]^{-1}$$

where ρ_u and ρ_b are the densities of the unburned and burned mixtures respectively whereas the parameter refers to the burned volume fraction[1].

Here, $y_b = {V_b}/{V}$, where V_b and V are the volume of burned gases and the total volume respectively.

In order to find out the changes in x_b according to time, this equation must be modified so that we get x_b as a function of time.While the density ratio (ρ_u/ρ_b) does depend on the air-fuel equivalence ratio, burned gas fraction in the unburned mixture, gas temperature and pressure, its value is close to 4, for most spark-ignition engine operating conditions [1].

$$\therefore x_b = \left[1 + 4\left(\frac{1}{y_b} - 1\right)\right]^{-1}$$

$$= \left[\frac{4}{y_b} - 3\right]^{-1}$$

$$x_b = \left[\frac{4V}{V_b} - 3\right]^{-1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

For both cases, the value of V remains the same, i.e. the swept volume $V_s = 4.728x10^5$ mm^3.

Now, the changes to be considered in terms of burned mass fraction are considered only till the flame front reaches the cylinder side walls. The position of the spark plug is significant only during the initial period of combustion when the flame gets a larger are of flame front A_f. This condition exists up to the point when the flame front reaches the cylinder walls. After this point, the area of flame front does not differ from that in the conventional case. Hence, the increase in x_b achieved in this latter part of combustion is negligible. The appreciable increase in x_b is seen in the initial part of combustion, and therefore, it is this part of the combustion that this paper focuses on.

Case 1: When the spark plug is in the conventional position:

Until the flame front reaches the cylinder side walls, it is assumed that it travels in a perfect hemispherical pattern in a laminar flow. Thus,

$$V_b = \frac{2}{3}\pi r^3$$

Assuming constant flame velocity, the radius up to which the flame reaches can be written as $r = S_L t$, where S_L is the flame velocity and t is the time taken by the flame to reach the point.

$$V_b = \frac{2}{3}\pi S_L^3 t^3 \quad \text{.. (2)}$$

Thus, from (1) and (2),

$$x_b = \left[\frac{4x4.728x10^5 x10^{-9}}{\frac{2}{3}\pi S_L^3 t^3} - 3\right]^{-1} \quad \text{................. (a)}$$

Case 2: When the spark position is changed:
When the combustion begins, until the flame front reaches the cylinder top wall, the flame front travels in a perfect sphere. But after this point, the flame front travels in a position between a sphere and a hemisphere. The clearance height is 10.133 mm.

Time taken to reach the cylinder top is $t_1 = \frac{h_2}{S_L}$.

Hence up to t_1,

$$V_b = \frac{4}{3}\pi r^3 = \frac{4}{3}\pi S_l^3 t^3$$

$$\therefore x_b = \left(\frac{4x4.728x10^{-4}}{\frac{4}{3}\pi S_L^3 t^3} - 3\right)^{-1} \quad \text{..................... (b)}$$

After t_1, the flame front has reached the clearance height and it is slowly turning from a sphere to a hemisphere (when it reaches the cylinder walls). Hence for simplicity, the volume paved by the flame is considered to be midway between $4/3\pi r^3$ and $2/3\pi r^3$, i.e. πr^3.

At a time after t_1,

$$V_b = \frac{4}{3}\pi(10.133x10^{-3})^3 + \pi(r^3 - 10.133^3)x10^{-9}$$

$$\therefore x_b = \left(\frac{4x4.728x10^{-4}}{\frac{4}{3}\pi(10.133x10^{-3})^3 + \pi(S_L^3 t^3 - 10.133^3)x10^{-9}} - 3\right)^{-1}$$

$$\text{................... (c)}$$

These are the values for x_b as they change with time. The values for flame velocities range from 15m/s to 30m/s [2]. Hence the changes of x_b with time have been plotted at flame velocities of 15m/s and 30m/s. By using equations (a), (b), and (c) for S_L=15m/s and 30m/s, the following graphs are plotted. These curves are important aids in interpreting flame geometry information.

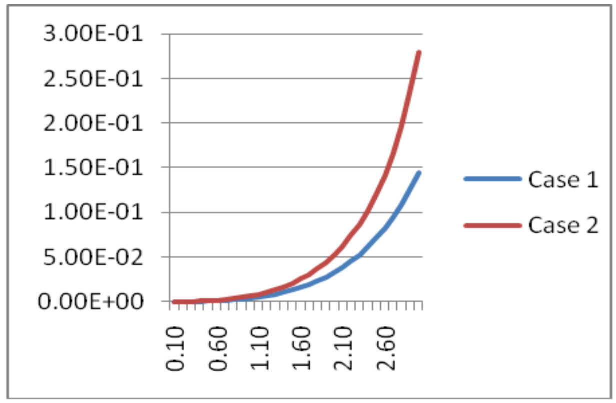

Fig. 1 Burned mass fraction comparison at flame velocity = 15m/s

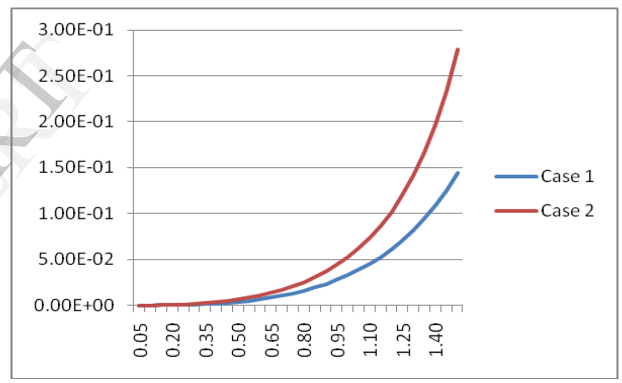

Fig. 2 Burned mass fraction comparison at flame velocity = 30 m/s

4. Effect on Parameters

The quality of combustion achieved in the changed format of the engine is better than the combustion quality of the original engine across the entirety of its working conditions. If implemented, this modification can cause a substantial improvement in the performance of the I. C. Engine. The implications of this modification include:

1. More stable and complete combustionat all speeds.
2. Higher pressure and temperature achievement at all loads for the same inlet conditions.
3. Less fuel wasted per cycle.
4. More useful power achieved per cycle.
5. Higher torque capacities for the engine through all load conditions.

Although this analysis conclusively proves the validity of the theory put forth in this paper, it is necessary to consider the improvements (or otherwise) in terms of the standard parameters of the engine. Some of these parameters are listed below.

1. Thermal efficiency η_{th}
2. Brake power as a function of fuel consumption.
3. $\frac{mep}{p_{\max}}$ for given maximum pressure
4. Torque speed characteristics
5. Composition of exhaust gases

The thermal efficiency η_f is given as $\eta_{th} = \frac{P}{\dot{m}_f Q_{HV}}$ [1]. Thus the thermal efficiency depends on the ratio of power generated by a constant amount of fuel. Thus the thermal efficiency increases. Brake power also increases for the same fuel consumption, as does the torque capacity of the engine. The mean effective pressure is a function of the brake power. An analysis of the composition of exhaust gases will give us an insight into the change in fuel wastage phenomena.

This paper contends that in all the parameters mentioned above the modified engine suggested will be better than the existing engine. However, this modification gives a lot of disadvantages to the existing design.

5. Disadvantages

The main disadvantages of this design are enlisted as follows:

1. Hard to manufacture
2. Maintenance of spark plug assembly is a problem
3. If current is given to the spark plug, the piston strength is compromised

Thus, if these problems are solved, this can be a valid hypothesis for an engine design.

6. Future Scope

1. Detailed FEM analysis (strength and stability analysis) augmented with CFD analyses using Gambit and Fluent to be conducted on the engine cases for design purpose.
2. Spark plug must be designed to accommodate in situ operation.
3. Engine design modifications to accommodate new design of spark plug position

7. References

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