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**Original Research Article**

# Experimental Study on Combustion Duration and Performance Characteristics of a Hydrogen-Ethanol Dual Fueled Engine

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## Abstract

This paper discusses the results of combustion duration and performance characteristics of a hydrogen-ethanol dual fueled engine. The tests were conducted on a compression ignition engine (modified to run on spark ignition mode) fueled with hydrogen–ethanol dual fuel combination with different percentage substitutions of hydrogen (0–80% by volume with an increment of 20%) at a constant speed of 1500 rpm and a load of 100% in kW by adjusting the loading switches. The various engine operating parameters like compression ratio, equivalence ratio, spark timing, and engine's performance parameters like brake power, brake specific fuel consumption, brake mean effective pressure, andbrake thermal efficiency effect on combustion duration were studied. The best operating conditions were obtained at a compression ratio of 11:1 and the optimum fuel combination was found to be 60–80% hydrogen substitution to ethanol.For better performance in terms of power and economy, it was found from the present study that the combustion duration has to be in between 35 to $42^0$.

**Keywords:** Hydrogen; Ethanol; Combustion duration; Compression ratio; Ignition timing; Spark ignition; Performance.

**Nomenclature**

| | |
|---|---|
| BMEP | brake mean effective pressure |
| BSFC | brake specific fuel consumption |
| BTDC | before top dead center |
| BTE | brake thermal efficiency |
| CA | crank angle |
| CHR | cumulative heat release |
| CI | compression ignition |
| CR | compression ratio |
| DC | direct current |
| HP | horse power |
| kW | kilo watt |
| MBT | maximum brake torque |
| SI | spark ignition |
| VRCS | variable rate cooling system |

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## 1. Introduction

Among the various alternative fuels hydrogen and alcohol are very attractive substances for many practical applications in the energy sector. While conventional energy sources such as natural gas and oil are non-renewable, hydrogen and alcohol can be coupled to act as renewable energy sources. Ethanol has a higher octane rating is considered a renewable fuel and the engine needs to be only slightly modified. The increase in compression ratio (CR) is possible with ethanol because of its high octane number. Operation at high CR reduces fuel consumption, which is relatively high because of alcohol's low heat of combustion. Hydrogen for applications in internal combustion engines is remarkably a light gaseous fuel having some unique and highly desirable properties like cleanest burning chemical fuel wider limits of flammability in air than propane, methane or gasoline and higher burning velocity of hydrogen air mixture which is about six times higher than that of the gasoline–air mixture.

Enriching ethanol with hydrogen could help solve the cold starting issue while adding many other advantages. Hydrogen enriching of fuels has been shown to extend the lean limit of the fuel and enable high dilution combustion regimes [1]. Therefore, hydrogen addition reduces the specific fuel consumption of the ethanol engine. The advantage of the hydrogen-supplemented fuel presents in requiring a smaller quantity of hydrogen, which considerably reduces the problems connected with hydrogen storage in the automobile [2].

Combustion duration is very important operating parameter that affects spark ignition engine performance and efficiency and not much previous work was done on hydrogen–ethanol dual fuel engine under actual operating conditions, thus this study in this paper concentrates on the effect of combustion duration (which was varied by varying the equivalence ratio) on engine performance parameters namely (BMEP, BP, BSFC and BTE), and spark timing of a hydrogen-ethanol dual-fuel combination with different percentage substitutions of hydrogen (i.e. 0-80% with an increment of 20% by volume) at three different compression ratios of 7:1, 9:1 and 11:1.These experimental studies were used to study the effect of various operating parameters on the combustion duration as well as the effect of combustion duration on the engine performance characteristics to try to get a better understanding of the interaction between these parameters.

## 2.Prior research

Bayraktar and Durgun [3] developed an empirical correlation for combustion duration in SI engines. In their study, they have determined the effects of variations in compression ratio, engine speed, fuel/air equivalence ratio and spark advance on combustion duration by means of a quasi-dimensional SI engine cycle model previously developed by the authors. Burn durations at several engine-operating conditions were calculated from the turbulent combustion model. Variations of combustion duration with each operating parameter obtained from the theoretical results were, expressed by second-degree polynomial functions. By using these functions, a general empirical correlation for the burn duration has been developed. In this correlation, the effects of engine operating parameters on combustion duration were taken into account. Combustion durations predicted by means of this correlation are in good agreement with those obtained from experimental studies and a detailed combustion model.

Zuohua Huang et al. [4] investigated the combustion characteristics of a direct-injection spark-ignited engine fueled with natural gas–hydrogen blends under various ignition timings and lean mixture condition. Their results showed that the ignition timing has significant influence on engine performance, combustion, and emissions. They noticed that the time intervals between the end of fuel injection and ignition timing are very sensitive to direct-injection gas engine combustion. They also found that the turbulence in combustion chamber generated by the fuel jet maintains high and relatively strong mixture stratification when decreasing the time intervals between the end of injection and the ignition timing, giving fast burning rate, high brake mean effective pressure, high thermal efficiency and short combustion durations. They noticed that for specific ignition timing, the brake mean effective pressure and the effective thermal efficiency increase and combustion durations decrease with the increase of hydrogen fraction in natural gas.

Moreno et al [5] in his paper discussed the experimental tests results carried out in a wide range of speeds at equivalence ratios of 1, 0.8 and 0.7 and at full load. The ignition timing was maintained for each speed, independently of the equivalence ratio and blend used as fuel. Four methane-hydrogen blends were used. In-cylinder pressure, mass fraction burned, heat released and cycle-by-cycle variations were analyzed as representative indicators of the combustion quality. It was observed that hydrogen enrichment of the blend improve combustion for the ignition timing chosen. This improvement is more appreciable at low speeds, because at high speeds hydrogen effect is attenuated by the high turbulence. Also, hydrogen addition allowed the engine to run stable in points where methane could not be tested.

A study by Pereira et al. [6] covered stoichiometric and lean mixtures ($\lambda = 1.0$ and $\lambda = 1.2$), various spark advances (30–50° CA), a range of engine temperatures (20–90°C), and diverse injection strategies (single and “split” triple). In-cylinder gas sampling at the spark-plug location and at a location on the pent-roof wall was also carried out using a fast flame ionization detector to measure the equivalence ratio of the in-cylinder charge and identify the degree of stratification. Combustion imaging was performed through a full-bore optical piston to study the effect of injection strategy on late burning associated with fuel spray wall impingement. Combustion with single injection was fastest for ethanol throughout 20–90°C, but butanol and methane were just as fast at 90°C; iso-octane was the slowest and gasoline was between iso-octane and the alcohols. At 20°C, relative air fuel ratio ($\lambda$) at the spark plug location was 0.96–1.09, with gasoline exhibiting the largest and iso-octane the lowest value. Ethanol showed the lowest degree of stratification and butanol the largest. At 90°C, stratification was lower for most fuels, with butanol showing the largest effect. The work output with triple injection was marginally higher for the alcohols and lower for iso-octane and gasoline (than with single injection), but combustion stability was worse for all fuels. Triple injection produced a lower degree of stratification, with leaner $\lambda$ at the spark plug than single injection. Combustion imaging showed much less luminous late burning with tripe injection. In terms of combustion stability, the alcohols were more robust to changes in fueling ($\lambda = 1.2$) than the liquid hydrocarbons.

Sun Bai et al. [7] in their paperstudied and discussed about the laminar flame speed difference between hydrogen and gasoline.A distinctive rule of combustion duration in hydrogen internal combustion engine was discovered by analyzing the experiment data.A new characteristic of the location of 50% mixture combust up was proposed and analyzed, this will be helpful for the calibration of optimum ignition timing.

Madhujit Debet al. [8] in their paper investigated the hydrogen-diesel dual fuel combustion in direct injection (DI) diesel engine. The investigation presented in their paper preferred hydrogen as a long-term renewable and least polluting fuel among various alternative fuels for internal combustion (IC) engines. In their study a diesel engine was made to run using hydrogen in dual fuel mode with diesel, where hydrogen is introduced into the intake manifold using an LPG-CNG injector and pilot diesel is injected using diesel injectors. The hydrogen energy contents of the total fuel were varied from 0%, 11%, 17%, 30%

and 42% (the 0% hydrogen energy content represents neat diesel fuel), were experienced at (1500 ± 10) rpm of invariable engine speed and 5.2 kW of consistent indicated power. The test results showed the improvement in brake thermal efficiency of the engine, reduction in brake specific energy consumption with an increasing hydrogen energy fraction. In addition to that, it was also observed by them that there was a sharp increase in peak in-cylinder pressure and the peak heat release rate values with the increasing hydrogen rate.

**3. Experimental setup and procedures**

The engine (Figure 1) used in the present study was a Kirloskar AV-1, single cylinder direct injection diesel engine modified to run at low compression ratios thus making it adaptable to run in spark ignition (SI) mode by replacing the diesel fuel system with carburetor, that was connected to the air-intake-manifold of the engine inlet system and a spark plug was located in place of the diesel injector. Also a provision was made to induct hydrogen gas in the inlet manifold. To vary the compression ratio from 7 to 11 different spacers were placed between the cylinder and the cylinder head thus increasing the clearance volume thereby reducing the compression ratio.

The engine was coupled to a DC dynamometer and all the experiments were carried out at a constant speed of 1500 rpm and 100% load in kW. The specifications of the engine are given in Table 1.To facilitate operating the engine under spark timing a varying timing arrangement is provided on the engine. By adjusting varying timing arrangement mechanically the spark timing could be varied at will even while the engine is in operation. For a given quantity and quality of the mixture under a given load setting the timing which gave the maximum speed is taken as the MBT spark timing. The contact breaker points and the condenser of the ignition circuit are fixed on a Bakelite disc. The disc is mounted on the engine camshaft over which a small cam is made to operate the contact breaker points. The angular position of the contact breaker points with respect to the cam decides the spark timing and it could be altered at will by a suitably designed linkage which is provided on the engine test rig. The special software stores the data of pressures and volumes corresponding to a particular crank angle location for plotting the P-V and P-θ curves. The software also provides the facility of analyzing the combustion data such as the rate of heat release, combustion duration period in degrees, peak pressures and stores it separately for analysis in the data acquisition system. The engine electronic control system provided access to all calibration parameters allowing the user to set a desired equivalence ratio (by adjusting fuel flow rate), combustion duration and ignition timing. The transducer was cooled by water to get accurate readings. The water supply to the engine was also properly maintained by a specially designed system known as variable rate cooling system (VRCS) that was provided on the engine to vary the cooling rate from1.5 liters/min to 5.5 liters/min to avoid over heating of the engine. The VRCS was operated with the help of an electrical pump of 1/4$^{th}$ HP capacity located outside the engine with a rheostat arrangement (i.e. step-less regulator arrangement). The temperature indicator on the engine control panel indicates the engine operating temperature hence when it is noticed that the temperature of the engine is rising then manually the VRCS is operated to increase the engine cooling rate and thereby maintaining the engine temperature in a safe range and hence avoiding the overheating of the whole system. Prevention of explosive atmosphere in the test bench room was taken care by means of monitoring leaks of the hydrogen supply line, continuous monitoring of the test bench, air and a powerful ventilation system. The hydrogen cylinder is placed at a safe distance from the engine to avoid heat transfer to the cylinder. Although it was possible to have higher compression operation of the engine, compression ratios were restricted in the range 7:1 to 11: 1 as higher compression ratios increases the problem of undesirable combustion such as pre-ignition and backfiring. Backfiring was prevented by keeping the engine scrupulously clean and by using a cold spark plug with an appropriate narrow gap.

The engine performance tests were carried out to study the effect of combustion duration on various operating and performance parameters of the engine. These tests were conducted at three different compression ratios of 7, 9 and 11. At each compression ratios constant speed test runs were made at 1500 rpm. During these performance tests the throttle was varied and the load was kept constant at 100% load in kWby adjusting the loading switch. At 100% load the speed was kept constant by controlling the hydrogen and ethanol flow rate (i.e. by adjusting the hydrogen substitutions from 0 to 80%by volume, an increment of 20%) and the spark timing were adjusted for best torque. The rates of air and fuel consumption were also measured. The hydrogen flow is measured using a specially designed hydrogen flow meter. To dampen the pressure fluctuations in the intake line which particularly occur with large displacement single cylinder engines a stabilizing tank is located at the inlet of the engine. The intake temperature and pressure were chosen to give stable and knock free engine operation. The hydrogen cylinder pressure was reduced before inducting the hydrogen into the inlet manifold. Safety devices like water trap and flame arrester were connected in the fuel supply line because of the hazardous nature of hydrogen.

The hydrogen gas was inducted into the intake-manifold and a thermal mass flow controller controlled its flow rate. The maximum amount of hydrogen supplied was limited by unstable operation at low outputs and by rough engine running due to knock at high outputs. When the hydrogen supply was increased the ethanol quantity was automatically decreased by the governor mechanism of the engine to maintain the speed constant. Ethanol flow rate is measured on volume basis using a burette and a stopwatch. The time consumed for 10 cc of fuel consumption is timed using a digital stopwatch with an accuracy of 0.1 s.

Table 1 - Engine specifications.

| **Engine specifications** | **Data** |
|---|---|
| Bore (mm) | 80 |
| Stroke (mm) | 110 |
| Cylinder Capacity ($cm^3$) | 552.64 |
| Compression ratio (made available) | 7:1 – 11:1 |
| Ignition source | Spark plug |
| Combustion Chamber | Disk-shaped combustion (with a flat piston and chamber ceiling) |
| Orifice diameter (mm) | 15 |
| Rated power (kW) | 3.7 |
| Rated speed (rpm) | 1500 |

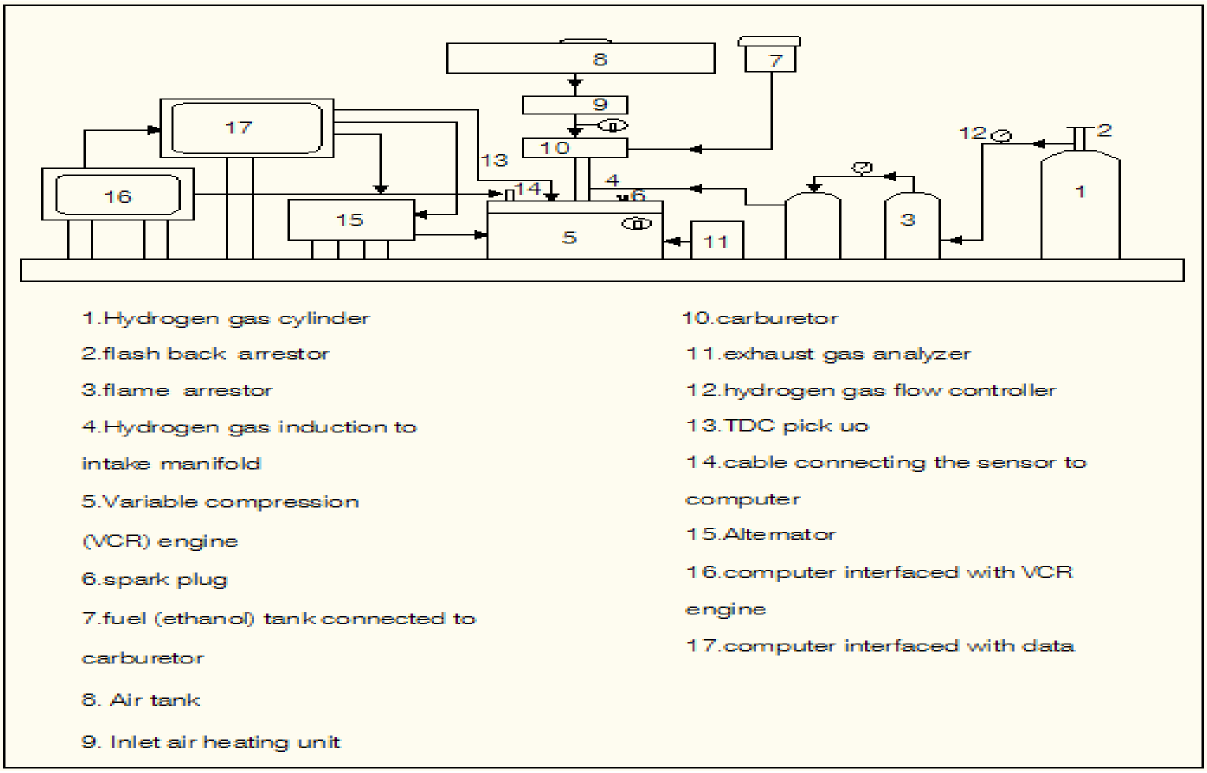


Figure 1 – Experimental Test Rig with necessary instrumentation.

## 4. Results and Discussion

4.1 Effect of combustion duration on engine's operating conditions

4.1.1 Effect of compression ratio on combustion duration

Increased turbulence in the unburned mixture at the time of combustion increases the burning rate. Turbulence is usually increased by generating swirl during the induction process. The duration of the early stage and main stage of the burning process decrease as the turbulent velocity at the start of combustion is increased. The faster combustion process comes primarily from the higher turbulence intensity; however, the decrease in characteristics turbulence scale that accompanies with the increase in turbulence is also significant since it results in a shorter characteristic burning time. The multitude of ignition centers from ethanol fuel flame will make flame path from each ignition source very short thus; the combustion time of the hydrogen fuel expected to be much shorter. Thus, the combustion process is so fast to appear as if it happened by auto ignition.

Figure 2 shows that the combustion duration decreases as the compression ratio increases with increasing substitution of hydrogen to ethanol. This is because of the increase in the end of compression temperature and pressure and decrease in the fraction residual gases. This creates a favorable condition for the reduction of ignition lag and increase in the flame speed. The percentage reduction in combustion duration in degrees for 100% ethanol was 30.98% for an increase of compression ratio from 7:1 to 11:1, whereas the average reduction for 80% hydrogen under identical conditions was 17.8%.

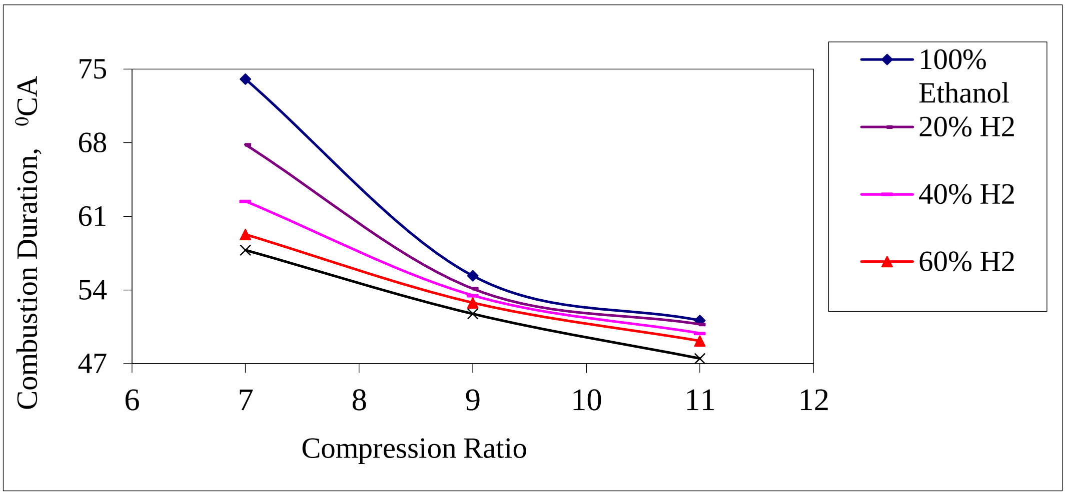


Figure 2 –Effect of compression ratio on combustion duration for various percentages of hydrogen substitution at 100% load and 1500 rpm.

4.1.2 Effect of combustion duration on equivalence ratio

As the stoichiometric equivalence ratio of hydrogen is more than that of ethanol addition of hydrogen to ethanol increases the stoichiometric equivalence ratio of hydrogen-ethanol fuel blends. From Figure 3, 4 and 5 it can be observed that for 100% ethanol (pure ethanol) with the increase in compression ratio the combustion duration decreases with an increase in equivalence ratio. With addition of hydrogen to ethanol, it is noticed from the Figure 4 and 5 for each CR that there is a decrease in combustion duration with a subsequent increase in equivalence ratio value. For 100% ethanol with the increase in compression ratio the combustion duration decreases with an increase in equivalence ratio. The average decrease in combustion duration for pure ethanol is 3.25% when the CR is changed from 7 to 11.

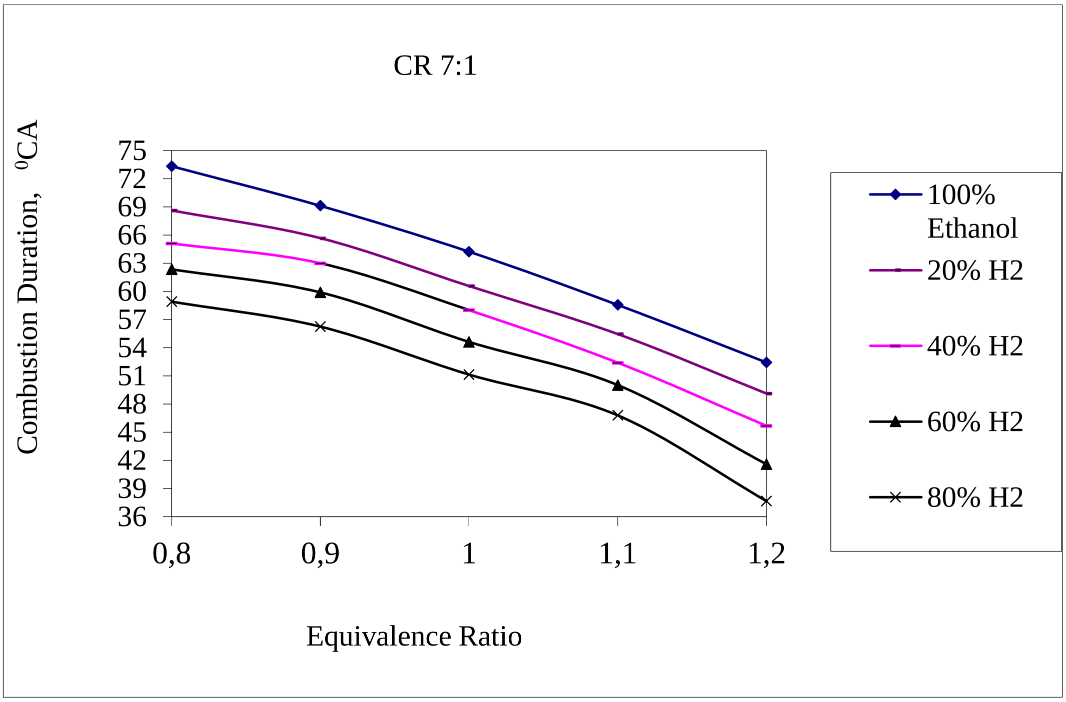


Figure 3 –Effect of equivalence ratio on combustion duration for various percentages of hydrogen substitution at CR 7:1, 100% load and 1500 rpm.

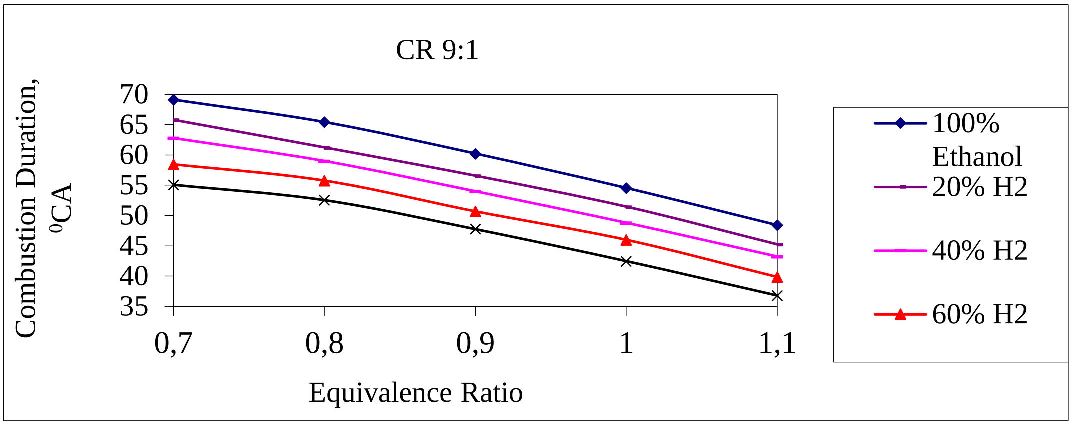


Figure 4 –Effect of equivalence ratio on combustion duration for various percentages of hydrogen substitution at CR 9:1, 100% load and 1500 rpm.

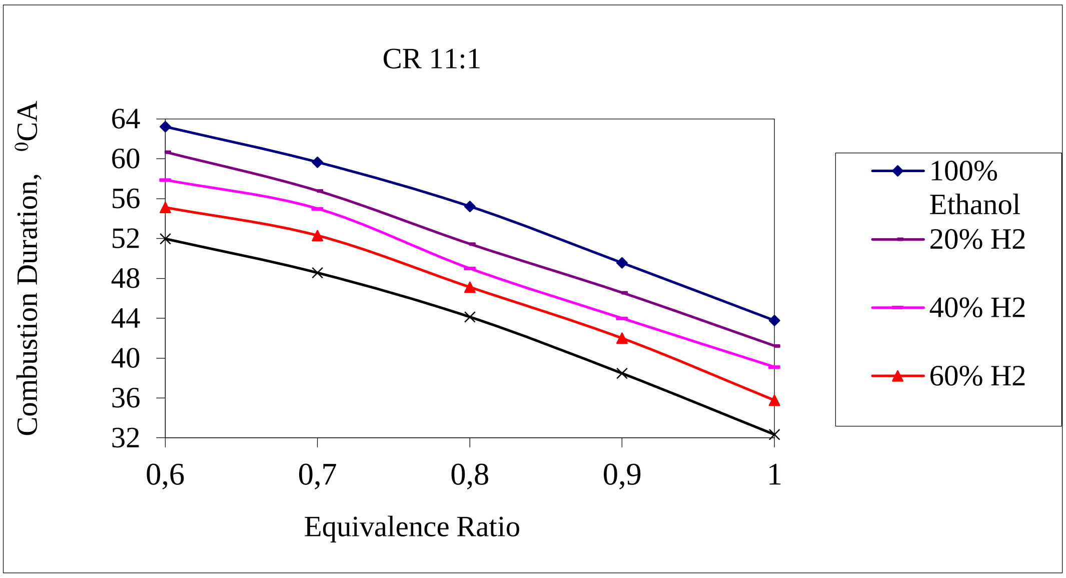


Figure 5 –Effect of equivalence ratio on combustion duration for various percentages of hydrogen substitution at CR 11:1, 100% load and 1500 rpm.

Figure 3, 4 and 5 shows a trend of decreasing as the hydrogen supplementation is increased. This is as expected because the faster combustion of the hydrogen will lead to a decrease in the burn duration. This increase is what allows the stable operation at this operating condition. Burn duration is decreased only slightly by the increase in pressure at the spark. In actuality, an increase in the initial pressure during combustion can greatly alter the way the fuel burns. Higher pressure in a limited volume means that the temperature in the cylinder is higher and that the fuel and air molecules are more reactive. The mixture burns faster so the burn duration becomes shorter. The delayed spark firing takes advantage of the more reactive nature of the fuel to get more power out of the stroke. Therefore, it is seen that as the combustion duration decreases with increasing substitution of hydrogen to ethanol it tends to increase the equivalence ratio thus resulting in increase in heat release rate. The heat release rate at a low equivalence ratio gives poor combustion stability. At high equivalence ratios the heat release and thus combustion rate increases and the heat release pattern becomes more or less smooth and gradual thus enhancing the combustion stability.

At higher equivalence ratios, the combustion is completed earlier due the faster flame speed. On the other handat low equivalence ratios the heat release rate may extends even in the exhaust stroke giving rise to more unburnt hydrogen and thus poor thermal efficiency. It is also observed from the above figures that operating at lean mixtures tends to increase the combustion duration. This effect is more predominant at higher speeds if considered. This is due to lesser thermal energy liberated from the leaner mixtures which increases the ignition delay and slows the flame propagation. The flame temperature is low at lean mixtures. Further, the incomplete combustion due to oxygen deficiency at rich mixtures also has an adverse effect over the flame speed. In addition, it can be seen from Figure 5 that the combustion duration is less for 80% hydrogen substitution at equivalence ratio equal to 1.0 and a CR of 11:1. From Figure 3, 4 and 5 we can see that at a given equivalence ratio, combustion duration shortened as hydrogen fraction increased and if the mixture is made progressively rich the combustion duration decreases. Thus, concluded that hydrogen addition could indeed speed up flame propagation.

4.1.3 Effect of combustion duration on spark timing

Figure 6, 7 and 8 shows the combustion durations versus the ignition timings for different percentage substitutions of hydrogen at three compression ratios of 7, 9 and 11.

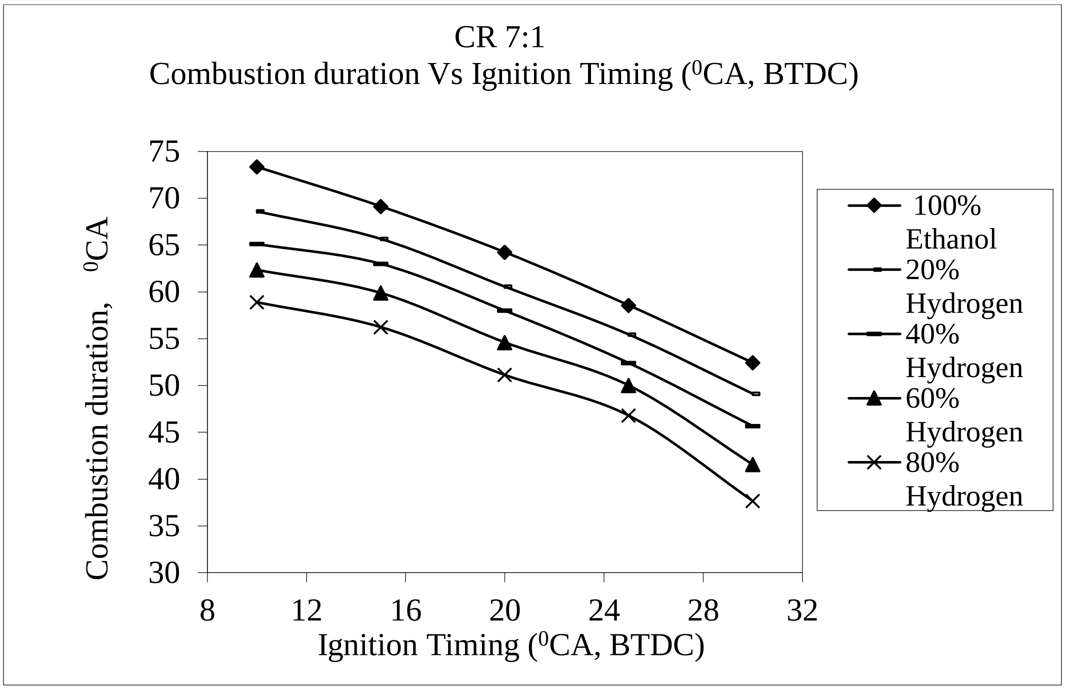


Figure 6 –Effect of combustion duration on Ignition timing for various percentages of hydrogen substitution at CR 7:1, 100% load and 1500 rpm.

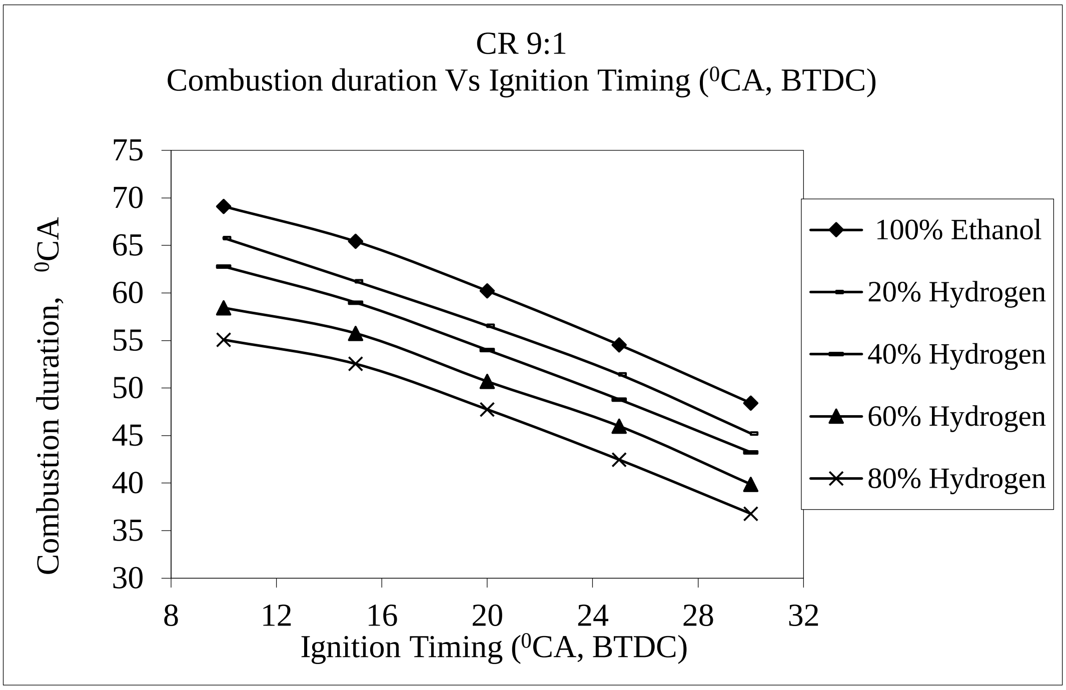


Figure 7 –Effect of combustion duration on Ignition timing for variouspercentages of hydrogen substitution at CR 9:1, 100% load and 1500 rpm.

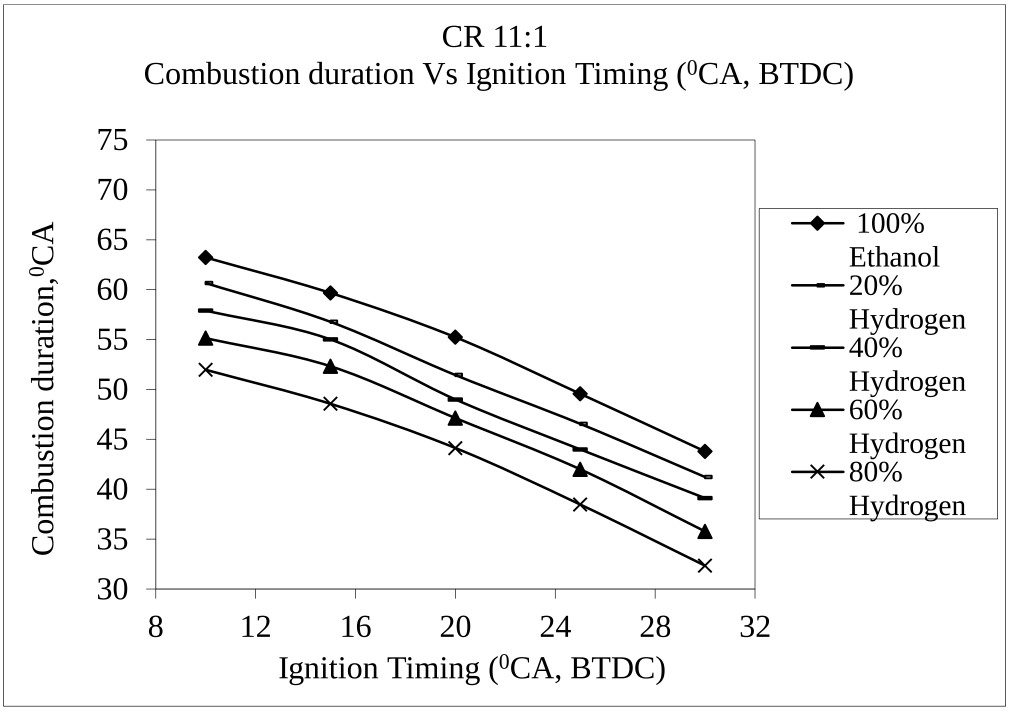


Figure 8 –Effect of combustion duration on Ignition timing for various percentages of hydrogen substitution at CR 11:1, 100% load and 1500 rpm.

Flame development duration decreases with advancing the ignition timings. The combustion duration seen to decrease as the spark timing is advanced. Advancing ignition timing will shorten the time duration between the ends of fuel admission, ignition timing and this will form the high mixture stratification in the combustion chamber, and relatively rich mixture near the spark plug making the mixture more easily to be ignited thus reduces the ignition delay and the flame development duration. With the advanced ignition timings by the time the ignition delay is over and a regular flame front is established the piston might still be before but close to the dead center and the thermal conditions are quite favorable for combustion. Retarded ignition timings make the combustion to be initiated after the TDC and naturally, it would take a longer time with the piston already on the expansion stroke. For specific ignition timing, the flame development duration decreases with the increase of hydrogen fraction indicating that addition of hydrogen can promote flame kernel formation and flame propagation at early stage of mixture combustion. The improvement of flame propagation speed and the increase in gas temperature of the mixture make quickly the combustion resulting in the decrease in combustion duration with advancing ignition timings. For lean mixture combustion, the decrease in time intervals between the end of fuel admission and ignition timing makes high mixture stratification and increases the burning velocity. Meanwhile, shortening the time intervals between the end of fuel admission and ignition timing and can maintain a high turbulence generated by high-speed hydrogen gas and this will increase the mixture burning velocity. For specific ignition timing the combustion duration decrease at all compression ratios with the increase of hydrogen fraction and this are resulted from burning velocity enhancement by hydrogen addition.

4.2. Effect of combustion duration on engine's performance parameters

Theoretically, the rate of combustion should be such that the combustion duration is minimum with high rate of pressure rise. Pressure should be maximum at TDC to produce greater force acting through a large period of the power stroke. This, however, means that the products will have enough time to lose some of its heat to the coolant resulting in poor performance in addition to the rough engine operation. Therefore, in practice, engines are so designed that only 50 % of the pressure rise is completed by the TDC resulting in peak pressure and temperature occurring at $10^{o}$ to $15^{o}$after TDC which reduces the heat loss and makes the engine operation smooth. Therefore, for best results, the combustion has to be completed within $15^{o}$after TDC.

4.2.1 Effect of combustion duration on brake mean effective pressure

From Figure 9, 10 and 11 it is seen that with an increase in the combustion duration period there is a decrease in brake mean effective pressure. This is because of the increased heat losses. For 80% hydrogen at CR of 7 the change in combustion duration was $21^{0}$ for a decrease in BMEP value by 23.3%. On the other hand, for CR 11 the increase in combustion duration was $15^{0}$ for a decrease in BMEP value by 29.3%.

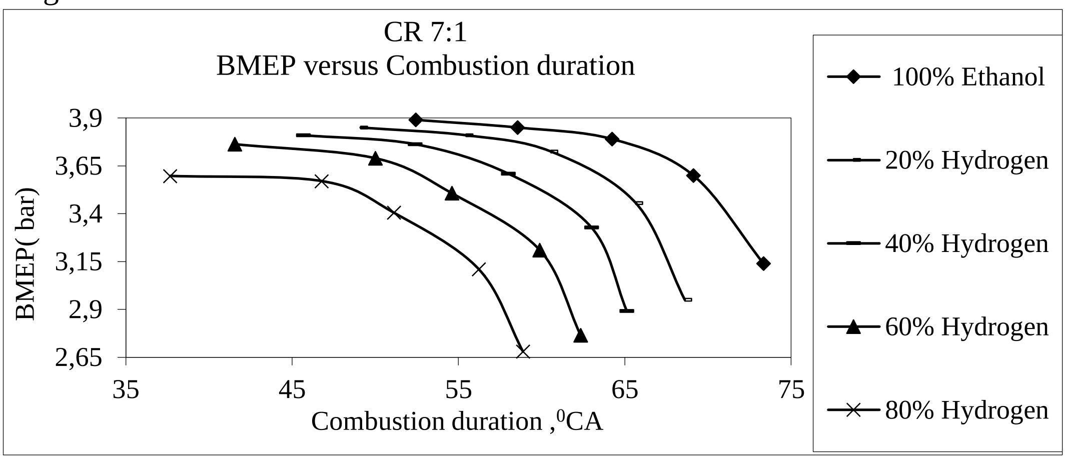


Figure 9 –Effect of Combustion duration on BMEP for various percentages of hydrogen substitutions at 7:1 CR and 1500 rpm.

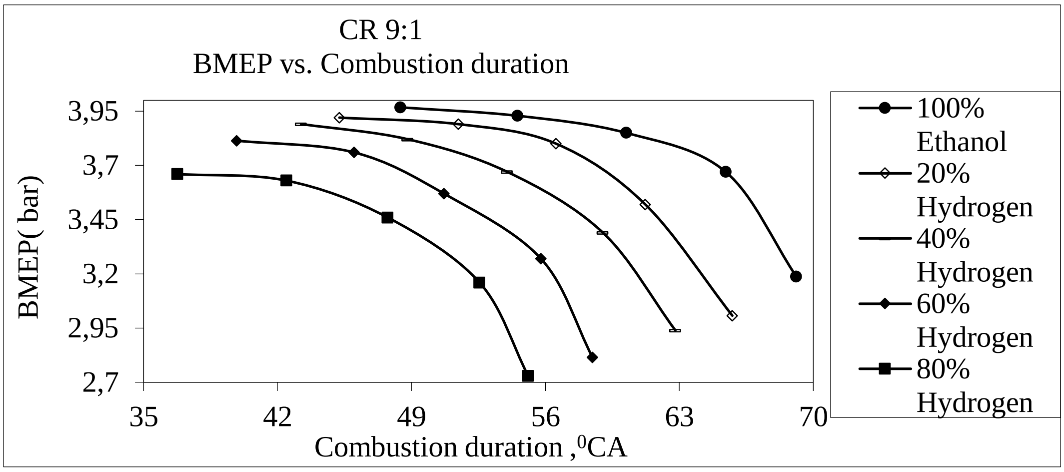


Figure 10 –Effect of Combustion duration on BMEP for various percentages ofhydrogen substitutions at 9:1 CR and 1500 rpm.

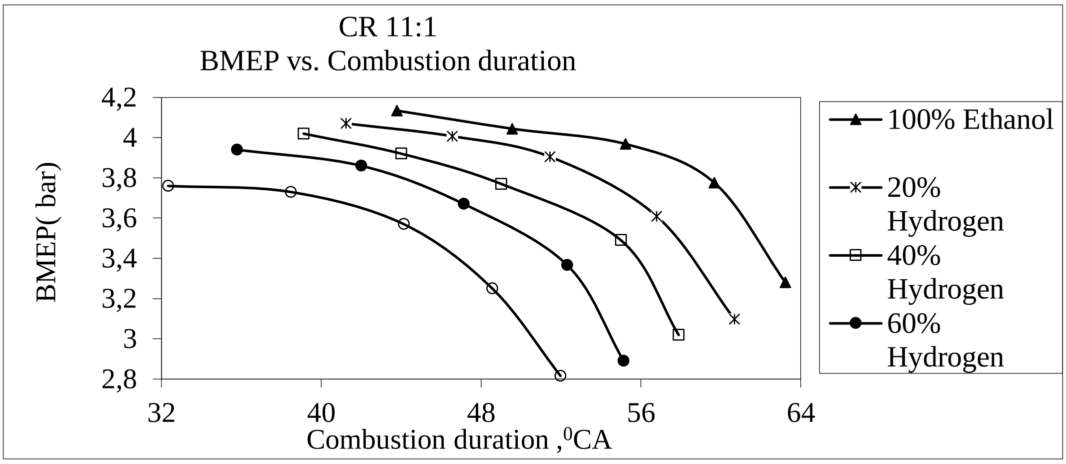


Figure 11–Effect of Combustion duration on BMEP for various percentages of hydrogen substitutions at 11:1 CR and 1500 rpm.

4.2.2 Effect of combustion duration on brake power

On the other hand because increasing combustion duration increases the lean misfire limit therefore there is a decrease in the BP as seen from Figure 12, 13 and 14.

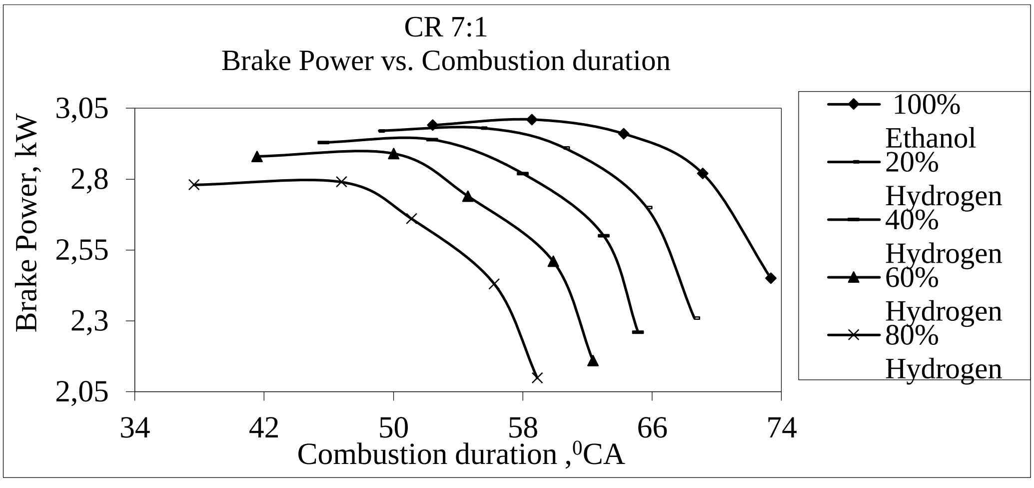


Figure 12–Effect of Combustion duration on Brake Power for various percentages of hydrogen substitutions at 7:1 CR and 1500 rpm.

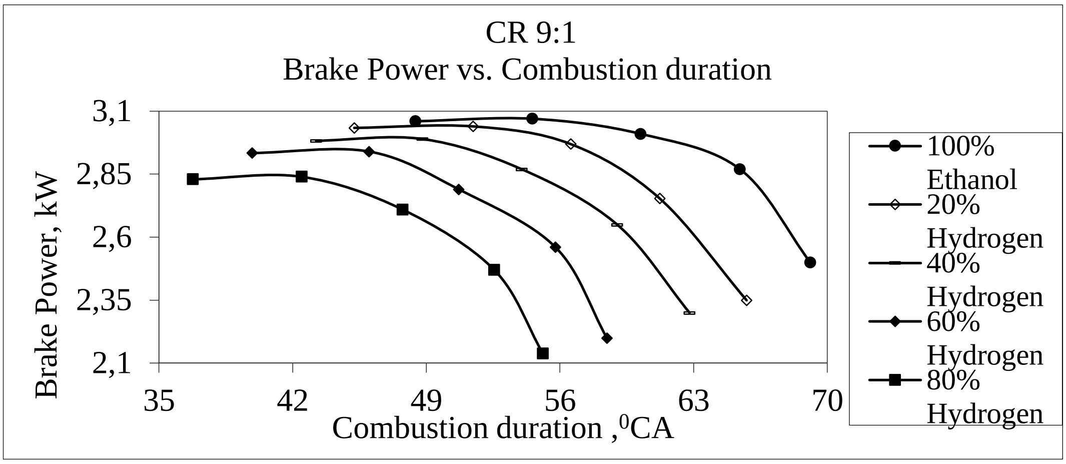


Figure 13–Effect of Combustion duration on Brake Power for various percentages ofhydrogen substitutions at 9:1 CR and 1500 rpm.

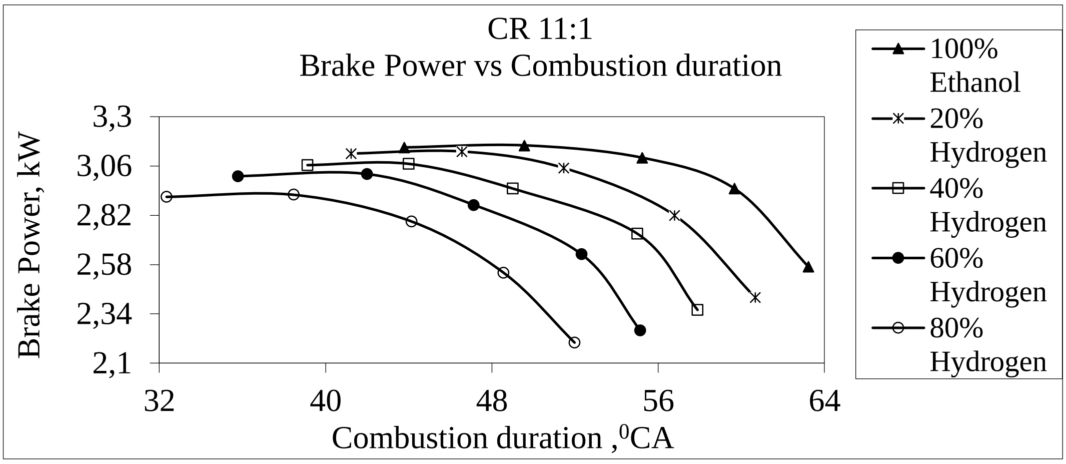


Figure 14–Effect of Combustion duration on Brake Power for various percentages of hydrogen substitutions at 11:1 CR and 1500 rpm.

4.2.3 Effect of combustion duration on brake specific fuel consumption and brake thermal efficiency

BSFC decreases to a certain limit as shown in Figure 15, 16 and 17 and there is an improvement of the brake thermal efficiency as shown in Figure 18, 19 and 20.

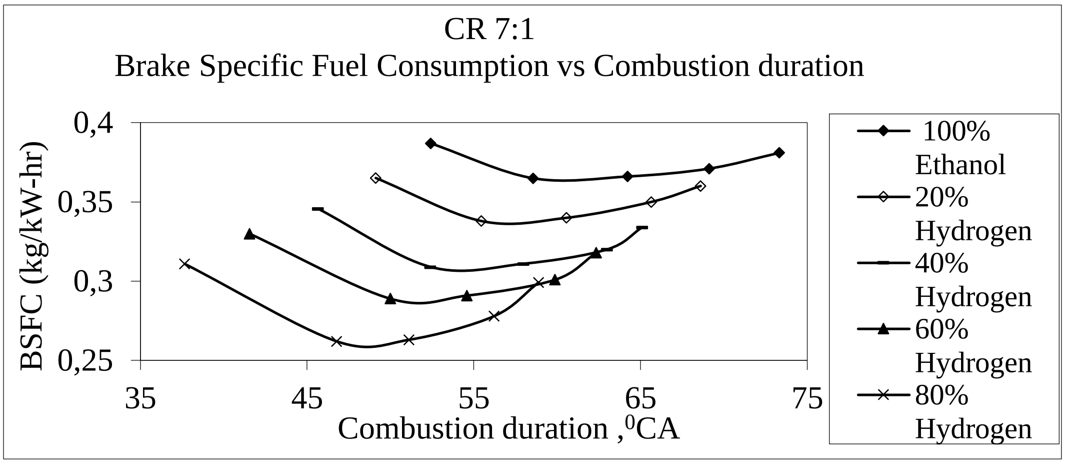


Figure 15–Effect of Combustion duration on BSFC for various percent hydrogen substitutions at 7:1 CR and 1500 rpm.

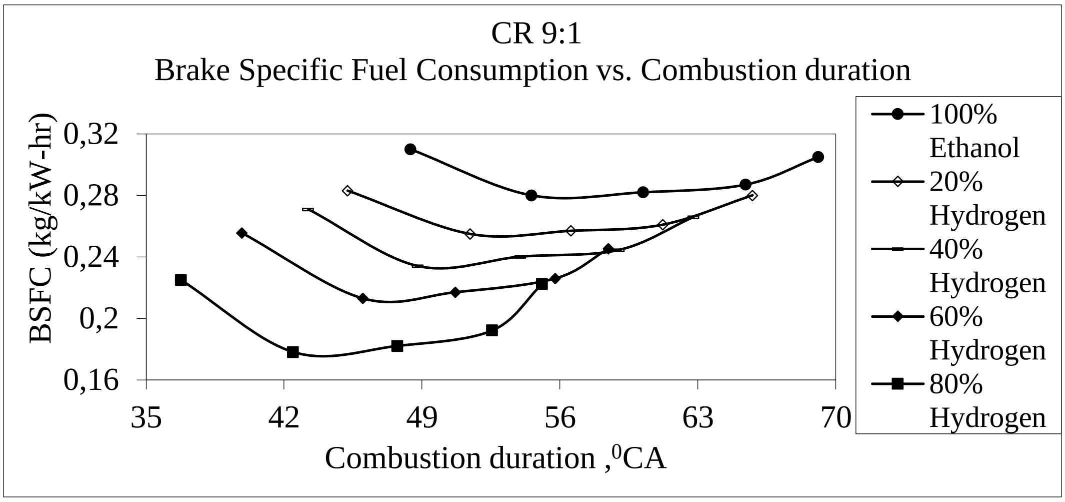


Figure 16–Effect of Combustion duration on BSFC for various percent hydrogen substitutions at 9:1 CR and 1500 rpm.

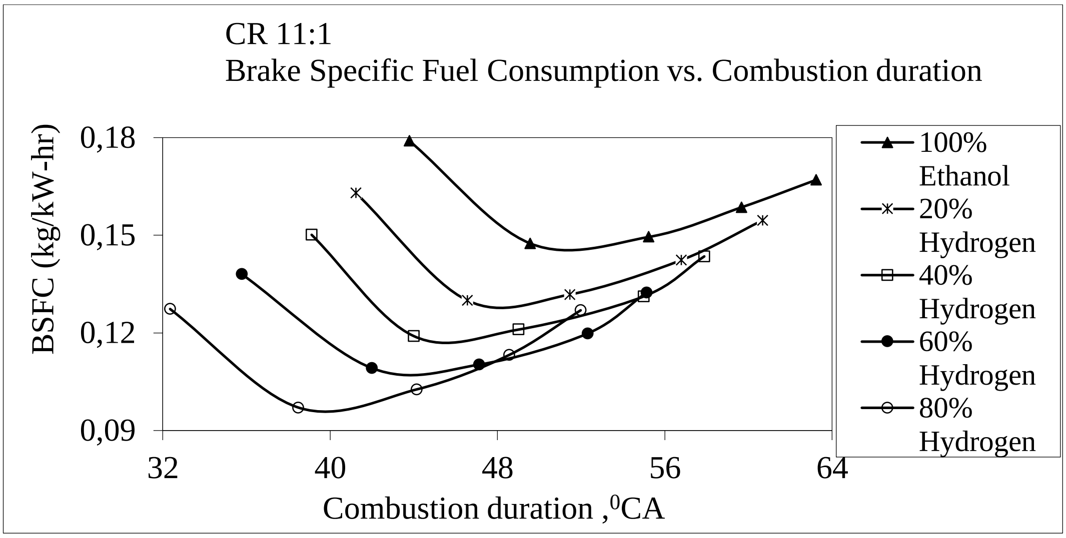


Figure 17 – Effect of Combustion duration on BSFC for various percent hydrogen substitutions at 11:1 CR and 1500 rpm.

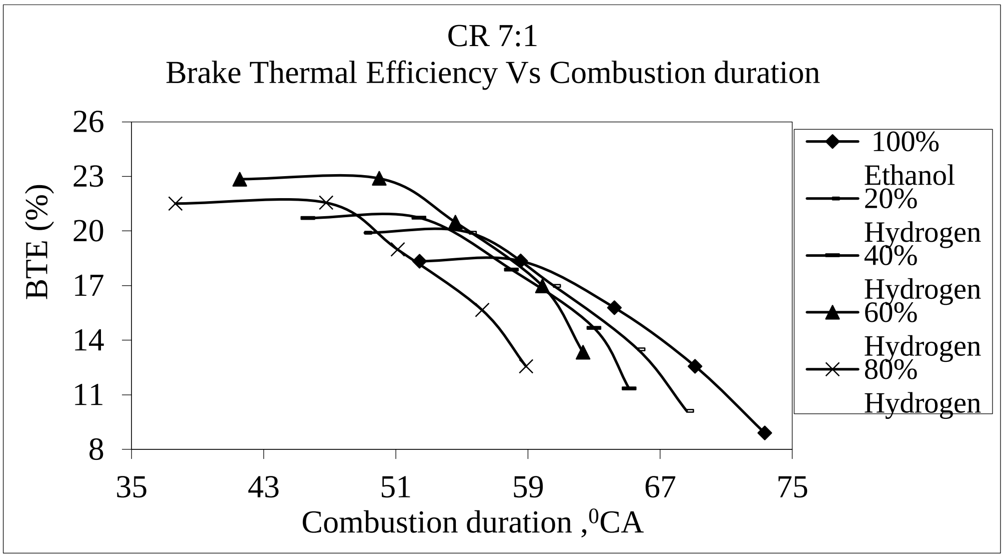


Figure 18 – Effect of Combustion duration on BTE for various percent hydrogen substitutions at 7:1 CR and 1500 rpm.

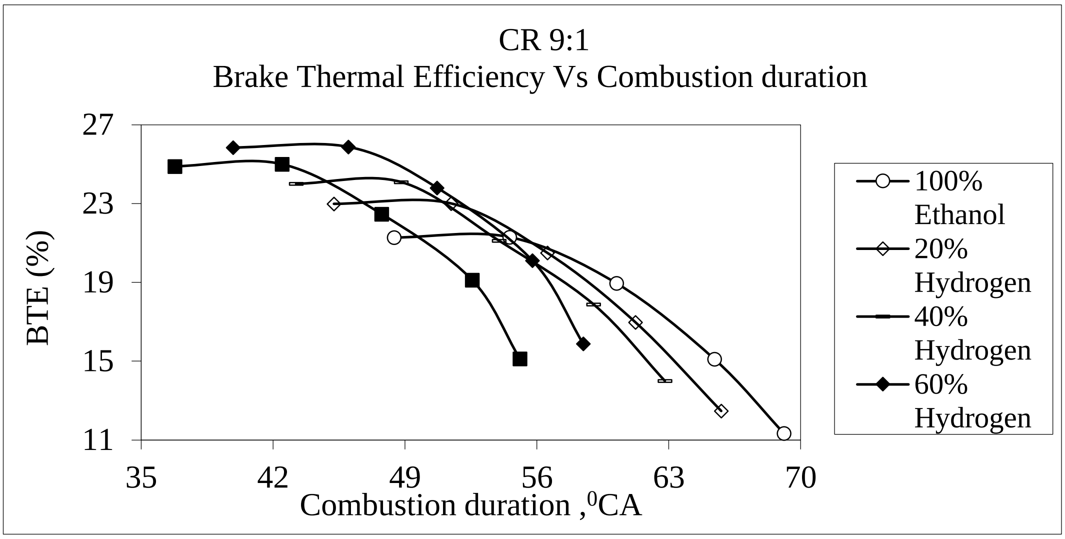


Figure 19 – Effect of Combustion duration on BTE for various percent hydrogensubstitutions at 9:1 CR and 1500 rpm.

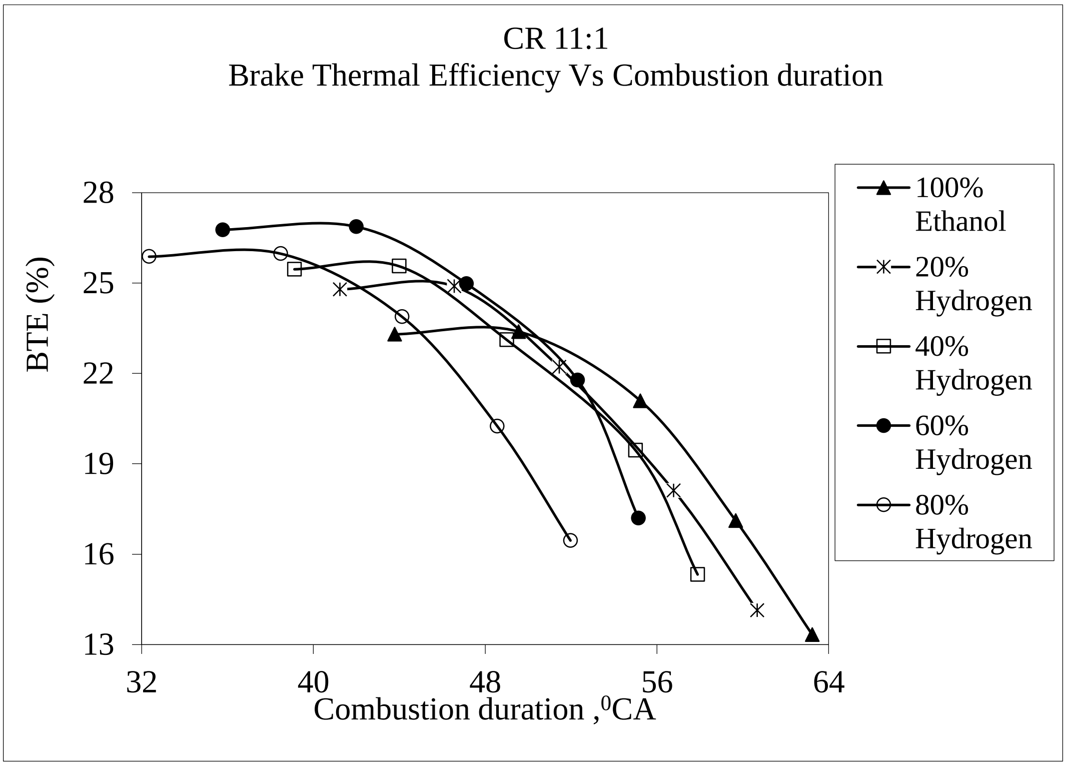


Figure 20 – Effect of Combustion duration on BTE for various percent hydrogensubstitutions at 11:1 CR and 1500 rpm.

Therefore, it has become clear that any attempt to increase the combustion duration by either reducing the compression ratio, or operating at leaner mixtures is going to improve the engine's economy but is going to reduce the power output. Further, examination of the Figure 12, 13 and 14 leads to a conclusion that for better performance in terms of power and economy the combustion duration has to be in between 35 to 42$^0$ All hydrogen substitutions showed the maximum increase in brake thermal efficiency and reduction in BSFC value at around 25$^0$ CA advanced ignition timing.

## 5. Conclusions

- The combustion duration decreases as the compression ratio increases with increasing substitution of hydrogen to ethanol because of the increase in the end of compression temperature and pressure.

- The percentage reduction in combustion duration in degrees for 100% ethanol was 30.98% for an increase of compression ratio from 7:1 to 11:1, whereas the average reduction for 80% hydrogen under identical conditions was 17.8%.
- For 100% ethanol with the increase in compression ratio the combustion duration decreases with an increase in equivalence ratio. The average decrease in combustion duration for pure ethanol is 3.25% when the CR is changed from 7 to 11.
- At a given equivalence ratio the combustion duration is shortened as hydrogen fraction increased and if the mixture is made progressively rich the combustion duration decreases.
- With an increase in the combustion duration period there is a decrease in brake mean effective pressure because of the increased heat losses. For 80% hydrogen at CR of 7 the change in combustion duration was $21^0$ for a decrease in BMEP value by 23.3%. On the other hand, for CR 11 the increase in combustion duration was $15^0$ for a decrease in BMEP value by 29.3%.
- Increasing combustion duration decreases the BP, BSFC to certain limit and increases the brake thermal efficiency.
- Any attempt to increase the combustion duration by either reducing the compression ratio or operating at leaner mixtures is going to improve the engine's economy but is going to reduce the power output.
- The best operating conditions were obtained at a compression ratio of 11:1 and the optimum fuel combination was found to be 60–80% hydrogen substitution to ethanol.
- For better performance in terms of power and economy, the combustion duration has to be in between 35 to $42^0$.
- Combustion duration has a significant effect on both performance and operating characteristics of the engine and has to be carefully designed for to achieve the best engine results.
- Engines operating parameters have to be carefully chosen by the designer taking into account their effect on the combustion duration.

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