

# Effect of Compression Ratio and Equivalence Ratio on the Emission Characteristics of a Hydrogen-Ethanol Fuelled Spark Ignition Engine

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## Abstract

*This paper discusses the results obtained from experimental investigations to evaluate the exhaust emission (i.e. CO, HC and NO<sub>x</sub>) characteristics of a hydrogen-ethanol dual fuel combination of 80 percent hydrogen and 20 percent ethanol to a spark ignition engine at the three different compression ratios of 7:1, 9:1 and 11:1, at varying equivalence ratios and an engine speed of 1500 rpm. The benefit of substituting 80% of hydrogen with ethanol deriving from the present study is a reduction of HC and CO emissions with increasing compression ratios. This however, is accompanied by a corresponding increase in NO<sub>x</sub> emissions with increasing compression ratios.*

**Keywords:** Compression ratio, Emissions, Ethanol, Hydrogen, Spark ignition

**Notations:** CO : Carbon Monoxide; HC : Hydrocarbon; NO<sub>x</sub> : Nitrogen Oxides

## 1. Introduction

Alternative fuels, hydrogen and alcohols are 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 [1]. The high heat of vaporization of Ethanol fuel gives rise to a reduction in the peak temperature inside the cylinder and hence reduced the NO<sub>x</sub> emissions and increased engine power [2]. Ethanol is a likely alternative automotive fuel in that it has properties that would allow its use in present combustion engines with minor modifications. As a fuel for spark-ignition engines, ethanol has some advantages over gasoline, such as better engine anti-knock characteristics and reduction of

CO and UHC emissions as highlighted from the work of Bang Quan et. al. in [3]. A small amount of hydrogen mixed with gasoline and air produces a combustible mixture, which can be burned in a conventional spark ignition, internal combustion engine at an equivalence ratio below the lean flammability limit of a gasoline-air mixture. The operation of engines on lean fuel mixtures has a number of positive features. It can, in principle, provide high thermal efficiency, low likelihood of an engine knock, reduced emissions especially of NO<sub>x</sub> and permits use of higher compression ratios while simultaneously reducing wasteful heat transfer away from the combustion cylinder. The resulting ultra-lean combustion of ethanol and

gasoline produces a low flame temperature that leads directly to lower heat transfer to the walls, higher engine efficiency and lower exhaust of CO and NO<sub>x</sub> [4].

The use of hydrogen as a primary or supplementary automotive fuel, is likely to give rise to a reduction of emissions in a spark ignition (SI), internal combustion (IC) engine. Besides being the cleanest burning chemical fuel, hydrogen can be produced from water (and is thus a non-fossil fuel) and, conversely, on combustion forms water again by a closed cycle [5]. Thus from the above literature it is clear that among the various alternative fuels, hydrogen and alcohols are attractive substances for many practical applications in the energy sector.

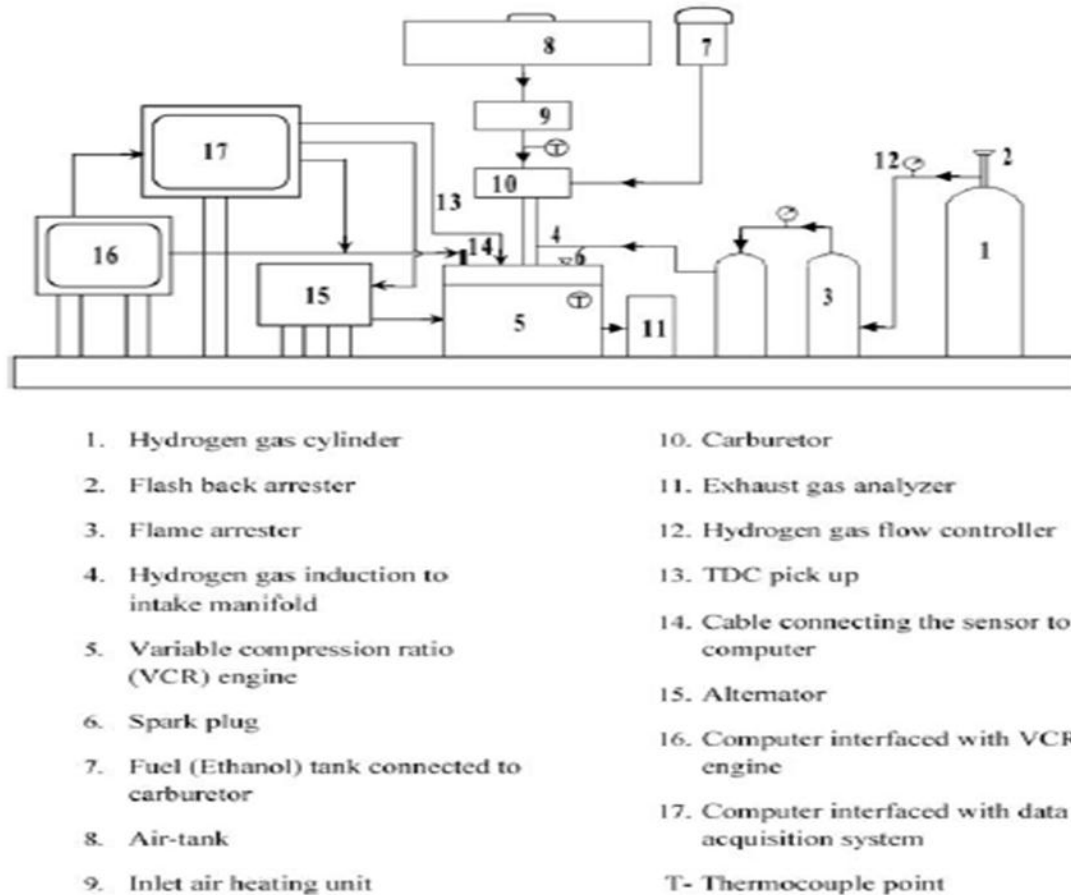
As not much work has been done previously on the emission characteristics of hydrogen-ethanol dual fuel internal combustion engines, this study attempts to fill this gap by investigating the variation of the emission characteristics of SI engine fuelled with an 80% hydrogen and 20% ethanol mixture, at the three different compression ratios of 7:1, 9:1 and 11:1, for different equivalence ratios.

## 2. Experimental Work

The engine 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 (Figure 1), 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. The technical details 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.

**Table 1. Technical Details of the Engine**

Table 1 – Technical details of the engine.	
Bore (mm)	80
Stroke (mm)	110
Displacement (cm <sup>3</sup> )	552.64 cc
Compression ratio (made variable)	7:1–11:1
Ignition source	Spark plug
Combustion chamber	Disk-shaped combustion chamber (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**

## 2.1 Measurement Systems

The following measurement systems are used in the engine-testing schedule.

### 1. Speed Measurement

On the shaft projection, a reference indicator was provided to mark the TDC, which was detected by a proximity sensor. Proximity sensor is placed very close to the flywheel. The proximity RPM sensor senses the speed and displays on the digital RPM indicator. The computer interfacing of the engine helps in easy and accurate interpretation of results. The accuracy of this sensor being one revolution per minute.

### 2. Pressure Measuring System

Crank-angle-resolved in-cylinder pressure was, measured. A computer interfaced piezoelectric sensor of Range 145 bar was used to note the in-cylinder pressure. Pressure signals were obtained at one-degree crank angle intervals using a digital data acquisition system. The average pressure data from 100 consecutive cycles were used for calculating combustion parameters. Special software was used to obtain combustion parameters. From the average pressure crank angle history the net heat release rate was obtained using the first

law of thermodynamics. The transducer is connected to the computer through a logic card that converts the pressure signals to digital signals. The computer is programmed using MATLAB and 'C' language for calculating the indicated power and plot the P- $\theta$  curve for different loads and hydrogen-ethanol dual fuel combinations.

### **3. Temperature Measurement**

The test rig equipped with thermocouples in conjunction with the digital temperature indicators. K-type of thermocouples are placed at different points to note the temperatures at the inlet, exhaust of the engine, engine head, cooling water inlet, cooling water outlet, and lubricating oil temperatures etc. To read directly the temperatures are displayed on the panel board digital display system.

### **4. Hydrogen Flow Measurement**

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 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.

### **5. Ethanol Fuel Flow Rate Measurement**

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 sec.

### **6. Water-cooling System**

The test engine is a water-cooled vertical engine. A good water circulation system is provided to cope with the cooling requirements of the engine, transducer and the supply to the exhaust gas calorimeter. The transducer and the engine require uninterrupted water supply for satisfactory performance. The transducer was cooled by water for accurate readings. A minimum of 3 litres/minute water supply has to be circulated through the transducer therefore a water circulating system with a ½ H.P motor and VRCS pump is provided to maintain the water flow.

### **7. Emission Measurement**

Engine exhausts emissions were measured using an advanced AVL five-gas analyzer (Figure2) which is a non-dispersive infrared gas analyzer. The sample to be evaluated is passed through a cold trap to condense the water vapors that influences the functioning of the infrared analyzer. The exhaust gas analyzer is calibrated periodically using standard calibration gas. The hydrocarbons and NO<sub>x</sub> are measured in terms of parts per million (ppm) as hexane equivalent and carbon monoxide emissions are measured in terms of percentage volume.



**Figure 2. AVL Five Gas Analyzer**

### **8. Dynamometer**

The engine is coupled to an AC alternator with a flexible tire coupling. Resistive type of loading is used. Operation of loading switches applies the load on the alternator in steps of 0.5 kW, which in turn applies load on the engine. An electrical wattmeter, voltmeter and an ammeter are provided to know the electrical current output.

### **9. Indicated Power Calculations**

Special software was prepared and used to calculate the indicated power by directly interpreting the inside cylinder pressures. This software runs a program, which constructs the P-V plot and by the trapezoidal rule calculates the area under the curve of the P-V plot and directly displays the indicated power values.

### **10. Pressure, Volume, and Combustion Data**

The special software stores the data of pressures and volumes corresponding to a particular crank angle location for plotting the P-V and P- $\theta$  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.

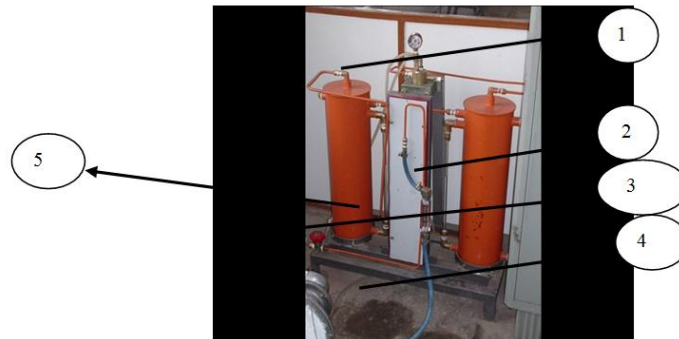
### **11. Hydrogen Storage System**

The Hydrogen is stored in the compressed gaseous form at a pressure of 150 bars in the commercially available cylinders with a pressure indicator to read the inside pressure. With a pressure reduction valve its pressure reduced to atmospheric before inducting into the inlet manifold. The outlet pressure indicated in a separate pressure gauge.

### **12. Flash Back Arrestor & Flame Trap**

Since hydrogen is a colorless, odorless gas and burns without flame. The flame sometimes travels along the hose, enters into the storage cylinder and proves to be hazardous. In order to have safe operations a safety device called flash arrestor is used. This prevents the flame from entering into the cylinder. This is installed near the gas cylinder. Along with the flash back

arrestor a Flame trap is also used in the hydrogen supply line. It consists of two inter-connected stainless steel cylinders half filled with water. The hydrogen has to pass through these traps before reaching the engine manifold. In case of any accidental flame generation in the supply hose, once it reaches the flame trap the flame is put off by the trap water and in case if the flame survives in the trap and tries to reach the cylinder the flashback arrestor (Figure3) in the supply line near the cylinder disconnects the hydrogen supply. The flash back arrestor senses the high-pressure pulse that gets generated upon the fire and immediately disconnects the hydrogen supply by closing its inner passage which is pressure sensitive.

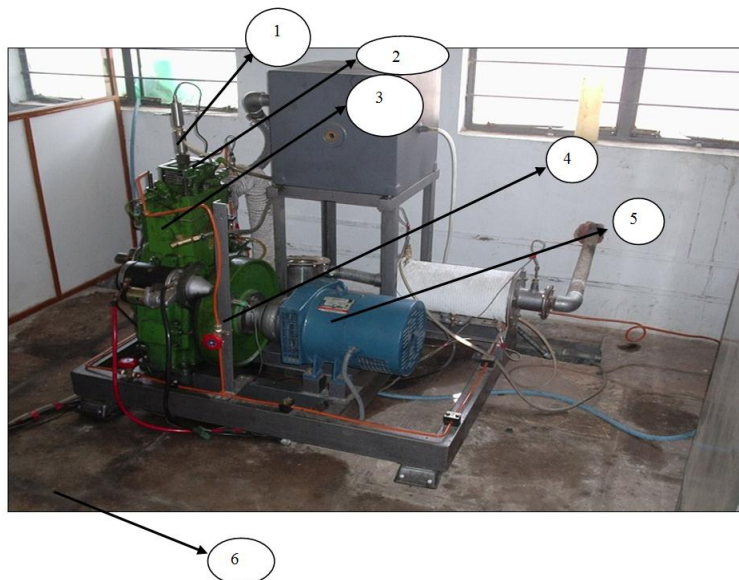


**Figure 3. Flame Trap with Flow Controller**

(1. Flow controller; 2.Flame trap; 3. Supply connection to engine from flame trap;  
4. Supply connection from hydrogen cylinder; 5. Flow meter)

## 2.2 Lay-out of the Experimental Setup

The photographic view of the experimental setup is shown in Figures 4 and 5.



**Figure 4: Experimental Setup**

(1. Pressure transducer ; 2. Water supply to pressure transducer; 3. CNG injector;  
4. Hydrogen supply to engine; 5. Arrangement for Bio-fuel heating; 6. Cable  
connections to computer)



Figure 5. Computer Interfaced VCR with Control Panel

### 3. Results and Discussion

HC Vs Equivalence Ratio (80% Hydrogen)

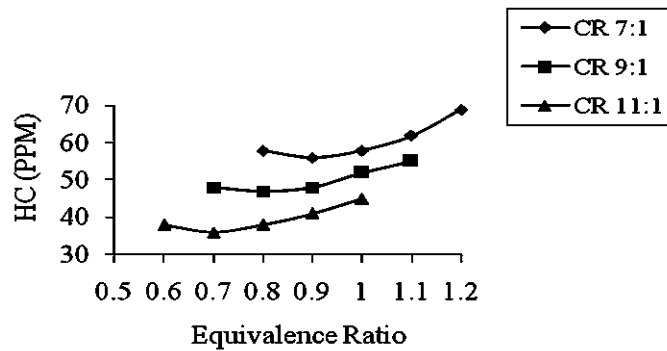
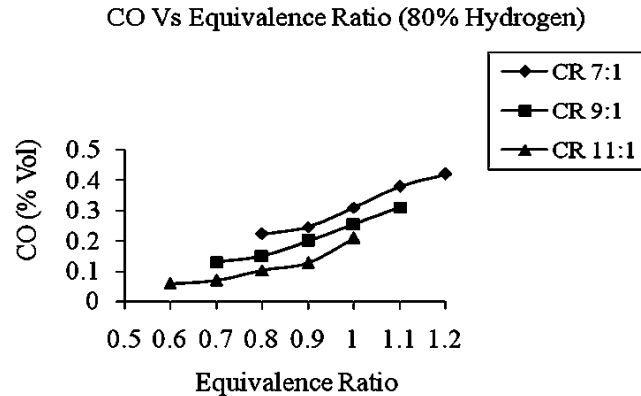


Figure 6. Variation of HC emissions with equivalence ratio for 80% hydrogen substitution at 7:1, 9:1, 11:1 compression ratios

Figure 6 demonstrates the various HC emission levels for 80% hydrogen enrichment of ethanol. It can be observed that the minimum HC emission for the three compression ratios

used in testing occurred at an equivalence ratio of about 0.7 and for a compression ratio of 11:1, and further increase of the equivalence ratio beyond this value resulted in a continuous increase of the HC emitted. As seen from the figure, the highest HC emission for the three compression ratios used in testing occurred at an equivalence ratio of about 1.2 for a compression ratio of 7:1. At equivalence ratio of 1.0, the percentage reduction in HC emission noticed when the compression ratio was increased from 7 to 11 was around 22.4%.

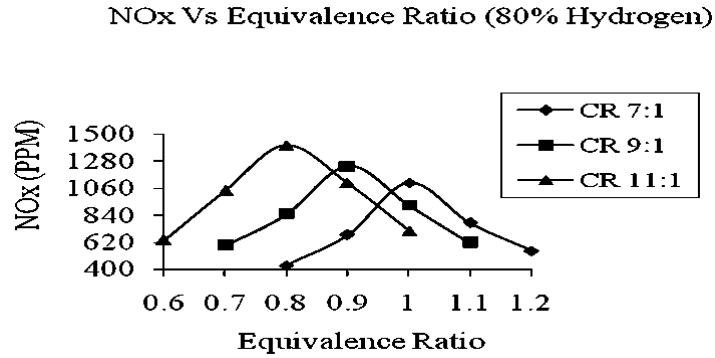


**Figure 7. Variation of CO emissions with equivalence ratio for 80% hydrogen enrichment of ethanol at 7:1, 9:1, 11:1 compression ratios**

Figure 7 shows the variation of various CO emission levels with equivalence ratio for 80% hydrogen enrichment of ethanol. It can be observed that the lowest CO emission for the three compression ratios used in testing occurred at an equivalence ratio of about 0.6 at a CR of 11:1, and that further increase in the CR beyond this value resulted in sustained increase of CO emission. As seen from the figure the highest CO emission for the three CRs used in testing occurred at an equivalence ratio of about 1.2 and for a compression ratio of 7:1. At an equivalence ratio of 1.0 the percentage reduction in CO emissions when the compression ratio was increased from 7:1 to 11:1 was around 32.58%. It is seen from Figure 7 that the CO emission increased steadily with increasing equivalence ratio, for all three CRs. With fuel lean mixtures CO emission varies little with equivalence ratio: It is interesting to note that at an equivalence ratio 0.8 the CO level is increased. This is the consequence of low temperatures freezing the oxidation. One other observation is that, the CO levels were nearly independent of the fuel used and depended almost exclusively on the equivalence ratio. This is to be expected if nearly complete combustion takes place. Alcohols are expected to produce a slightly lower CO level in combustion, when compared to petrol, because of a lower ratio of carbon in their fuel molecules, and their more favorable dissociation properties. The HC and CO emissions seen in the Figures 6 and 7 are extremely low as expected in an engine using hydrogen-ethanol as a fuel. This is because the hydrogen fuel has no hydrocarbons, and the fact that HC emissions arise mainly from unburned fuels. The absence of hydrocarbons in hydrogen fuel also keeps the CO emissions very low. Ideally, all of the carbon in the hydrocarbons should be converted to  $\text{CO}_2$  in complete combustion. Incomplete combustion on the other hand leads to the generation of some CO. The reason for the presence of some CO and HC emissions in the engine fuelled with hydrogen-ethanol is due to combustion of the lubricating oil in the engine. The oil is not intended for combustion, and there are ways of minimizing its ingress into the combustion chamber. The oil can make its way into the combustion chamber past the piston rings, through leakage at the intake valve guide, or through the crankcase ventilation system. Some solutions to minimize these highlighted



problems include: ensuring a smaller piston ring gap, ensuring the sustenance of a higher piston ring to cylinder wall pressure; improving the valve guide seals to prevent oil from getting past the guides; and improving the oil trap for the crankcase ventilation system. Using synthetic engine lubricating oils has been said to decrease the emissions. These oils are not made with hydrocarbons, and therefore do not contribute to those emissions. They also are slicker, decreasing the internal friction in the engine.



**Figure 8. Variation of NO<sub>x</sub> with equivalence ratio for 80% hydrogen enrichment of ethanol at 7:1, 9:1 and 11:1 compression ratios**

As shown in Figure 8, NO<sub>x</sub> emission reaches maximum values at equivalence ratios of 1.0, 0.9, and 0.8 for 7:1, 9:1, and 11:1 compression ratios, respectively. In the lower equivalence ratio range, NO<sub>x</sub> concentration is negligibly small. These trends can be explained by the fact that NO formation reactions depend upon temperature in the combustion chamber, mixture strength, and available oxygen, and they occur primarily in the post flame gases. The type of the fuel used affects the flame temperatures and the sufficiency of the available oxygen is affected by the stoichiometry, which is in turn a function of the type of fuels used. As the mixture air-fuel ratio gets leaner, the temperature prevalent in the combustion chamber drops thus leading to a weakening of the NO<sub>x</sub> formation kinetics. As the compression ratio increases, it is observed in Figure 8 that the peak NO<sub>x</sub> emission occurs at equivalence ratios that are leaner. At higher compression ratios, the charge condition at the start of combustion would be more homogeneous and this helps in shifting the peak NO<sub>x</sub> occurrence points to the leaner side. At equivalence ratio of 1.0 the percentage increase in NO<sub>x</sub> emissions noted when the compression ratio was increased from 7 to 11 was around 27.8%. For equivalence ratios below 0.8, NO formation is restricted due to thermal quenching during the formation process. For mixtures richer than 0.8, thermal dissociation of NO is the limiting factor. As seen from Figure 8, variation in NO<sub>x</sub> concentration levels was a function of the equivalence ratio for all compression ratios. It increased for all compression ratios initially with increasing equivalence ratios, reached a peak value, and declined with increasing equivalence ratio thereafter. As seen in Figure 8, peak NO<sub>x</sub> emissions were produced at equivalence ratios that were less than unity, dropping off on either side of this peak. For the same mixture strength, NO<sub>x</sub> emission is expected to be slightly higher for higher engine speed because of reduced effective heat loss. Therefore, it is apparent that the chemical interactions controlling formation of oxides of nitrogen in hydrogen-ethanol fuelled engine are the same as in other combustion processes. Thus, engines operated on hydrogen-ethanol dual fuel using near stoichiometric mixtures can be expected to produce substantial NO<sub>x</sub>, emissions quite similar to those obtained with other fuels. Well-known control techniques such as spark retardation, exhaust gas recirculation, and water addition are effective in controlling NO<sub>x</sub>, emissions and

are recommended for use in the Hydrogen/ethanol fuelled Internal combustion SI engine. In addition, the extremely low value of the lean flammability limit of hydrogen-air mixtures can be used to reduce NO<sub>x</sub> emissions to almost negligible amounts by restricting operation to equivalence ratios less than 0.7. Direct cylinder fuel injection may be employed to boost power without sacrificing emissions control at these lean mixtures. Ethanol contains an oxygen atom in its basic form; it therefore can be treated as a partially oxidized hydrocarbon. When ethanol is added to the hydrogen fuel, it can provide more oxygen for the combustion process and leads to the so-called “leaning effect”. Owing to the leaning effect, CO and HC emissions are expected to decrease tremendously; and so NO<sub>x</sub> emission, under the operating conditions mentioned above.

#### 4. Conclusion

The important conclusions, which emerged from the experimental investigations conducted in this work to evaluate the exhaust emissions (i.e. CO, HC and NO<sub>x</sub>) characteristics of a single cylinder internal combustion diesel engine modified into an SI internal combustion engine, fuelled with 80% of hydrogen enrichment of ethanol include:

1. At an equivalence ratio of 1.0 the percentage reduction in HC emission when the compression ratio was increased from 7:1 to 11:1 was around 22.4%.
2. At an equivalence ratio of 1.0 the percentage reduction in CO emission when the compression ratio was increased from 7:1 to 11:1 was around 32.58%.
3. As the compression ratio increased, the peak NO<sub>x</sub> emission occurred at lower equivalence ratios and therefore leaner mixtures.
4. At an equivalence ratio of 1.0 the percentage increase in NO<sub>x</sub> emission when the compression ratio was increased from 7:1 to 11:1 was around 27.8%.
5. The important improvement of 80% hydrogen enrichment of ethanol is the reduction of the HC and CO emissions with a slight increase in NO<sub>x</sub> emissions, while increasing the higher useful compression ratio of a hydrogen supplemented fuel engine.

#### Acknowledgement

The authors sincerely acknowledge “*The All India Council for Technical Education*” (AICTE), New Delhi, India for the financial support in carrying out this research work.

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