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A Study of Spark Ignition Engine Fueled with Methanol and Ethanol Fuel Blends with Iso-Octane

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Abstract

Alternative fuels are derived from resources other than petroleum. The benefit of these fuels is that they emit less air pollutant compare to gasoline and most of them are more economically beneficial compared to oil and they are renewable. In addition, ethanol has higher evaporation heat, octane number and flammability temperature therefore it has positive influence on engine performance and reduces exhaust emissions. In this study, the effects of unleaded iso-octane, unleaded isooctane–ethanol blend (E5) and iso-octane-methanol (M5) blends on engine performance are investigated experimentally in a single cylinder four-stroke spark-ignition engine at a constant 8 Kg load. The engine speed was changed from 1100 to 1800 rpm. The results of the engine test showed that ethanol addition to unleaded iso-octane increases the value of IP, FP and IMEP with E5 fuel. The results also showed that the indicated power, brake power, friction power, indicated mean effective pressure, torque, exhaust temperature, and thermal efficiency increases with the increase in engine speed at a constant load of 8 Kg for E5, M5 and iso-octane fuels. Thermal efficiency was maximum for E5 fuel (38.13%) at a speed of 1750 rpm.

Keywords: *Thermal efficiency, volumetric efficiency, exhaust temperature, indicated power, bsfc, engine speed*

1. Introduction

Ethanol and methanol have been used as a fuel for engines since 19th century. Among the various alcohols, ethanol and methanol are known as the most suited renewable, bio-based and eco friendly fuel for spark-ignition (SI) engines. The most attractive properties of ethanol as an SI engine fuel are that it can be produced from renewable energy sources such as sugar, cane, cassava, many types of waste biomass materials, corn and barley. Ethanol and methanol are alcohol- based fuels made by fermenting and distilling starch crops, such as corn. Both ethanol and methanol produce less emission than gasoline. In Brazil, ethanol is well known as a clean, economic and available fuel for vehicles. But engines work on alcoholic fuels will experience a decrement in brake torque and power compared to gasoline.

Increasing air pollution is one of the most important problems of developed countries today. Exhaust emissions from motor vehicles has a main role in this pollution. It is not sufficient to change the design of motor to cope with the legal regulations, so it is necessary to continue to work on alternative fuel technologies. The engine thermal efficiency can be improved with increasing of compression ratio. Alcohols burns with lower flame temperatures and luminosity owing to decreasing the peak temperature inside the cylinder. So that the heat losses and NO_x emissions are lower. Both methanol and ethanol have high latent heat of vaporization. The latent heat cools the intake air, so the increased charge density and increases volumetric efficiency. However the oxygen content of methanol and ethanol reduces their heating value compared to gasoline. As a disadvantage for methanol and ethanol which reduce the vehicle range per liter of fuel tank capacity [1-5].

Hsieh [6] investigated experimentally the engine performance and emission of a spark ignition engine, using ethanol–gasoline blend fuels in ratios of 5%, 10%, 20% and 30%. The results showed that with increasing the ethanol rate, the heating value of the blended fuel decreased, while the octane number of the blended fuels increased. By using the ethanol–gasoline blended fuels, the engine torque and specific fuel consumption slightly increased.

Abdel-Rahman and Osman [7] conducted performance tests on a variable compression ratio engine using different percentages of ethanol in gasoline fuel up to 40%. With increasing of ethanol amount in the blend, the octane number also rise up, but decrease the heating value. The power increment was observed with the addition of the ethanol up to 10% at compression ratio of 10:1. The best compression ratios were

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found to be 10, 11 and 12 for 20%, 30% and 40% ethanol to give maximum indicated power, respectively.

Palmar [8] found that the octane number had an increment of five and the engine output increased 5% for every 10% ethanol addition to gasoline.

Badwan [9] studied blended fuels ranging from E10 to E70 and concluded that the highest antiknock capability was obtained with E50. One major objective of using ethanol gasoline blended fuel is its ability to lower the emissions of CO and UHC.

In the experimental study of Al-Hasan [10], the effects of usage of unleaded gasoline–ethanol blends on spark ignition engine performance and exhaust emission were investigated. The results showed that ethanol addition leads to an increase in brake power, brake thermal efficiency, volumetric efficiency and fuel consumption by about 8.3%, 9%, 7% and 5.7% mean average values, respectively. The best result at the engine performance and exhaust emissions was obtained by usage of 20% ethanol fuel blend.

In Bayraktar [11], the most suitable blend for SI engines, from the engine performance and CO emissions points of view, was determined to be 16.5% theoretically and 7.5% experimentally. CO emission was reduced while NO emission was found to increase due to the rising cylinder temperature.

Taylor [12] compared the performance of four alcohols. They found little difference in combustion efficiency of the four alcohols from gasoline. However, using alcohol can increase charge density because of the evaporative cooling in the intake manifold.

W. Dai [13] did the study of engine cycle simulation of ethanol gasoline blends. In this study, an ethanol model has been developed using a Ford propriety engine CAE tool, GESIM (General Engine Simulation Program) for the simulation of ethanol and ethanol – gasoline blends. GESIM was then validated against experimental data in a 3.0 L V6 2-valve engine. It was concluded that GESIM has successfully predicted the trends of engine burn rates, fuel consumption, exhaust temperature, and various exhaust emissions for E22 and E85 fuels.

C. Wei [14] did the study of air – fuel ratio on engine performance and pollutant emission of an SI engine using ethanol – gasoline blended fuels. It was concluded that torque output increases slightly at small throttle valve opening when ethanol - gasoline blended fuel was used and CO and HC emission were reduced with the increase of ethanol content in the blended fuel. It was also found that CO_2 emission per unit horse power output was similar or less than that for gasoline fuel.

H. S. Yucesu [15] did comparative study of mathematical and experimental analysis of spark ignition engine performance used ethanol – gasoline blend fuel. In this study, ethanol-unleaded gasoline blends (E10, E20, E40, and E60) were tested in a single cylinder, four stroke spark ignition and fuel injection engine by varying the ignition timing, relative air fuel ratio and compression ratio at a constant speed of 2000 rpm and wide open throttle. It was concluded that torque with blended fuels was higher than that of base gasoline in all the speed range and a significant reduction in HC emissions was observed as a result of the leaning effect and additional fuel oxygen caused by the ethanol addition. It was suggested that higher compression ratios can be used with ethanol gasoline blends without knock.

L. Shenghua [16] did the study of spark ignition engine fuelled with methanol gasoline blends. The engine was three cylinders with a bore of 68.5 mm. The methanol was blended with gasoline containing 10, 15, 20, 25, and 30% in volume. Engine power and torque decreases while the brake thermal efficiency is improved. The maximum pressure is higher than that of pure gasoline operation under the same engine speed and throttle opening (50% WOT) when engine fuelled with M20.

M. KOc [17] evaluated the effects of ethanol – unleaded gasoline blends on engine performance and exhaust emissions in a spark- ignition engine. In this study, the effects of unleaded gasoline and unleaded gasoline ethanol blends (E50 and E85) on engine performance and pollutant emissions were investigated experimentally in a single cylinder four- stroke spark ignition engine at two compression ratios (10.1 and 11.1) at varying engine speed 1500 rpm to 5000 rpm. It was concluded that ethanol addition to unleaded gasoline increase the engine torque, power and fuel consumption and reduce carbon monoxide (CO), nitrogen oxides (NO_X) and hydrocarbon (HC) emissions.

M. Eyidogan [18] In this study, the effects of ethanol–gasoline (E5, E10) and methanol–gasoline (M5, M10) fuel blends on the performance and combustion characteristics of a spark ignition (SI) engine were investigated. The results indicated that when alcohol–gasoline fuel blends were used, the brake specific fuel consumption increased; cylinder gas pressure started to rise later than gasoline fuel. Almost in the all test conditions, the lowest peak heat release rate was obtained from the gasoline fuel use.

R. H. Chen [19] evaluated gasoline displacement and NO_x reduction in an SI engine by aqueous alcohol injection. In this experiment, two engine speeds (2000 and 3000 rpm), three throttle openings (40%, 60% and 80%) and three purities of alcohols (99.7%, 75%, and 50%) were utilized to test the engine performance and emissions. The author suggested that engine ran normally with a gasoline – alcohol fuel spray containing upto 30% ethanol and 16% water.

D.R. Prajapati et al [20] studied the effect of blended fuels on specific fuel consumption at varying engine loads using CVCRM engine test rig. The experimental results shows that out of the two mustard oil blends, 20-PRM shows the lowest fuel consumption at the engine loads of 2.5 Kg and 5.0Kg, whereas 15-PRM shows the lowest specific fuel consumption at the engine loads of 7.5 Kg.

Z.M.Hasib et al [21] evaluated Performance characteristic analysis of small diesel engines fuelled with different blends (B20, B30, B50) of mustard oil bio-diesel. A comparison of engine performance for different blends was carried out under different operating conditions.

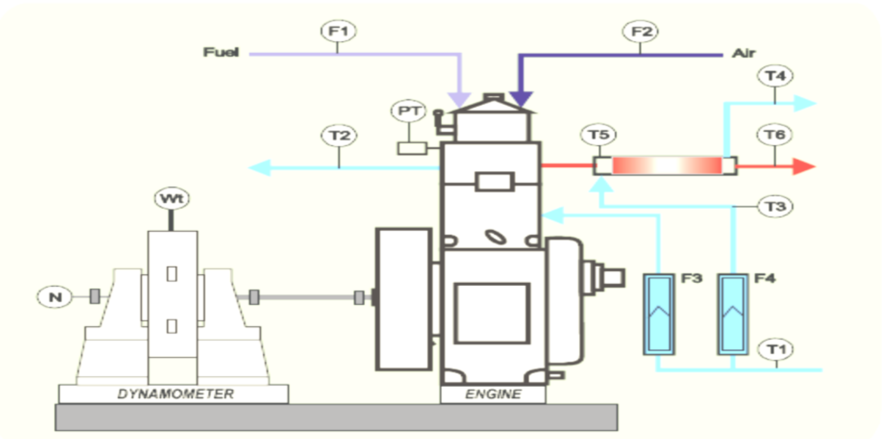
The literature survey shows that using ethanol and methanol in SI engines by blending with iso-octane is more practical than using it alone. If ethanol and methanol production can meet the demand and cost of blended fuels can compete with that of conventional iso-octane, widespread use of E5 and M5 blends can be possible. For this reason, the present study is focused on this topic. Here, the effects of ethanol and methanol addition to iso-octane in 5% concentrations on engine performance are examined by conducting experimental studies. This study is not carried out earlier in one paper under the similar conditions.

Table 1 Comparison of fuel properties

Property	Methanol	Ethanol	Iso-octane
Chemical formula	CH_3OH	C_2H_5OH	C_8H_{18}
Molecular weight(Kg/kmol)	32.04	46.07	114.228
Oxyzen present (wt %)	49.9	34.8	-
Density (g cm^{-1})	792	789	740
Freezing point at 1 atm (^{0}C)	-97.778	-80.0	-107.378
Boiling temperature at 1 atm (^{0}C)	64.9	74.4	99.224
Auto-ignition temperature(^{0}C)	463.889	422.778	257.23
Latent heat of vaporization at 20^{0}C (KJ/Kg)	1103	840	349
Stoichiometric air/fuel ratio (AFR)	6.47	9.0	15.2
Lower heating value of the fuel (KJ/Kg)	20000	26900	44300
Research octane number (RON)	111	108	100
Motor octane number (MON)	92	92	100

2. Experimental setup

The experiments were conducted at four different engine speeds ranging between 1100 rpm to 1800 rpm at a constant load of 8 kg at different throttle openings. At each of these engine speeds, three different fuels were tested. These fuels were iso-octane, iso-octane ethanol blend (E5) and iso-octane-methanol blend (M5), the numbers following E and M indicate percentage of volumetric amount of ethanol, methanol. Properties of ethanol, methanol, and gasoline are shown in table1. Engine indicated power, brake power, friction power, indicated mean effective pressure, brake mean effective pressure, torque, brake specific fuel consumption, exhaust gas temperature, volumetric efficiency and thermal efficiency were measured during the experiments. No data was taken until speed and load were maintained at 1% of the fluctuation. For each experiment, air fuel ratio was changed to maintain the same speed and load. The specifications of engines are shown in table 2.The setup consists of single cylinder, four stroke, Multi-fuel, research engine connected to eddy current type dynamometer for loading. The operation mode of the engine can be changed from diesel to Petrol or from Petrol to Diesel with some necessary changes. In both modes the compression ratio can be varied without stopping the engine and without altering the combustion chamber geometry by specially designed tilting cylinder block arrangement. The injection point and spark point can be changed for research tests. Setup is provided with necessary instruments for combustion pressure, Diesel line pressure and crank-angle measurements. These signals are interfaced with computer for pressure crank-angle diagrams. Instruments are provided to interface airflow, fuel flow, temperatures and load measurements. The set up has stand-alone panel box consisting of air box, two fuel tanks for duel fuel test, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and hardware interface. Rotameters are provided for cooling water and calorimeter water flow measurement. A battery, starter and battery charger is provided for engine electric start arrangement. The setup enables study of VCR engine performance for brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, Mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio, heat balance and combustion analysis. Lab view based Engine Performance Analysis software package "Enginesoft" is provided for on line performance evaluation.



F1	Fuel consumption kg/hr	F2	Air consumption kg/hr
F4	Calorimeter water flow kg/hr	T3	Calorimeter water inlet temperature ºK
T4	Calorimeter water outlet temperature ºK	T5	Exhaust gas to calorimeter inlet temperature. ºK
T6	Exhaust gas from calorimeter outlet temperature. ºK		

Fig.1 Schematic arrangement of the test engine

Table 2 Specification of the engine

Type	1 cylinder, 4 stroke, water cooled
Cylinder bore and stroke	110 mm, 87.5 mm
Compression ratio	10
Maximum power	4.5 kW at 1800 rpm
Spark variation range	0 – 70 deg btdc
Dynamometer	Eddy current, water cooled, with loading unit
Air Box	M S fabricated with orifice meter and manometer
Fuel tank	Capacity 15 ltr, dual compartment, with fuel metering pipe of glass
Calorimeter	Pipe in pipe
Piezo sensor	Combustion 5000 psi
Crank angle sensor	1 deg, speed 5500 rpm with tdc pulse
Temperature sensor	Type RTD, PT 100 and Thermocouple, type k
Load indicator	Digital, range 0-50 kg, supply 230 V AC
Fuel flow transmitter	DP transmitter, Range 0- 500 mm WC
Air Flow Transmitter	Pressure transmitter
Rotameter	Engine cooling 40- 400 LPH; Calorimeter 25-250 LPH

3. Results and Discussions

3.1 Indicated power (IP)

Under the same engine speed and load, see in figure 2, when the engine is fuelled with different fuels (E5, M5, & iso-octane). The indicated power of oxygenated fuel is higher than that of pure iso-octane. It is maximum for E5 and minimum for iso-octane. It is further observed that maximum increment is 6.72% at low speed to minimum increment 1.73% at 1400 rpm for E5. It is because of the reason that oxygenated fuels have better combustion efficiency and E5 fuel have higher lower heating value than M5 fuel. It is also shown in figure that indicated power increases with the increase in engine speed at the same load for all the tested fuels and it is because of the reason that when engine speed increases, less quantity of heat flows through the cylinder wall which results in higher indicated power.

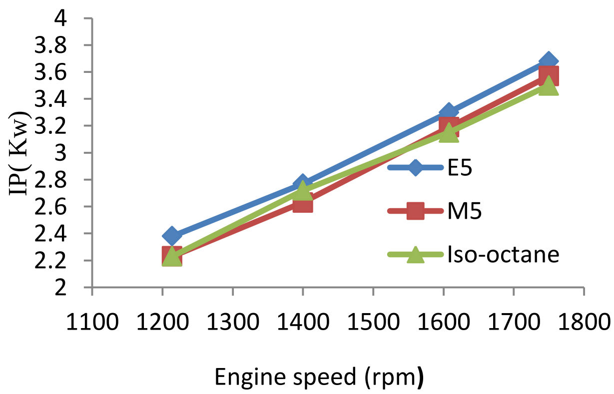


Fig. 2 Effect of engine speed on IP (Kw) at constant load (8 Kg)

3.2 Brake power (BP)

Figure 3 shows the variation in brake power with respect to engine speed at the same load for all the tested fuels. It is shown in the figure that brake power increases with the increase in engine speed for all the tested fuels and it is because of the reason that indicated power increases so the brake power increases. E5 fuel has highest IP and highest friction power so its brake power reduces

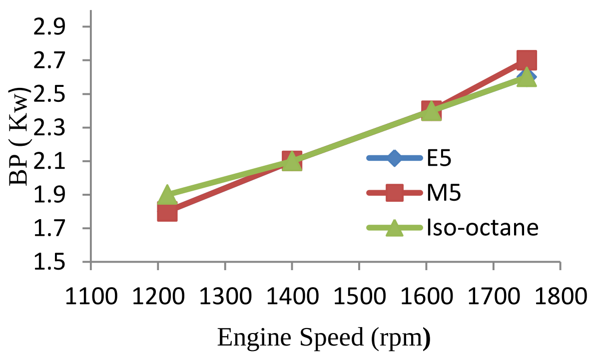


Fig. 3 Effect of engine speed on BP at constant load (8 Kg)

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3.3 Friction power

Figure 4 shows the Friction power includes pumping work, rubbing friction work and accessory work. The gross indicated mean indicated pressure is obtained from $\int P\,dv$ over the compression and expansion process for a four stroke engines. By subtracting the brake mean effective pressure from indicated mean effective pressure, rubbing friction and auxiliary frictions are obtained.

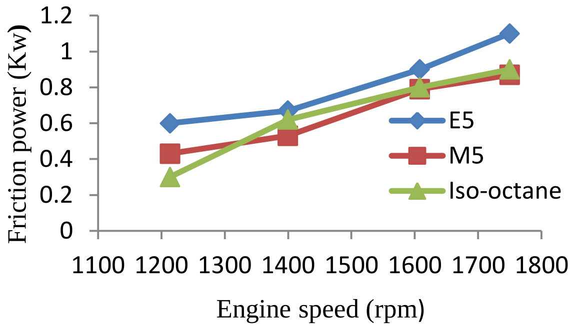

Fig. 4 Effect of engine speed on friction power at constant load (8 Kg)

The total friction mean effective pressure is given by the correlation

tfmep = 0.97 + 0.15 (N/1000) + 0.05(N/1000)^2

The first term represents boundary friction which is independent of speed, the second term represents hydrodynamic friction and third term represents turbulent dissipation.

The effect of ethanol and methanol addition (5% in volume) iso-octane on friction power between 1200 rpm to 1800 rpm engine speed is shown in figure 4 at a constant load of 8 kg. The friction power increases with the increase in engine speed and It is because of the reason that as engine speed increases, the number of cycles per second increases which results in more frictional power. Friction power is maximum for the E5 and minimum for the M5.

3.4 Indicated mean effective pressure (IMEP)

Figure 5 shows the variation in indicated mean effective pressure with the change in engine speed at the same load. Figure shows that IMEP increases with the increase in engine speed for all the tested fuels. IMEP is maximum for the E5 fuel than the iso-octane and this increment is 12.5, 2.8, 2.7 and 5.5% at 1200, 1400, 1600 and 1750 rpm, respectively. It is because of the reason that indicated power is maximum for the E5.

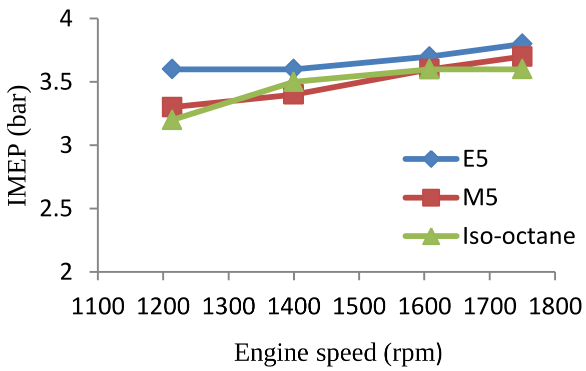

Fig.5 Effect of speed at IMEP at constant load (8 Kg)

3.5 Brake mean effective pressure (BMEP)

In naturally aspirated engines, brake mean effective pressure is not stress limited, it then reflects the product of volumetric efficiency(ability to induct air), fuel air ratio (effectiveness of air utilization in combustion and fuel conversion efficiency). In supercharged engines BMEP indicates the degree of success in handling higher gas pressures and thermal loading. Figure 6 shows the variations in BMEP with respect to engine speed at the same load. The variation in BMEP is between 2.7 and 2.8 for all the tested fuels.

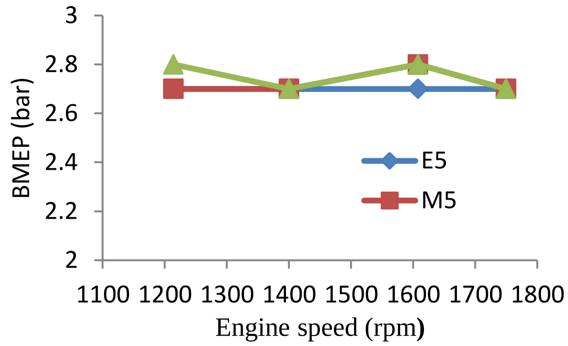

Fig. 6 Effect of speed on BMEP at constant load (8 Kg)

3.6 Torque

Figure 7 shows the variations in torque with the change in speed for the E5, M5 and iso-octane at constant load of 8 kg. The maximum torque is for the iso-octane and minimum for the E5 but at the 1600 rpm, maximum torque is for the methanol, followed by ethanol and minimum for iso-octane. It is also observed that torque increases with the increase in engine speed. This is explained with several reasons. Beneficial effect of ethanol as an oxygenated fuel is a possible reason for more complete combustion, thereby increasing the torque. In addition, a larger fuel for the same volume is injected to the cylinder due to higher density of ethanol. This results in increase in torque and power. And finally, the latent heat of evaporation of blended fuels is higher than that of base gasoline; this provides lower temperature intake manifold and increases volumetric efficiency. The charge into the cylinder directly affects on torque and power. The average increment in engine torque compared with E0 was about 1.39% with M5and 0.7% with E5 at compression ratio of 10:1.

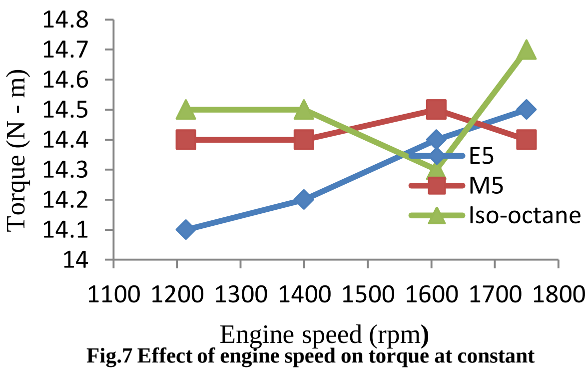

Fig.7 Effect of engine speed on torque at constant load (8 kg)

3.7 Exhaust temperature

Figure 8 shows the variations of exhaust gas temperature corresponding to various tested fuels at a constant load of 8 kg. It is shown in the figure that exhaust temperature increases with the increase in engine speed for all the tested fuels. This is explained with several reasons. With the increase in engine speed, combustion gases gets less time to remain in contact with cylinder wall and therefore more energy is released with exhaust gases which increases the temperature of exhaust gases. The highest temperature is observed with the M5 and lowest for the E5. These variations in exhaust temperature can be attributed to increase in thermal efficiency or A/F ratio which affects the combustion temperature. Higher combustion temperature leads to higher exhaust temperature.

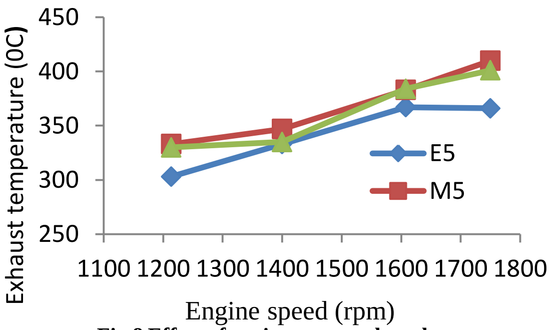

Fig.8 Effect of engine rpm on the exhaust temperature at constant load(8Kg)

3.8 Volumetric efficiency

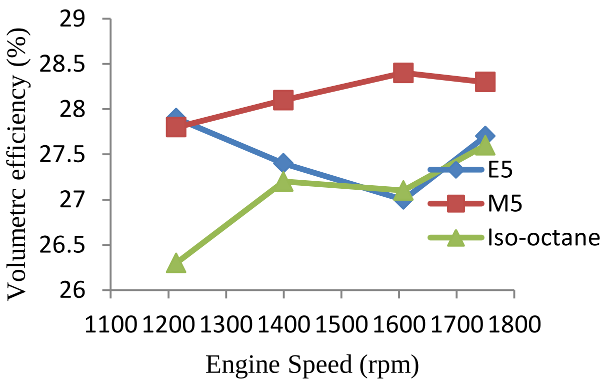

Fig.9 Effect of speed on volumetric efficiency at constant load (8 Kg)

Figure 9 represents the change in volumetric efficiency with the change in engine speed at the constant load of 8 kg for all tested fuels. It is shown in the figure that volumetric efficiency of M5 is maximum among all the fuels and it is because of the reasons that as liqid fuels have high latent heat of vaporization, they produce a cooling effect on the intake charge during vaporization. Therefore, there will be an increase in intake charge density and consequently in volumetric efficiency. A/F ratio is another important parameter that affects volumetric efficiency. When stoichiometric A/F ratio is high that means there is more quantity of air injected in inlet air and results in increased volumetric efficiency. The methanol has highest latent heat of vaporization and A/F ratio is high therefore volumetric efficiency of methanol is maximum.

3.9 Brake specific fuel consumption (BSFC)

Figure 10 shows the variations in brake specific fuel consumption with respect to engine speed at constant load of 8 kg. It is shown from the figure that ethanol addition causes 15.1% and 39.3% increment in BSFC with E5 at 1400 and 1600 rpm. It is well known fact that heating value of fuel affects the BSFC. The lower energy content of ethanol-iso-octane and methanol iso-octane fuels causes some increment in BSFC of the engine when it is used without any modification. The increment mainly depends upon the percentage of ethanol and methanol addition in iso-octane. The heating value of ethanol & methanol are approximately 39.27% & 54.85% less than the values of iso-octane. Therefore, more blends of fuel are required to produce the same power at the same operating conditions due to its lower heating value in comparison to base fuel. As a result, BSFC increases. The BSFC is lower at higher speed because of high thermal efficiency.

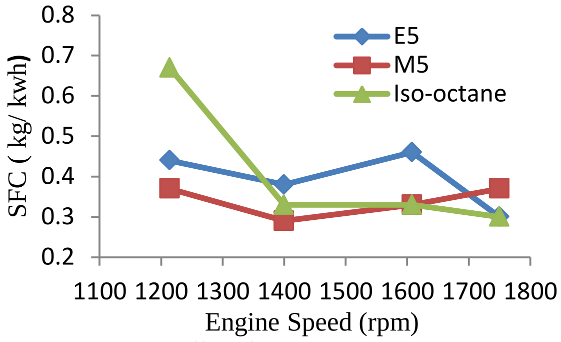

Fig.10 Effect of engine speed on SFC at constant load (8 kg)

3.10 Thermal efficiency

Figure 11 shows the effect of engine speed on the thermal efficiency at a constant load of 8 Kg. It is shown in the figure that thermal efficiency increases with the increase in engine speed and it is because of the reason that less quantity of heat is being lost through the cylinder wall. The thermal efficiency variations are (23.1 to 38.13%) for E5, (18.63 to 36.12%) for M5, and (14.3 to 35.9%) for iso-octane. Maximum thermal efficiency was obtained for E5 at a engine speed of 1750 rpm. It is interesting to note that thermal efficiency of M5 fuel is 35% higher than the base fuel and nearly 5% higher than the E5 fuel at a speed of 1400 rpm. It is because of the reason that M5 fuel contains nearly 50% more oxyzen than the base fuel and 5% more oxyzen than the E5 fuel and fuel was supplied at the stoichiometric condition.

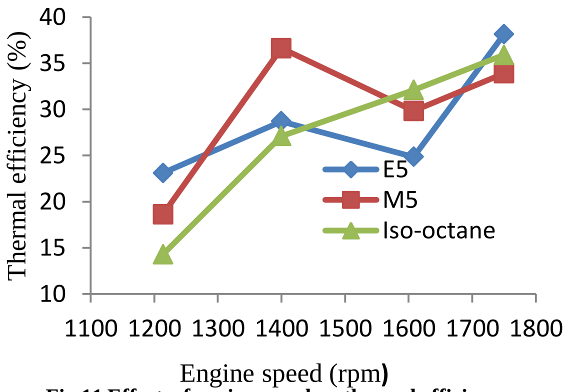

Fig.11 Effect of engine speed on thermal efficiency at constant load of 8 Kg

4. Conclusion

In this paper, the effect of engine speed on the indicated power, brake power, friction power, indicated mean effective pressure, BMEP, torque, exhaust temperature, volumetric efficiency ,BSFC, and thermal efficiency is experimentally evaluated at a constant load of 8 Kg for E5, M5 and iso-octane fuels. The following conclusions have been drawn from this study.

(1) The indicated power, brake power, friction power, indicated mean effective pressure, torque, exhaust temperature and thermal efficiency increases with the increase in engine speed at a constant load of 8 Kg for E5, M5 and iso-octane fuels.

(2) The value of IP, FP, IMEP, BSFC was higher for E5 fuel among all the tested fuels in all engine speed range.

(3) The value of volumetric efficiency and exhaust temperature was higher for M5 fuel among all the tested fuels.

(4) The value of thermal efficiency was higher for E5 fuel at a speed of 1750 rpm.

(5) The value of IP, IMEP and torque was minimum for the iso-octane.

(6) The value of friction power and BSFC was minimum for the M5.

(7) The value of torque and exhaust temperature was minimum for E5.

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