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EXPERIMENTAL STUDY ON HYDROUS METHANOL FUELLED HCCI ENGINE USING AIR PRE-HEATER ASSISTED CONTROLLED AUTOIGNITION

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Summary

The present study investigates the performance and emission characteristics of a hydrous methanol fuelled homogeneous charge compression ignition (HCCI) engine using the air preheater assisted controlled auto-ignition method. The HCCI engine has the ability to ignite all kinds of engine fuels irrespective of their octane and cetane number and it has a great potential to reduce NO_x and smoke emissions. In this study, a regular diesel engine has been modified to work as an HCCI engine. Hydrous methanol with 15% water is used in this HCCI engine and its performance and emission characteristics are investigated and analyzed. The intake air is preheated by a heater located on the suction side of the engine. The heater helps to raise the temperature of the air which in turn raises the temperature of the fuel-air mixture up to its autoignition point. The temperature of the preheated air is decided upon the load conditions. The experiments are conducted with hydrous methanol as engine fuel to determine the operating limit, heat release rate and exhaust emissions at different load conditions. The investigation reveals that the hydrous methanol operation reduces NO_x and smoke significantly more than the direct injection CI engine. Thus, hydrous methanol with 15% water operates well in an HCCI engine without any additional operational problems.

Key words: *hydrous methanol, air pre-heater, performance, emission and combustion*

1. Introduction

The rise of industrialization and modernization leads to a steep rise in the consumption of and demand for petroleum-based fuels every year. It is also known that fuel is critical to any strategic plan for the economic development and energy security of a nation. Particularly in developing countries like India, fuel imports cause serious economic consequences in the form of budget deficits. On the other hand, the continuous addition of greenhouse gases to the atmosphere by fossil fuel combustion increases global warming. Considering these disastrous effects, many alternative fuels have been identified in the past and tested successfully in the existing engines with and without engine modifications. So far many alternative fuels have been suggested by researchers but each has one or more undesirable characteristics. This prevents the complete substitution of existing fuels with alternative fuels. However, various fuel admission techniques and engine modifications experimented on earlier give a good solution to applying a larger fraction of alternative fuels in existing engines.

In the present study, hydrous methanol with 15% water content was used in a modified engine running on the HCCI combustion mode. It has been observed that methanol has not fully replaced the existing SI and CI engine fuels. Therefore, an alternative internal combustion engine technology that has an ability to offer higher thermal efficiency and reduced vehicle emissions has been used. HCCI combines the advantages of both SI and CI engines. The regular diesel engine used for diesel fuel operation is not suitable for autoigniting all kinds of fuel. Hence, a new kind of engine that has a facility for fuel preparation, control and assistance for autoignition was prepared and fuelled by mydrous methanol. Methanol is an octane fuel having high self-ignition temperature and is not autoignitable in ordinary CI engines, even at higher compression ratios. Hence, the HCCI engine designed for using methanol fuel requires assistance and control for autoignition. Several potential control methods have been proposed to control the HCCI combustion such as varying the rate of exhaust gas recirculation (EGR), Zhao et al [1], using a variable compression ratio (VCR), Christensen et al [2] and using variable valve timing (VVT), Yang et al [3]. In addition, various methods are available to autoignite the low cetane fuel in an HCCI engine. Among these, air-pre heater assisted autoignition was used in the present study.

The intake charge temperature affects the combustion and formation of emissions via two distinct pathways. When temperature is lowered, ignition delay is extended. This leads to enhanced air/fuel premixing. Additionally, the adiabatic flame temperature of a fuel parcel which has a particular equivalence ratio decreases. Li et al [4] showed that the increase in the intake charge temperature from 31°C to 54°C increased NO_x emissions linearly from approximately 10 ppm to 50 ppm when n-heptane fuel was combusted at a fixed fuel delivery rate, engine speed and 30% EGR. This is in agreement with another HCCI work whereby NO_x increased when the inlet air temperature increased from 35°C to 80°C, Akagawa et al [5]. Both unburned HC and CO were observed to be unaffected by the intake temperature.

Experiments conducted on a high speed direct injection (HSDI) diesel engine revealed that peak soot luminosity was markedly reduced when the intake temperature decreased from 110° C to 30°C under a load condition of 3 bar of the indicated mean effective pressure (IMEP), Choi et al [6]. This was attributed to both lower soot temperatures and reduced soot formation. However, the in-cylinder soot luminosity was clearly observed even at 30°C, which indicated that complete eradication of soot formation with typical fuel injection system parameters was difficult to be achieved.

The intake charge temperature has been demonstrated to affect the IMEP produced under the HCCI combustion mode. Experimental findings indicated that regardless of the EGR rate, lowering the intake charge temperature increased the IMEP due to an increase in the in-cylinder charge mass. When the temperature decreased from 105°C to 30°C, the maximum IMEP increased from 2.7 bar to 3.7 bar.

For early injection HCCI diesel combustion, EGR is used as a means of diluting the gas mixture in an HCCI diesel engine thereby retarding the ignition timing. In a premixed charge compression ignition (PCCI) combustion system described by Kondo et al [7] a large amount of EGR (54%) was introduced to retard the ignition timing toward the TDC and improve the IMEP. In another PCI combustion system, it has been shown that high EGR rates up to 68% are used to effectively control the start of combustion (SOC), Boyarski and Reitz [8]. High EGR rates can also be used to counter combustion noise by controlling the start of combustion, Masahiko [9]. However, the drawbacks of such high rates of EGR include problems with transient response and temperature stability characteristics. Therefore, for early injection HCCI combustion, EGR should be combined with some other combustion control technology such as modification of fuel properties or adoption of some other chemical approach. In the case of late injection such as the modulated kinetics (MK) combustion system, EGR is typically utilised as a NO_x reduction measure with typical levels of approximately 40%, Wimmer and Eichlseder [10]. NO_x is reduced because of the lowering of the flame temperature due to the charge dilution and higher heat capacity of the cylinder charge when EGR is introduced. The latest study by Kumaran et al [14] used one such type of oil called turpentine in a regular DI diesel engine along with blends (60% diesel, 40% turpentine) and acetylene gas of fixed quantity supplied to the inlet manifold at a point closer to the intake valve. The temperature of the combustion chamber gases is such that air-fuel mixture autoignites at the desired crank angle position in the combustion cycle Maurya et al [15]. Suyin et al [16] investigated the diesel HCCI technology and discussed various control and performance parameters affecting HCCI. They concluded that for long term development of HCCI diesel combustion systems and superior mixture formation and control, flexible fuel injection strategies and EGR control will be the most critical approaches. Visakhamoorthy et al (2012) used a parallel computing multi-zone combustion model for stimulating operational characteristics of an n-butanol/n-heptane fuelled HCCI engine utilizing the negative valve overlap (NVO) technology. The model precisely predicted the unburned hydrocarbon emission and under-predicted NO_x emission. The model captured the trend of increasing NO_x levels with increasing the fraction of butanol. The model was also able to predict the pressure, heat release rates, and combustion phasing of all the three fuel blends tested [17].

The motivation for the present study is to operate a single cylinder engine in the hydrous methanol fuelled HCCI mode in a wide load range by using air preheater assisted controlled autoignition. In this study, the effect of the air preheater temperature on the performance, emissions and operating load range of the hydrous methanol fuelled HCCI mode at a constant speed of 1500 rev/min is investigated. Finally, the hydrous methanol mass flow rate for the best brake thermal efficiency point at each load condition is found.

2. Experimental setup

A 3.7 kW single-cylinder, water-cooled, direct injection CI engine was modified to operate in the HCCI mode. An eddy current dynamometer was coupled to the engine to absorb the power produced by the engine. The specification of the engine is shown in Table 1. The engine used in the test rig is a regular diesel engine. The manifold at the inlet side of the engine has two electronic fuel injectors.

An anti-pulsating drum and an air temperature indicator are also provided at the suction side. The outlet side of the engine consists of an exhaust gas temperature indicator, a gas analyzer and a smoke sampler. The fuel used, hydrous methanol, was stored in separate fuel tanks and was injected into the intake manifold by using two separate electronic fuel injectors. The test setup was equipped with a diesel consumption measuring facility. The cylinder pressure data and crank angle data were acquired by a 16 bit data acquisition system. The schematic of the experimental setup is shown in Figure 1.

Table 1 Specification of the engine

Make and Model	Kirloskar, AVI
General details	four stroke, compression ignition, constant speed, vertical, air-cooled, direct injection
Number of cylinders	one
Bore	87.5 mm
Stroke	110 mm

Cubic capacity	661 cc
Compression ratio	17.5:1
Rated speed	1500 rpm
Rated output	4.4 kW@ 1500 rpm
Fuel injection timing	23^{o} before TDC
Diesel injector opening pressure	180 bar
Type of combustion chamber	hemispherical open combustion chamber

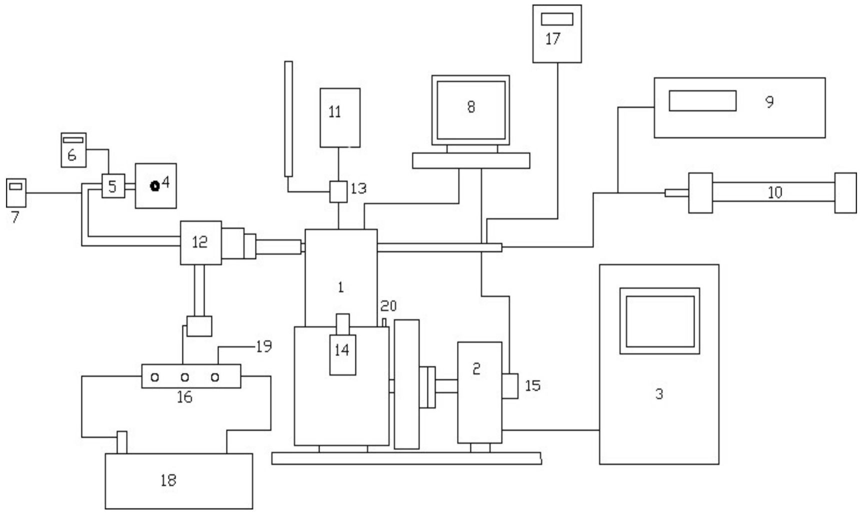

Fig. 1 Schematic of the experimental setup
1. Diesel engine, 2. Eddy current dynamometer, 3. Dynamometer control, 4. Anti-pulsating drum, 5. Air pre-heater, 6. Watt meter, 7. Inlet temperature indicator, 8. Computer with DAQ, 9. Gas analyser, 10. Smoke sampling pump, 11. Diesel tank, 12. Methanol injector, 13. Three-way cock 14. Fuel injection pump, 15.Crank angle encoder 16. Engine control module (ECM), 17. Exhaust gas temperature indicator, 18. Battery, 19. Input signal for ECM operation, 20. Inductive pickup for ECM operation

3. Methodology

This study uses hydrous methanol (15% by volume of distilled water mixed with 85% methanol by volume) in a DI diesel engine under the HCCI mode to study its feasibility, performance and emission characteristics. The engine was started by using diesel fuel and kept under various load conditions to obtain performance and emission readings. These readings were considered as base line readings and were subsequently used for comparison. The suction side of the engine - hydrous methanol dual fuel engine - consists of a hydrous methanol spray injector which slowly admits methanol into the engine and converts the sole fuel engine into diesel. A further increase in the methanol admission led to misfiring of the mixture. Once the misfire has been identified, the air heater attached to the suction side of the engine started functioning and eliminated the misfiring. Consequently, the admission of hydrous methanol increased further and the heater was also tuned up in such a way that any further misfiring is avoided. The range of the preheat temperature selected for this experiment varied from 115 ºC to 155 ºC. Temperatures higher than 155 ºC resulted in knocking and temperatures lower than 115 ºC resulted in misfiring of the fuel.

Under this condition, the premixed methanol air mixture prepared by the vaporizer is autoignited inside the engine. The heat required for autoigniting the methanol air mixture is shared by the adiabatic compression by the engine and the air heater attached to the suction side of the engine. This particular state of operation is called HCCI operation. After the HCCI operation had been achieved, the intake air temperature was optimized by considering brake thermal efficiency. That is, the intake air temperatures were varied in steps of 5^{o}C at each and every step, the BTE was also observed. The temperature at which BTE showed the highest value was considered as the best intake air temperature or the optimum intake air temperature. The same procedure was repeated for all load conditions. For each and every load step, the optimum intake temperature, hydrous methanol consumption, heat release rate, air inflow rate, emission parameters such as CO, HC, CO_2, NO_x and smoke were recorded and compared.

4. Results and discussion

This technique permits 100% diesel replacement by using hydrous methanol after slight engine modification. The factors that prevent hydrous methanol autoignition in a CI engine are its low cetane number and high self-ignition temperature. To overcome this, the in-cylinder temperature of the engine must be maintained well above the self-ignition temperature of methanol. This can be provided by an air pre-heater housed in the suction side of the engine. Hence, the air enters the engine with sufficiently high temperature to enable the autoignition of the hydrous methanol-air mixture. The heat required for the autoignition of the hydrous methanol–air fuel mixture is provided by external heating and adiabatic compression in the engine. The performance, emission and combustion characteristics of the hydrous methanol fuelled HCCI engine are discussed in the following section.

4.1 Operating range

The temperature of the intake air plays a vital role in the hydrous methanol combustion. Since methanol has higher autoignition temperature, the heat generated at the end of compression will not be sufficient to autoignite the methanol vapour. To overcome this, hot air is supplied during the suction stroke. This supply helps to raise the temperature of the mixture well above the autoignition temperature. The fixed temperature of the intake air is capable enough to support a short range of methanol energy share. To cater for a wide range of methanol share the intake temperature must be changed. High temperature was used at lower brake mean effective pressures (BMEPs) and lower temperature was used during higher BMEPs. This is mainly due to the change in combustion temperature with the change in the BMEP.

Higher BMEP produces higher combustion temperature. This is due to a higher energy share of the hydrous methanol fuel. Lower BMEP offers lower combustion temperature. This is due to a lesser energy share of the methanol fuel. Since methanol combustion in the HCCI engine depends upon the temperature of the intake air, one particular intake air temperature will not support all methanol flow rates. Hence, an operating range was determined for each and every preheat temperature. The operating range of the methanol fuel was limited by knocking and misfiring. To determine the knock limit and the misfire limit for every preheat temperature, the hydrous methanol flow rate was increased in steps and its corresponding BMEP was measured. The knock limit was measured by the rate of pressure rise and the misfire limit was measured by the coefficient of variance (COV) of the BMEP. When the rate of pressure rise (RPR) exceeded 10 MPa, a rapid change in engine noise was observed. This particular limit was defined as knock limit and when the COV exceeded 10%, an unstable operation was felt in the HCCI operation. This limit was defined as the misfire limit. Similarly, the knock limit and the misfire limit were determined for all other preheat temperatures and are shown in Figure 2. The COV and the rate of pressure rise were determined from the recorded combustion pressure and the crank angle data.

The operating range of APH at a higher temperature is shorter than the operating range of APH at a lower temperature. The reason for the shorter operating range is lower combustion temperature and leaner air-fuel mixture. Similarly, the reason for a longer operating range is higher combustion temperature.

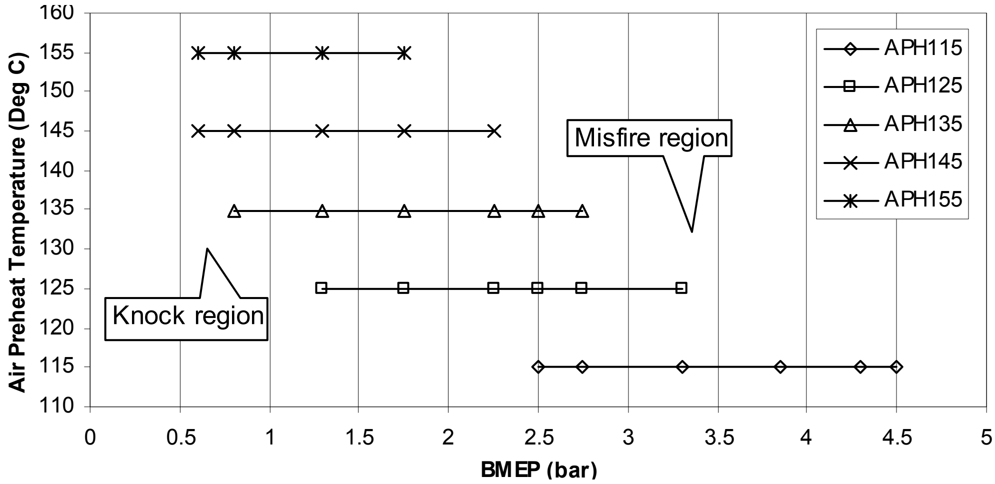

Fig. 2 Operating range of hydrous methanol HCCI engine at various APH temperatures

4.2 Energy share

Figure 3 shows the range of the methanol energy share at various intake air temperatures. A higher preheat temperature permits a wider range of the methanol energy share due to more supply of external heat through the intake air. A lower preheat temperature permits a narrower range of the methanol energy share due to less supply of external heat through the intake air. The higher preheat temperature tolerates the wider range of the methanol energy share without misfiring. This was achieved by substituting the loss of heat due to evaporation with the heat of the intake air supplied. This substitution did not yield the desired result at a wider range of the methanol energy share. Hence, the narrower range of the methanol energy share was tolerated during lower preheat temperatures.

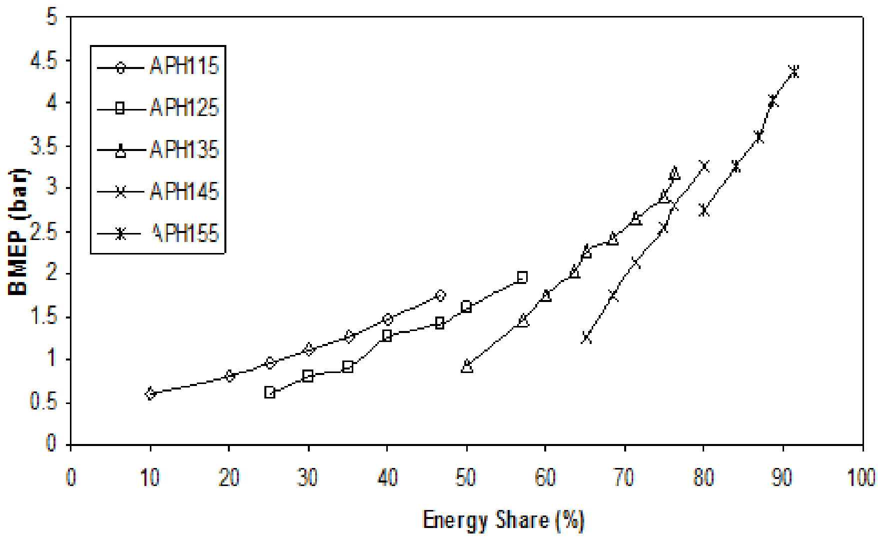

Fig. 3 Energy share of methanol at various APH temperatures

During the higher preheat operation more fraction of heat is supplied through the intake air and less fraction of heat is derived from the methanol fuel. Hence, higher preheat temperature tolerates a wider range of the methanol energy share. However, this reverses during lower preheat temperatures. During a lower preheat temperature higher energy share is derived from the methanol fuel and hence the lower preheat temperature tolerates a narrower range of the methanol energy share. In addition, during higher intake temperatures, due to the supply of more heat through the intake air, a wider range of the methanol operation is permitted. This trend reverses during lower preheat temperatures. At 0.55 bar of BMEP, the energy supplied by methanol is about 23.5% of total energy and it increases as load increases. The energy share of methanol at 4.52 bar of BMEP is around 82%.

4.3 Brake thermal efficiency

Figure 4 shows the variation of brake thermal efficiency at various intake air temperatures and methanol flow rates. The brake thermal efficiency of one particular air flow rate increases with an increase in the methanol flow rate. The efficiency reaches its maximum and then decreases. The change in the methanol flow rate changes the cylinder temperature by the evaporation and combustion. At the time of lower methanol flow rates the heat available in the hot air was more than sufficient to ignite methanol efficiently. Hence, the brake thermal efficiency increases with respect to the increase in the methanol flow rates. This trend ceases at one particular methanol flow rate and then the efficiency decreases. The reason for this trend was insufficient air temperature and reduction in peak pressure.

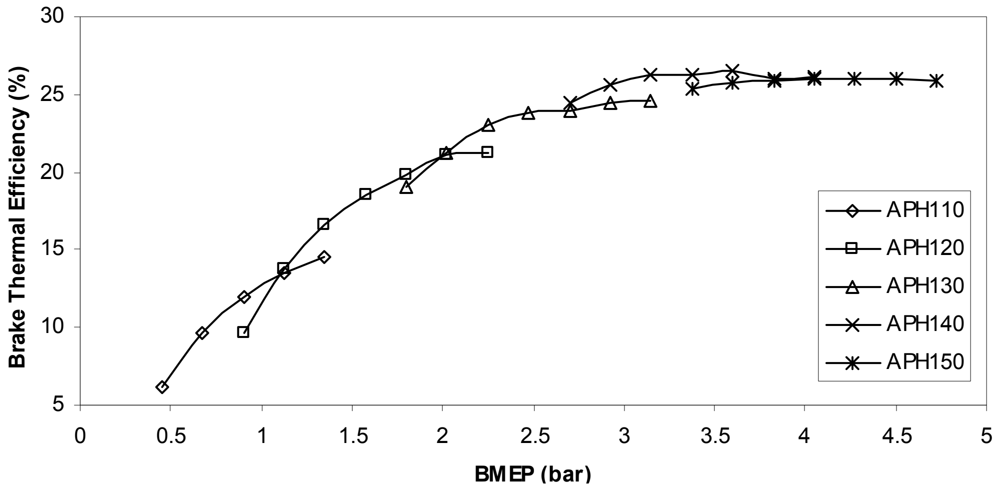

Fig. 4 Variation of brake thermal efficiency at various APH temperatures

The high latent heat of vapourization of methanol also leads to the lowering of performance at higher flow rates of methanol. The increasing trend of the brake thermal efficiency observed at the beginning is due to the rapid charge preparation and better combustion phasing. The maximum brake thermal efficiency obtained at one particular APH temperature is called optimum brake thermal efficiency.

The air preheating helps to vapourize the methanol fuel and autoignite the fuel-air mixture. In the procedure, one particular temperature is maintained constant for a range of hydrous methanol flow rates. Hence, the heat supplied through the intake air is constant for a range of methanol flow rates. This is not sufficient for higher methanol flow rates as these quantities are demanding higher heat for vapourization. Therefore, the performance at the tail end of the flow rates for one particular APH temperature is lowering.

4.4 Oxides of nitrogen emission

The production of NO_x emission is directly proportional to the combustion temperature and pressure. The combustion temperature and pressure in turn are directly proportional to the quantity of fuel used for combustion. For any particular APH temperature, if the methanol flow rates rise, NO_x emission also increases proportionally. In the procedure, higher APH temperatures are used for lower BMEPs and lower APH temperatures are used for higher BMEPs.

Figure 5 shows the variation of NO_x emissions for various APH temperatures with respect to the BMEP. It is found that for all APH temperatures the NO_x emission increases when the hydrous methanol flow rate increases. The overall trend also shows that NO_x

emission increases when the BMEP increases. This is mainly due to the changes in the combustion temperature and the change in the methanol flow rates. Lower combustion temperature produces lower NO_x emission and higher combustion temperature produces higher NO_x emission. However, the observed NO_x emission at full load of 15% of hydrous methanol fuelled HCCI engine is 67 ppm. The NO_x emission of a diesel engine at full load was 1200 ppm. It has been measured during the experiment at the engine speed of 1500 rpm. This may be due to the instantaneous and low temperature combustion in the HCCI mode. Due to the instantaneous combustion, local high temperature regions are eliminated. This elimination in turn reduces the formation of thermal NO, Christensen et al [11].

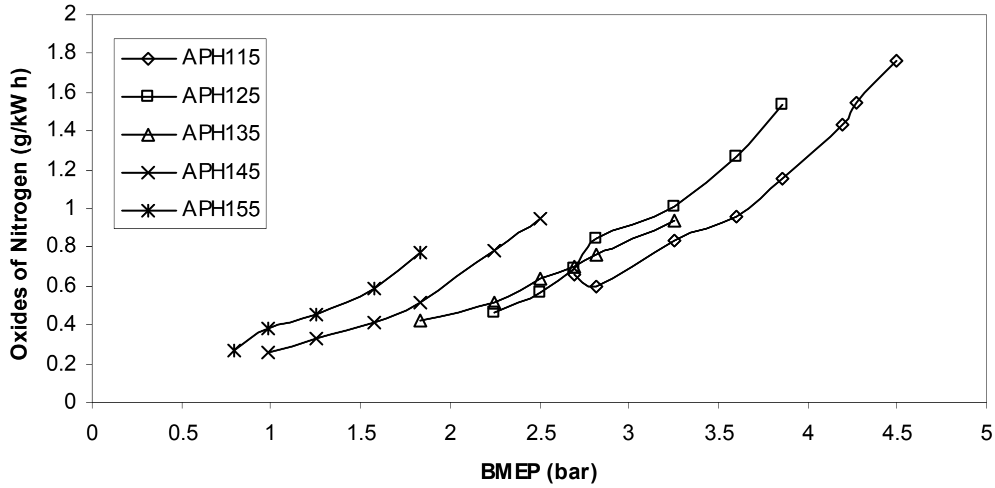

Fig. 5 Variation of oxides of nitrogen emission at various APH temperatures

4.5 Carbon monoxide emission

Figure 6 shows the variation of CO emissions in various APH temperatures with respect to the BMEP.It is found that for all APH temperatures the CO emission decreases when the methanol flow rate increases. The overall trend also shows that the CO emission decreases when the BMEP increases. These conditions are mainly a result of the changes in the combustion temperature that also change the methanol flow rates. A lower combustion temperature produces higher CO emission and a higher combustion temperature produces lower CO emission. A lower combustion temperature causes incomplete oxidation and a higher combustion temperature causes complete oxidation of the fuel. The flame quenching near the cylinder wall also results in an increased emission of CO.

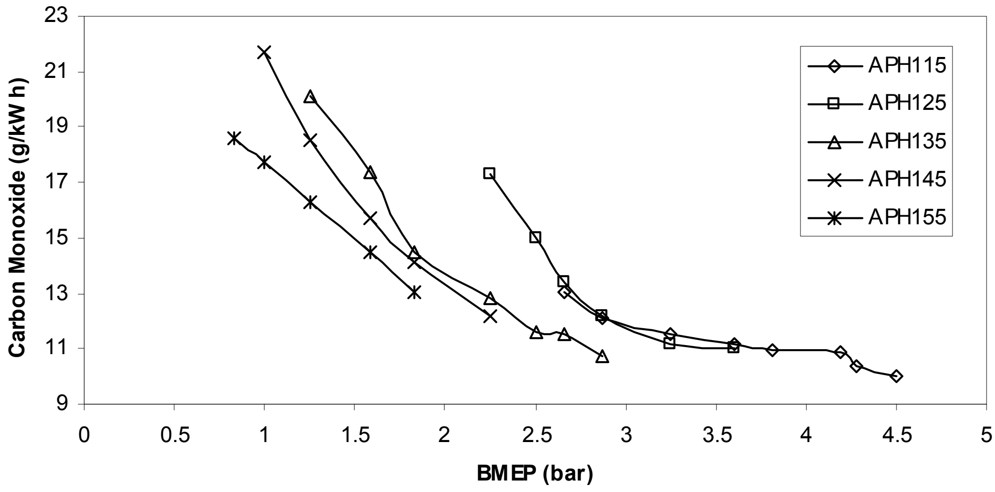

Fig. 6 Variation of carbon monoxide emission at various APH temperatures

4.6 Unburned hydrocarbon emission

Figure 7 shows a variation of HC emissions for various APH temperatures with respect to the BMEP. It is found that for all APH temperatures the HC emission increases when the methanol flow rate increases. The overall trend of HC emission corresponds to all APH temperatures. This trend also shows that the HC emission increases when the BMEP increases. During the lower methanol flow rate and the lower BMEP, a higher amount of energy share was supplied through the intake air, which combusts all fuel-air mixtures without leaving unburnt HC. When the BMEP increases, the amount of methanol flow increases and the heat supplied through the intake air finds it difficult to oxidize the fuel air mixtures fully. This is the main reason for the higher HC emission at higher methanol flow rates.

In addition to that, heat energy required for the charge preparation and autoignition was derived from the methanol fuel. Hence, more fraction of charge finds insufficient heat for oxidation and flame quenching occurs near the cylinder wall. This is the main reason for higher HC at higher hydrous methanol flow rates and higher BMEP. The non-penetration of flame into the crevice and cylinder corners is another reason for the higher HC emission.

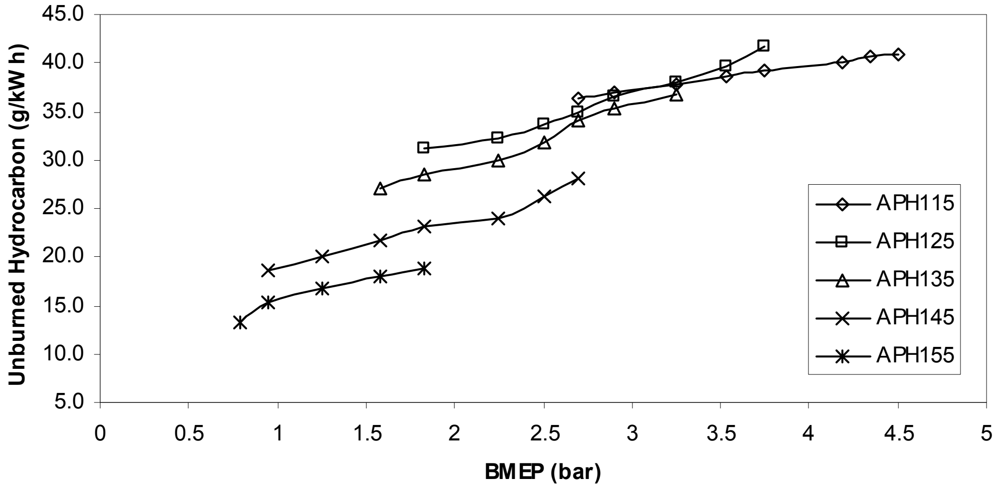

Fig. 7 Variation of unburned hydrocarbon emission at various APH temperatures

4.7 Maximum rate of pressure rise

Figure 8 shows the variation of the maximum rate of pressure rise for various APH temperatures with respect to the BMEP. It is found that for all APH temperatures, the maximum rate of pressure rise increases when the methanol flow rate increases. The overall trend also shows that the maximum rate of pressure rise increases when BMEP increases. During a lower methanol flow rate and lower BMEP, a higher amount of heat energy is supplied through the intake air, which helps to combust the methanol fuel with a lower rate of pressure rise. During higher methanol flow rates, due to the rapid rate of heat release, the combustion occurs with a higher rate of pressure rise. This is the main reason for a higher rate of pressure rise for all APH temperatures. The figure also shows that for all APH temperatures the observed maximum rate of pressure rise is not more than 8.5 bar per crank angle degree. The water content present in hydrous methanol absorbed a considerable amount of heat from the combustion and caused the lower rate of pressure to rise. This helps smooth engine operations without knocking and also extends the operating range of the HCCI mode.

The rate of pressure rise is an important parameter for the HCCI investigation as it is used to define the upper boundary of the HCCI combustion. When the fuelling rate increases, the combustion rate increases and causes an unacceptable rise in engine noise, which may eventually lead to a higher level of NO_x emission. The maximum rate of pressure rise is very high for richer fuel–air mixtures and is low for leaner fuel–air mixtures, Duret [12].

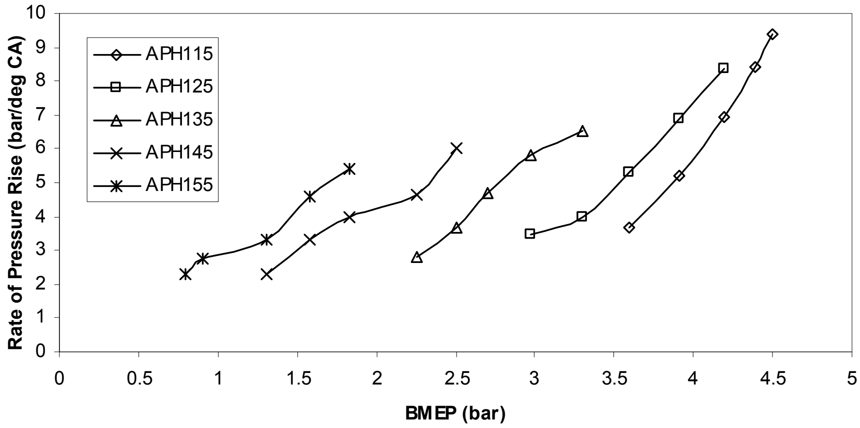

Fig. 8 Variation of rate of pressure rise at various APH temperatures

4.8 Influence of APH in HCCI operation

The primary objective of air-preheating (APH) in the methanol fuelled HCCI engine is to provide assistance for methanol autoignition. The experiment has been carried out for the range of preheat temperatures from 115 ºC to 155 ºC for different BMEP values. To explore the effect of the APHT in the methanol fuelled HCCI engine the engine was run at one particular methanol flow rate at 3.0 bar of BMEP. The compression ratio and other specifications are given in Table 1. At this stage, the APH temperature was changed from 115 ºC to 155 ºC and its corresponding performance and emission parameters were observed and they are presented in the following discussion.

4.9 Cylinder pressure diagram

Figure 9 shows the in-cylinder pressure traces for the methanol HCCI combustion at different APHTs. It can be noticed that the maximum pressure increases with a decrease in the APHT and the crank angle position of the maximum pressure advances. The maximum pressure is observed for the lowest APHT and the minimum pressure is observed for the highest APHT.

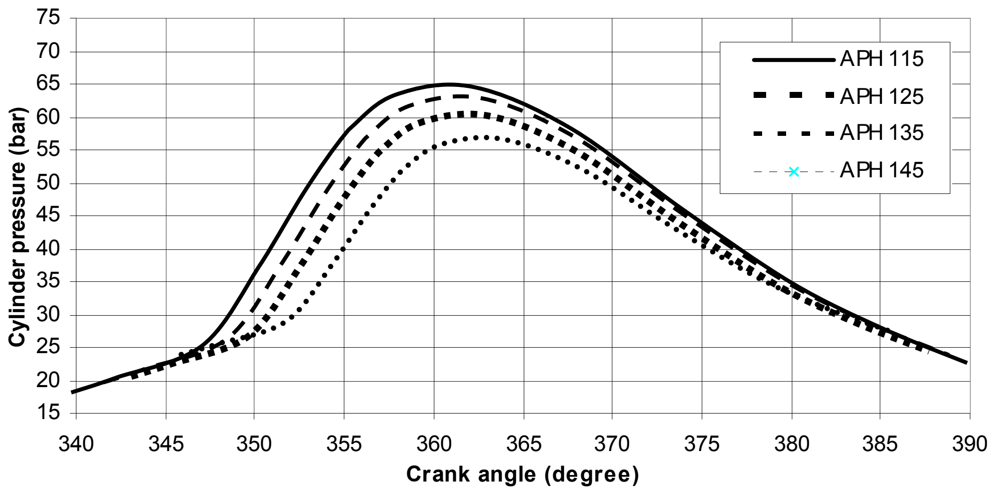

Fig. 9 Variation of cylinder pressure at 3 bar BMEP for various APH temperatures

4.10 Rate of heat release

Figure 10 shows the rate of heat release for the hydrous methanol HCCI combustion at different APHTs. The highest rate of heat release occurs when the highest APH temperature is present and the lowest rate of heat release occurs when the lowest APHT is present, Aroonsrisopon et al [13]. The start of combustion was found to advance with an increase in the APH temperature.

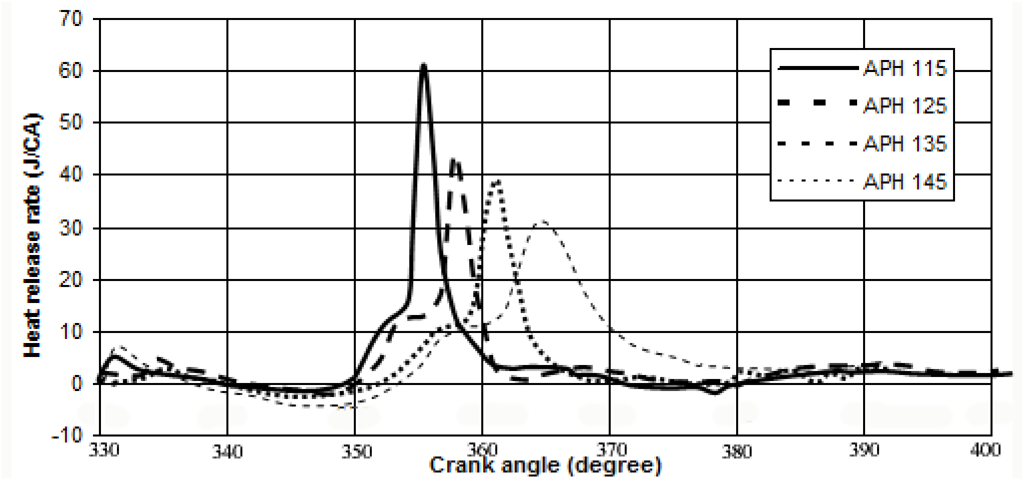

Fig. 10 Variation of heat release rate patterns at 3 bar BMEP for various APH temperatures

4.11 Effects of intake air temperature on hydrous methanol HCCI combustion

The NO_x levels increase with an increase in the APH temperature (higher APH temperatures). This condition can be attributed to the increased rate of heat release and a consequent rise in the mean gas temperature. However, the NO_x levels are extremely lower than in the DI CI mode. The trend of HC shows that the the HC level declines due to the improved combustion and increased rate of heat release. Similarly, better combustion and higher combustion temperature reduce CO emissions. The change in the APH temperature also affects the BTE. A higher APH temperature increases BTE and a lower APH temperature decreases BTE. The highest BTE was observed close to the knocking conditions. The variation of BTE, NO_x, CO, HC and RPR at various APH temperatures are shown in Figures 11, 12, 13, 14, and 15 respectively.

Finally, the performance, combustion and emission characteristics of the hydrous methanol fuelled HCCI engine are investigated. It is observed that due to the preheating of air by an electric air heater, the power consumption increases and the brake thermal efficiency decreases.

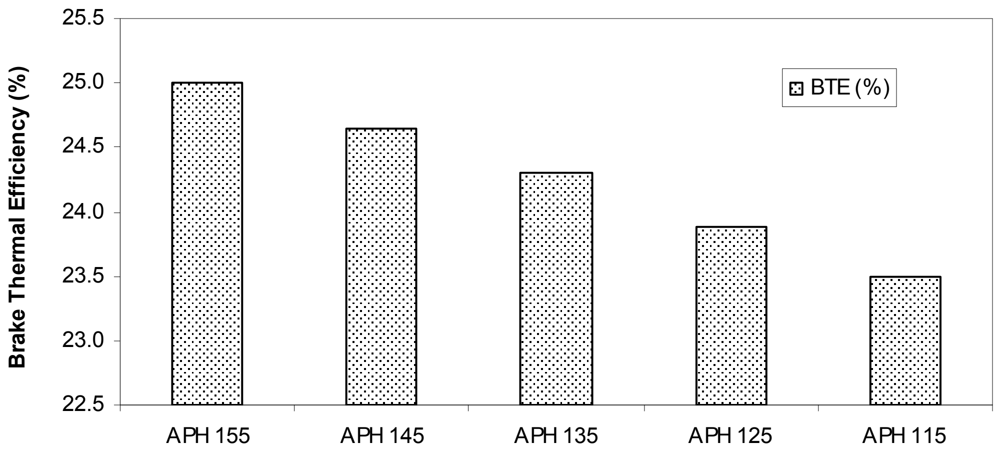

Fig. 11 Variation of brake thermal efficiency at 3 bar BMEP for various APH temperatures

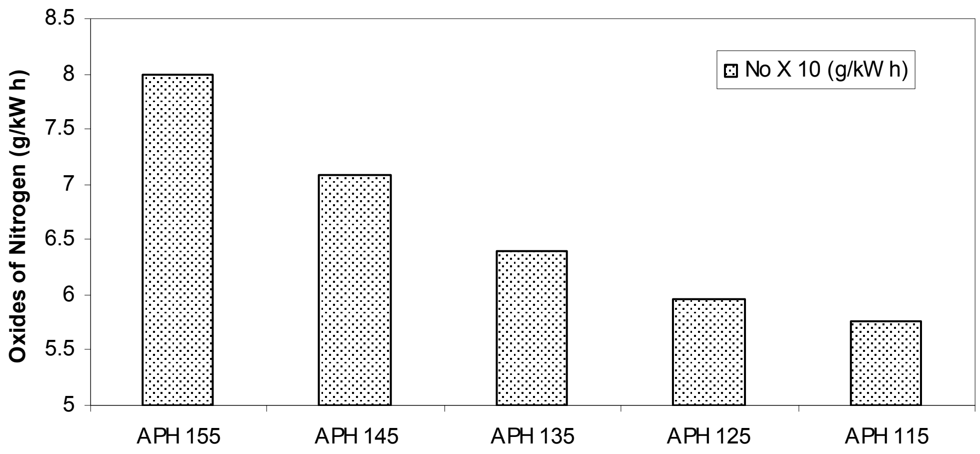

Fig. 12 Variation of oxides of nitrogen at 3 bar BMEP for various APH temperatures

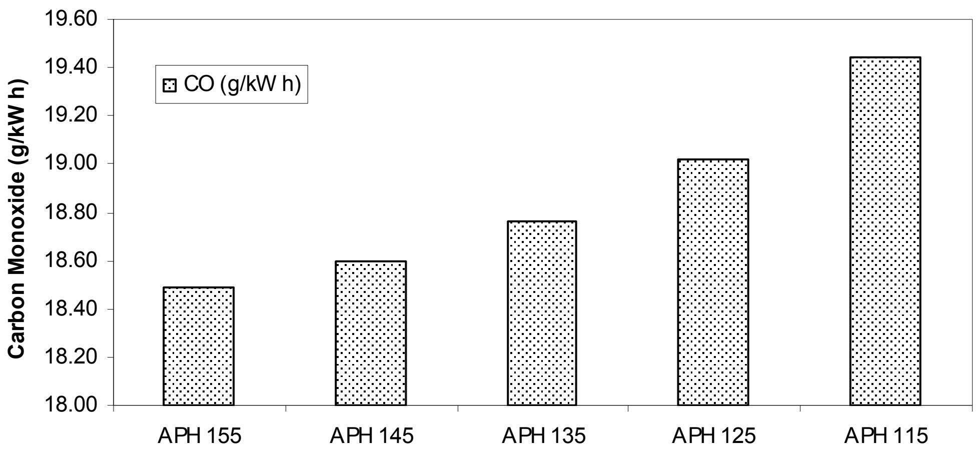

Fig. 13 Variation of carbon monoxide at 3 bar BMEP for various APH temperatures

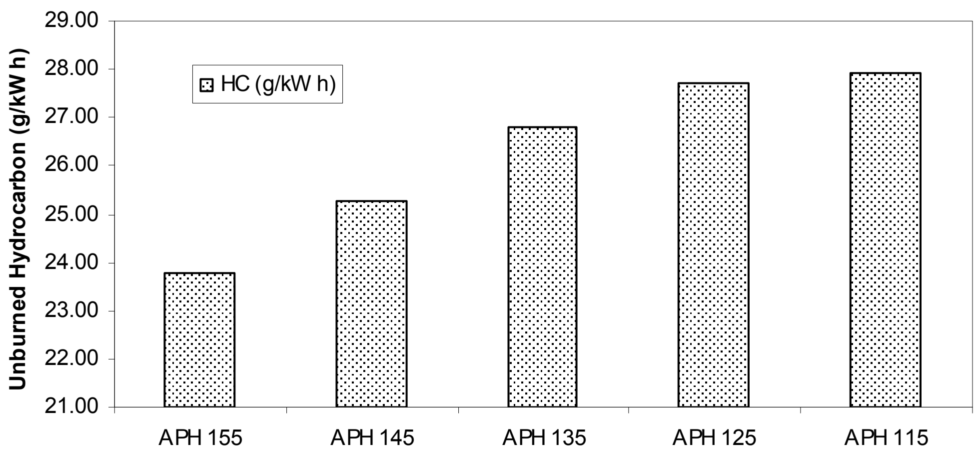

Fig. 14 Variation of unburned hydrocarbon at 3 bar BMEP for various APH temperatures

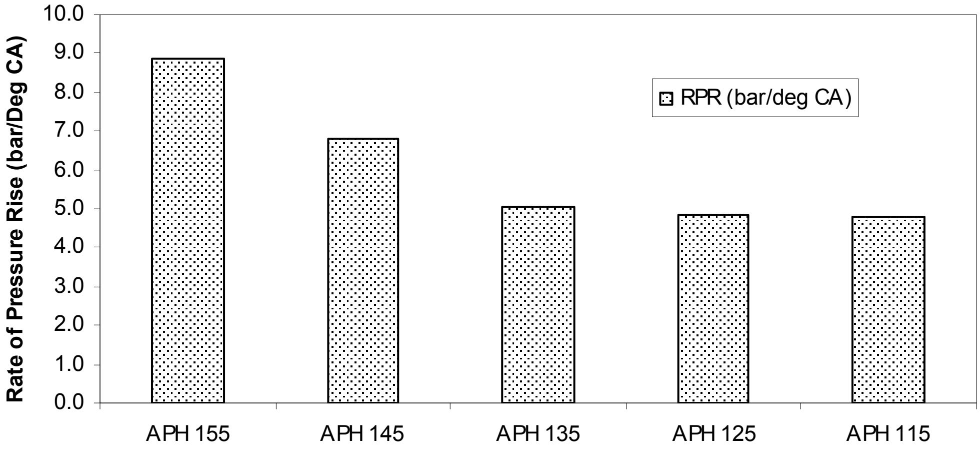

Fig. 15 Variation of rate of pressure rise at 3 bar BMEP for various APH temperatures

5. Conclusion

Based on the experimental investigations conducted on a single cylinder diesel engine modified for the hydrous methanol fuelled HCCI engine, the following major conclusions are made.

1. The results show that the air pre-heater assisted autoignition in the hydrous methanol fuelled HCCI engine is possible after minor engine modification.
2. The investigation shows that hydrous methanol suits perfectly the HCCI engine and the water content present in methanol helps to phase the combustion perfectly and to change the rate of combustion.
3. The investigation proves that the hydrous methanol operation reduces NO_x and smoke emission to extremely lower levels than the DI CI operation, i.e. it is approximately 18 times lower than the DI CI mode of operation.
4. This investigation shows increased CO and HC emissions. However; this is conventional behaviour of the HCCI operation.
5. The maximum brake thermal efficiency of the hydrous methanol operated HCCI engine is 27% at full load condition which is 3.5% lower than the diesel operated DI CI mode of operation.

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