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Experimental Investigation on Hydrous Methanol Fueled Homogeneous Charge Compression Ignition Engine Using Spark Assisted Method

M. Venkatesan[a*], N. Shenbaga Vinayaga Moorthi[b], R. Karthikeyan[c], A. Manivannan[d], V.C.Sathish Gandhi[a]

[a]*Department of Mechanical Engineering, University College of Engineering Nagercoil, (A Constituent College of Anna University Chennai), Tamil Nadu, India*
[b]*Department of Mechanical Engineering, University V.O.C College of Engineering Thoothukudi, (A Constituent College of Anna University Chennai), Tamil Nadu, India*
[c]*Department of Mechanical Engineering, Shri Andal Alagar College of Engineering, Mamandur, Tamil Nadu, India*
[d]*Department of Mechanical Engineering, Regional Centre, Anna University, Tirunelveli RegionTirunelveli, Tamil Nadu, India.*

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ABSTRACT

The present work investigates the performance and emission characteristics of hydrous methanol fuelled Homogeneous Charge Compression Ignition (HCCI) engine. In the present work, a regular diesel engine has been modified to work as HCCI engine. Hydrous methanol is used with 15% water content in this HCCI engine and its performance and emission behavior is documented. A spark plug is used for assisting auto-ignition. The spark timings are changed in steps of 3 degrees and the suitable timing that offers better phasing is called optimum spark timing. From the investigation, it is found that the hydrous methanol suits perfectly with HCCI engine and the water content present in the hydrous methanol helps to phase the combustion perfectly and to change the rate of combustion. The investigation also proves that the hydrous methanol operation reduces NO and smoke to extremely low level which is not possible by the direct injection CI engine. The water content present in the hydrous methanol helpes to control the timing of auto ignition and to run HCCI engine smoothly. Therefore, it is beneficial to use hydrous methanol in internal combustion engines.

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1. INTRODUCTION

The increase in research conducted on alternative fuel sources is driven not only by the adverse effect of hydrocarbon fossil fuels (HCFF) on the environment, but also by the depletion of HCFFs and the consequent rise in the price of petroleum products [1]. Biofuels cover a range of alternative fuels which can be produced from plant seed and vegetable oils as well as animal fats. Performance of biofuels can be affected by poor low temperature performance and oxidation stability and high NO_x emissions [2, 3]. Homogeneous charge compression ignition engine (HCCI) takes advantage of favorable characteristics inherent to both the spark ignition (SI) and the compression ignition (CI) engines [4-7]. HCCI engines are premixed like a SI engine and compression ignited like a diesel engine. Because of the typically lean fuelling rates, the low final flame temperatures result in significantly reduced emissions of oxides of nitrogen (NO_x). Since the fuel and air in an HCCI engine are premixed and lean, there is no rich combustion (as in a diesel engine), so particulate matter (PM) emissions are also quite low. HCCI engines can be run with any type of fuels as long as the fuel can be fully vaporized, sufficiently mixed with air, and achieve sufficient temperature during the compression stroke to reach the autoignition conditions. Therefore, HCCI is often referred to as controlled auto ignition (CAI). Methanol is an alcohol containing one carbon atom per

*Corresponding Author's Email: mechvenkatesan2013@gmail.com (M. Venkatesan)

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molecule. It is commonly known as wood alcohol. Methanol has been widely used as an industrial chemical since the 1980s. Methanol is produced from steam-reformed natural gas and carbon dioxide using a copper based catalyst. Currently, methanol is used as a feedstock for a variety of widely used organic chemicals, including formaldehyde, acetic acid, chloromethane and methyl tertiary butyl ether. The important properties of hydrous methanol are given in Table 1. In conventional HCCI combustion, fuel is injected early in the compression stroke, which allows sufficient mixing time for the formation of a homogeneous mixture. Takeda et al. [8] described one of the earliest implementation of this strategy in the premixed lean diesel combustion (PREDIC) mode operated on a DI four-cycle naturally aspirated single-cylinder engine.

Using two side injectors and one centre injector, different injector configurations are tested in combination with varying injection timings and excess air ratios (k). For fixed excess air ratios, the ranges of operational injection timings are limited to misfiring (too early) and knocking (too late). For instance, at k = 2.7, injection timing has to be between approximately 80° Before Top Dead Centre (BTDC) 60°BTDC. Nakagome et al. [9] found that NO_x emissions are significantly lower under PREDIC operation compared to conventional diesel combustion, with values reaching as low as 1/10 of the minimum concentrations emitted by standard diesel operation. However, this is accompanied by increased HC and CO concentrations due to over-leaning of air fuel mixture. The shortcomings of the PREDIC combustion system included limited partial load operation and lack of ignition timing control**.** Wimmer and Eichlseder [10] introduced that the Homogeneous Charge Late Injection (HCLI) and the Highly Premixed Late Injection (HPLI) are the latest HCCI diesel combustion concepts featuring late injection timing**.** Hashizume et al. [11] found that multiple injection strategies have been developed subsequently to single early injection strategy for diesel combustion in HCCI mode.

TABLE 1. Properties and composition of hydrous methanol

Chemical structure	CH_3OH
Fuel material	Natural gas, Coal or woody biomass
Physical state	Liquid
Molecular weight	32.04
Pump octane number	112
Auto ignition temperature	897^0F
Weigh composition t (%):	
Carbon	37.05
Hydrogen	12.6
Oxygen	49.9

The systems featuring multiple injections include the Multiple Stage Diesel Combustion (MULDIC) and Homogeneous charge intelligent Multiple Injection Combustion System (HiMICS). Su et al. [12] reported that with an early injection at 150°BTDC, a second injection occurs within the range of 2°BTDC to 30° After Top Dead Centre (ATDC). The first stage of combustion is premixed lean combustion which lowers NO_x emissions, while the second stage is diffusion combustion which occurs under high temperature and low oxygen conditions. Toyota Motor Corporation developed a new combustion technology known as the Uniform Bulky Combustion System (UNIBUS) which applies a double injection technique on a common rail injection system [13]**.** This technology was implemented in a new version of four-cylinder DI diesel engine in August 2000 by Jacobs et al. [14]. They demonstrated that the effects of EGR on emissions during HCCI combustion are tied closely to injection timing. Experiments performed on a four-cylinder 1.7 L diesel engine showed that when EGR rate increased from 42 to 45%, NO_x reduced within the late injection timing range from 18°BTDC to 9°BTDC. However, within this range, for earlier injection timings, the reduction in NO_x is accompanied by increased soot – thus exhibiting soot-NO_x trade-off. Only at retarded injection timings (9°BTDC), and at high enough EGR rates (44–45%), soot emission was lowered.

These findings are similar to another low temperature combustion work carried out by Kook et al. [15] on a single-cylinder optical diesel engine. It was found that at fixed injection timing within the range of 30.25–7.75°BTDC, increasing EGR from 0 to 65% decreased NO_x emissions. This was caused by the decrease in the adiabatic flame temperature because of the additional heat capacity of diluent charge at fixed SOI. Yun et al. [16] suggested that at moderate to high EGR rates, retarding injection resulted in decrease in soot emissions. The result of using such high EGR rates is increased HC and CO emissions due to slow oxidation at lower flame temperatures. Abd-Alla [17] reported that the long-term, excessive EGR can degrade engine durability and performance due to increased piston-cylinder wearing, as EGR contains abrasive and corrosive components such as sulphur oxide**.** Bunting [18] evaluated four gasoline range fuels in a spark assisted HCCI engine and showed the combustion and performance characteristics more on Motor Octane Number (MON). General trends show that the higher the MON, NO_x and indicated specific fuel consumption (ISFC), the lower are the combustion pressures and heat release rates.

Reuss et al. [19] used spark-assisted compression ignition mechanism to control the combustion phasing of compression-ignition gasoline engines. This study employed a combination of pressure-based heat-release analysis, spark-discharge voltage and current measurements, and cycle-resolved combustion imaging to define four periods of combustion process.

Oakley et al. [20] demonstrated that ethanol and especially methanol can achieve a larger lambda-EGR operating region than gasoline because the combustion phasing is less sensitive to lean mixtures and increased EGR. Likewise ethanol and methanol showed advanced combustion phasing compared to gasoline. Yousuf Ali et al. [21] studied a 4-stroke 5hp diesel engine tested with diesel oil plus cottonseed oil blends. The blends can profitably be employed in an existing CI engine without major engine modifications, further it can be an immediate solution for the development of rural areas, and for the emergency use in the event of severe diesel fuel shortage.

Seyyedvalilu et al. [22] presented exergy and exergoeconomic analysis and parametric study of a diesel engine based Combined Heat and Power (CHP) system that produces 277 kW of electricity and 282 kW of heat. Increasing compressor pressure ratio led to increase in the work output, heating power, exergetic efficiency, and exergy destruction cost and exergoeconomic factor of the CHP system in all environment temperatures. Also increasing turbine inlet temperature decreased the work output, exergetic efficiency and exergoeconomic factor while increasing the heating power as well as exergy destruction cost in all environment temperatures. Semin et al. [23] investigated the effect of fuel injection pressure on power performance and fuel consumption of diesel engine. The experiment results showed that the increasing injection pressures increased the engine power and fuel consumption. The motivation for the present work is to operate a single cylinder engine in hydrous methanol fuelled HCCI mode in a wide load range using spark assisted controlled autoignition. In this study, the effect of hydrous methanol flow rate on the performance, emissions and operating load range of a hydrous methanol fuelled HCCI mode, at a constant speed of 1500 rev/min is investigated.

2. EXPERIMENTAL SETUP

A single-cylinder, water-cooled, direct injection CI engine was modified to operate in a HCCI mode. A suitable eddy current dynamometer was coupled to the engine for loading and measurement purposes. The engine specifications are shown in Table 2. The engine used in the test rig is a regular diesel engine. The inlet side of the engine has two electronic fuel injectors to supply hydrous methanol into the engine separately. The inlet side of the engine also consists of anti-pulsating drum and air temperature indicator.

TABLE 2. Specifications 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} BTDC
Diesel injector opening pressure	180 bar
Type of combustion chamber	Hemispherical open combustion chamber

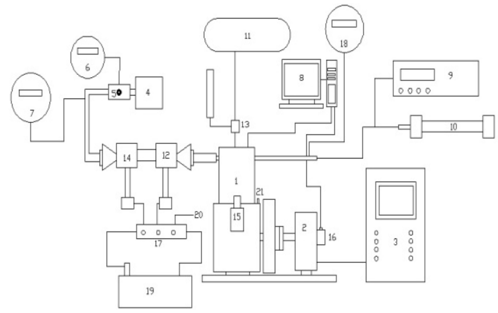

Figure 1. Schematic representation of experimental setup
1. Diesel engine, 2. Eddy current dynamometer, 3. Dynamometer control, 4. Anti pulsating drum, 5. Air preheater, 6. Watt meter, 7. Inlet temperature indicator, 8.Computer with DAQ, 9.Gas analyser, 10. Smoke sampling pump, 11. Diesel tank, 12 Hydrous methanol injector, 13. Three way cock 14. Injector, 15. Fuel injection pump, 16. Crank angle encoder 17. ECM, 18. Exhaust gas temperature indicator, 19. Battery, 20. Input signal for ECM operation and 21. Inductive pickup for ECM operation

The outlet side of the engine consisted of an exhaust gas temperature indicator, gas analyser and smoke sampler. The fuel used, hydrous methanol, was stored in separate fuel tanks and injected into the intake manifold using two separate electronic fuel injectors. The setup also consisted of diesel consumption measuring facility. The cylinder pressure data and crank angle data were acquired by a 16 bit data acquisition system. This system analysed the combustion parameters. The schematic representation of the experimental setup is shown in Figure 1.

3. METHODOLOGY

In this technique, a spark plug was used for ignition, which burned a fractional quantity of mixture and produced a compression wave. The combined effect of adiabatic compression and the compression produced by flame propagation (compressing wave) raised the temperature of the mixture well above the auto-ignition temperature and ignited the same.

The performance of the spark assisted HCCI engine is more dependent on spark timing. The spark plug functioning in this method was releasing the spark in advanced spark timing, where the spark was released well before the completion of compression stroke. The added spark ignited a little fraction of mixture and caused the flame propagation. This flame compresses the remaining mixture in addition to the compression offered by the raising piston. The combined effect of both compressions leads to auto-ignition of hydrous methanol fuel.

The spark timing was varied from - 41º CA and - 33 ºCA (negative sign indicates the angle measured from TDC). The spark timings lower than 33 converts the HCCI into SI mode. Hence, the lower limit of HCCI operation was limited as -33 and the higher limit was limited as -45. The higher limit is the limit at which the spark discharge does not change the cylinder pressure. The spark timings are changed in steps of 3 degrees and the suitable timing that offers better phasing is called optimum spark timing.

4. RESULTS AND DISCUSSION

This technique permits 100% diesel replacement using hydrous methanol after slight engine modification. The factors that prevent hydrous methanol auto ignition in CI engine are its low cetane number and high self-ignition temperature. To overcome this, the cylinder temperature of the engine must be maintained well above the self ignition temperature of hydrous methanol. This can be provided by the spark plug housed in the cylinder head.

4. 1. Operating Range The operating range for each and every load steps was limited by knocking and misfiring. The lower and upper limit of operating range for various spark timings was identified by increasing and decreasing hydrous methanol flow rates, respectively. The point at which engine misfires is called misfire limit (upper limit) and the point at which the engine produces unwanted noise is called knock limit (lower limit). The range obtained in-between misfire and knock limit is called the operating range. The same steps were repeated for all spark timings.

Figure 2 shows the operating range of spark assisted hydrous methanol fueled HCCI engine for various spark timings. From the figure, it is seen that the operating range of advanced spark timings are longer than retarded spark timings. The reason for the shorter operating range is lower combustion temperature and leaner air-fuel mixture.

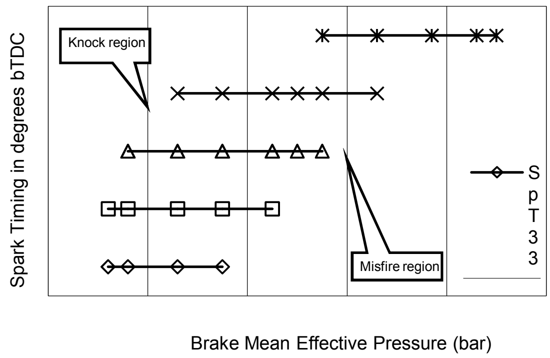

Figure 2. Operating range of hydrous methanol HCCI engine at various spark timings

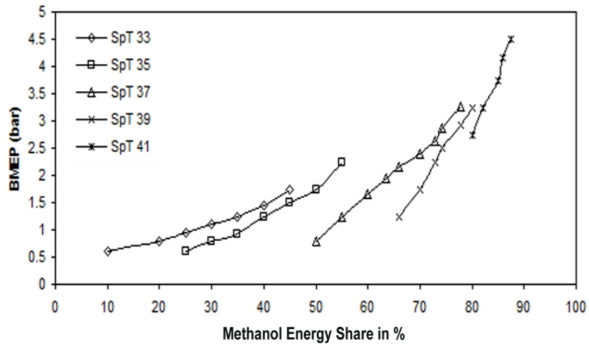

Figure 3. Energy share of hydrous methanol at various spark timings

Similarly, the reason for the longer operating range is higher combustion temperature and richer air fuel mixture.

4. 2. Energy Share Figure 3 shows the range of hydrous methanol energy share for various spark timings. From the figure, it is seen that the longer range of hydrous methanol energy share is tolerated during retarded spark timings and shorter range of hydrous methanol energy share is tolerated during advanced spark timings. The advanced spark timings are used for higher hydrous methanol flow rates and it develops higher combustion temperature. Further addition of hydrous methanol increases the combustion temperature and hence, it does not tolerate longer range of hydrous methanol energy share.

4. 3. Brake Thermal Efficiency Figure 4 shows the variation of brake thermal efficiency for various spark timings. From the figure, it is seen that the brake thermal efficiency increases with respect to hydrous methanol flow rates. However, after certain flow rate it starts to decrease due to sluggish burning and misfiring of hydrous methanol fuel. The rising trend of brake thermal efficiency at the beginning is due to the better combustion phasing caused by advanced spark timing. The decreasing trend of brake thermal efficiency after the maximum value is due to the sluggish burning and misfiring of hydrous methanol by retarded spark timing.The measured brake thermal efficiency shows that the values are increasing gradually, reaching a maximum and then decreases. The maximum brake thermal efficiency obtained at one particular spark timing is termed as optimum brake thermal efficiency.

4. 4. Oxides of Nitrogen Emission Figure 5 shows a variation of NO emission for various spark timings with respect to BMEP. From the figure, it is found that for all spark timings the NO emission increases when hydrous methanol flow rate increases. The temperature of combustion plays a vital role in the production of NO emission. The low temperature combustion produces lower NO emission and the high temperature combustion produces higher NO emission. However, the NO emission of hydrous methanol fueled HCCI engine is much lower than DI CI mode. This may be because of instantaneous combustion in HCCI mode. The instantaneous combustion eliminates local high temperature regions and reduces the formation of thermal NO.

4. 5. Carbon Monoxide Emission Figure 6 shows a variation of CO emission for various spark timings with BMEP. From the figure it is found that for all spark timings the CO emission decreases when hydrous methanol flow rate increases. The overall trend also shows that the CO emission decreases when BMEP increases. These are mainly due to the changes in combustion temperature with change in hydrous methanol flow rates. The lower combustion temperature produces higher CO emission and higher combustion temperature produces lower CO emission. The lower combustion temperature causes incomplete oxidation and higher combustion temperature which causes complete oxidation of fuel. The flame quenching near the cylinder wall also causes the liberation of more CO.

4. 6. Unburned Hydrocarbon Emission Figure 7 shows a variation of HC emission for various spark timings with respect to BMEP. From the figure it is found that for all spark timings the HC emission increases when hydrous methanol flow rate increases. The overall trend also shows that the HC emission increases when BMEP increases. During lower hydrous methanol flow rate more fraction of charge was combusted by retarded spark timing and hence it leaves lesser amount of HC. This trend reverses when hydrous methanol flow rate increases and the HC emission increases. The non-penetration of flame into the crevice and cylinder corners is also caused for the more HC emission.

From figure 7 it is also seen that the HC emissions is lower at lower loads and higher at higher loads. That is HC emission increases as load increases. This is mainly due to the increase of energy share of hydrous methanol, flame quenching near cylinder wall and fuel vapour present in the crevice volume. The higher latent heat of vaporization of hydrous methanol is also one of the reasons for the higher HC emission.

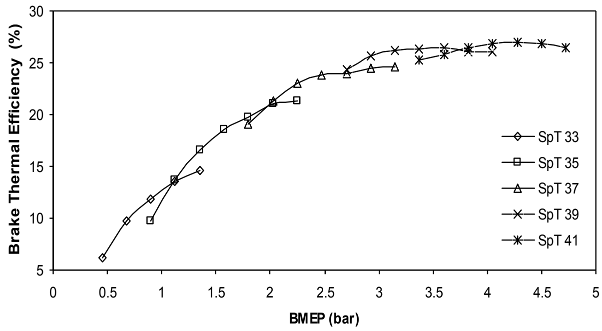

Figure 4. Variation of brake thermal efficiency at various spark timings

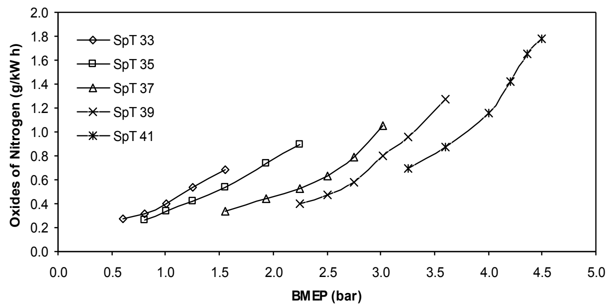

Figure 5. Variation of oxides of nitrogen emission at various spark timings

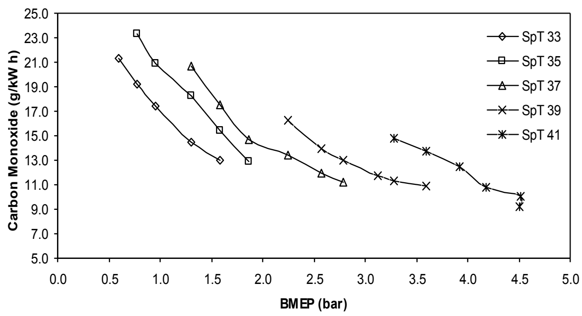

Figure 6. Variation of carbon monoxide emission at various spark timings

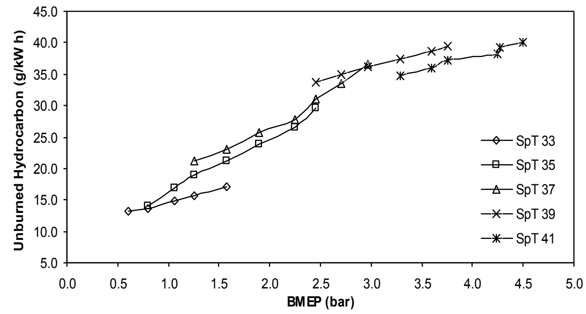

Figure 7. Variation of unburned hydrocarbon emission at various spark timings

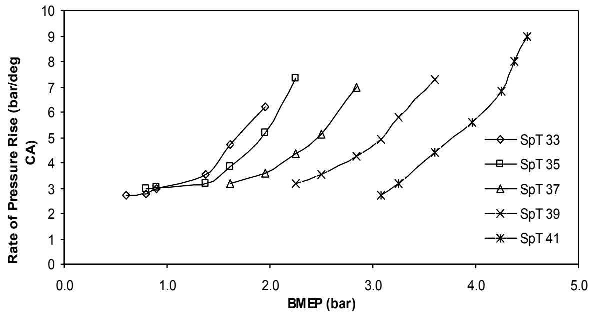

Figure 8. Variation of rate of pressure rise at various spark timings

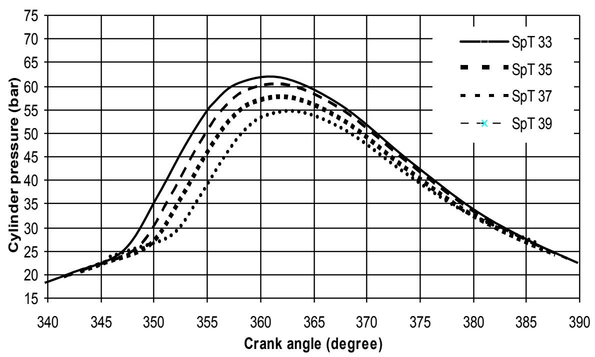

Figure 9. Variation of cylinder pressure at 3 bar BMEP for various spark timings

4. 7. Maximum Rate of Pressure Rise Figure 8 shows a variation of maximum rate of pressure rise for various spark timings with BMEP. From the figure, it is found that for all spark timings the maximum rate of pressure rise increases when hydrous methanol flow rate increases. The overall trend also shows that the maximum rate of pressure rise increases when BMEP increases.

The rate of pressure rise increases with respect to hydrous methanol flow rate and advance spark timing. In addition to that the maximum rate of pressure rise is higher for richer fuel–air mixtures and is rather lower for leaner fuel–air mixtures. From the figure it is also seen that for all load conditions the maximum rate of pressure rise is below 8.5 bar/CAD. The water content present in the wet hydrous methanol absorbs considerable amount of heat form the combustion and causes the lower rate of pressure rise.

4. 8. Influence of Spark Assistance in HCCI Operation The primary objective of providing spark assistance in hydrous methanol fuelled HCCI engine is to auto-ignite the hydrous methanol fuel-air mixture present in the cylinder. The spark plug functioning in this method does not follow the regular SI engine spark timing. Instead, it functions with very advanced spark timing.

The spark discharged in the engine varies between the - 41º CA and - 33ºCA. The spark timing lowers than 33 converts the HCCI into SI mode. Hence, the lower limit of HCCI operation is 33 and the higher limit is 45. The higher limit is the limit at which the spark discharge does not change the cylinder pressure. The spark timings are varied in steps of 3 degrees and the suitable timing that offers better phasing of stable HCCI operation is called optimum spark timing. To explore the effect of spark timing in an hydrous methanol fueled HCCI engine, the engine was kept at one particular hydrous methanol flow rate, which corresponds to 3.0 bar BMEP. At this stage, the spark timings were changed from lower limit to higher limit and its corresponding performance and emission parameters were observed and are presented in the following discussion.

4. 9 Cylinder Pressure Diagram Figure 9 shows the in-cylinder pressure traces for hydrous methanol fuelled HCCI combustion at various spark timings. It can be noticed that the maximum pressure decreases with respect to advance of spark timing and the crank angle position of maximum pressure retards. The maximum pressure occurs for the lowest spark timing and the minimum pressure occurs for the highest spark timing.

4. 10. Rate of Heat Release Figure 10 shows the rate of heat release of hydrous methanol fuelled HCCI combustion at various spark timings. The highest rate of heat release occurs for the lowest spark timing and the lowest rate of heat release occurs for the highest spark timing. The start of combustion was found to advance with respect to decrease of spark timing.

4. 11. Effects of Spark on Hydrous Methanol Hcci Combustion The NO_x levels increases with respect to decrease of spark timing, which can be attributed to the increased rate of heat release and consequent rise in the mean gas temperature. The trend of HC shows that the HC level declines due to the improved combustion and increased rate of heat release. Similarly, the better combustion and higher combustion temperature reduces the CO emissions. The change of spark timing also affects the brake thermal efficiency. The retarding of spark timing increases the brake thermal efficiency and advancing of spark timing decreases the brake thermal efficiency. The variation of brake thermal efficiency, NO_x, CO, HC and RPR at various spark timings are shown in Figures11, 12, 13, 14, and 15, respectively.

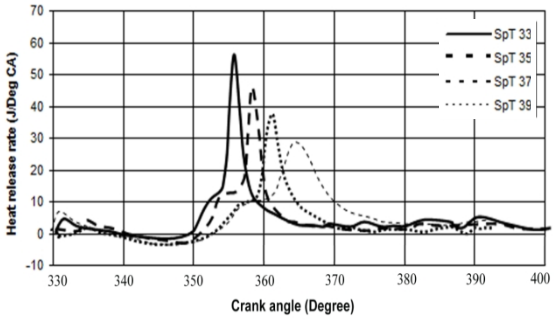

Figure 10. Variation of heat release rate patterns at 3 bar BMEP for various spark timings

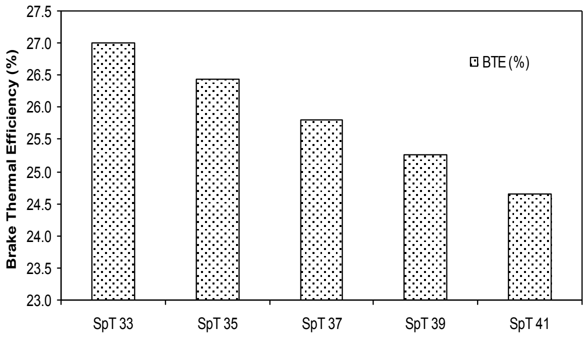

Figure 11. Variation of brake thermal efficiency at 3 bar BMEP for various spark timings

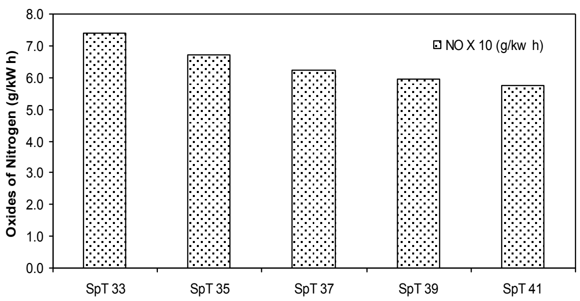

Figure 12. Variation of oxides of nitrogen at 3 bar BMEP for various spark timings

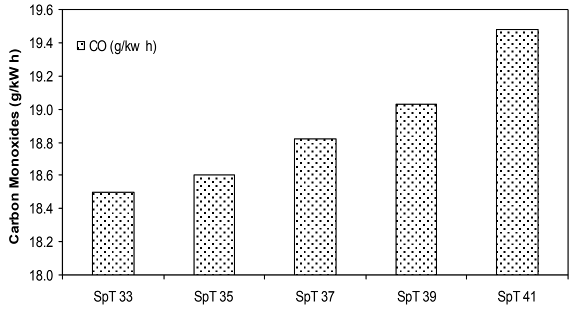

Figure 13. Variation of carbon monoxide at 3 bar BMEP for various spark timings

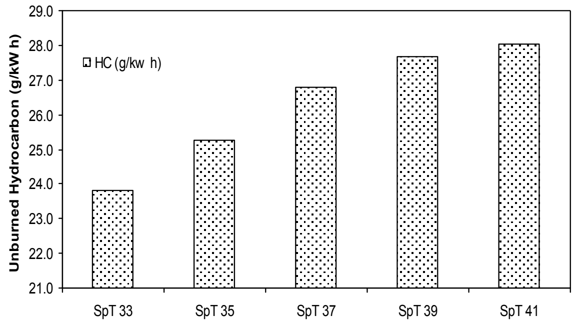

Figure 14. Variation of unburned hydrocarbon at 3 bar BMEP for various spark timings

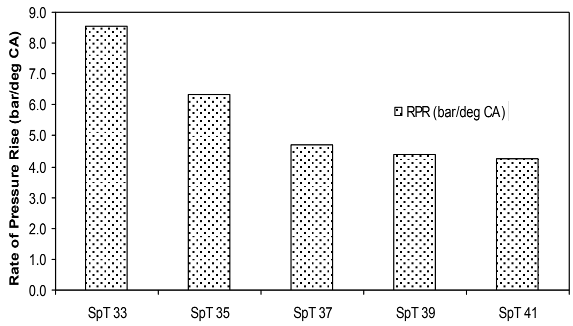

Figure 15. Variation of rate pressure rise at 3 bar BMEP for various spark timing

5. CONCLUSION

From the detailed experimental investigation, conducted using spark assisted hydrous methanol fuelled HCCI engine, the following major conclusions are drawn.

1. Results show that spark assisted hydrous methanol performed well in HCCI engine after little engine modifications.
2. The water content present in the hydrous methanol plays a vital role to phase the triggering of

combustion effectively and to alter the rate of combustion.

3. The maximum brake thermal efficiency obtained by this method at full load is 27%, this is 3.8% lower than that of diesel fuelled DI CI mode of operation.
4. The hydrous methanol HCCI operation reduces NO_x and smoke emission to extremely low level. It is approximately 16 times lower than that of diesel fuelled DI CI mode of operation.
5. This investigation shows increased CO and HC emission is 3.5% higher than that of diesel fuelled DI CI mode of operation. However, this is a conventional behavior of HCCI operation.

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Experimental Investigation on Hydrous Methanol Fueled Homogeneous Charge Compression Ignition Engine Using Spark Assisted Method

M. Venkatesan[a], N. Shenbaga Vinayaga Moorthi[b], R. Karthikeyan[c], A. Manivannan[d], V. C. Sathish Gandhi[a]

[a]*Department of Mechanical Engineering, University College of Engineering Nagercoil, (A Constituent College of Anna University Chennai), Tamil Nadu, India*
[b]*Department of Mechanical Engineering, University V.O.C College of Engineering Thoothukudi, (A Constituent College of Anna University Chennai), Tamil Nadu, India*
[c]*Department of Mechanical Engineering, Shri Andal Alagar College of Engineering, Mamandur, Tamil Nadu, India*
[d]*Department of Mechanical Engineering, Regional Centre, Anna University, Tirunelveli RegionTirunelveli, Tamil Nadu, India*

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چکیده

این پژوهش عملکرد و ویژگی های آلایندگی موتور احتراق همگن فشرده (HCCI) با سوخت متانول حاوی آب را مورد بررسی قرار می دهد. در این تحقیق، یک موتور دیزل معمولی اصلاح شده است تا به عنوان موتور HCCI کار کند. متانول حاوی آب با مقدار ۱۵٪ آب در این موتور HCCI مورد استفاده قرار گرفت و عملکرد و آلایندگی آن ثبت گردید. یک شمع برای کمک به احتراق خودکار مورد استفاده قرار گرفت. زمانبندی جرقه در ۳ نوبت تغییر داده شد و زمانی که بهترین فاز احتراق از آن بدست آمد، زمان جرقه مطلوب نامیده شد. از این تحقیقات، متانول حاوی آب عملکرد مناسبی با موتور HCCI داشته و محتوای آب موجود در متانول به مرحله احتراق کمک می کند و نرخ احتراق را تغییر می دهد. تحقیقات همچنین ثابت کرد که متانول حاوی آب NO و دود را به سطح بسیار پایین که توسط موتور CI تزریق مستقیم امکان پذیر نیست کاهش می دهد. آب موجود در متانول به کنترل زمان بندی سیستم جرقه زنی اتوماتیک و عملکرد روان موتور HCCI کمک می کند. بنابراین، استفاده از متانول حاوی آب در موتورهای احتراق داخلی سودمند است.

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