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PHONE : - + 91-8877340707
E-Mail : info@enrichedpublications.com

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Whenoutstandingadvancesaremadeinexistingareasorwhennewareas havebeendevelopedtoadefinitivestage,specialreviewarticleswillbe considered by the editors.

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Contents

An Approach Of Experimental Study On HCCI Engine

P. V. Ramana[1], D. Maheswar,[2] B. Umamaheswar Gowd[3]

[1]Research Scholar, JNT University, Anantapur, India

[2]Professor & Principal, KMIT, Hyderabad, India

[3]Professor, Mechanical Dept, JNTUA, Anantapur, India

ABSTRACT

Spark ignition engine (S.I) and compression ignition engine (CI) development is reached to peak stage from the last decades even it may have the substantial advantages in efficiency and exhaust emissions still it may need to go for further advancement by research activities for the day to day changes for the future requirements . The CI engine has a fuel efficiency advantage over the SI engine due to higher thermodynamic efficiency and lower pumping losses. In regard to the exhaust emissions the SI engine holds an advantage over the CI engine. From researches the new combustion concept is the homogenous charge compression ignition (HCCI), is a promising technology that combines elements of the diesel and gasoline engine. Engine operations in HCCI mode for the advanced reciprocating internal combustion engine allows for improvement of thermal efficiency and substantial reduction NOx emission. The attractive properties are increased fuel efficiency due to reduced throttling losses, increased expansion ratio and higher thermodynamic efficiency. The most production feasible solution for gasoline HCCI engine is application of exhaust gas recirculation. This technique increases thermal energy of a mixture, thus allowing for autoignation at moderate compression ratios. However, high exhaust gas recirculation rate decreases be applied in order to improve volumetric efficiency and extend high load limit. However increase the amount of intake air can lead to reduction of start of compression temperature via decreases of residuals in a mixture .in order to achieve HCCI mode of combustion, temperature of start of compression must be kept within narrow limits. HCCI is a way to increase the efficiency of the gasoline engine. With the advantages there are some mechanical limitations to the operation of the HCCI engine. The implementation of homogenous charge compression ignition (HCCI) to gasoline engines is constrained by many factors. The main drawback of HCCI is the absence of direct combustion timing control. Therefore all the right conditions for auto ignition have to be set before combustion starts In this experimental study investigations and modeling are studied. Experiments were carried out using four stroke single cylinder and six cylinder research engines. The single cylinder engine was equipped with fully variable valve train and direct gasoline injection. Application of mechanical boosting allowed for widening achievable load range in HCCI mode of operation. Numerical calculations allowed for determination of admissible valve train setting and intake pressure, which guarantee proper temperature of start of compression.

Keywords: HCCI, diesel engine, combustion, CFD CI, SI, NOx, CR

1. Introduction

As we know there are basically two modes different combustions in SI and CI engines. One is petrol and other Diesel fuel consumption modes. The combustion processes of them are very different one is homogeneous and other is heterogeneous. In the Diesel engine the combustion is initiated by auto ignition (spontaneous ignitions) because of high pressure and temperature at the end of compression stroke. However, in the petrol engine the combustion is caused by a spark that ignites a mixture that has been premixed before in the carburetor. Due to these different kinds of combustion, the two engines have different characteristics. The CI has a high efficiency, but it is very contaminating. Contrarily, the SI is not very efficient because of its low compression ratio but it has low emissions However, the IC engine is certainly not the best apparatus in every aspect but it seems to be a good on overall considerations. It is possible to develop further as the IC engine to be the better in some property but this should usually cost in another angle.

Till today internal combustion engines (ICEs) are playing important role in the automobile field because of their simplicity, robustness and high thermal efficiency and easy to use. Homogeneous charge compression ignition engine have been often considered for development substitution to the IC engines as an equivalent alternative engine for transportation and stationary applications in near future. Automobile and engine manufacturers are interested in homogeneous charge compression ignition engine due to their potential for high efficiency and low emissions. Homogeneous charge compression ignition combustion is a thermal auto ignition of a premixed fuel air mixture, with no flame propagation. The combustion temperature is low enough that the engine produces extremely low nitrogen oxide (NOX) emissions (a few parts per million) with no need for after treatment. Also, lean, premixed combustion results in near zero particulate matter emissions finally homogeneous charge compression ignition engines do not require spark plugs or a three way catalyst and are therefore expected to have lower maintenance requirements.

The reason why it's so well accepted can be explained by its overall appearance regarding properties like performance, economy, durability, controllability but also the lack of other competitive alternatives. Research engine was fuelled with gasoline with the use of direct injection. Analysis of attainable increase of maximum engine load via boost application wasperformed on the base of measurement data and supported by engines cycle modeling. The main advantage of the homogeneous charge compression ignition (HCCI) combustion systems versus spark ignition and diesel engines is a sustainable reduction of cylinder out NOx emission. Additionally fast heat release rate allows for an increase of thermal efficiency in comparison to spark ignition engines (5). In order to obtain the auto-ignition temperature it is necessary providing additional energy to the in-cylinder load. This can be achieved in several ways.

In early experiments on this combustion system intake air preheating was widely used often combined with elevated compression ratios (3). However this technique is not applicable in production automotive engines. The most production viable solution for introducing additional energy into the cylinder is internal exhaust gas re circulation(EGR) utilizing negative valve overlap (NVO) however this technique application is limited to low and medium loads(7). Substantial dilution of the in-cylinder load by re circulated exhaust provides reduction volumetric efficiency, and therefore limits high load boundary in HCCI combustion mode. Moreover, engine operation in HCCI mode at loads above 0.5 MPa of indicated mean effective Pressure (IMEP) is associated with high mechanical and thermal loads of combustion chamber parts (6). Mechanical loads are the result of relatively high pressure rise rate. The thermal loads are increased by higher mean in-cylinder temperatures due to the large amount of re-circulated exhaust. In case of negative valves overlap, in cylinder load is compressed twice during a single engine cycle. Supercharging of the HCCI engine allows for increase of the permissible engine load if limitations come from insufficient amount of the intake air. Increase of intake pressure allows for application of lower EGR rates. However, fast heat release and large amounts of fuel at higher engine loads can result with excessive pressure rate rise. Application of boost and increase of in cylinder charge mass allows substantial reduction of NOX emission and pressure rate rise simultaneously

2. Research Aspects

With improvement in fuel efficiency and combustion stability, Combustion of HCCI was first applied to two-stroke engines [1], [2]. On four-stroke engine, when HCCI combustion is applied seems the fuel efficiency could be improved up to 50 % compared to the SI engine [3].Applying the technical knowledge of HCCI for the development of homogeneous charge compression ignition engine that meets the advanced reciprocating internal combustionengine program targets. Many tasks were involved in the development of the homogeneous charge compression ignition engine for the advanced reciprocating internal combustion engine. Four main timing control areas were identified by the available literature investigations: thermal control through variable compression ratio (VCR), variable valve timing (VVT), exhaust gas recirculation (EGR), and fuel mixtures or additives. CFD (Computational Fluid Dynamics) approach will be used to investigate HCCI Combustion Process and to know the detail of its combustion limits and its drawbacks of HCCI Engine combustion. As estimation of influence of boost pressure on HCCI engine working process. In order achieve auto-ignition in cylinder temperature was elevated with the use of internal EGR obtained via the NVO technique.

The research aspects for the study considered as

- Ignition Timing Control
- Engine Cold-Start

- Release rate
- Multi-Cylinder Engine Effects
- Fuel System
- Control Strategies and Systems
- Transient Operation

The progress towards development of practical homogeneous charge compression ignition engine system for stationary power generation was made in this experimental study as under

- Importance
- Working principle
- Starting of HCCI engines
- Control methods of HCCI
- Dual mode transitions
- Characteristics
- Recent developments
- Conclusion

3. Importance Of Research

As compared with Carnot cycle Otto cycle (SI) engine and diesel cycle (CI) engines are less efficient and developed equivalent efficiencies as Carnot efficiency by intensive researches from Stirling cycle and Ericsson cycle does not find practical applications due to their practical operational difficulties. So SI and CI engines are in use form last decades. Spark Ignition (SI) and the Compression Ignition (CI) engine reached almost saturated in the in the development in last decades. Due to various limitations of most developed and widely used petrol and diesel engines, Researchers are continued in different direction for further development to overcome the present problems ; SI engines have very high NOx and PM emissions and CI engines have high efficiency because of it difficulties in the long run the homogenous charge compression ignition (HCCI) is a promising new engine technology that combines elements of both the diesel and gasoline engine operating cycles with alternative combustion technology and also with high efficiency and lower NOx and particulate matter emissions. Hence HCCI concept given importance for further study to overcome the problems with both SI and CI engines and also it has the following overcomes on comparison

- High efficiency, no knock limit on compression ratio.
- Low NOx and no NOx after treatment systems required.
- Low PM emissions, no need for PM filter.

- HCCI provides up to a 15-percent fuel savings, while meeting current emissions standards.
- HCCI engines can operate on gasoline, diesel fuel, and most alternative fuels.
- In regards to CI engines, the omission of throttle losses improves HCCI efficiency.

4. Concept Of HCCI And Its Working

HCCI is characterized by the fact that the fuel and air are mixed before combustion starts and the mixture auto-ignites as a result of the temperature increase in the compression stroke. It is neither SI engine combustion nor CI engine combustion but it is new combustion technique between SI and CI combustion. HCCI is a relatively new combustion technology. It is a hybrid of the traditional spark ignition (SI) and the compression ignition process (such as a Diesel engine). Optical diagnostics research shows that HCCI Combustion initiates simultaneously at multiple sites within the combustion chamber and that there is no discernable flame propagation show in figure 1 below.

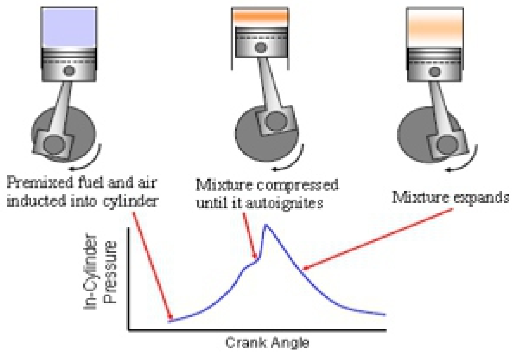

Figure: 1 cylinder pressure versus crank angle

Homogeneous Charge Compression-Ignition engines have the potential to provide dramatic increases in fuel efficiency over standard stoichiometric SI engines, while dramatically reducing NOx emissions

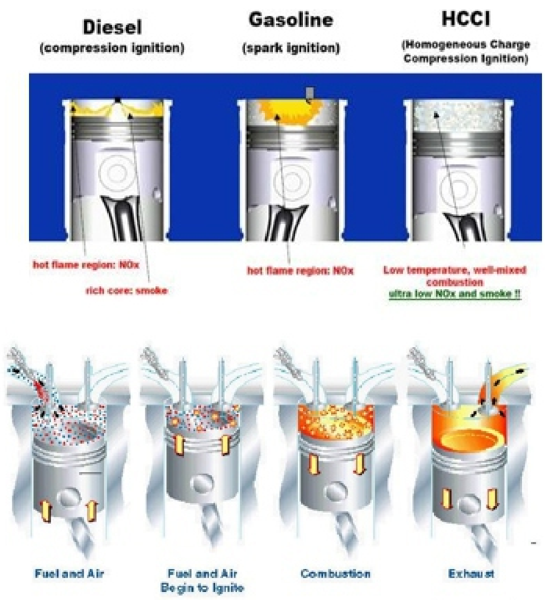

Figure: 2 compression HCCI ignition with SI and CI

Starting HCCI engines

HCCI engines are often difficult to start, at cold start charge does not readily auto ignite , the compressed gas temperature in an HCCI engine is reduced because the charge receivesno preheating from the intake manifold and the compressed charge is rapidly cooled by heat transfer the cold combustion chamber walls. Without some compensating mechanism, the low compressed charge temperatures could prevent and HCCI engine form firing. So early proposal was to start in SI mode or CI mode and then run in HCCI mode. A common approach has been to start. The engine ignition mode or diesel mode and transition to HCCI mode after warm up. It involves at high compression ratios the risk of knocking and cylinder failure. Ian auxiliary injector was installed in the cylinder head in addition to the original injector to achieve engine cold-start and warm-up and used to inject the pilot diesel fuel before the Top dead centre of compression stroke. The pilot fuel was used to ignite the premixed fuel injected by the original injector near the TC of intake stroke. When the engine warmed up, the engine was changed to HCCI mode, which used only one pulse fuel injection by the main injector before the TC of exhaust stroke. However, successful transition typically, requires advanced engines equipped with variable compression ratio (VCR) or variable valve timing (VVT), which may be expensive or difficult to implement for heavy duty engines. In practice operation in SI mode requires equivalence ratio of 0.6-0.65 or greater (Flynn et al. 200), which is high enough to damage the engine if thermal auto ignition or knock occurs during the transition. Instead of attempting to start the engine in SI mode and transition to HCCI mode, a brand new approach is used: start the engine directly in HCCI mode by preheating the intake with a gas fired burner. This was easy to implement by adding a burner to the pre-heater. The burner is run for a period of time (30 minutes) until the pre-heater reaches a high temperature (300oC). At this condition, running the intake charge through the pre-heater while simultaneously spinning the engine with an air starter is enough to achieve HCCI ignition. Thus the engine is started in homogeneous charge compression ignition mode by running a natural gas fueled combustor that heats the preheated. The intake gases are then circulated through the hot pre heater reaching a high enough temperature for homogeneous charge compression ignition to occur. Once combustion starts itself sustaining and therefore the burner can be turned off quickly after ignition. Now in take air preheating with hot exhaust (HE) and burner system allows startup in HCCI mode with conventional starter .figure 3 shows the curve of starting mode in HCCI .the burner is a source of emission and a consumer of fuel and as such in practical development of an HCCI engine for stationary power generation. This would have to b considered as a contributor to the overall system emission and fuel consumption. Starting in HCCI mode instead of (late fire) SI to HCCI transition avoid potential risk of knock damage that could occur with SI operation at high CR near Stoichiometric ratio.

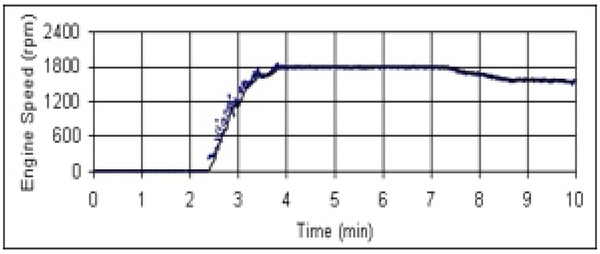

Figure:3 Engine speed versus time for HCCI mode startup

fueling system

The fueling system presents several changes because HCCI combustion is extremely sensitive to equivalence ratio. Just a few cycles of HCCI combustion at high equivalence ratio (φ>0.5) are enough to cause physical damage to the engine .therefore the fuelling system has to guarantee that no equivalence ratio excursions will occur “safe equivalence ratio (φ ?0.45)under any circumstances. It may also be desirable to run at low equivalence ration for low load operation. these difficulty requirement where met with a novel solution: the stock carburetor tuned for natural gas was replaced with a carburetor tuned for liquid petroleum gas (LPG) .considering that average composition of natural gas is approximately C1.2H3.5 and average composition of LPG is C3.5H8.5 a carburetor tuned for operating at equivalence ratio 0.9 on LPG will run at φ 0.3 -0.5 when fueled with natural gas . This is ideal for HCCI as the carburetor is quite efficient at maintaining the equivalence ratio. The equivalence ratio is reduced below 0.4 with an electronic control valve that reduces pressure in natural gas line.

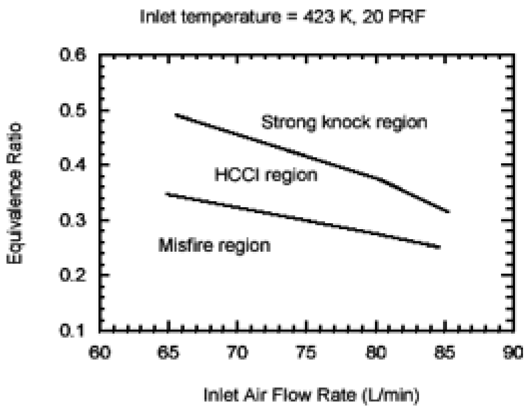

Figure 3.1 inlet air flow rate versus equivalence ratio.

control methods of HCCI combustion

To commercialize of the HCCI engine it has to overcome certain challenges which are existing at present. Low combustion temperatures, though conducive for low NOx emissions, lead to high HC and CO emissions. This is because of incomplete conversion of fuel to CO2 [7] in complete combustion causes CO emissions, and also it is difficult to control ignition timing and the rate of combustion for a required speed and power range [8]. The control over ignition timing is achieved by a spark plug or fuel spray in gasoline engines and diesel engines, respectively. Absence of such mechanisms makes it difficult to directly control ignition in HCCI and therefore, indirect methods are adopted. The auto-ignition events in HCCI is difficult to control, unlike the ignition even in spark-ignition (SI) and diesel engines which are controlled by spark plugs and in cylinder fuel; injectors, respectively but in HCCI engine auto-ignition challenges are controlling ignition timing over a range of speeds and loads, extending the operating range to high loads, achieving cold start capability and reducing hydrocarbon and carbon monoxide emissions at low loads. HCCI engines have a small power range, constrained at low loads by lean flammability limits and high loads by in- cylinder pressure restrictions so it requires the spontaneous and simultaneous combustion of fuel-air mixture need to be controlled and no direct control methods possible as in SI or CI engines. As stated above there are four main key areas were identified for timing control from the available literature: like thermal control through exhaust gas recirculation (EGR), variable compression ratio (VCR), variable valve timing (VVT), and fuel mixtures or additives. In HCCI mode, combustion initiation has to be controlled indirectly, via in-cylinder temperature at the start of compression. Some of the controlling parameters are

- Variable compression ratio(VCR)
- Variable induction temperature
- Variable valve actuation(VVA)
- Variable valve timing(VVT)
- Exhaust gas recirculation(EGR)

Variable compression ratio method (VCR)

The geometric compression ratio can be changed with a movable plunger at the top of the cylinder head. This concept used in “diesel” model aircraft engine. This could be achieved through a couple of different methods. One method would be to place a plunger within thecylinder head that could vary the compression ratio. Another option would be to have an opposed-piston design which would include variable phase shifting between the two crankshafts. Other possibilities exist as well but the key is to develop these in order to have excellent response time to handle transient situations. In order to study the VCR (Variable Compression Ratio) effect on the engine performance we could change the amount of compression for each cylinder and can study the effect. This could change the engine characteristics. By incorporating a device that could change cylinder volume rapidly, individual control of each cylinder could be conceivably achieved

Variable induction temperature

Pre-heating of intake air, intercooler by-pass, etc.are the Intake air thermal management- The simplest method uses a resistance heater to vary inlet temperature but this method is slow. Now FTM (Fast

Thermal Management) is used. It is accomplished by rapidly varying the cycle to cycle intake charge temperature by rapid mixing as shown in figure-4

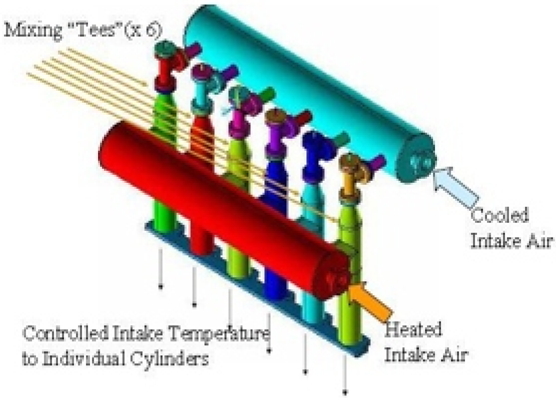

Figure: 4 fast thermal management

Rapid mixing of cool and hot intake air takes place by FTM system and can achieve optimal temperature as demanded and hence better control is possible. In FTM Control method Combustion timing can be controlled by adjusting balance of hot and cold flow

4.3.3. Variable valve actuation (VVA)

Early exhaust valve closure –internal exhaust gas recirculation. High positive valve overlap, Late intake valve opening, Variable Valve Actuation (VVA) system, VVA, irreversible expansion on intake valve are various controlling methods.

Within combustion chamber VVA method gives finer control and it involves controlling the effective pressure ratio. It controls the point at which the intake valve closes. If it closesafter the BDC, the compression ratio and the effective volume will changes. Negative valve overlap combined with fuel injection heat supply during gas exchange phase.

Variable Valve Timing (VVT)

VVT allows variation of the compression ratio not through geometric means but through timing of the opening and closing of the intake and exhaust valves. In addition, this system can act as a more direct method of EGR by controlling the amount of trapped residual gases thus allowing temperature and mixture control. As VVT allow the variation of compression ration not through geometric means but through timing of valve opening hence by changing the timing of the intake/exhaust valve changes the amount of combustible air in the cylinder thus controlling combustion strength and timing. This could be used to change the cylinder performances individually if a good control method is found. It should be noted that typical VVT schemes run cylinders with set timing, whereas these could need flexibility not only in timing but per cylinder.

Exhaust gas recirculation (EGR)

It is the process of recycling exhaust gases and adding them to the intake air. With EGR it is possible to control temperature, mixture, pressure, and composition. In comparison to the other control methods EGR is relatively simple, and has a great benefit. EGR can produce more power in an engine because more fuel could be pumped into the cylinder without spontaneous ignition due to the relative inertness of the emissions gas compared to air. It also could be used to control individual cylinder performance.

5. HCCI Combustion Experiments Works

Researchers at the Lund institute in Sweden have done a great deal of experimental work on 4-stroke single cylinder HCCI combustion [6-7-29-35] both naturally aspirated and supercharged operation have been studied using natural gas, isooctane and ethanol [35].moderate to high loads were achieved in this study: 14 bar IMEP for natural gas, 12 bar IMEP for ethnol and 10 bar IMEP for Isooctane. NOx was very low over the entire operating range,but HC and CO emissions high. Another study looked at the same fuels in naturally aspirated mode but with variable EGR [34].this work showed that increasing EGR for each of these fuels could further reduced NOx,HC and CO emissions.

Recently the Lund group has operated a six cylinder engine (in this case 1.95L/cylinder) in HCCi mode31. The combustion process was adjusted using variable intake temperature anda dual fuel configuration that allowed for variation of the fuel blends auto ignition characteristics (eg octane number) engine speed also varied. Brake mean effective pressure (BMEP) between 1.5 and 6 bar was achieved and brake thermal efficiency ranged between 26 and 43% nox emissions of under 20 mg/kWh were achieved up to 5 bar BMEP, but rose rapidly at higher load (up to 250-450 mg/kW-h)

Current-HCCI developed work Engine (Experimental Engine setup)

A cooperative Fuels Research (CRF) single cylinder engine and a single cylinder Ricardo Mark III engine with a Rover K7 head are used to carry out HCCI experiments. The experimental data for the HCCI experiments done in this study and the experimental data for the CFR engine and geometrical specifications of both the CFR and Ricardo engines shown in table-1

Parameters	CFR engine	Ricardo engine
Bore x stroke (mm)	83	80
Stroke(mm)	114	88.9
Compression Ratio	12	10
Displacement [L]	0.622	0.447
Number of valves	2	4
IVO,IVC[aBDC]	-170^0, 34^0	-175^0, 55^0
EVO, EVC [aBDC]	-40^0, 165^0	-70^0, -175^0

The operating parameters of the single cylinder CRF engines is shown in the following table- 2

Parameter	CFR engine
Engine speed [rpm]	700
Manifold temp[^{0}C]	88
EGR (%)	1-31
Equivalence Ratio	0.45 – 1.1
Manifold pressure [kpa]	89-92
Fuel[prf]	20,40,60
Oil Temp[^{0}C]	70

PRF number is defined as the volume percentage of iso-Octane in the fuel mixture of n- Heptane (PRF0) iso –Octane (PRF100)

Experiments have been performed on a single cylinder cooperative fuels research (CRF) engine modified for HCCI operation. The engine is naturally aspirated and an intake manifold heater has been installed to allow for preheating the intake air. The engine characteristics and operating parameters used in these experiments are listed in tableThe CFR engine has been fitted with an optrand Auto PSI-S(200 bar full scale range)combustion pressure sensor. The signal is acquired with a National Instruments PCI-6110E data acquisition (four input channels with a 5 mega-sample per second per channel maximum acquisition rate) board in Windows NT computes system. The pressure is acquired at every 0.1 crank angle degrees (CAD) using a 3600 /rev crankshaft encoder. Significant noise was present in the pressure data, despite extensive efforts to suppress it. An eighth- order Butterworth digital low pass filter has been used to filter the raw pressure data in post processing. The raw pressure signals are filtered forward and backward to eliminate any phase shift. The pressure data is filtered and then averaged. The rate of heat release was calculated from the average pressure trace using the method describes in the Heywood text.

- Variable compression Waukesha CFR engine
- 20HP general electric Dc motor Dynamometer
- Cooling system
- Air intake System
- Exhaust System
- Data Acquisition
- Gas Analysis/Sampling

Figure: 5 Research engine with variable valve actuation system

Pure propane and a blend of 15% by volume dimethyl- ether (DME) in methane were the fuels tested. These tests have been designed to characterize the operating parameters that influence HCCI engine emissions and performance. The fuel, intake air temperature, and equivalence ratio were varied in this experiment.

Testing was also conducted with pure methane fuel, but stable HCCI operation was only achieved for one operating point at the upper limit of preheating capacity and compression ratio. Operation in HCCI mode with pure methane was achieved initially using blend of methane and DME. The flow rate of each fuel was independently controlled and once stable operation was achieved, the DME flow rate was gradually reduced to zero

Propane and the DME in methane blend have similar reaction characteristics in an HCCI engine cycle, exhibiting cool flame heat release [41-40] the low temperature reaction increases charge temperature (and generate a radical pool initiating further chemical reactions) as the charge is compressed. Pure methane has very little cool flame chemistry causing the greater difficulty achieving conditions for auto ignition in an engine cycle relative to the other fuels [63]Simulations have been performed to look at the effect of several different control parameters on HCCI combustion

By a computational fluid dynamics (CFD) simulation using KIVA-3V code coupled with detailed chemistry[14] was found a multi pulse injection strategy for premixed charge compression ignition(PCCI)combustion fuel splitting proportion, injection timing, spray angles, and injection velocity effects are were examined. As the focus of the research the mixing process and formation of soot and nitrogen oxide (NOx) emissions were investigated. Due to the considerable changes of the mixing process and fuel distribution in the cylinder the result showed that the fuel splitting proportion and the injection timing impacted significantly on the combustion and emissions. Appropriate injection timing and fuel splitting proportion must be jointly considered for optimum combustion performance and the spray, inclusion angle and injection velocity at the injector exit can be adjusted to improve mixing, combustion and to minimize emissions.

On different fuels many experimental investigations were conducted with regard to homogeneous-charge compression- ignition. Dual fuel is the one of the approach, in dual fuel approach N-heptane and n-butane was considered for covering an appropriate range of ignition behavior typical for higher hydrocarbons [15]. For both fuels starting from detailedchemical mechanisms, reaction path analysis was used to derive reduced mechanisms, which were validated in homogeneous reactors and showed a good agreement with the detailed mechanism. Through the Conditional Moment Closure (CMC) approach the reduced chemistry was coupled with multi zone models (reactors network) and 3D-CFD. In a model based control strategy, in 2002 a study introduces a modeling approach for investigating the effects of valve events to adapt the injection settings according to the air path dynamics on a Diesel HCCI engine; based on a Knock Integral Model and intake manifold conditions the start of injection is adjusted. Researcher complements existing air path and fuel path controllers, and aims at accurately controlling the start of combustion [16].Experimental results were presented, which stress the relevance of the approach.

Modeling approach study was introduced in 2002 for investigating the effects of valve events HCCI engine simulation and gas exchange processes in the framework of a full-cycle HCCI engine simulation [17]. KIVA-3V is a multi-dimensional fluid mechanics code, which was used to simulate exhaust, intake and compression up to a transition point, before which chemical reactions become important. To compute the combustion events and the part of expansion the results are then used to initialize the zones of a multi-zone, thermo-kinetic code, the application of the method was illustrated in the context of variable valve actuation after the description and the validation of the model against experimental data,. It has concluded that early exhaust valve closing, accompanied by late intake valve opening, has the potential to provide effective control of HCCI combustion. With appropriate extensions, that modeling approach can account for mixture in homogeneities in both temperature and composition, resulting from gas exchange, heat transfer and insufficient mixing.

A multi-dimensional CFD code, KIVA-ERC-Chemkin that is coupled with Engine Research Center (ERC)-developed sub-models and the Chemkin library, was employed. Simulations of combustion of direct injection gasoline sprays in a conventional diesel engine were presented and emissions of gasoline fueled engine operation were compared with those of diesel fuel [18]. Using a reduced mechanism for primary reference fuel the oxidation chemistry of the fuels was calculated, which was developed at the ERC. With available experimental measurements for a range of operating conditions, the results show that the combustion behavior of DI gasoline sprays and their emission characteristics are successfully predicted and are in good sense. It seems that gasoline has much longer ignition delay thandiesel for the same combustion phasing, thus NOx and particulate emissions are significantly reduced compared to the corresponding diesel cases. The results of parametric study indicate that expansion of the operating conditions of DI compression ignition combustion is possible. Further investigation of gasoline application to compression ignition engines is recommended.

For an idealized engine configuration under HCCI-like operating conditions [19] Three- dimensional time-dependent CFD simulations of auto ignition and emissions were reported. The main focus is on NOx emissions and detailed NOx chemistry as an auto ignation mechanism of n-heptane and iso–octane. To accelerate the computation of chemical source terms a storage/retrieval scheme is used, and turbulence/chemistry interactions were treated using a transported probability density function (PDF) method. Simulations that include the direct in-cylinder fuel injection, and feature direct coupling between the stochastic Lagrangian fuel-spray model and the gas-phase stochastic Lagrangian PDF model. For the conditions simulated, consideration of turbulence/chemistry interactions is essential. Simulations that ignore these interactions fail to capture global heat release and ignition timing, in addition to emissions. For these lean, low-temperature operating conditions, engine-out NOx levels are low and NOx pathways other than thermal NO are dominant. In some cases Engine-out NO2 levels exceed engine-out NO levels. For accurate emission predictions, In-cylinder in homogeneity and unmixedness must be considered. These findings are consistent in recently reported literature of HCCI engine with results.

With CHEMKIN-PRO's HCCI Combustion Model the effect of EGR on HCCI engine operation application of many automotives can be modeled. Always user needs more accurate emission results that can be by the multi zone model. It allows specifying non-uniform initial conditions and heat transfer for regions within the cylinder [20].

Motion planning in the control of the coupled air path dynamics of turbocharged Diesel engines using Exhaust Gas Recirculation was demonstrated in the research [21] of 2007. Very large rates of burned gas need to be considered for HCCI combustion mode and proven on realistic test-bench cases that the proposed approach can handle such situations. The air path dynamics has a strong coupling; nice properties that make it easy to steer through control strategy. Over a wide range of set points it triangular form yields exponentialconvergence. To satisfy operational constraints it can also shown through simple analysis, provided transient are chosen sufficiently smooth.

For a Stochastic Reactor Model (SRM) for HCCI engines was suggested [22] a storage/retrieval technique. This technique enables fast evaluation in transient multi-cycle simulations. The Stochastic Reactor Model uses chemical kinetics, accounts for turbulent mixing and convective heat transfer in detailed, and predicts ignition timing, cumulative heat release, maximum pressure rise rates, and emissions of CO, CO2, unburnt hydrocarbons, and NOx. When coupled to a commercial one as an example of research, dimensional CFD engine modeling package, the scheme enables convenient simulation of transient control, using a simple table on a two-dimensional parameter space spanned by equivalence ratio and octane number. Developed computational tool was believed that it will be useful in identifying parameters for achieving stable operation and control of HCCI engines over a wide range of conditions. Furthermore, a tabulation tool enables multi-cycle and multi-cylinder simulations, and thereby allows studying conveniently phenomena like cycle-to-cycle and cylinder-to-cylinder variations. Optimization of engine operating parameters becomes feasible in particular, simulations of transient operation and control, design of experiments.

5. 2 Dual mode transitions

When auto-ignition occurs too early or with too much chemical energy, combustion is too fast and high in-cylinder pressure can destroy the engine. For this reason, HCCI is typically operated at lean overall fuel mixtures so this restricts engine operation at high loads practical HCCI engine will need to switch to a conventional SI or diesel mode at very low and high load conditions due to dilution limits there are two modes: one is HCCI-SI dual mode and other is HCCI-DI dual mode. SI mode transitions equips VVA and spark ignition system and Operates in HCCI mode at low to medium loads and switches into SI mode at higher loads this transition is not very stable and smooth . DI-HCCI mode has long ignition delay and rapid mixing is required to achieve diluted homogeneous mixture. Combustion noise and NOx emissions were reduced substantially without an increase in PM and combustion facing is controlled by injection timing. Thus DI-HCCI proves to be promising alternative for conventional HCCI with good range of operation.

6.0 Combustion Characteristics

Following figure 6 shows the details of heat release rate versus crank angle.

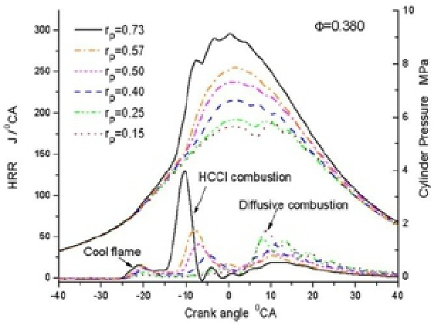

Figure: 6 Heat release rate versus crank angle

Emission Characteristics

Emission characteristics curves are drawn for different pressure ratios by taking indicated mean effective pressure (MPa) and NOx emissions in ppm and the details are shown below

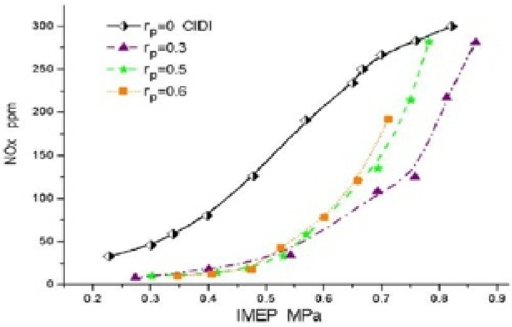

Figure 7 shows NOx emissions in parts per million versus indicated mean effective pressure in mega Pascal

Approaches to Reduce the Emission

In present days Technology changes to reduce the exhaust emissions by introducing the engine modifications, using exhaust gas recirculation, and catalytic converters after treatment, it takes slow fleet turnover due to fully longer implementation. However, they eventually result in significant emission reductions worldwide and will be continued on anever-widening basis in the United States. New technologies, like hybrids, solar photo voltaic cells, batteries and fuel cells, show significant promise in reducing emissions from current sources dominated by diesel fuel usage. Some program are underway in California like the turnover of trucks and especially off-road equipment is slow; awareness in the pollution control agencies need to be addressed to the existing emissions with in-use programs, such as exhaust trap retrofits and smoke inspections and pollutions norms. Other steps that can be continued with improved technology for reduction of emissions ,that will allow the use of the diesel engine, with its superior fuel consumption, to continue to benefit of society which will greatly reduces the negative environmental and health impacts to maintain the ecological balance.

The further development of combustion engines continues to be driven by the following legal, social and economic factors: legislation on exhaust gas is becoming more and more restrictive; fuel consumption needs to be reduced in view of global CO2 emission and the limited fossil resources. In the commercial vehicle segment, the diesel engine has always been prevalent due to its robustness and unequalled efficiency. In the years to come, however, future emission limits will require the simultaneous reduction of nitrogen oxides (NOx) and particulate emissions to extremely low values throughout most of the world.

Emission characteristics of premixed ratio on CO of HCCI-DI

shows the emission characteristics of premixed ratio versus CO% of different equivalence ratios

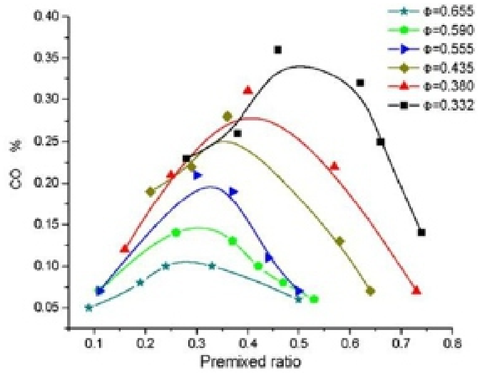

Figure: 8 Effect of premixed ratio on CO emission of HCCI-DI Engine performance

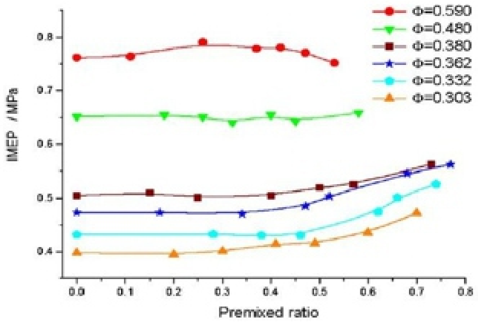

Figure: 9 Effect of premixed ratio on indicated mean effective pressure of HCCI-DI

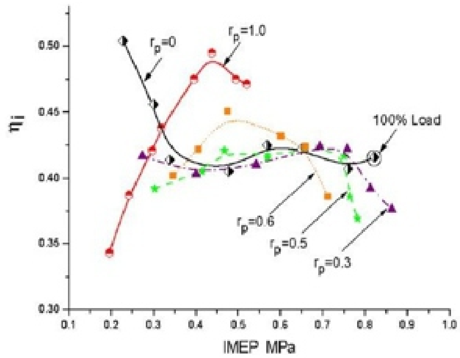

Figure: 10 Comparison of indicated thermal efficiency between HCCI-DI, HCCI and CIDI

Recent developments in HCCI

Solution for turbo charging includes Use VGT (Variable Geometry Turbine) which allows for a greater range of turbine nozzle area, better chance to achieve high boost. And also Combining turbo charging and super charging may be beneficial.EGR (Exhaust Gas Re- circulation) Can be adopted for higher efficiencies and lower HC and CO emissions. The exhaust has dual effects on HCCI combustion. It dilutes the fresh charge, delaying ignition and reducing the chemical energy and engine work. And also reduce the CO and HC emissions. Many companies recently developed HCCI prototypes .following are the details of some of the companies

a) General Motors has demonstrated Opel Vectra and Saturn Aura with modified HCCI engines.

b) Mercedes-Benz has developed a prototype engine called Dies Otto, with controlled auto ignition. It was displayed in its F-700 concept car and the 2007 Frankfurt Auto show.

c) Volkswagen is developing two types of engine for HCCI operation. The first, called combined combustion system (CCS) is based on VW group 2.0 litre diesel engine but uses homogeneous intake charge rather than traditional diesel injection. It requires the use of synthetic fuel to achieve maximum benefit. The second is called Gasoline Compression Ignition (GCI); It uses HCCI when cruising and spark ignition when accelerating. Both engines have been demonstrated in Touran prototypes, and the company expects them to ve ready for production in about 2015.

d) In May 2008, General Motors gave auto express access to a Vauxhall insignia prototype fitted with a 2.2 litre HCCI engine, which will be offered alongside their eco FLEX range of small capacity, turbocharged petrol and diesel engines when the car goes into production. Official figures are not yet available, but fuel economy is expected to be in the region of 43mpg(Miles per gallon) with carbon dioxide emissions of about 150g/km(grams per kilometer), improving on the 37mpg(Miles per gallon) and 180g/km produced by the current 2.2 litre petrol engine. The new engine operates in HCCI mode at low speeds or when cruising, switching to conventional spark ignition when the throttle is opened.

8. Conclusions

HCCI-DI combustion with n-heptane/diesel dual fuel is a 3-stage combustion process consisting of cool flame, HCCI combustion and diffusive combustion. Increase of premixed ratio, shortens the NTC (Negative Temperature Coefficient), increases the peak in-cylinder pressure and temperature and rises the highest heat release rate of HCCI combustion phase. NOx emissions decreases firstly at low premixed ratios and exhibit at eng of increasing high of premixed ratios. Pre-mixed ratios has no significant effect on soot emission and the soot emission could remain at the same level but then have a peak value with a certain higher premixed ratio relating to the equivalence ratio. The change of CO with premixed ratio is mainly depending on whether the premixed equivalence ratio exceeds the critical value. UHC increases almost linearly with the premixed ratio mainly due to incomplete oxidation in the boundary layer and the crevices. The IMEP increases with the increase of premixed ratioat low to medium loads. The indicated thermal efficiency shows deterioration at high load with large premixed ratios. Reduce unwanted emissions Unburned Hydrocarbons (UHC) Carbon Monoxide (CO) Nitrogen Oxides (NOx) Particulates (soot) that come from internal combustion engines (ICE) and increase fuel economy

9. Summary

HCCI engines are a promising technology that can help reduce some of our energy problems in the near term. However, control remains a challenge because HCCI engines do not have a direct means to control the combustion timing. Many concepts to be consider various parameters like, compression ratio, intake/exhaust temperature, intake mass, intake air pressure, composition could be controlled. With the generation of different concept a CFD simulation will be carried out to study the effect of various parameters on engine performance. A few degrees of difference in intake temperature can have significant effects on combustion strength. By varying intake temperatures for individual cylinders, combustion could be controlled, and also based on design of HCCI engine on feasible results. For determine the design parameters at full load / part load conditions CFD package analysis with 3D model could be applied for improving of combustion system. Till today the success of HCCI development is tempered by challenges that must be overcome before it hits the primetime of production. Control of the combustion process over the wide range of operating conditions experienced in everyday driving is the greatest challenge, because unlike a conventional-ignition engine, HCCI's combustion is not controlled by precisely timed spark events. Ensuring auto ignition at extreme temperatures and in the thinner air of high altitudes are the tallest hurdles to overcome.

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Investigation Of Optimum Operating Temperature Range For CI Engine Fuelled With Biodiesel

Karsoliya R.P.[1], Mandloi R.K. [2]
M. A. National Institute of Technology, Bhopal, India

ABSTRACT

The atmospheric temperature in various regions of India varies from 0°C to 48°C and due to this vast varying temperature range it is difficult for the engine to perform uniformly. Therefore it is essential to investigate the temperature that is most suited for operating parameters of engines and will give the best performance levels. The aim of this study is to analyze the performance of a compression ignition (CI) engine using Jatropha biodiesel and its blends with diesel as fuel on changing engine temperature to obtain best performance levels. It has been tried to investigate the optimum operating temperature of engine that will deliver the best performance level. The blends of Jatropha with diesel in varying proportions (B10, B20 and B40) are prepared and are investigated in single cylinder, four stroke CI engine computerized test rig. The results obtained on engine performance parameters i.e brake power (bp), brake thermal efficiency (ηth) and volumetric efficiency (ηv) and presented graphically with respect to the engine coolant temperature at different engine loads. The present study inferred that it is preferable to operate the engine at temperature 65˚C-75˚C to obtain best performance indicators.

Keywords: CI engine, Jatropha biosiesel, diesel, smoke intensity, operating temperature.

1. Introduction

The ongoing diminishing of fossil fuel reserves and consequently dependency on foreign energy resources and also destruction of the environment caused by burning fossil fuels triggered the research of biodiesel as a substitute for petroleum based diesel fuel in the recent years.

Today research and development in the field of internal combustion (IC) engines have to face a double challenge; on the one hand fuel consumption has to be reduced, while on the other hand ever more stringent emission standards have to be fulfilled.. The development of engines with its complexity of in-cylinder processes requires modern development tools to exploit the full potential in order to reduce fuel consumption.

Sharp hike in petroleum prices and increase in environmental pollution jointly have necessitated exploring some renewable indigenous alternatives to conventional petroleum fuels. Also, depletion of fossil fuels, vehicular population, increasing industrialisation, extra burden on home economy, growing energy demand, explosion of population, environmental pollution, stringent emission norms (Euro I, II, III, IV), etc emphasize on the need for alternative fuels. The alternative fuels must be technically feasible, environmentally acceptable, readily available and economically competitive.

Bio-diesel, which can be used as an alternative fuel is made from renewable biological sources such as vegetable oils and animal fats. It is bio-degradable, non-toxic and possesses low emission profiles [1]. Also, the use of bio-fuels is environmental friendly. Significant researches have already been put forward in investigating the performance of biodiesel in diesel engine application.

The researchers have reported varying results on power delivered by diesel engine with the use of biodiesel. Some authors have shown power loss while others have shown an increase in rated power and torque. Cetinkaya et al. in 2005 [2] performed experimentation on a 75 kW four-cylinder common rail engine. They observed that the reduction of torque was only 3 to 5 % with waste oil biodiesel compared to petroleum diesel. Lin et al. in 2006 [3] also carried out experiments on a 2.84 L naturally aspirated engine and they found 3.5% less power using pure palm oil biodiesel than that of petroleum diesel at full load condition. Hansen et al. in 1997 [4] also studied the break torque of test engine by varying viscosity,density and heating value of the fuel. They have shown that the break torque loss was 9.1% when 100 % bio-diesel was used as fuel instead of petroleum diesel at 1900 rpm.

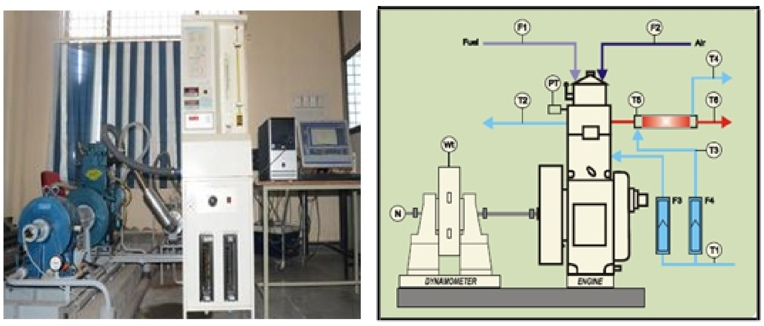
Many researchers have reported an increase in Brake Specific Fuel Consumption (bsfc) when using biodiesel and its blends compared to petroleum diesel fuel. These increases were basically the result of the loss of heating value in the biodiesel fuel blends. [5, 6, 7, 8, 9] In the present scenario the designs of CI engine being used in automotives by various manufacturers are not properly suitable to Indian climate condition. Looking in to the vast varying atmospheric temperature range in the country it is very difficult to say that which temperature is most suited to operating condition of engines and will give the best performance levels as far as ηth and bp is concerned.

After reviewing the literature, applying real experiences, in the experimental investigations and observations it is inferred that the engine systems can be optimized and evolved to provide precision cooling with necessary changes in engine cooling system, to reduce excessive heating and irregular temperature gradient for better performance. A lot of work has been done on the study of performance and emission characteristics of alternate fuels in IC engines. Limited amount of work is done related with temperature effect on CI engines running on bio-diesel. As per the author's point of view, it has been observed that no literature is available on the effect of CI engine operating performance parameters for bio- diesel with changing engine operating temperature related to ambient atmospheric temperatures. The authors are in strong opinion to evaluate the research on engine operating temperature with biodiesel and its blends as this would have definite impact on engine performance.

2. Engine Test Set Up And Methodology

A computerized C I engine test rig used for present experimental investigation. This experimental test rig consists of a single cylinder, four strokes, constant speed; direct injection diesel engine is used for the experiments having a rated power output of 5.2 kW at a constant speed of 1500 rpm. The test rig also have eddy current dynamometer as loading system, water cooling system, lubrication system and various sensors and instrumentation integrated with computerized data acquisition system for online measurement of load, air & fuel flow rate, instantaneous cylinder pressure, position of crank angle, exhaust emissions and smoke opacity etc. Figure 1 (a) shows the photographic image of the experimentalsetup used in the laboratory to conduct the present study and Figure 1 (b) shows the schematic representation of the experimental test setup.

Commercially available lab-view based engine performance analysis software package “Enginesoft” is used for on line performance data storage. The smoke intensity is measured in terms of Hartridge Smoke Unit (HSU in %) or in terms of K (the light absorption coefficient (m-1).



a) (b)

Figure 1. (a) Experimental Set up (b) Schematic representation of set up

Cylinder Heat Transfer Measurements

There are a wide range of temperature and heat fluxes in an internal combustion engine. The values of local transient heat fluxes can vary by an order of magnitude depending on the spatial location in the combustion chamber and the crank angle. The source of the heat flux is not only the hot combustion gases, but also the engine friction that occurs between the piston rings and the cylinder wall. When an engine is running at steady state, the heat transfer through out most of the engine structure is steady.

The maximum heat flux through the engine components occurs at fully open throttle and at maximum speed. Peak heat fluxes are in order of 1to10 MW/m.2 the heat flux increase with in increases with increasing engine load and speed. The heat flux is largest in the centre of the cylinder head, the exhaust valve seat and the centre of the piston. About 50% of the heat flow to the engine coolant is through the engine head and valve seats, 30% through the cylinder sleeve or walls and remaining 20% through the exhaust port area

Effect of Engine Temperature

Temperature control is very important for combustion engines as temperature is a critical; factor both for chemical reactions and mechanical stresses. Traditionally, temperature control is performed by feedback of a global quantity, the coolant temperature, which however is a poor indicator of specific temperatures. The use of pumps opens flew possibilities for thermal control, in particular in terms of efficiency, but also of pollution, especially in the cold start phase. It shows that predictive control and the use of coolant pumps allow to regulating specific temperatures. [11].

Heat Release and Component Temperature in CI Engines

The heat flux to the combustion chamber walls varies with engine design and operating condition. Also the heat flux to the various parts of the combustion chamber is not the same. As a result of this nonuniform heat flux and the different thermal impedances between locations on the combustion chamber surface and the cooling fluid, the temperature distribution within engine components is nonuniform. [11].

Effect of Engine Variables

The following variables affect the magnitude of the heat flux to the different surfaces of the engine combustion chamber and the temperature distribution in the components that comprise the chamber, engine speed, engine load, overall equivalence ratio, compression ratio, injection timing, swirl motion, wall material, mixture inlet temperature, coolant temperature and composition. These variables with speed and load have the greatest effect. [11].

Engine Cooling Systems

There are two types of engine cooling systems used for heat transfer from the engine block and head; liquid cooling and air cooling. With a liquid coolant, the heat is removed through the use of internal cooling channels with in the engine block. Liquid systems are much quieter than air systems, since the cooling channel absorbs the sounds from the combustion process. However, in this experimentation the engine temperature was artificially maintained by controlling the flow rate of coolant to the required fixed temperature.

3. Results And Discussion

The performance parameters considered in the present study are bp, ηv and ηth responding to coolant temperatures considered as engine operating temperature. Thermal Performance Evaluation is carried out in following three different experimental stages;

1. Diesel as fuel at different coolant temperatures and loads.

2. Jatropha bio-diesel as fuel at different coolant temperatures and loads.
3. Blends of Jatropha biodiesel and diesel as fuel at different coolant temperatures and loads.

The blend proportions used to conduct the experiments are B10, B20 and B40. The blends are prepared by direct mixing of both the fuels in required proportions. Mixing is done with the help of a magnetic mixer. Blends used are as follows;

- B10: 10% biodiesel and 90% diesel
- B20: 20% biodiesel and 80% diesel
- B40: 40% biodiesel and 60% diesel

Figure 2 shows graph between brake power (bp) and coolant water temperature (T2) at part load. It is seen from the graph that bp tends to increase with temperature from 50˚C. This is because the higher wall temperature delays flame quenching on the wall as the quench layer thickness gets reduced and hence the bp increases. The maximum bp occurs at around 65 ˚C-75˚C for all test fuels considered. With diesel the maximum bp is 2.36 kW at 65˚C and at 75 ˚C while with biodiesel the maximum bp is 2.18 kW at 75 ˚C. Beyond 75 ˚C the bp reduces. This is attributed to the fact that volumetric efficiency (ηv) reduces with increase in operating temperature due to the decrease in air density at higher temperature [11]. It is also seen from the figure that the increase of biodiesel percentage in the blends (B40 and B100) resulted in a decrease of bp over the entire temperature range. This is due to the fact that the higher viscosity and lower heating value of biodiesel reduces bp. The higher viscosity results in power losses, because the higher viscosity decreases combustion efficiency due to poor fuel injection atomization. It was also found that the B20, B10 and B0 have almost similar bp values. This could be attributed to additional lubricity and presence of oxygen provided by the biodiesel in blends B20 and B10 resulting improved combustion and mitigates the effect of higher viscosity and lower heating value of biodiesel.

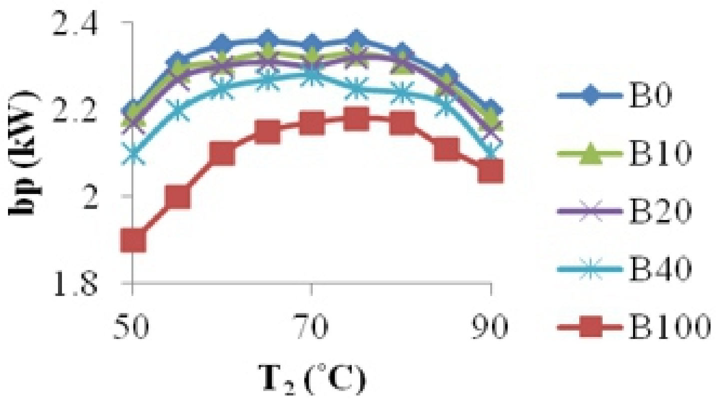


Figure 2. Brake power (bp) of B0, B10, B20, B40 and B100 with different coolant temperatures (T2) at part load on engine

Figure 3 shows graph between volumetric efficiency (η_v) and coolant water outlet temperature (T_2) at part load on engine. It is seen from the graph that η_v reduces with increase in coolant temperature. This is attributed to the fact that there is decrease in air density with increase in operating temperature due to which the η_v decreases (Heywood 1988[11]).

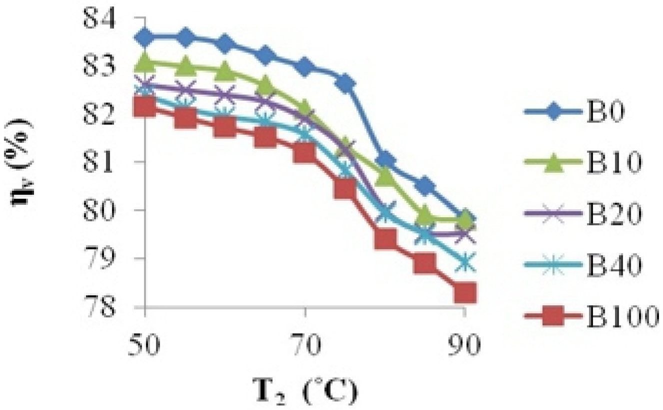


Figure 3. Volumetric efficiency (η_v) of B0, B10, B20, B40 and B100 with different coolant temperatures (T_2) at part load

Figure 4 presents the variation of brake thermal efficiency (η_{th}) for biodiesel and its blends with diesel with different operating temperatures (T_2) at part load on test engine. It can be observed that η_{th} increases continuously with increase in T_2 for all the fuels. The increasing trend is due to higher temperature results in better combustion of fuel. It is also observed that ηth is decreased with increase in biodiesel content in the blend at a constant T_2. The decrease may be due to higher viscosity of biodiesel which hinders the fuel evaporation due to poor atomization during combustion process. The maximum η_{th} for B0, B10, B20, B40 andB100 are 30%, 29%, 28%, 25% and 24% respectively. Beyond 80 ˚C the η_{th} reduces. This is attributed to the fact that bp and volumetric efficiency (η_v) reduces with increase in operating temperature due to the decrease in air density at higher temperature.

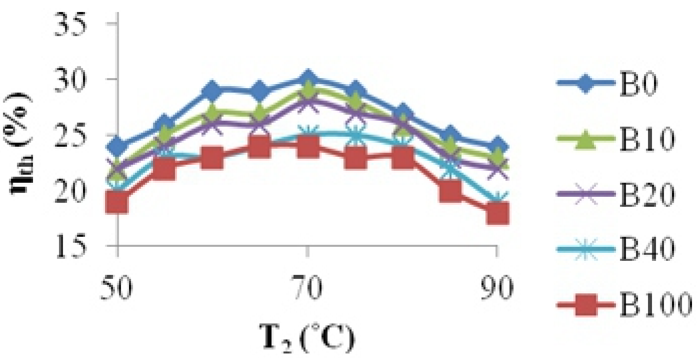


Figure 4 Brake thermal efficiency (η_{th}) of B0, B10, B20, B40 and B100 with different coolant temperatures (T_2) at part load

Figure 5 shows the graph of bp versus coolant temperature (T_2) at full load on test engine. It is observed that bp increase with the increase in load as expected. As the load increases from part load to full load the approximate percentage increase in bp for all test fuels is approximately 22 - 25 %. With diesel the maximum bp is 4.54 kW at 65 ˚C while with biodiesel the maximum bp is 4.3 kW at 65 ˚C. The minimum bp occurs at temperature 90˚C for all fuels tested. It is also seen from the figure that the increase of biodiesel percentage in the blends resulted in a decrease of power over the entire temperature range.

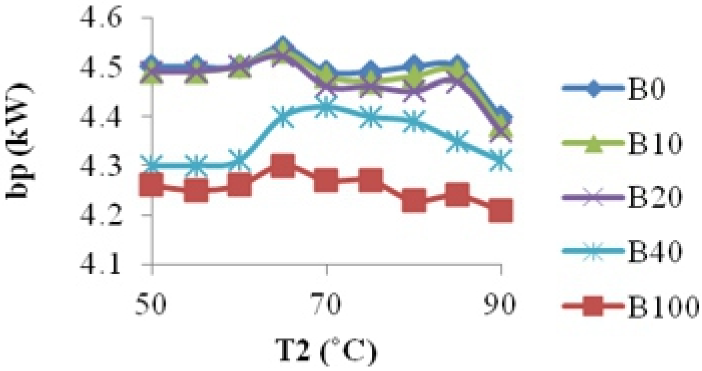


Figure 5. Brake power (bp) of B0, B10, B20, B40 and B100 with different coolant temperatures (T_2) at full load on engine

Figure 6 shows the graph of variation of η_v of diesel and biodiesel with different operating temperatures at full load on test engine. As the load increases from part to full the approximate percentage decrease in η_v for all test fuels is approximately 1 - 3 %. It is also seen from the graph that η_v reduces with increase in coolant temperature due to decrease in air density

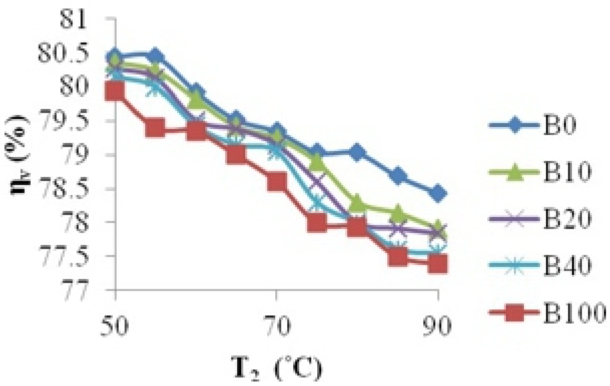


Figure 6 Volumetric efficiency (η_v) of B0, B10, B20, B40 and B100 with different coolant temperatures (T_2) at full load

Figure 7 shows the graph of variation of η_{th} of diesel and biodiesel with different operating temperatures at full load on test engine. It is observed that η_{th} increases with increase in load for all the blends tested. The trend may be due to the reason that relatively less portion of power is lost with increasing load. It can be observed that η_{th} decreases with the increase in blend proportion at constant temperature. The reduction in η_{th} can be attributed to lower heating value of the blends. Also, the higher viscosity of the blend may result in slightly reduced atomization and poorer combustion. The early initiation of combustion for biodiesel and early pressure rise before TDC contributes to increased compression work and heat loss resulting in a decrease in $_{th}\eta$

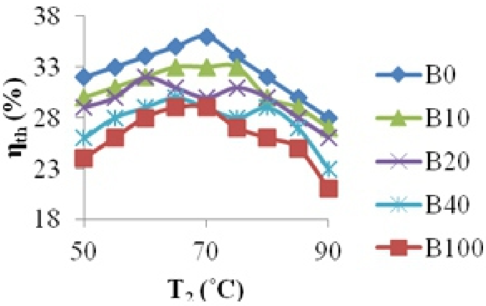


Figure 7 Brake thermal efficiency (η_{th}) of B0, B10, B20, B40 and B100 with different coolant temperatures (T$_2$) at full load

4. Conclusions

The experimental study is conducted on a single cylinder, four stroke, constant speed, water-cooled, direct injection diesel engine using Jatropha biodiesel and its blends with diesel. The thermal performance and smoke characteristics were evaluated by running the engine at different combinations of preset engine loads, ranging part to full load, with various coolant temperature at exit from 50˚C to 90˚C in steps of 10˚C.

From the experimental investigation on CI engine following conclusions can be drawn;

1. A single cylinder, four stroke, constant speed, water-cooled, direct injection CI engine originally designed to operate on diesel as fuel can also be operated on pure jatropha biodiesel without any system hardware modifications.

2. Based on the observation of graphs of bp versus coolant temperature, it can be concluded that with the increase in coolant temperature, the bp of diesel engine operated using diesel, biodiesel and its blends tends to increase with temperature from 50˚C. This is because the higher wall temperature delays flame quenching on the wall as the quench layer thickness gets reduced. With diesel the maximum bp is 2.36 kW at 65˚C and at 75 ˚C while with biodiesel the maximum bp is 2.18 kW at 75 ˚C. Beyond 75 ˚C the bp reduces due to reduction in volumetric efficiency (ηv).

3. The increase of biodiesel percentage in the blends resulted in a decrease of bp over the entire temperature range attributed to the fact that the higher viscosity and lower heating value of biodiesel reduces bp.

4. Based on the observation of graphs of ηv versus coolant temperature, it can be concluded that ηv tends to reduce with temperature due to decrease in air density.

5. The ηth increases continuously with increase in T2 due to higher temperature results in better combustion of fuel but beyond 80 ˚C the ηth reduces due to the fact that bp and volumetric efficiency (ηv) reduces.

Based upon the performance characteristics of CI engine under investigation it is inferred that engine should operate at coolant temperature of 65˚C-75˚C. This experimental result shows that there is a requirement to think about modifications in existing engine cooling system design as per India's climatic condition.

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Fuzzy Logic Approach Of Sensorless Vector Control Of Induction Motor Using Efficency Optimization Technique

G. Srinivas[1], Dr. S. Tarakalyani [2]

[1]Research scholar, J.N.T.U.H. Hyderabad.

[2]Professor, E.E.E.Dept, J.N.T.U.H. Hyderabad.

A B S T R A C T

This paper deals with the fuzzy logic approach of sensor less vector control of induction motor using efficiency optimization technique. The efficiency optimization control on the basis of search, where the flux is decremented in steps until the measured input power settled down to the lowest value. The control does not require the knowledge of machine parameters, is completely insensitive to parameter changes, and is applicable universally to any arbitrary machine. a fuzzy logic based on-line efficiency optimization control is proposed for an indirect vector controlled drive system. Energy optimizing controllers interface with the ASD to minimize line power consumption and that controller is on-line efficiency optimization controller using fuzzy logic. In this analysis the performance of the drive without fuzzy controller & when it is incorporated is analyzed & compared. The above analysis is done in MATLAB /SIMULINK/ using fuzzy logic toolbox.

Keywords: vector control, fuzzy logic, speed estimator, induction motor, and efficiency optimization

1. Introduction

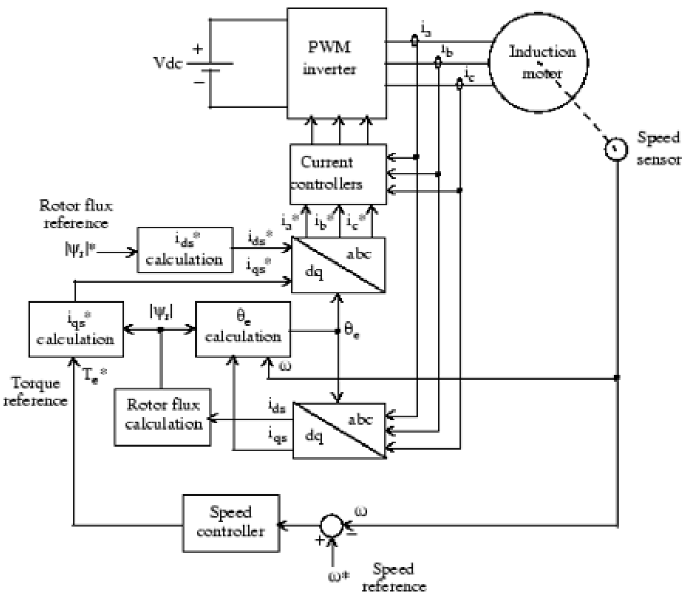
Induction motor is the work horse in industry due to its rigid construction & can work under all conditions of environment. But due to the factor that flux & torque cannot be controlled individually as the stator current is a combination of both it is not popular like D.C. Motor. But with the development of power electronics & Vector control concept the three phase stator current can be resolved into two phase components by orthogonal transformation by using Clarke's transformation& to rotor reference frame by parks transformation. To do this the position of flux vector is important. This position of flux vector can be found by direct & indirect methods where direct method employs sensors incorporated in stator which adds to cost, size & induction of harmonics. Hence in indirect control this flux vector can be found by machine parameters & modeling equations governing its performance. Hence sensor less vector control has gained importance. Basically there are various methods of indirect vector control of which Kalman filter, MRAS, sliding mode observer are in major use in earlier days, and hence these methods are prone to numerical & steady errors due to large calculations involved. Hence with the development of software's like Matlab/Simulink, & computer methods like fuzzy logic, neural networks the complications have been resolved.

The Indian power sector has come long way in power generation from 1300MW capacity during independence to 102907MW at present. However in spite of government's plans, the present power availability is not good enough to cater to the needs of the country, as there is a peak shortage of the power of around 10,000MW (13%) and 40,000 million units deficit (7.5%). Unless the system efficiency improves in terms of technical improvements, the crisis will still continue. Energy savings possible due to some major energy equipments such as transformers, motors etc.

The present paper deals with calculation of torque, speed of induction motor without optimization controller & comparing it with after installing controller using fuzzy logic approach.

$$\begin{pmatrix} V_{ds} \\ V_{ds} \\ V_{qr} \\ V_{ds} \end{pmatrix}^{*} \begin{pmatrix} (R_s + L_s P) & \omega_s L_s & L_m P & \omega_s L_m \\ -\omega_s L_s & (R_s + L_s P) & -\omega_s L_m & L_m P \\ L_m P & (\omega_{sl}) L_m & (R_r + L_r P) & \omega_{sl} L_r \\ -(\omega_{sl}) L_m & L_m P & -\omega_{sl} L_r & (R_r + L_r P) \end{pmatrix} \begin{pmatrix} I_{qs} \\ I_{ds} \\ I_{qr} \\ I_{ds} \end{pmatrix}$$

Modeling of induction motor



A. Dynamic Modeling of induction motors

The dynamic model of the induction motor is derived by transforming the three-phase quantities into two phase direct and quadrature axes quantities. The equivalence between the three-phase and two-phase machine models is derived from the concept of power invariance [16]. Induction motor modeling the synchronous reference frame is shown in equation (1)

Electromagnetic Torque:

$$=\frac{3}{2}\frac{P}{2}L_m\left(i_{qs}\ \ i_{dr}-i_{ds}\ \ i_{qr}\right) \qquad (2)$$

The dynamic equations of the induction motor in synchronous reference frames can be represented by using flux linkages as variables. This involves the reduction of number of variables in dynamic equations, which greatly facilitates their solution. The flux-linkages representation is used in motor drives to highlight the process of the decoupling of the flux and torque channels in the induction machine. The stator and rotor flux linkages in synchronous reference frame are shown in equations (3)-(8)

$$\lambda_{qs} = L_s i_{qs} + L_m i_{qr} \qquad (3)$$

$$\lambda_{ds} = L_s i_{ds} + L_m i_{dr} \qquad (4)$$

$$\lambda_{qr} = L_r i_{qr} + L_m i_{qs} \qquad (5)$$

$$\lambda_{dr} = L_r i_{dr} + L_m i_{ds} \qquad (6)$$

$$\lambda_{qm} = L_m (i_{qs} + i_{qr}) \qquad (7)$$

$$\lambda_{dm} = L_m (i_{ds} + i_{dr}) \qquad (8)$$

B. State Space model of induction motors

The space phasor model of the induction motors can be presented in state space equations from previous equation, so it can be expressed in the synchronously rotating d-q reference frame as shown in equations (9) to (17).

$$\dot{X} = AX + Bu \qquad (9)$$

$$X = [i_{ds} \quad i_{qs} \quad \lambda_{dr} \quad \lambda_{qr}]^T \qquad (10)$$

$$u = [V_{ds} \quad V_{qs}]^T \qquad (11)$$

$$\begin{pmatrix} a_1 & w_e & a_2 & \frac{L_m w_r}{\sigma L_s \quad L_r} \\ -w_e & a_1 & \frac{L_m w_r}{\sigma L_s \quad L_r} & a_2 \\ \frac{L_m R_r}{L_r} & 0 & \frac{-R_r}{L_r} & w_e - w_r \\ 0 & \frac{L_m R_r}{L_r} & -w_e + w_r & \frac{-R_r}{L_r} \end{pmatrix} \qquad (12)$$

$$B = \begin{pmatrix} a_3 & 0 \\ 0 & a_3 \\ 0 & 0 \\ 0 & 0 \end{pmatrix} \qquad (13)$$

$$a_1 = \frac{-R_s}{\sigma L_s} - \frac{(1-\sigma) R_r}{\sigma L_r} \qquad (14)$$

$$a_2 = \frac{L_m \; R_r}{\sigma L_s \; L_r 2} \qquad (15)$$

$$a_3 = \frac{1}{\sigma L_s} \qquad (16)$$

$$\sigma = 1 - \frac{L_m 2}{L_S \; L_r} \qquad (17)$$

Fuzzy efficiency Controller

A. Efficiency Optimization Control

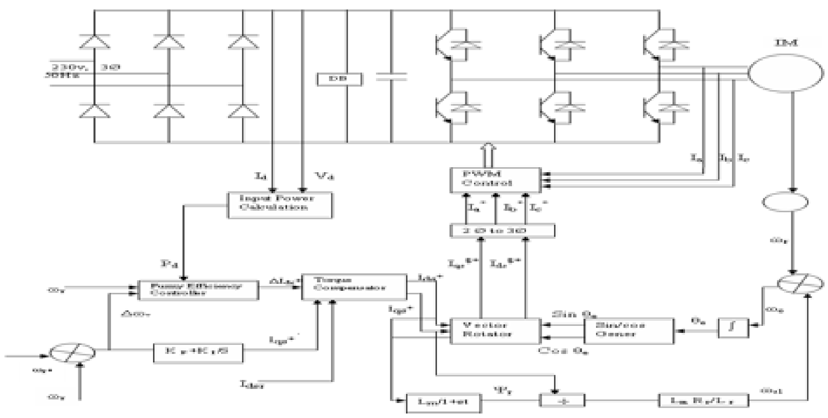


Fig. 1. shows the indirect vector controlled induction motor with efficiency optimization controller block diagram

The principle of efficiency optimization control with rotor flux programming at a steady- state torque and speed condition is explained in Fig.: 1 The rotor flux is decreased by reducing the magnetizing current, which ultimately results in a corresponding increase in the torque current (normally by action of the speed controller); such that the developed torque remains constant. As the flux is decreased, the iron loss decreases with the attendant increase of copper loss.

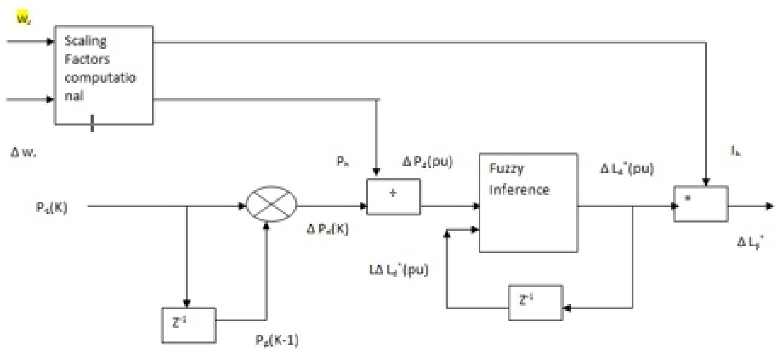


Fig.2.Efficiency optimization control block diagram

The above figure2 explains the fuzzy efficiency controller operation. The input dc power is sampled and compared with the previous value to determine the incrΔement Pd. In addition, the last excitation current decrement (L Δids) is reviewed. On these bases, the decrement step oΔf ids* is generated from fuzzy rules through fuzzy inference and defuzzification. The adjustable gains Pb and Ib, generated by scaling factors computation block, convert the input variable and control variable, respectively, to per unit values so that a single fuzzy rule base can be used for any torque and speed condition. The input gain Pb as a function of machine speed w can be given as P=a w+b Where the coefficients a and b were derived from simulation studies. The output gain Ib is computed from the machine speed and an approximate estimate of machine torque T_e $I_b = c_1\omega_r - c_2 T_e + c_3$ Again, the appropriate coefficients c_1, c_2, and c_3 were derived from simulation studies. In the absence of input and output gains, the efficiency optimization controller would react equally to a specific value of ΔP_d resulting from a past action Δ ids*(k-1), irrespective of operating speed. Since the optimal efficiency is speed dependant, the control action could easily be too conservative, resulting in slow convergence, or excessive, yielding an overshoot in the search process with possible adverse impact on system stability. As both input and output gains are function of speed, this problem does not arise. The above equation also incorporates the a priori knowledge that the optimum value ofΔ ids* is a function of torque as well as machine speed. In this way, for different speed and torque conditions, the same Δids*(p.u) will result in differeΔnt ids*, ensuring a fast convergence. One additional advantage of per unit basis operation is that the same fuzzy controller can be applied to any arbitrary machine, by simply changing the coefficients of input and output gains.

The membership functions for the fuzzy efficiency controller are shown below. Due to the use of input and output gains, the universe of discourse for all variables are normalized in the [-1, 1] interval. It was verified that, while the control variable Δids*, required seven fuzzy sets to provide good control sensitivity, the past control action L Δi_{ds}* (i.e. Δi_{ds}* (k - 1)) needed only two fuzzy sets, since the main information conveyed by them is the sign. The small overlap of the positive (P) and negative (N) membership functions is required to ensure proper operation of the height defuzzification method, i.e., to prevent indeterminate result in case L Δids* approaches zero. The rule base for fuzzy control is given below. The basic idea is that if the last control action indicated a decrease of dc link power, proceedsearching in the same direction and the control magnitude should be somewhat proportional to the measured dc link power change. In case the last control action resulted in an increase of Pd (ΔPd > 0), the search direction is reversed, and the Δids*, step size is reduced to attenuate oscillations in the search process.

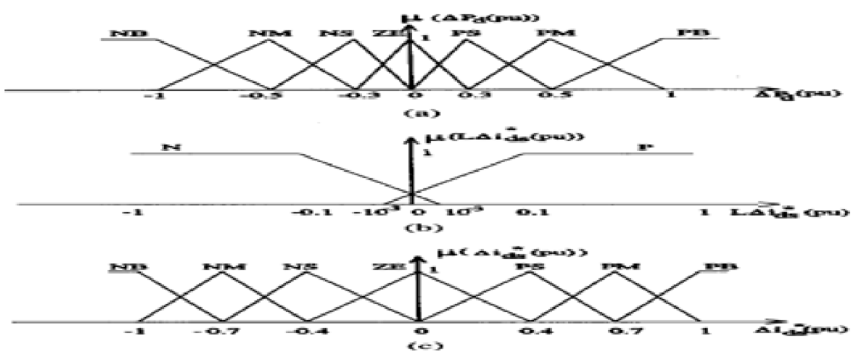


Fig.3a,b&c Membership functions for efficiency controller change of DC link power (ΔP_d(pu)) ,Last change in excitation current (LΔids*(pu)) & Excitation current control increment(Δids*(pu)).

Table I

$\Delta_{()}/\Delta_{()}$	N	P
PB	PM	NM
PM	PS	NS
PS	PS	NS
ZE	ZE	ZE
NS	NS	PS
NM	NM	PM
NB	NB	PB

Rules base for Fuzzy Efficiency Controller:

As the excitation current is decremented with adaptive step size by the fuzzy controller, the rotor flux Ψdr will decrease exponentially which is given by equation (18)

$$\frac{d}{dt}\psi_{dr}=\frac{L_m i_{ds}-\psi_{dr}}{T_r} \qquad (18)$$

Where λr = Lr/Rr is the rotor time constant and Lm the magnetizing inductance. The decrease of flux causes loss of torque, which normally is compensated slowly by the speed control loop. Such pulsating torque at low frequency is very undesirable because it causes speedripple and may create mechanical resonance. To prevent these problems, a feed forward pulsating torque compensator has been proposed.

Under correct field orientation control, the developed torque is given by equation (19)

$$T_e=K_t i_{qs}\psi_{dr} \qquad (19)$$

For an invariant torque, the torque current Iqs, should be controlled to vary inversely with the rotor flux. This can be accomplished by adding a compensating sΔiIgnal qs* to the original Iqs*' to counteract the decrease in flux ΔΨ dr (t) where t € [0, T] and T is the sampling period for efficiency optimization control. Let iqs (0) and Ψdr (0) be the initial values for iqs and Ψdr, respectively, for the k-th step change of ids*. For a perfect compensation, the developed torque must remain constant, and the following equality given by equation (20) holds good.

$$[\psi_{dr}(0)+\Delta\psi_{dr}(t)]\left[i_{qs}(0)+\Delta i_{qs}(t)\right]=\psi_{dr}(0)i_{qs}(0) \qquad (20)$$

Solving for ΔI_{qs} (t) yields

$$i_{qs}(t)=\frac{\psi_{dr}(t)i_{qs}(0)}{\psi_{dr}(0)+\psi_{dr(t)}} \qquad (21)$$

Where ΔΨdr (t) is governed by above equation with, substituted for Δ ids*. To implement such compensation, above equations are adapted to producΔe Iqs (t), using flux estimate Ψdr and command in Iqs* place of actual signals. A good approximate solution forΔI qs(t) can be obtained by replacing the denominator of the above equation by its steady-state value estimate ΔΨdr (t).In this case the compensation can be implemented in two steps as shown in Fig. 4. First, the value for the compensating torque current step is computed by discret $i_{qs}^{*}(k)$ given by equation (22) as

$$i_{qs}^{*}(k)=\frac{\psi_{dr}^{*}(k-1)\ \psi_{dr}(k)}{\psi dr^{*}}\quad i_{qs}(k-1) \qquad (22)$$

Next, the current step is processed through a first order low pass filter of rotor time constant, and then added to the previous compensating steps. This current is added to the original speed loop generated current Iqs*' so that, at any instant, the product Iqs Ψdr remains essentially constant.

II. Results

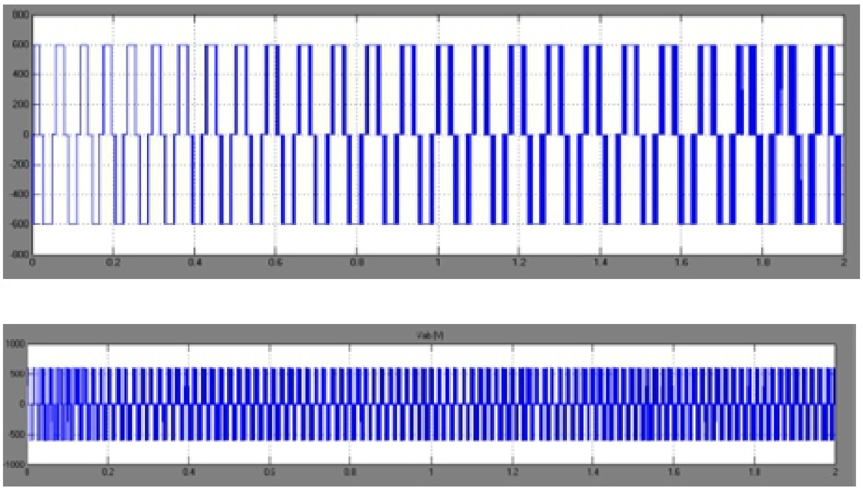
Table II

Specifications of the Induction Motor:

HP=5	Power rating of motor
V=440v	Voltage applied
F=50HZ	Frequency
N=1500RPM	Speed in RPM
P=4	No of poles
R_s = 0.406Ω	Stator Resistance
R_r = 0.478 Ω	Rotor Resistance
L_{ls} = 2.13mH	Stator Leakage Resistance
L_{lr} = 2.13mH	Rotor Leakage Resistance
L_m = 49.4mH	Mutual Inductance

The above block diagrams represent the Simulink model of the proposed methods. Figure 4 shows the model without fuzzy controller & only using PI control & Fig 5 shows the proposed fuzzy control model incorporated in vector control method. The simulation of the induction motor was done with fuzzy controller & without fuzzy controller for a simulation time of 2 seconds & the graphs for speed; voltage & torque with & without controller incorporated are as shown below.

Wave forms of voltages with PI Controller &fuzzy controller



From the simulation studies of voltage wave forms it can be concluded that the voltage is smooth & ripple free with more pulses in a given interval. Hence by using fuzzy controller the voltage is maintained at highest value with less distortion hence better performance.

Rotor speed wave forms with PI controller & with fuzzy controller

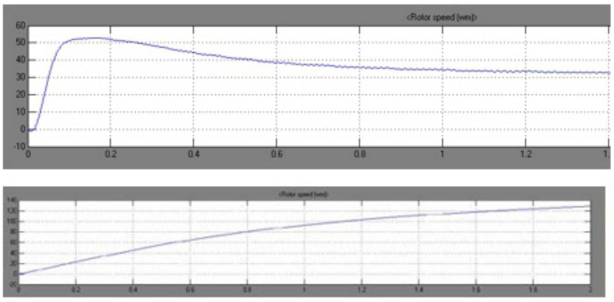


Fig. 7.

From the analysis of rotor speeds it is observed that the maximum speed that can be attained is more with controller & it is found that this speed is maintained constant. Whereas without controller the maximum speed attained is less & after a short while speed drops drastically . Hence with fuzzy controller maximum speed can be increased & maintained constant.

Wave form for torques with PI controller & with fuzzy controller

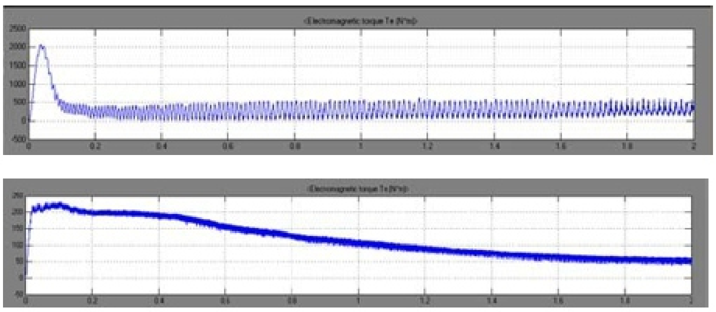


Fig. 8

The above simulation wave forms for torque show that the behavior of induction motor ie as torque decreases speed increases. The torque with controller is less distorted, almost all constant without jerky operation. Hence torque is improved & almost all constant with fuzzy controller

Conclusion:

This was observed mainly through the graphs obtained as outputs from the MATLAB simulink the graphs were obtained for the system with and without fuzzy controller The performance of the drive with fuzzy optimized controller is better when compared to without fuzzy controller. It is found from the graphs for voltage ,speed ,torque that voltage wave form is improved ,torque is stabilized & speed is increased with fuzzy controller . from this it is concluded that this controller gives better performance & it is superior ..It was further concluded that that for all load torques the output power is maintained almost constant. Input power minimization has been done at all load torques and the input power decreases i.e. the efficiency is increased at all loads.

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Effect Of EGR On HCCI Engine With Hydrogen As Fuel

P. V. Ramana[1], Dr. D. Maheswar,[2] Dr. B. UmaMaheswar Gowd [3]

[1]Asst. Professor, HI_Tech CE, Hyderabad

[2]Professor, MNREC, Sanga Reddy

[3]Professor, JNT University, Ananthapur

ABSTRACT

In recent years a great deal of work has been done and the research area has extended to all aspect of the combustion process. It has been gradually presenting a picture of energy saving and cleaner exhaust emissions. Increasing environmental concerns regarding the use of fossil fuels and global warming have prompted researchers to investigate alternative fuels. All engine manufacturers of today are seriously challenged, not only by legislative demands of low emissions, but also by the need to decrease the dependency on non-renewable fuels, such as oil. Hydrogen (H2) has been suggested as a possible replacement for the fuels used today ,the way to reach this goal is to using new combustion concepts, such as Homogeneous Charge Compression Ignition Homogeneous Charge Compression Ignition (HCCI) engines promise high thermal efficiency combined with low levels of nitric oxide and particulate matter emissions where HCCI engines can operate on gasoline, diesel fuel, and most alternative fuels and studies suggest that the concept of an HCCI engine running on H2 would result in a locally emission free engine. The scope of this proposal is to study the key processes involved with H2 HCCI combustion and examine whether the mechanisms of these processes can be easily incorporated into current engines further more to investigate broaden the stable operations range for HCCI a series of experiments were analyzed using hydrogen fuel adaptation and their composition and the exhaust mechanism of the exhaust gas.

Keywords: HCCI, combustion, fuel efficiency, pollutant emission, alternate fuels, Hydrogen fuel, EGR

1. Introduction

Compression ignition engine are preferred prime movers due to excellent drivability and higher thermal efficiency. Despite their advantages they produce higher levels of NOx and smoke emissions which will more harmful to human health. Hence stringent emission norms have been imposed. In order to meet the emission norms and also the fast depletion of petroleum oil reserves lead to the research for alternative for diesel engines.

Diesel engines currently offer significant fuel consumption benefits relative to other power plants for on and off road applications; however, costs and efficiency may become problems as the emissions standards become even more stringent. Homogeneous charge compression ignition (HCCI) combustion offers a solution to this problem as it achieves gasoline-like emissions while keeping diesel-like fuel-efficiency by operating at lower combustion chamber temperatures. Furthermore, HCCI combustion works with gasoline, diesel and most alternative fuels, giving it a major advantage for future developments.

The concept of homogeneous charge compression ignition (HCCI) has been described before by a number of researchers. Onishi, et al. (1979) experimentally studied homogeneous charge compression ignition in a two-stroke engine and achieved low cyclic variation at idle and up to 40% load. as reported in [22]. Investigators worldwide are developing HCCI engines as this technology has not matured sufficiently. They can be used in either SI or CI engine configurations with a high compression ratio (CR). HCCI engines work without the help of diesel injectors or spark plugs and can achieve high engine efficiency with low emission levels. General Motors Corporation (GM) has unveiled a prototype car with a gasoline HCCI engine and it was claimed that it could cut fuel consumption by 15% [23]. The engine is able to virtually eliminate NOx emissions and lowers throttling losses which assists better fuel economy [24].

In HCCI engines, the fuel and air are premixed to form a homogeneous mixture before the compression stroke. As a result, the mixture ignites throughout the bulk without discernable flame propagation due to occurrence of auto ignition at various locations in the combustion chamber (multi-point ignition). This may cause extremely high rates of heat release, and consequently, high rates of pressurization [3-5]. In HCCI engines, auto-ignition and combustion rate are mainly controlled by the fuel chemical kinetics, which is extremely sensitive to the charge composition and to the pressure and temperature evolution duringthe compression stroke, therefore HCCI combustion is widely assumed to be kinetically controlled [3, 6, 7]. The main objective of HCCI combustion is to reduce soot and NOX emissions while maintaining high fuel efficiency at part load conditions [2, 8]. In some regards, HCCI combustion combines the advantages of both spark ignition (SI) engines and compression ignition (CI) engines [8, 9]. The results from experiment and simulation show that the HCCI combustion has a low temperature heat release and a high temperature heat release, and both heat releases occur within certain temperature ranges. The low temperature heat release is one of the most important phenomena for HCCI engine operation and the occurrence of it depends chemically on the fuel type [10-12].

However there are certain number of obstacles and problems in its application that have not been resolved. These problems are the control of ignition and combustion, difficulty in operation at higher loads, higher rate of heat release, higher CO and HC emissions particularly at light loads, difficulty with cold start, increased NOX emissions at high loads and formation of a completely homogeneous mixture [13-15]. The lack of a well-defined ignition timing control has led a range of control strategies to be explored. Numerous studies have been conducted to investigate HCCI combustion control methods such as intake air preheating [14, 16, 17], Variable Valve Actuation (VVA) [4], Variable Valve Timing (VVT) [1], Variable Compression Ratio (VCR) [18] and EGR rate [10]. Moreover many studies also focused on the effects of different fuel physical and chemical properties to gain control of HCCI combustion [9, 19-21].

Fuel flexibility in HCCI can be applied for a wide range with different octane/cetane numbers. The combustion process of a HCCI engine has little sensitivity to fuel characteristics such as lubricity and laminar flame speed. Fuels with any Octane or Cetane number can be burned, although the operating conditions must be adjusted to accommodate different fuels, which can impact efficiency. An HCCI engine with variable compression ratio or variable valve timing could, in principle, operate on any hydrocarbon or alcohol liquid fuel, as long as the fuel is vaporized and mixed with air before ignition. Besides gasoline[25] and diesel fuel [26], a variety of alternative fuels, such as methanol [27], ethanol [28,29], natural gas [30], biogas [31], hydrogen [32], DME [27] and their mixtures [33-35], including also gasoline and diesel mixtures and different mixtures of iso-octane with heptane [36], have been experimentally proved as possible fuels for HCCI combustion in both two-stroke and four-stroke engines.

The purpose of the present study is to examine the effect of various operating variables of a homogeneous charge compression ignition (HCCI) engine fueled with hydrogen.

Hydrogen As A Fuel

Since the HCCI engine depends on the cylinder charge auto igniting, the use of high compression ratios is required. With no charge heating, this was found to be between 18:1 and 25:1 when carrying out simulation studies for the engine. Conventional spark ignition engines are typically limited to a compression ratio of approximately 10:1. The combination of fast heat release in HCCI mode and the use of a lean cylinder charge, gives close to constant volume combustion with low peak gas temperatures leading to reduced heat transfer losses through the cylinder walls and high indicated thermal efficiencies.

Hydrogen possesses some features that make it attractive for use as a fuel in internal combustion engines, enabling fast, close to constant volume combustion, high combustion efficiency and low emissions. Numerous authors have investigated the use of hydrogen in spark ignition (SI) engines, and the feasibility of hydrogen as a fuel in [37]such engines is well established. An overview of the characteristics of hydrogen as a fuel for SI engines was presented by Karim [38].

The flame speed of hydrogen is higher and hydrogen allows operation at significantly higher excess air ratios than conventional hydrocarbon fuels. This enables extended lean burn operation of the engine, potentially leading to a drastic reduction of NOx emissions. High diffusivity and low quenching distance avoids poor vaporization problems. Emissions of carbon monoxide and un burnt hydrocarbons are practically eliminated with a hydrogen fuelled engine, as the only source of carbon will be the lubricating oil. For the same reason the engine does not emit carbon dioxide. The only non-trivial exhaust gas emissions will be nitrogen oxides, which result from the oxidation of atmospheric nitrogen under high temperatures. It will be shown below that with HCCI operation and a very lean mixture this pollutant can be reduced to near-zero levels. The ignition energy for hydrogen is low, however the temperature required for auto ignition is significantly higher than that of conventional hydrocarbon fuels. Therefore, CI engines using hydrogen fuel require high compression ratios and/or pre-heating of the inlet air to ensure auto ignition.The engine used on this research work was a four-stroke, single cylinder, direct injection, naturally aspirated, air-cooled CI engine. The engine was coupled to a hydraulic pump system so that the engine could be operated at varying load conditions. The engine was fitted with an injection system allowing hydrogen to be mixed with the inlet air. Using this setup, the performance of the engine operating in hydrogen fuelled HCCI mode and normal diesel fuelled mode investigated and compared. With the engine running at 2200 rpm, air inlet temperature set at 93°C and a hydrogen flow rate of 90dm3/min, HCCI mode of operation was tested.

With the objective of determining the leanest cylinder charge that the hydrogen fuelled HCCI engine can operate smoothly, a series of test were carried out with varying fuel-air ratio. Figure 1 illustrates the brake thermal efficiency as a function of the excess air ratio, λ. It can be seen that the engine is able to operate with extremely lean cylinder charges and still maintain a relatively high thermal efficiency when compared to conventional diesel engine operation. Figure 2 shows the in-cylinder gas pressure for 10 consecutive engine cycles with the engine running at 2200 rpm with an air excess ratio of 3. The engine brake thermal efficiency under these conditions was found to be approximately 45%. This is a significant increase in the thermal efficiency compared to its value under conventional operation using diesel fuel.

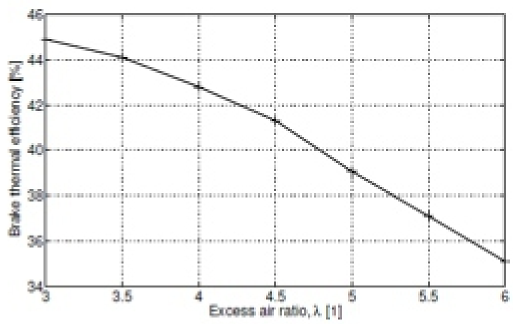

Figure 1: Engine brake thermal efficiency.

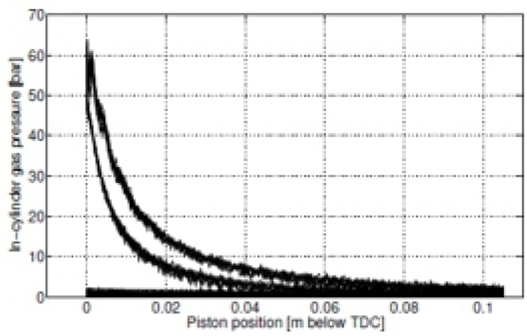

Figure2:Cylinder pressure plots in HCCI

The ignition angle (αign), governs the combustion process, and control of the ignition timing is of high importance in order to optimize engine operation. Figure 3 shows how the ignition timing is nearly linearly dependent on the inlet air temperature for the HCCI operational mode. This indicates that control of the inlet air temperature can be used to control ignition timing for the hydrogen fuelled HCCI engine. Figure 4 indicates that an increase in the air inlet temperature results in a decrease of engine power output. The brake thermal efficiency and the indicated mean effective pressure decrease with an increase in the air inlet temperature.

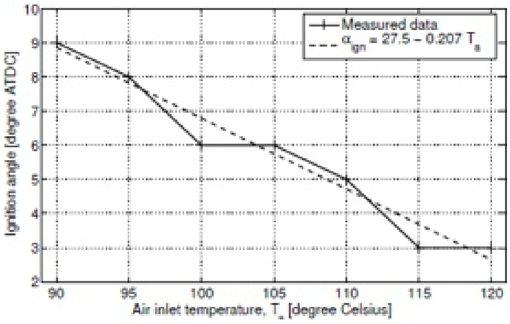

Figure 3: Ignition angle for Varying engine power out put.

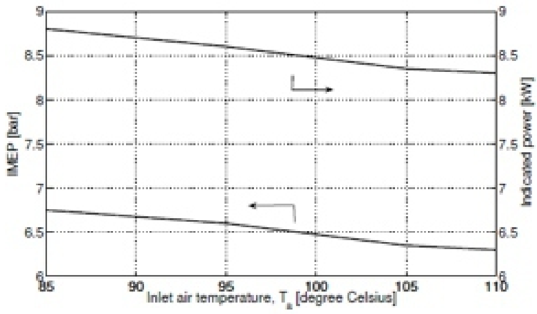

Figure 4: Effect of inlet air temperature on inlet air temperature

The exhaust emissions were measured while the engine was operated in hydrogen fuelled HCCI mode at a speed of 2200 rpm and Ta of 100◦C. The results of the test are presented in Figure 5 As can be seen, the NOx emissions increase sharply for λ > 3.5, due to the increasing in-cylinder gas temperatures, and become negligible for higher values of λ. The NOx levels are considerably lower that what would be expected for conventional diesel engine operation for all the cases investigated.

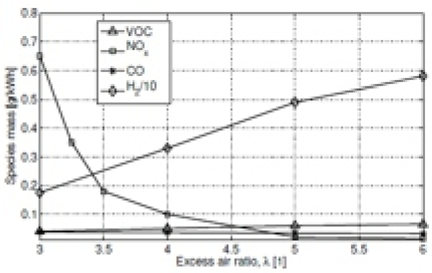

Figure 5: H_2 HCCI engine exhaust gas emissions levels.

The Experimental results[39] that presents that, the peak in-cylinder pressures and the rates of pressure rise were higher in the HCCI hydrogen engine than for conventional operation on diesel fuel, limiting the HCCI engine to part load operation and potentially requiring design changes to maintain engine reliability. The fuel efficiency obtained was, however, significantly higher than that obtained when operating as a conventional diesel fuelled engine, and high efficiency was obtained even with very lean cylinder charges. The inlet air had to be heated in order to ensure auto ignition and it was demonstrated that the inlet air temperature is the most useful variable to control ignition timing. Engine emissions were measured shown that negligible levels of all exhaust emissions were produced, including nitrogen oxides, compared to conventional diesel-fuelled operation.

Better understanding of the HCCI combustion process can be greatly aided by exploration of the chemical processes occurring in the combustion process, such as the effect of fuel structure on combustion timing. It is possible to observe combustion characteristics of the fuel-in-air charge by collecting exhaust samples at different combustion timing.

Combustion timing is determined by a number of different parameters, such as equivalence ratio, intake manifold pressure, and intake manifold temperature. The primary influence on combustion timing is the intake manifold temperature of the fuel-in-air mixture inducted into the engine combustion chamber.

The devices such as electrical heaters, heat exchangers, and exhaust gas recirculation (EGR) control the intake manifold temperature(Tin).The composition of the fuel also plays a major part in the ignition process, as different fuels possess different auto ignition characteristics. Hountalaous et al using 3D-multi dimensional model to examine the effect of EGR temperature on a turbocharged DI diesel engine with three different engine speeds, and they reported that high EGR temperature affects the engine brake thermal efficiency, peak combustion pressure, air fuel ratio and also soot emissions, and the combined effect of increased temperature and decreased O_2 concentration resulted low NOx emissions. Also they suggested that EGR cooling is necessary to retain the low NOx emissions and prevent rising of soot emissions without affecting the engine efficiency at high EGR rates. Ken Satoh et al [4] investigated on a naturally aspirated single cylinder DI diesel engine with various combinations of EGR, fuel injection pressures, injection timing and intake gas temperatures affect exhaust emissions and they found that NOx reduction ratio has a strong correlation with oxygen concentration regardless of injection pressure or timing. NOx reduction ratio is in direct proportion to intake gas temperatures. EGR may adversely affect the smoke emission because it lowers the average combustion temperatures and reduces the oxygen intake gases, which in turn keeps soot from oxidizing.

Effect Of EGR In HCCI Engine

According to different methods of the exhaust gas recirculation, EGR technique can be divided into internal EGR and external EGR. The internal EGR rate can be obtained by changing valve overlap period to negative valve overlap (NVO) and the external EGR rate can be adjusted by using an EGR valve. For high-octane fuel (such as: gasoline) HCCI, negative valve overlap (NVO) is recognized as one of the possible implementation strategies of HCCI closet to production. The effect of inlet air temperature through NVO is insignificant when the engine runs well inside of the HCCI operating range [40, 42]. The effect of EGR on HCCI combustion can be divided into three parts: a dilution effect (inert gasses present in the EGR), a thermal effect (heat exchange, thermal loss to the wall, EGR ratio mixture quality, EGR temperature, heat capacity), and a chemical effect. The chemical effect influences not only the overall kinetics, but it also can change a specific reaction path, which makes this effect particularly interesting for the investigation of the auto ignition process [43]. The effects of EGR that have been investigated are: increase in intake charge temperature (heating effect), reduction of oxygen concentration (dilution effect), increase in specific heat of the mixture (heat-capacity effect), chemical interactions involving the CO2 and H2O species of the recycled burned gases (chemical effect), and stratification of the recycled burned gases (stratification effect) [41,44]. The dilution and heat capacity effects are responsible for reducing the heat-release rates and delaying HCCI combustion. The heating effect is mainly responsible for the advance timing of auto-ignition, and the residuals stratification effect facilitates HCCI combustion. Reactive species, present in the residuals, facilitate auto-ignition, and the inert species slow down the combustion rate (dilution effects) [41, 43, 44]. EGR provides the appropriate temperature to enhance auto- ignition, while maintaining the combustion temperature sufficiently low.

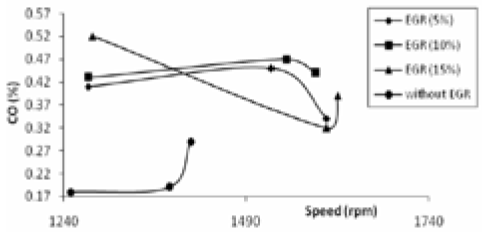

Figure 6: Effect of engine speed variations on CO emission of emission of dual fuel HCCI-DI

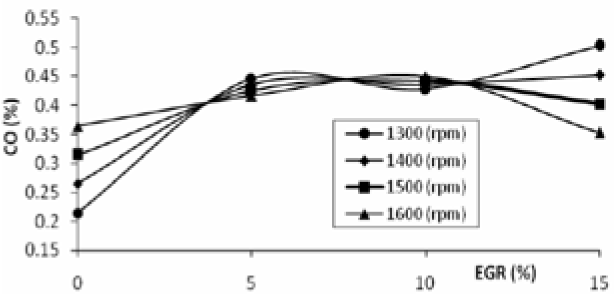

Figure7: Effect of EGR on CO on dual fuel HCCI-DI engine at different Engine speeds

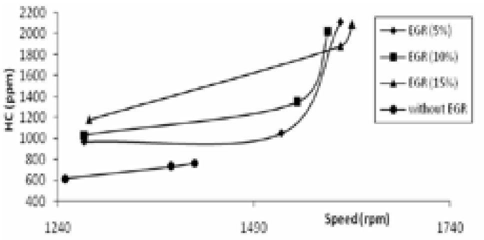

Figure8: Effect of engine speed variations on HC emissions of dual fuel HCCI-DI engine

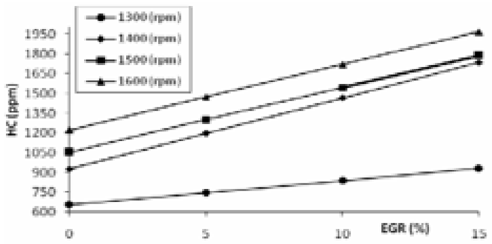

Figure9: Effect of EGR on HC emissions of dual fuel HCCI-DI engine at different engine

Speeds

The research was done to study the effects of EGR rate and engine speed variation on CO and HC emissions of gasoline-diesel dual fuel, Hydrogen fuelled HCCI engine & [45,46]. Results can be summarized as:

1) An advantage of dual fuel engine, which was noticed in all tests, was fast and easy transition to HCCI mode.
2) Increasing EGR dilutes the intake charge and reduces the amount of oxygen. Dilution also decreases combustion temperature and leads to incomplete HCCI combustion and therefore increases CO emission.
3) The insufficient time for formation of homogeneous charge mixture, caused by increasing engine speed, results in increase of CO emissions.
4) High engine speeds in HCCI mode results in insufficient time for formation of homogeneous mixture cause more HC emission due to incomplete HCCI combustion.
5) Increasing EGR rate dilutes the intake charge and reduces its oxygen. Dilution also decreases combustion temperature, which results in reduction of the amount of burnt fuel thus HC emission increases in comparison with no EGR.
6) HCCI can be induced and controlled by varying the mixture temperature, either by Exhaust Gas Recirculation (EGR) or intake air pre-heating.
7) A combination of HCCI combustion with hydrogen fuelling has great potential for virtually zero CO 2 and NO X emissions. Nevertheless, combustion on such a fast burning fuel with wide flammability limits and high octane number implies many disadvantages, such as control of backfiring and speed of auto

ignition and there is almost no literature on the subject, particularly in optical engines.

8) Experiments were conducted [46] in a single-cylinder research engine equipped with both Port Fuel Injection (PFI) and Direct Injection (DI) systems running at 1000 RPM. Optical access to in-cylinder phenomena was enabled through an extended piston and optical crown. Combustion images were acquired by a high-speed camera at 1° or 2° crank angle resolution for a series of engine cycles.

9) Spark-ignition tests were initially carried out to benchmark the operation of the engine with hydrogen against gasoline. DI of hydrogen after intake valve closure was found to be preferable in order to overcome problems related to backfiring and air displacement from hydrogen's low density.

10) HCCI combustion of hydrogen was initially enabled by means of a pilot port injection of n-heptane preceding the main direct injection of hydrogen, along with intake air preheating.

Conclusion

From the study it was observed that the peak in-cylinder pressures and the rates of pressure rise were higher in the HCCI hydrogen engine than for conventional operation on diesel fuel, limiting the HCCI engine to part load operation and potentially requiring design changes to maintain engine reliability.

As a consequence of the higher rates of pressure rise and peak pressures, additional design considerations must be given to the piston pin, crank bearings and piston rings, since their load carrying capacity should be taken into account if the reliability of the engine is to be maintained.

Sole hydrogen fuelling HCCI was finally achieved and made sustainable, even at the low compression ratio of the optical engine by means of closed-valve DI, in synergy with air-pre- heating and negative valve overlap to promote internal EGR. Various operating conditions were analyzed, such as fuelling in the range of air excess ratio 1.2–3.0 and intake air temperatures of 200–400 °C. Finally, both single and double injections per cycle were compared to identify their effects on combustion development.

Further the study analyzed that compression ratios beyond 12 are likely to produce severe knock problems for the richer mixtures used at high load conditions. It seems that the best compromise is to select the highest possible CR to obtain satisfactory full load performance. The choice of optimum compression ratio is not clear; and it may have to be tailored to the fuel and other techniques used for HCCI control. So , a variety of physical control methods (e.g., EGR) have been examined in an effort to obtain wider stable operation. From these investigations and many others in the past few years it appears that the key to implementing HCCI is to control the charge auto ignition behavior which is driven by the combustion chemistry.

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An Introduction To The Decision Trees And Its Types

Navid Samimi Behbahan[1], Milad Samimi Behbahan [2]
Department of Computer Engineering, Behbahan Branch, Islamic Azad University, Behbahan, Iran

A B S T R A C T

The structure of the decision tree in machine learning is a predictive model which connects all of the observed facts about a phenomenon to the amount of the goal of that phenomenon. The technique of machine learning in order to conclude a decision tree from data is called "decision tree" which is one of the most common ways of data analysis.

1. Introduction

Any corresponding internal node of a variable and any arc to a child demonstrate a possible amount for that variable. By having the amount of variables, which is demonstrated by a route from the root to the node, a loaf demonstrates the predicated amount of the target variable. A decision tree shows a structure in which loafs indicate categories and branches represent the seasonal composition of traits that result in these categories.

Any attributes in decision tree are classified into two types of classification attributes and true attributes in which classification attributes are kind of values which accept two or more discrete values (symbolic characteristics) while real qualities are only from the real numbers [1].

2- The Main Objectives Of The Classifying Decision Trees

- ✓ To classify the input data as accurate as possible
- ✓ To universalize the learnt knowledge from the the training data in a way that it can classify the unseen data as accurate as possible.
- ✓ In case of addition of new training data, training tree should be able to expand
- ✓ The structure of the resulting trees should be the simplest form possible.

3 - Types Of Decision Trees

- ✓ When the outcome of a tree, is a discrete set of possible values, it is called classification tree (for e.g. male or female, winner or loser). These trees represent the function X→C in which C accepts discrete values.
- ✓ When we are able to consider the outcome of the tree a "real value", we call it regression tree. (For

e.g. price of a house or length of stay of a patient in a Hospital). These trees predict the numbers in the node of a loaf and they can use a linear regression or fixed model (mean) or other types of models.

- ✓ CART stands for Classification and Regression Tree. The name refers to both procedures mentioned above.
- ✓ Cluster trees only classify the samples in nods of the loafs

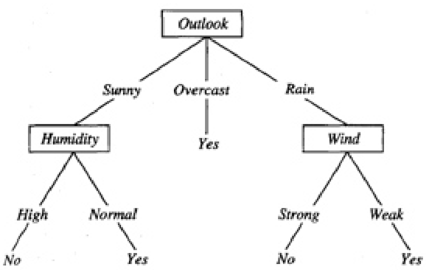
Most of the researches in machine learning are focused on classification trees [2].

4 - Representation Of Decision Tree

Decision trees, classify the samples by arranging them in the tree from the nod of the root to the nod of loafs. Any internal nod in a tree examines a quality of a sample and any branch coming out of a node corresponds to a possible value for that quality. Furthermore each node of a loaf is attributed to a category. Each sample is classified with starting from the node root of the tree and examining the quality by the node and moving forward along the branch corresponding to given quality value in the sample. This process is repeated for each sub tree whose root is a new node [3].

Example:

The following decision tree corresponds to the phrase below:



(Outlook=Sunny $\wedge$ Humidity=Normal) $\vee$ (Outlook=Overcast) $\vee$ (Outlook=Rain $\wedge$ Wind=Weak)

Figure 1: Converting a mathematical equation to a corresponding decision tree

Another way to show the above decision tree

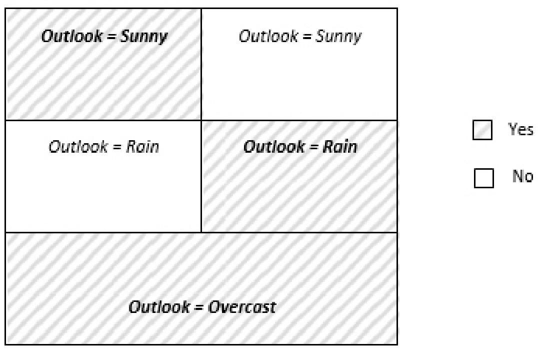


Figure 2: showing the decision tree

Example : representation of the decision tree for the for the function $A \wedge \sim B$

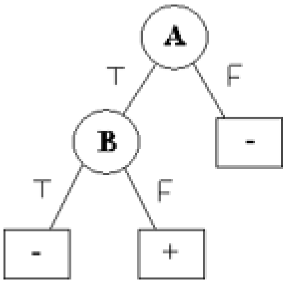


Figure 3: representing the decision tree for mathematical expression

5 - Expanding Decision Trees With Decision Graphs

Decision graphs are generalizations of decision trees which have leafs and decision nodes. A characteristic that distinguishes decision tree from decision graph is that decision graphs can have linkage. Linkage is a situation in which two nodes have an in common child and this situation represents two subsets which have common features, because of that they are considered to be one set [4].

In decision tree all the paths from the nod of the root to the nod of the loaf, move forward by the conjunction AND. In a decision graph it is possible to use disjunctions or OR to connect two or more paths.

For example classifying for the function $(A \wedge B) \vee (C \wedge D)$ is different. Corresponding decision tree and decision graph of this function is shown in the Fig. below. Decision tree divides the object's space into seven categories while the decision graph divides it into two categories [5].

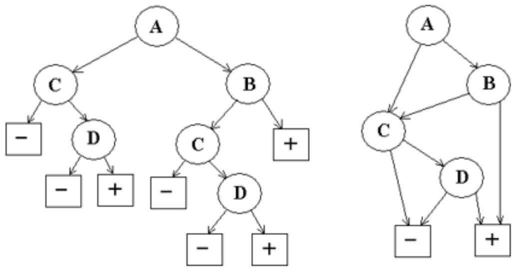


Figure 4: converting the decision tree to the decision graph

6 - Issues Suitable For Decision Tree Learning

There are many types of methods developed to learn decision trees with different abilities and requirements [6]. But in general, learning it is suitable for issues with the following features:

- ✓ Issues in which samples are represented in form of attribute-value pairs. In this kinds of issues samples are expressed with fixed set of attributes and values. The simplest situation to learn a decision tree is when any attributes take small number of discrete amounts. However, by developing to basic algorithms, it is also possible to use attributes with continuous values.

Example: attribute: temperature   quantity: {warm, temperate, cool}

- ✓ Issues in which the target function has discrete output values_ the decision tree methods can be expanded in such a way that they can learn to function with output of more than one value.
- ✓ Example: the output of a supposed target function {Yes, No}
- ✓ Seasonal attributes might be required_ decision trees are essentially representing seasonal phrases.
- ✓ Educational data may contain errors_ methods of learning a decision tree are resistant to errors in training data.
- ✓ Training data may have attributes which lack values_ the decision tree methods may be used when some of the training examples have unknown amounts.

7 - Conclusion:

Decision trees are capable of producing human-understandable descriptions from the data sets and can be used for classification and prediction. This technique can be widely used in different fields like diagnosing classifications of plants and customer's marketing strategies.

8. References

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