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ABSTRACT

The Homogeneous Charge Compression Ignition (HCCI) engine combustion uses heat energy from trapped exhaust gases enhanced by the piston compression heating to auto ignite a premixed air/gasoline mixture. As the HCCI combustion is controlled by the charge temperature, composition and pressure, it therefore, prevents the use of a direct control mechanism such as in the spark and diesel combustion. Using a large amount of trapped residual gas (TRG), is seen as one of the ways to achieve and control HCCI in a certain operating range. By varying the amount of TRG in the fresh air/fuel mixture (inside the cylinder), the charge mixture temperature, composition and pressure can be controlled and hence, the auto ignition timing and heat release rate.

The controlled auto ignition (HCCI) engine concept has the potential to be highly efficient and to produce low NOx, carbon dioxide and particulate matter emissions. It has however been found that the TRG promoted HCCI combustion mainly depends on the quantity and quality of TRG, that on the other hand depend on the combustion quality of the previous cycle, valve timing, engine load and speed. In that way, the operating range in terms of engine load and speed, for a naturally aspirated HCCI engine, is restricted by a misfire at low load and by fierce (knocking) combustion at high load.

One possible approach to extend the operating range of the HCCI combustion is to influence quality of the TRG by adjusting the coolant temperature. The engine coolant temperature influences the in-cylinder heat transfer process, which in turn influences the charge mixture temperature and therefore the HCCI combustion process itself.

The aim of this paper is to present tests and results obtained on the single cylinder research engine, equipped with a Fully Variable Valve Train (FVVT) run over a range of coolant temperature in the HCCI combustion mode and fuelled with gasoline fuel. The results obtained suggest that with reducing the coolant temperature, the high load limit can be extended up to 14%, while with increasing the coolant temperature the low load limit can be extended up to 28%.

INTRODUCTION

The HCCI combustion is a process that combines features of the spark ignition (SI) and compression ignition (CI) processes. In a HCCI engine the air and fuel are premixed prior to ignition and then ignited by the compression from the piston motion. The ignition is provided in multiple points and therefore the charge gives a parallel energy release. This results in the uniform and simultaneous auto-ignition and chemical reaction throughout the whole charge without flame propagation. In the HCCI combustion, chemical kinetics of the air/fuel mixture play the crucial role with no requirements for turbulence and mixing [1-3].

The HCCI engine concept has the potential to be highly efficient and to produce low NOx, carbon dioxide and particulate matter emissions. However, problems with cold start, running at idle and high loads, together with controlling the combustion over the entire load/speed range limits its practical application.

Using a large amount of trapped residual gas-TRG, is seen as one of the ways to achieve and control HCCI in a certain operating range. By varying the amount of TRG in the fresh air/fuel mixture (inside the cylinder), the charge mixture temperature and composition can be controlled and hence, the auto ignition timing and heat release rate [4].

The operating range in terms of engine load and speed, for a naturally aspirated HCCI engine, is restricted by a misfire at low load and by a high rate of pressure rise (knock) at high load. As a consequence, a possible area for HCCI operation, in a naturally aspirated, port injection engine for passenger car application is fairly small. A major problem in applying HCCI for commercial use is how to expand the operating range in order to cover entire or great parts of the NEDC (in Europe), FTP-75 (in USA) and Japan 10-15 driving cycles.

The objectives of this study are to investigate possibilities for extending the operating range of the HCCI combustion by adjusting the engine coolant temperature. The coolant temperature influences the in-cylinder heat transfer process, which in turn influences the charge mixture temperature and therefore the HCCI combustion process itself.

The experiment is performed on the single cylinder research engine equipped with a Fully Variable Valve Train (FVVT) system-Lotus Active Valve Train (AVT) and fuelled with commercial gasoline fuel. The results obtained are presented and discussed.

EXPERIMENTAL APPARATUS AND SET UP

A single cylinder research engine, with variable valve timing and measurement equipment for emissions, fuel consumption, air fuel ratio, temperatures and cylinder pressure monitoring is used for testing.

ENGINE

The engine employed in this research is a single cylinder, 4 stroke research engine based on a GM Family One 1.8 litre series architecture. In Fig. 1 a photograph of the engine is shown. It has a production piston and stroke, with a standard 4-cylinder head on top of a water cooled barrel to join the family one parts to the custom made bottom end. Only the front cylinder of the head is operational. The water jacket uses a combination of machined modifications and brackets. Unnecessary water transfer ports are blanked off. The engine has a specially designed single cylinder bottom end, with a capacity to cancel primary and secondary forces thus allowing either pure combustion work or optical access versions to be built.

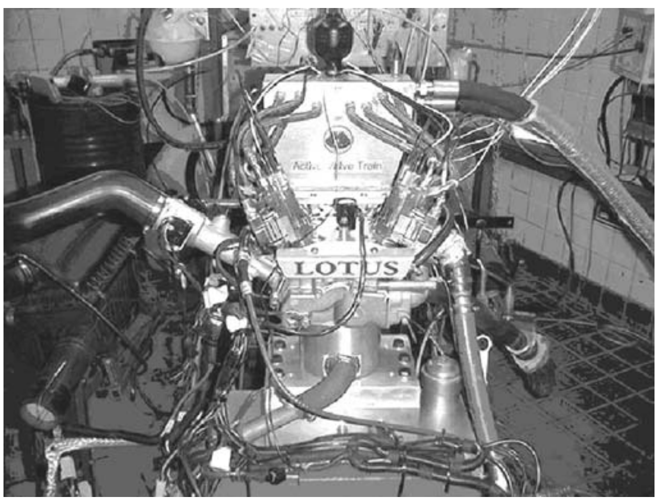

Fig. 1 Single-cylinder research engine with the AVT system

The use of conventional parts in the combustion system wherever possible ensures that the cost of rebuild is low in case of any component failures. The compression ratio can easily be changed in this engine, both because of the separate barrel and, more importantly, because of the AVT system which overrules the need for modifications of belt runs etc. Any change in compression ratio is achieved by means of the deck height being moved up and down by spacers or special short liners, or a combination of the two. The bottom end can accept various strokes up to and including 100mm, and is capable of running to 7000rpm (depending on stroke).

The major engine specifications and tests conditions are shown in Table 1. The detail description of the engine can be found in [5].

Table 1 Single cylinder engine specification

Bore	80.5 mm
Stroke	88.2 mm
Swept volume	450 cm^3
Compression ratio	10.5
Number of valves per cylinder	4
Valve control	Electro-hydraulic Lotus AVT system
Fuel injection	Port fuelled
Fuel	Standard gasoline (95 RON)
Equivalence air/fuel ratio	Stoichiometric
Engine speed	2000 rpm
Intake temperature	25^0 C
Inlet pressure	Naturally aspirated
TRG	Up to 80%

The research AVT system is fitted to allow the variable valve timing strategy to be used to trap the pre-defined quantity of TRG. The open and closing timings of the four electro-hydraulically driven valves are independently variable and are digitally controlled. Valve opening profiles can be selected and entered into the software by the user. The control software uses inputs from a crankcase encoder and valve linear displacement transducers to facilitate a closed-loop control to satisfy a `desired versus actual' position control until the required profiles are achieved. Fine tuning of valve profiles is accomplished by using valve-specific gain controllers.

The engine was connected to a Froude AG30, 30KW eddy-current dynamometer. A redline ACAP data acquisition system from DSP Technologies Inc. was used together with Horiba MEXA 7100 DEGR heated line emission analyser. The fuel was port injected and the engine management system was a conventional Lotus V8 controller.

MEASUREMENT PROCEDURE

Before each experiment it was necessary to ensure that the engine was fully warm, the valve actuators and measurement equipment calibrated and there were no changes in behaviour since the previous experiments.

The cycle resolved data was sampled for 300 consecutive cycles with a resolution of one crank-angle (CA) degree. The various measured and calculated variables can be displayed for each individual cycle as well as those averaged from over 300 cycles.

The data presented for the peak cylinder pressure (PCP), IMEP gross, coefficient of variation (COV IMEP), crank angle of 50% mass fraction burned (CA50), maximum pressure rise rate (dP/dCA) and heat release rate (HRR) were averaged from over 300 cycles for each step (by ACAP data acquisition system from DSP Technologies). The in-cylinder and coolant temperatures, HC and NOx emissions were time based values from Texcell system. Air mass flow rate was calculated from the fuel flow and air/fuel ratio measurement.

TECHNIQUES FOR HCCI COMBUSTION

The technique used to initiate and to control the HCCI combustion relies on the trapping of a pre-determined quantity of TRG by closing the exhaust valves relatively early in the exhaust stroke and by opening the inlet valves relatively late (usually symmetrically about TDC) in the intake stroke. The general principle can be seen in Fig. 2.

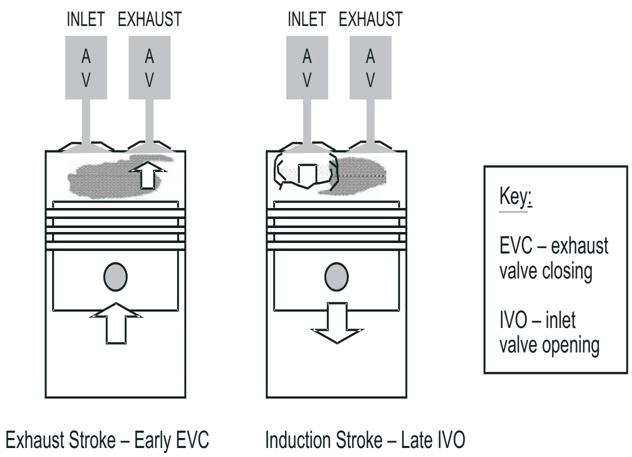

Fig. 2 The sequential valve event strategy

Following exhaust valve closure the TRG is then compressed during the final stage of the exhaust stroke. As the piston descends on the intake stroke, the inlet valves are opened and fresh charge is drawn into the cylinder which is partially filled with exhaust gases. At the end of the intake stroke the inlet valves are closed and the mixture of a fresh charge and residual gas is then compressed in the next compression stroke. The HCCI combustion occurs as the mixture temperature increases in the final stage of the compression stroke. Once the HCCI has occurred, the power stroke drives the piston down and the cycle is thus repeated. This method is named the *sequential method* (often refer in the literature as the recompression method). One more method for achieving HCCI combustion, the *simultaneous method* (often refer in the literature as the re-breathing method), has also been derived [5]. Generally, in this method, as the piston reaches BDC from the power stroke, the exhaust valves are opened and all of the exhaust gases are expelled from the cylinder. As the piston passes TDC, on the induction stroke, both inlet and exhaust valves are opened simultaneously and fresh charge and exhaust gas are together drawn into the cylinder. Again, the HCCI combustion occurs as the mixture temperature increases in the final stage of the compression stroke. Once HCCI has occurred, the power stroke drives the piston down and the cycle is thus repeated. Detail explanations of these two methods can be found in [5, 6].

In this experiment the engine is started in SI combustion mode. When a sufficient engine temperature is reached (90^0C coolant temperature) the transition from SI to HCCI mode is achieved by increasing the negative valve overlap, i.e. by using the sequential method. In all experiments the basic coolant temperature is 90 ^{0}C, and it is reduced for the upper operating range limit extension and increased for the lower operating range limit extension.

RESULTS AND DISCUSSION

The experimental study performed in this research can be generally divided into two groups: the extension of the upper operating range limit and the low operating range limit. For both studies, engine parameters during these investigations are kept constant at values specified in Table 1. During the HCCI combustion, the spark plug is left activated at TDC position for cleaning purpose only and it has no clear influence on the HCCI combustion performance.

EXTENDING THE UPPER OPERATIONAL RANGE LIMIT

The first group of engine tests are centred on the influences of coolant temperature on extending the upper limit of a stable HCCI combustion mode. At upper limit a HCCI engine can be restricted by a fast burn rate that causes a knocking combustion. To prevent this, either a higher dilution of the engine charge could be used (assuming the balance between dilution rate and increased heat input is in favour of slowing down the combustion) or a higher rate of heat transfer (between the engine charge and the cylinder walls) has to be implemented. The latter approach is chosen and it is done by decreasing the coolant temperature.

The coolant temperature is decreased from 90 ^{0}C to 65 ^{0}C and the engine is set at the wide open throttle (WOT) position. The timings for exhaust valve event are set at EVO 148 ^{0}CA, EVC 298 ^{0}CA whilst for the inlet valve event at IVO 422 ^{0}CA, IVC 578 ^{0}CA with the valve lifts of 2.5 mm for both valve events. The valve events and therefore the TRG amount (~40% by volume) are kept unchanged during the coolant sweep. The engine parameters are kept constant at values specified in Table 1.

With decreasing the coolant temperature the heat transfer between the hot gases inside cylinder and the cylinder wall increases due to higher temperature difference, and hence a larger amount of heat energy is removed. Consequently this leads to a lower in-cylinder temperature[1] (Fig. 3), which in turn increases the engine charge density and therefore offers the possibility to increase air flow through the engine.

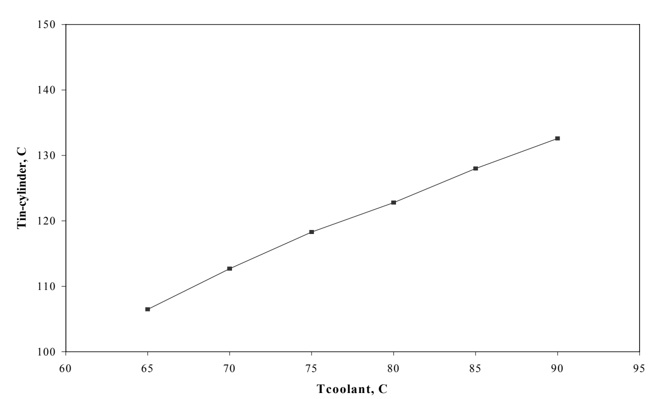

Fig. 3 In-cylinder temperature as a function of the coolant temperature at upper limit

[1] The in-cylinder temperature presented is the average value measured during the 4-stroke cycle by the sensor mounted into the cylinder head.

With increasing the air flow rate the power density increases and hence the engine power output (Fig. 4). It can be seen in Fig. 4 that reducing the coolant temperature from 90 ^{0}C to 65 ^{0}C increases the IMEP from 4.85 bar to 5.6 bar (for ~14%).

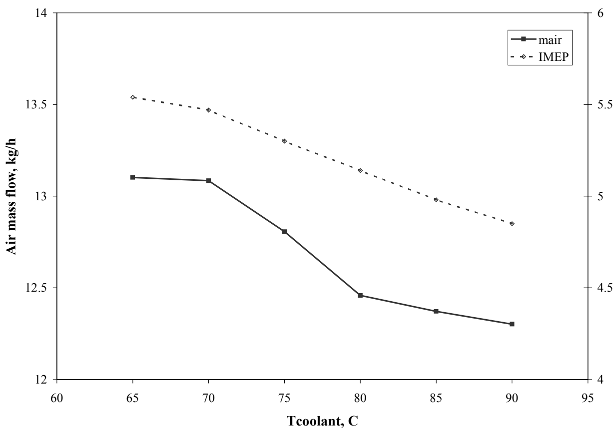

Fig. 4 Air-flow rate and IMEP as a function of the coolant temperature at upper limit

In the same time the coefficient of variation of the IMEP (COV IMEP) as an indicator of the combustion stability, the maximum rate of pressure rise (dP/dCA) and peak cylinder pressure (PCP) are reduced (Fig. 5). This is a likely consequence of enhanced heat transfer which reduces the temperature near the cylinder walls and therefore prevents development of high temperatures zones or ‘hotspots’ that are the main source for the knock [7, 8]. As a consequence of this the rate of heat release rate[2] (Fig. 6), pressure rise and peak cylinder pressure are reduced and the combustion stability is improved.

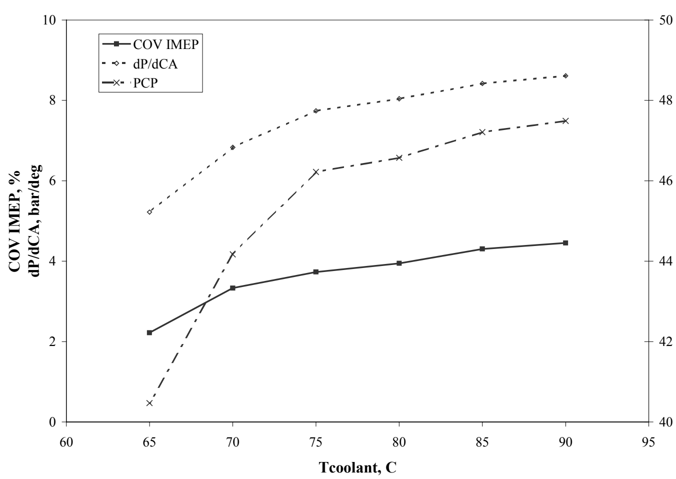

Fig. 5 COV IMEP, dP/dCA and PCP as a function of the coolant temperature at upper limit

[2] The reduction in HRR is also due to its later phasing which leads to reduction in the combustion temperature and hence slows down the combustion speed.

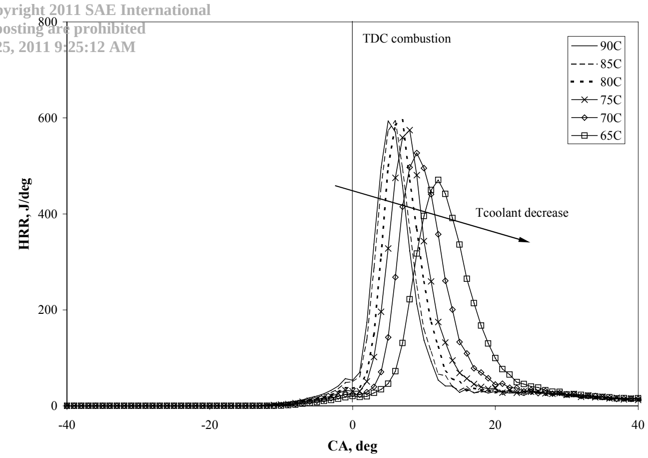

Fig. 6 Heat release rate (HRR) a function of the coolant temperature at upper limit

The reduced fuel consumption (ISFC), with a lower coolant temperature (Fig. 7), is most likely a result of improved combustion and thermal efficiencies (due to improved combustion stability).

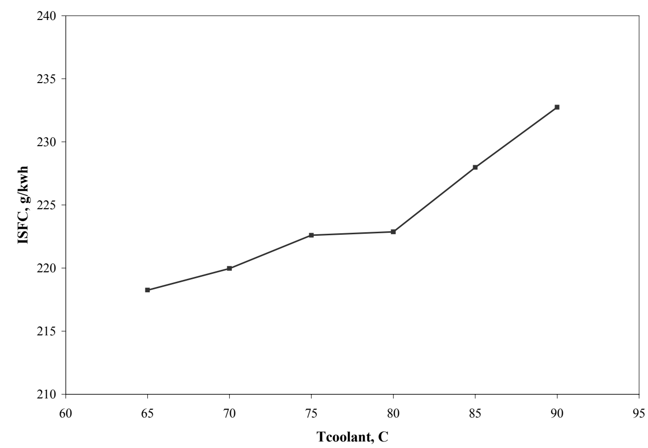

Fig. 7 The ISFC as a function of the coolant temperature at upper limit

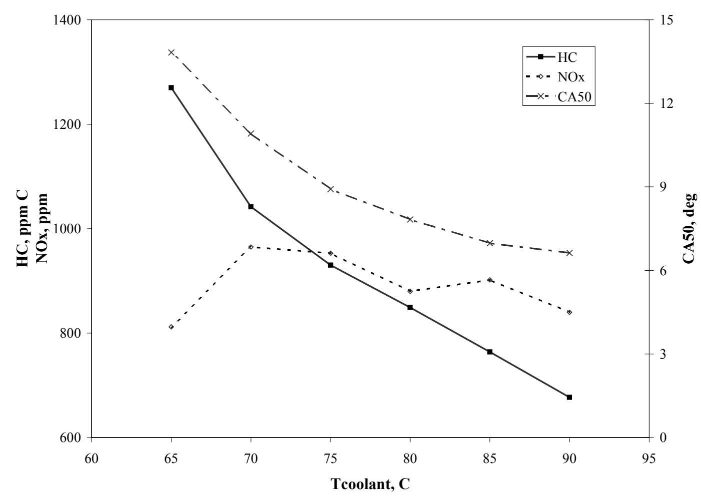

Fig. 8 The NOx, HC emissions and CA50 as a function of the coolant temperature at upper limit

It can be seen in Fig. 8 that with decreasing the coolant temperature the emission of HC increases. This is due to enhanced heat transfer and thus reduced temperature near the cylinder walls and hence the existence of colder boundary layers that suppress the fuel oxidation process. Parallel with this, a late combustion phasing (late CA50) decreases the in-cylinder gas temperature and thus quenches bulk gas oxidation which also increases HC emissions. The results obtained are in agreement with the test results reported in [9].

It can be noted that the NOx emission is almost constant for the investigated coolant temperature sweep, however at a relatively high level for the HCCI combustion mode (~ 900 ppm). The reasons for such a high NOx emission are highly likely that valve events were not optimised and the chosen initial point for the coolant temperature sweep (at 90 ^{0}C) has already a high NOx emission (850 ppm)[3].

EXTENDING THE LOW OPERATING RANGE LIMIT

The other side of the HCCI boundary, a lower limit of the operating range, is restricted by the misfire, since the compression temperature becomes insufficient to complete combustion. To overcome this, either the spark has to be used (to increase the temperature of engine charge) or the rate of heat transfer has to be reduced. The later approach is adopted and it is done by increasing the coolant temperature from 90 ^{0}C to 125 ^{0}C. To give better control of air-to-fuel ratio (AFR) at the minimum possible load, the engine throttle is completely closed and the PFI is air-assisted by a controlled amount of air (‘shop air’) being supplied to the intake manifold through a by-pass. The timings for the exhaust and inlet valve events are set at EVO 139 ^{0}CA, EVC 290 ^{0}CA, IVO 430 ^{0}CA and 581 ^{0}CA, respectively, with the valve lifts of 1.8 mm for both events and kept unchanged during the coolant temperature sweep. The engine parameters are kept constant at values specified in Table 1, whilst the TRG amount is ~50%.

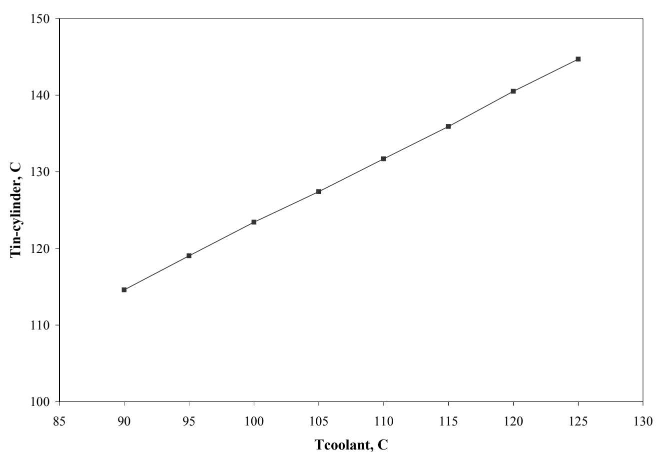

Fig. 9 In-cylinder temperature as a function of the coolant temperature at lower limit

With increasing the coolant temperature the heat transfer between the hot gases inside cylinder and the cylinder wall decreases due to a lower temperature difference, and hence less heat energy is removed. Consequently this leads to a higher in-cylinder temperature (Fig. 9) that provides an amount of heat necessary to sustain the HCCI combustion. The higher temperature of the charge and less dilution (due to reduced air flow through the engine) accelerates the overall chemical kinetics of the air/fuel mixture and therefore promotes its auto ignition.

With decreasing the airflow rate the power density decreases and therefore the engine power output (Fig. 10). It can be seen in Fig. 10 that increasing the coolant temperature from 90 ^{0}C to 125 ^{0}C reduces the IMEP from 2.11 bar to 1.51 bar (for ~ 28%).

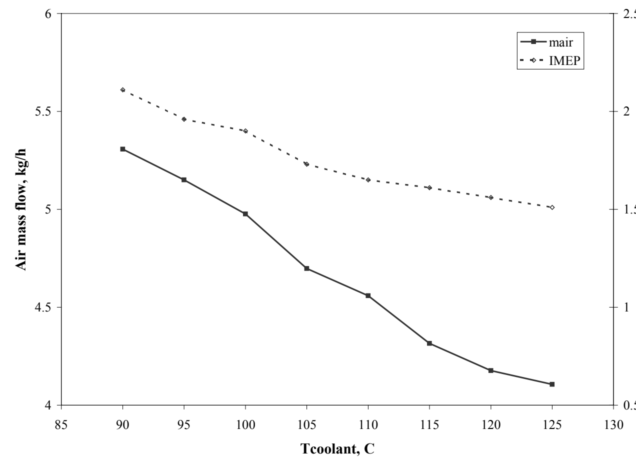

Fig. 10 Air-flow rate and IMEP as a function of the coolant temperature at lower limit

In the same time the COV IMEP slightly increases (Fig. 11), but the values are significantly lower from those acceptable for the misfire limits (COV < 10%). This indicates that a stabilised combustion is achieved throughout the coolant temperature sweep. The maximum rate of pressure rise, peak cylinder pressure rate (Fig. 11) and rate of heat release (Fig.12) are reduced, as expected, since the load output decreases.

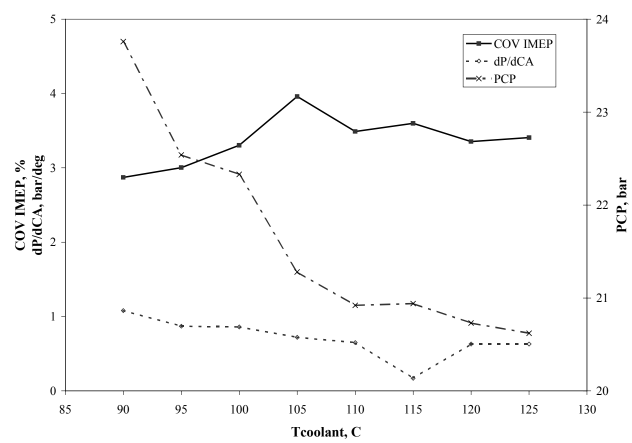

Fig. 11 COV IMEP, dP/dCA and PCP as a function of the coolant temperature at lower limit

The fuel consumption (ISFC) deteriorates with a higher coolant temperature (Fig. 13)[4]. This is an inevitable

[3] Authors acknowledge fact that the level of NOx emission is high at a given load. This is a consequence of running the PFI engine at stoichiometric AFR near the upper boundary of HCCI region (i.e. high load).

[4] The ISFC obtained for Tcoolant=125^{0}C is however, still proportionally lower compared to that obtained when the engine operated in SI mode (~ 20%) at given load [7].

consequence of running the engine at very low loads and hence at a lower thermal efficiency.

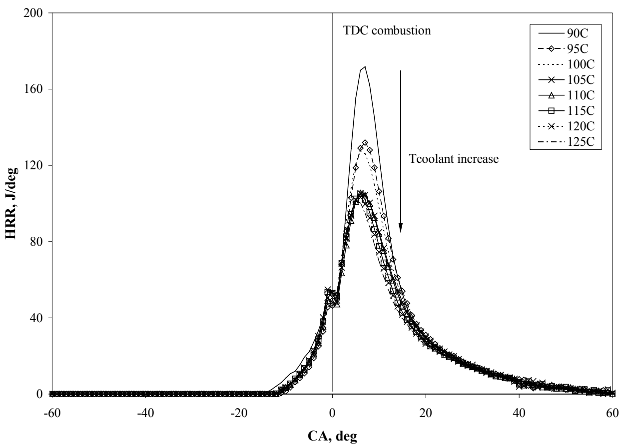

Fig. 12 Heat release rate (HRR) a function of the coolant temperature at lower limit

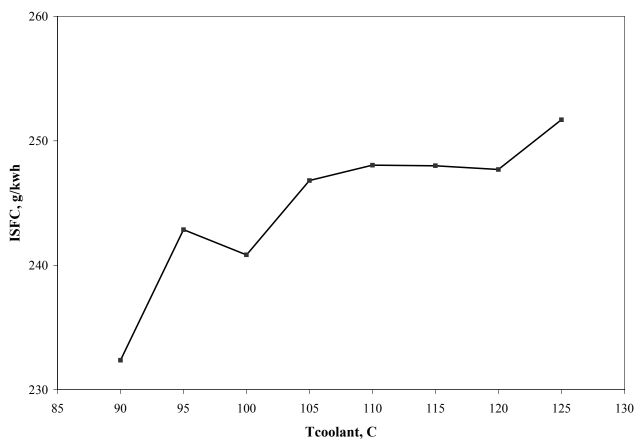

Fig. 13 The ISFC as a function of the coolant temperature at lower limit

It can be seen in Fig. 14 that with increasing the coolant temperature the emission of HC first increases and then decreases (for Tcoolant >105 ^{0}C). The increase in HC emission is likely due to a quenched bulk gas oxidation caused by the low combustion temperature produced at a low engine load. The decrease in HC emission with increasing the coolant temperature above 105 ^{0}C is due to a higher charge temperature and its less dilution that accelerates chemical reactions and therefore enhances the bulk gas oxidation.

It can be noted that the NOx emission for the investigated coolant temperature sweep is at a very low level (< 20 ppm). Increasing the coolant temperature above 105 ^{0}C decreases the NOx emission, likely a result of a lower combustion temperature obtained at a reduced load. The lower combustion temperature reduces the NOx emission production.

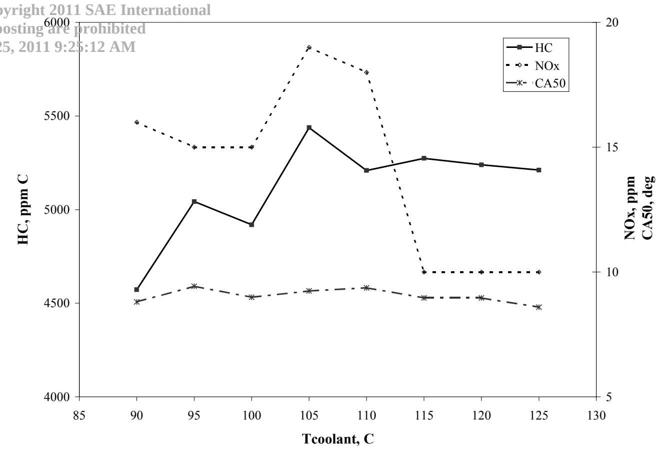

Fig. 14 The NOx, HC emissions and CA50 as a function of the coolant temperature at lower limit

It is also interesting to note that NOx and HC emissions decrease simultaneously for the coolant temperature higher than 105 ^{0}C. This is a quite different from the common experience obtained from SI combustion and also from experimental HCCI combustion results published world-wide to date. The trend of simultaneous reducing the NOx emission and HC emission are most likely a consequence of the more optimised valve events used that were on the other hand enabled by the flexibility of the AVT system.

PRACTICAL IMPLEMENTATION OF A VARIABLE COOLANT TEMPERATURE SYSTEM

The practical implementation of a new coolant system (thermostat and electric water pump) in an HCCI engine is current under investigation. It is likely that a new BMW's engine control unit (ECU) controlled thermostat and electric water pump approach [10] will be implemented. The operation of this system is governed by a control algorithm which manages the coolant flow rate to regulate engine temperature in response to engine speed, load and outside temperature, whilst offsetting possible abnormal combustion effects (knock and misfire), oil lubrication issues and anticipated driving demands [11].

CONCLUSIONS

Enlarging the HCCI operating range by controlling the coolant temperature was experimentally investigated on a single cylinder 4 stroke research engine equipped with a Lotus AVT system and fuelled with the standard gasoline fuel. The AVT system allowed different valve events strategies to be used for the initiation and control of HCCI combustion, the *recompression* strategy and *rebreathing* strategy, the former strategy being used in this research.

The coolant temperature was decreased from the nominal operational value of 90 ^{0}C to 65 ^{0}C to investigate the enlarging of the upper operating range limit, while it was increased from 90 ^{0}C to 125 ^{0}C to reduce the lower operating range limit.

The results obtained indicate that with reducing the coolant temperature, the upper limit can be extended up to 14%, while with increasing the coolant temperature the low limit can be extended up to 28%, while keeping the combustion stability, the rate of pressure rise and peak cylinder pressure in acceptable levels.

The fuel economy can be improved for the upper limit with reduced coolant temperature, but it deteriorates for the lower limit with increased coolant temperature. The simultaneous reduction in the NOx and HC emissions was obtained for one part of the low operating range. This unexpected behaviour was a result of the flexibility of AVT system that enabled more optimised valve events and hence strategies to be used.

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NOMENCLATURE

Definitions

^{0}CA – Degree crank angle (deg)
CA50 – Crank angle of 50% mass fraction burned (deg ATDCcombustion)
IMEP gross – Indicated mean effective pressure gross value (bar); (obtained from the integration of cylinder pressure trace over 360 ^{0}CA)
TDC combustion- Top dead centre combustion (0^0CA)
dP/dCA – Maximum rate of pressure rise (bar/^{0}CA)

Abbreviations

ATDCcombustion – After top dead centre combustion
BDC – Bottom dead centre
COV (IMEP) – Coefficient of variations of IMEP
HCCI - Controlled auto ignition
EVC - Exhaust valve closure
EVO – Exhaust valve open
THC – Total hydrocarbons
IVC – Inlet valve closure
IVO – Inlet valve open
ISFC – Indicated specific fuel consumption
NOx – Nitride oxide
PCP – Peak cylinder pressure
PFI – Port fuel injection
TRG – Trapped residual gas
WOT – Wide open throttle

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