Preprint Cambridge Centre for Computational Chemical Engineering ISSN 1473 – 4273

Dual-Fuel Effects on HCCI Operating Range: Experiments with Primary Reference Fuels

Ali Aldawood [1,2], Sebastian Mosbach [1], Markus Kraft [1], Amer Amer [2]

released: 05 April 2013

[1] Department of Chemical Engineering
and Biotechnology
University of Cambridge
New Museums Site
Pembroke Street
Cambridge, CB2 3RA
United Kingdom
E-mail: mk306@cam.ac.uk

[2] R&D Center
Saudi Aramco
Dhahran, Saudi Arabia

Preprint No. 125

Keywords: HCCI control, dual-fuel HCCI, HCCI range, intake temperature effects

Edited by
Computational Modelling Group
Department of Chemical Engineering and Biotechnology
University of Cambridge
New Museums Site
Pembroke Street
Cambridge CB2 3RA
United Kingdom

Fax: + 44 (0)1223 334796
E-Mail: c4e@cam.ac.uk
World Wide Web: http://como.ceb.cam.ac.uk/

Abstract

Results from a large set of HCCI experiments performed on a single-cylinder research engine fueled with different mixtures of iso-octane and n-heptane are presented and discussed in this paper. The experiments are designed to scrutinize fuel reactivity effects on the operating range of an HCCI engine. The fuel effects on upper and lower operating limits are measured respectively by the maximum pressure rise rate inside the cylinder and the stability of engine operation as determined by cycle-to-cycle variations in IMEP. Another set of experiments that examine the intake air heating effects on HCCI engine performance, exhaust emissions and operating envelopes is also presented. The effects of fuel reactivity and intake air heating on the HCCI ranges are demonstrated by constructing the operating envelopes for the different test fuels and intake temperatures. The paper discusses, in the light of the results, how the nonlinearity in fuel effects makes the dual fuel control approach less effective in extending the lower end of the HCCI load range. It also discusses how intake air heating affects the engine operation stability at low loads, and how varying fuel reactivity and intake heating can complement each other as an integrated control approach to extend both ends of the HCCI load range.

Contents

1 Introduction

The lack of external control in HCCI engine results in difficulties at low and high load operation. The low combustion temperature at low loads becomes insufficient to sustain a complete combustion which results in increased emission of carbon monoxide and hydrocarbons. Misfire eventually takes place limiting the low load operating range of the engine. At high loads, the rapid heat release causes the engine operation to become very noisy and unstable. Knocking starts to occur and thermal efficiency deteriorates as a result of increased heat losses, setting a limit to the upper range of operation. Misfire at low load and knocking at high load result in a narrow HCCI operating envelope.

Several combustion control strategies have been suggested and studied over the last decade to overcome the HCCI limited operating range. Examples of these strategies include using intake air heating, internal/external exhaust gas recirculation, variable compression ratio, variable dual fuel mixtures, and charge and thermal stratification. In general, these methods try to control the chemical reactions by controlling the combustion temperature and pressure or by altering the species involved in the reactions.

The dual fuel approach for controlling the combustion in HCCI engine has received great attention in recent years, and many control strategies based on this approach are under investigation. In this approach, two fuels with sufficiently different reactivity and combustion characteristics are used, and the combustion control is achieved by mixing the two fuels according to the conditions inside the cylinder.

Previous studies that investigated the dual-fuel approach include the work reported by Olsson et al. [9], Lu et al. [6], Atkins and Koch [3], and Ogura et al. [8]. These relatively early studies used primary reference fuels in their investigations. More recent studies used commercial fuels or mixtures of single-component and commercial fuels. Examples of fuels considered in these studies include mixtures of ethanol and n-heptane [10]; mixtures of n-heptane and diesel [7]; and mixtures of gasoline and diesel [4, 5]. All of these studies reported some positive results and varying levels of feasibility.

The current paper describes results from a set of HCCI experiments performed using two primary reference fuels, namely the iso-octane and n-heptane. A limited part of these results was presented in previous papers focusing on HCCI model development and optimization [1, 2]. The experiments aimed to examine the feasibility of the dual fuel control approach for extending the HCCI operating range. The fuel effects on operating limits were characterized through the variations in maximum pressure rise rate and the stability of engine operation as determined by cycle-to-cycle variations in indicated mean effective pressure. The results were used to construct engine operating envelopes for the different fuels, providing a visual demonstration and a quantification of the potential of the dual fuel approach to extend the HCCI operating window.

In addition to the dual-fuel effects, the paper also discusses the effects of intake air temperature on engine operation stability and the lower load limit. It also discusses how varying fuel reactivity and intake heating can complement each other as an integrated control approach for extending both ends of the HCCI load range.

While the current study tries to provide a useful insight on the potential of the dual-fuel approach, and while the use of primary reference fuels is convenient to generally represent

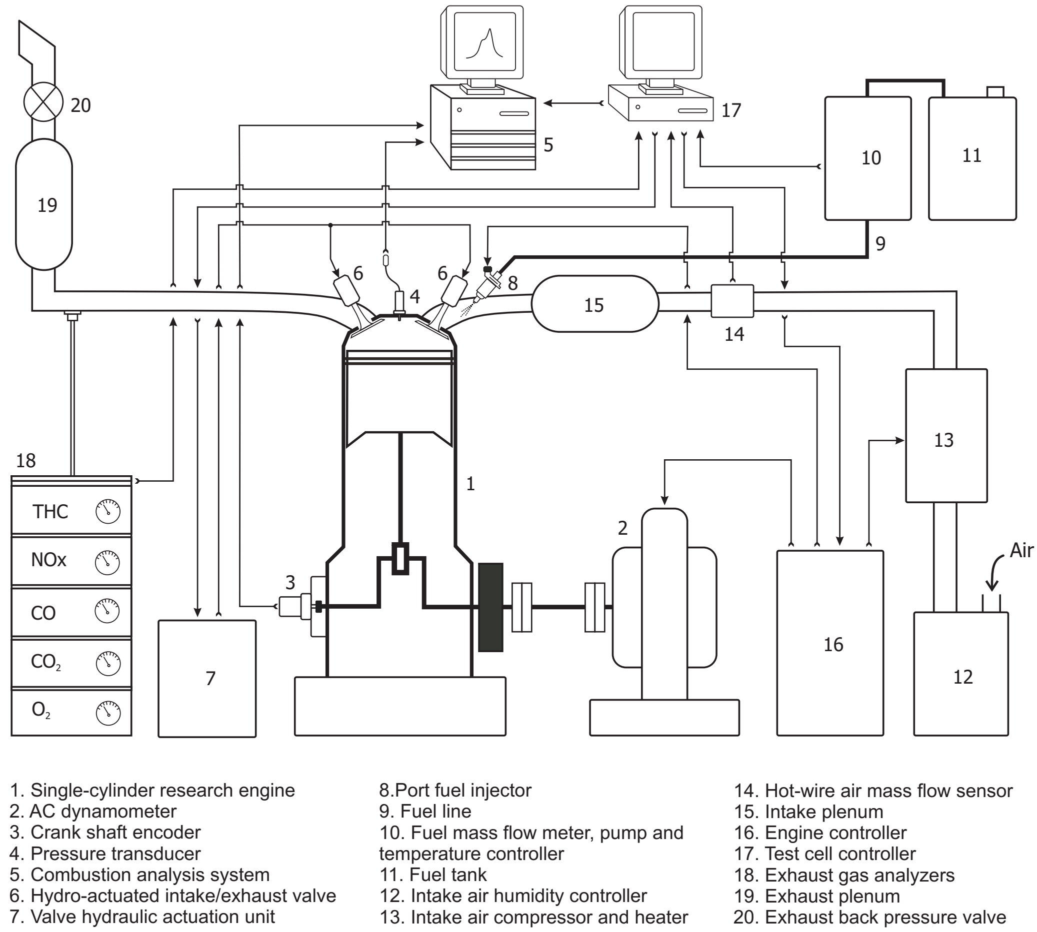

Figure 1: *The single-cylinder research engine test setup used in the current study.*

gasoline-like fuels, it is understood that the performance of actual gasoline or gasoline-like fuels could be noticeably different in HCCI engine. Being alkanes, the primary reference fuels might not be able to capture the combustion behaviors of other constituents, like aromatics, that exist in real fuels. The results presented hereafter therefore can only provide a qualitative indication of the potential of the dual fuel approach.

2 Experimental Setup

Figure 1 shows a schematic of the engine test facility used in this work. The setup consists of a single-cylinder research engine coupled to a 45 kW AC dynamometer, a fast-response engine controller, a combustion analysis system, exhaust gas analyzers, air and fuel mass flow meters, intake air boosting unit, and conditioning units for intake air and fuel. The test facility also contains heat exchangers for cooling water and lubrication oil. Both the coolant water and the lubrication oil were kept at 90^{o}C throughout the experiments.

The main specifications of the single-cylinder engine are listed in Table 1. The engine has a pent-roof combustion chamber head fitted with four hydro-actuated valves (see Figure

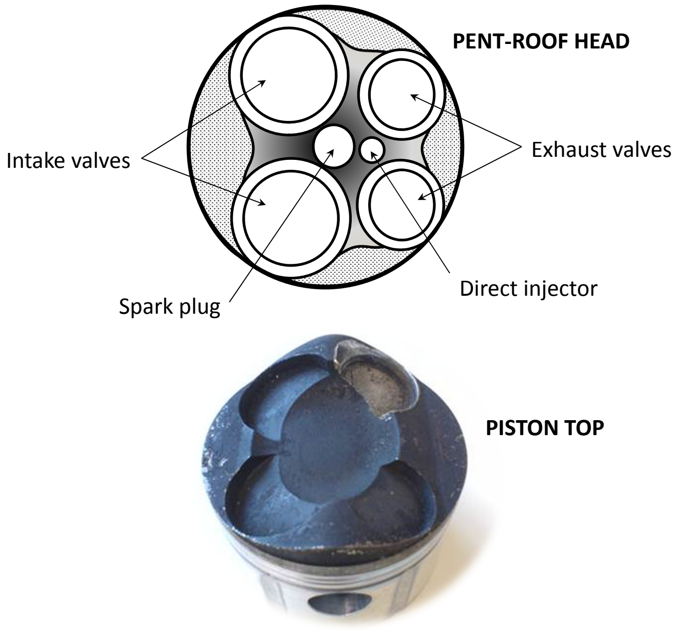

Figure 2: *Combustion chamber head and piston top of the single-cylinder test engine.*

2). The piston top contains the combustion chamber bowl along with deep valve recesses to allow increased range of authority.

Table 1: *Single-cylinder engine specifications and valve timing. Crank angles here are measured with respect to the firing TDC.*

Cylinder displacement	499 cm^3
Stroke	90 mm
Bore	84 mm
Connecting rod length	159 mm
Compression ratio	12:1
Number of valves	4
Fuel delivery system	PFI
Exhaust valve open	170^o
Exhaust valve close	-352^o
Intake valve open	-356^o
Intake valve close	-156^o

Fuel is delivered to the engine via an electromagnetic port fuel injector located at the end of the intake port just above the inlet valves. The fuel mass flow rate is measured continuously using a Coriolis flowmeter, and the fuel temperature is kept at 20^oC by a temperature controller integrated with the flowmeter and fuel circulation unit. The temperature and humidity of the intake air are controlled by a dehumidifier and a heater, while the pressure is controlled by a pressure boosting unit. The air mass flow rate is measured using a hot-wire flowmeter.

The combustion analysis system uses a piezo-resistive pressure sensor to measure the cylinder pressure, and a high-resolution crank angle encoder to determine the crankshaft position. The exhaust gas analyzers consist of non-dispersive infrared units to measure carbon monoxide and carbon dioxide, a heated flame ionization detector to measure total hydrocarbons, a chemiluminescence detector to measure nitrogen oxides, and a magneto-pneumatic detector to measure the oxygen.

3 Dual-Fuel Effects

All experiments were performed at a constant boosted intake pressure of 1.5 bar and an intake air temperature of 75^{o}C, and without using exhaust gas recirculation. While maintaining a constant turbocharging pressure might not be normally possible at low engine load, the boosting was found to be necessary to operate the current, relatively low compression-ratio, engine in HCCI mode at low load. The engine backpressure was kept at 1.5 bar in all experiments.

The engine was fueled with primary reference fuels at three different volume ratios: PRF40 (i.e. 40% iso-octane and 60% n-heptane), PRF60 and PRF80. The three test fuels were blended externally, and sufficient purging was conducted before running each fuel. Load sweeps were performed at three constant speeds of 1200, 1500 and 1800 rpm. The upper and lower bounds of the load range (i.e. by knocking and misfire limits) are first identified at each speed, and then a number of load points are determined along the identified range. Cylinder pressure data for 300 consecutive cycles are collected at each test point after allowing sufficient operation settling time, and associated operational data at each test point are collected at 0.1 second intervals for a period of 60 seconds.

3.1 Results and Discussion

Figure 3 through Figure 5 show cylinder pressure histories and heat release rates for load sweeps at 1500 rpm with the three different fuels. The heat release curves of PRF40 exhibit the expected strong two-stage ignition characteristics. The first-stage heat release becomes weaker as the iso-octane ratio increases in PRF60 and PRF80. Also as the ratio increases, the load range becomes narrower and the sensitivity of pressure history to the equivalence ratio becomes stronger.

Figure 6 shows the PRF ratio effect on the gross indicated mean effective pressure (IMEPg) at the three engine speeds. Increasing the PRF octane rating extends the upper operation boundary and preserves the engine ability to deliver more work as the equivalence ratio increases. The maximum IMEPg of 7.2 bar is obtained with PRF80 at 1800 rpm. The results also show the benefit of varying the PRF ratio on the engine positive work at similar operating conditions as well as the critical points when the change must take place to realize such benefit. Such information is of critical importance for developing a control strategy based on dynamic change of fuel ratio.

The extension of the operation range is further reflected by the gross indicated specific fuel consumption (ISFCg) and indicated thermal efficiency results (also shown in Figure

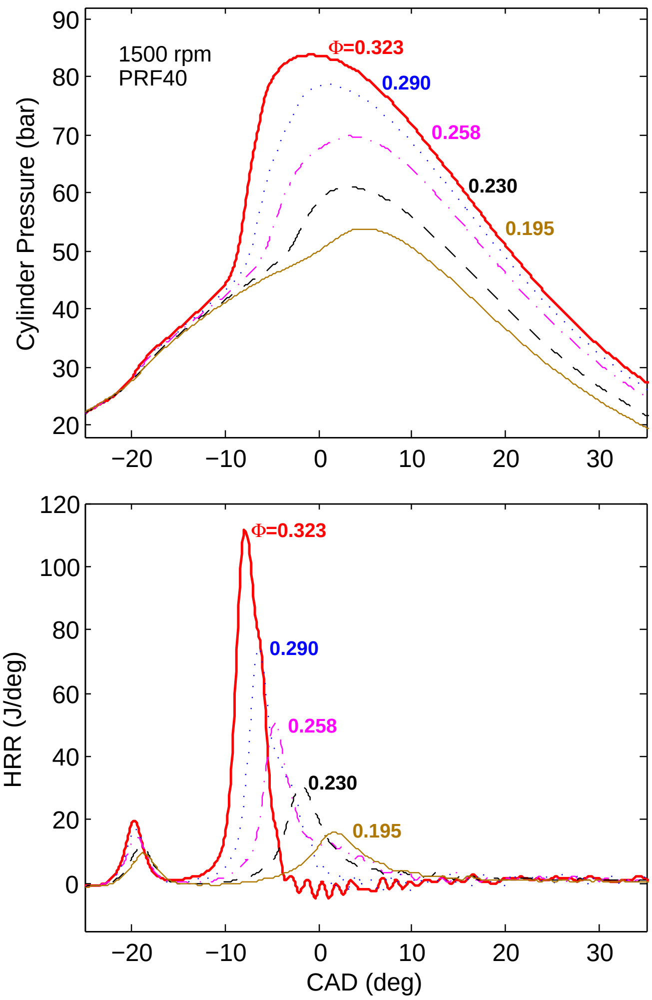

Figure 3: *Cylinder pressure histories and heat release rates for a load sweep at 1500 rpm with PRF40 fuel. The heat release curves show the expected strong two-stage ignition characteristics of PRF40.*

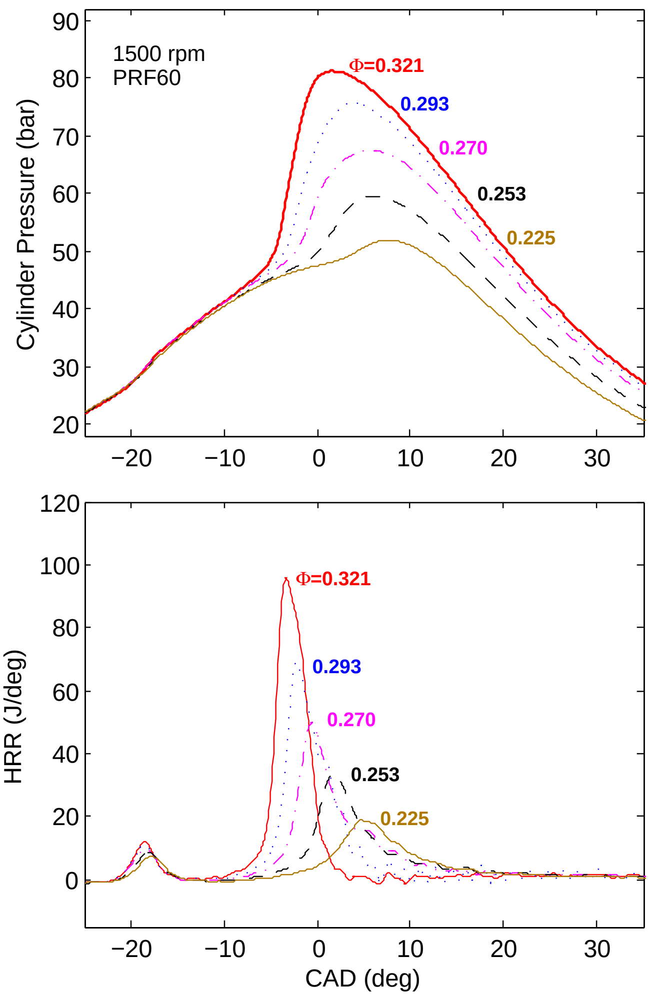

Figure 4: *Cylinder pressure histories and heat release rates for a load sweep at 1500 rpm with a PRF60 fuel.*

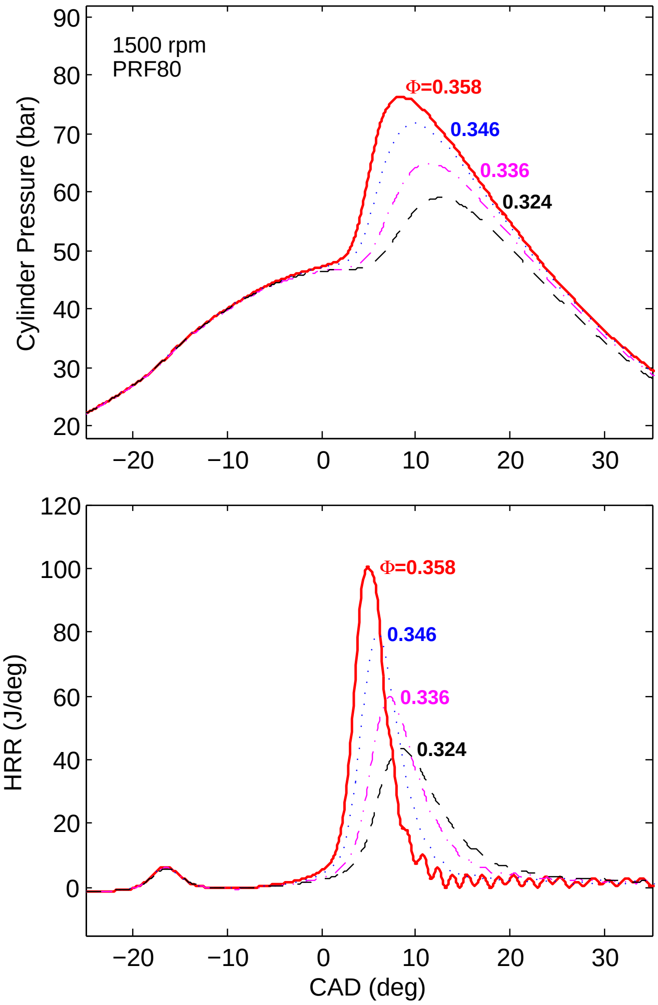

Figure 5: *Cylinder pressure histories and heat release rates for a load sweep at 1500 rpm with a PRF80 fuel. The pressure history is strongly sensitive to equivalence ratio.*

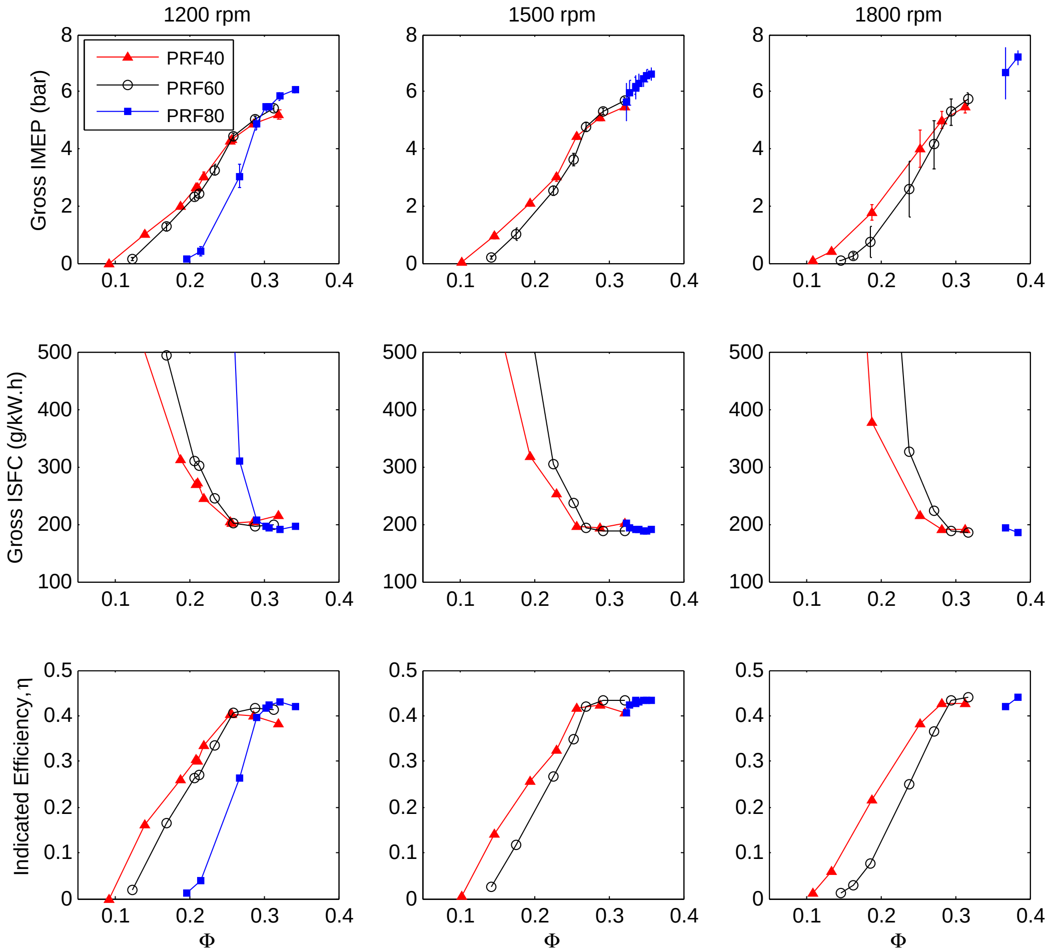

Figure 6: *Gross IMEP, ISFC and indicated efficiency for the three fuels. Increasing the octane rating (i.e. decreasing the fuel reactivity) extends the upper operation boundary and preserves the engine ability to deliver more work as the equivalence ratio increases. It also extends the plateau where the specific fuel consumption is minimal and indicated efficiency is maximal. The results in general reflect the expected low specific fuel consumption and high efficiency characteristics of the HCCI engine.*

6). Increasing the octane rating extends the plateau where the ISFCg is minimal and indicated efficiency is maximal. This plateau, as explained later, defines the operation range of the HCCI engine. A minimum ISFCg of 185.5 g/kW.h and a maximum indicated efficiency of 44% are obtained with PRF80 at 1800 rpm. A slight increase in ISFCg and decrease in indicated efficiency are observed with decreasing engine speed, but the results in general reflect the expected low specific fuel consumption and high thermal efficiency characteristics of the HCCI engine.

Figure 7 shows the results of CO, HC, and NOx emissions for the three speeds. HC emissions plateau around 1500 ppm through most of the operating range, but start to increase sharply as the engine approaches the misfire region that extends over a wide equivalence ratio range until the motoring limit. As the occurrence of misfiring cycles increases, more hydrocarbons escape the combustion chamber unburned. HC emissions normally originate both in the bulk gas and in the piston crevices, but the complex combustion chamber and piston design of the current engine appear to be specially conducive for increased crevice unburned hydrocarbons.

A slight increase in HC and CO emissions is observed with higher PRF ratios. This may be attributed to the increasing cycle to cycle variations observed generally with higher PRF ratio fuels (as can be seen in Figure 8). It is also observed that the climax of CO emissions within the misfire region corresponds to the start of increasing trends for both HC and NOx emissions. This may indicate a special significance of this inflection point where the CO starts to decrease and both HC and NOx start to increase. Such point could be used to mark the lower end of the practical load range of the HCCI engine. This inflection point is found to correspond here to about 6% IMEP COV. However as this point is characterized by significantly high CO levels, the lower load limit is set in this work to correspond to 5% IMEP COV as discussed below.

The boundaries of operation at the tested speeds can be inferred from Figure 8 which shows the readings of maximum pressure rise rate (MPRR) and coefficient of variation in the indicated means effective pressure (IMEP COV). The HCCI operating window is limited at the high end by the occurrence of knock and at the lower end by misfire. These two limits can be determined respectively in terms of MPRR and IMEP COV. The selection of specific values for these quantities is based in this work on observing the efficiency curve obtained with each PRF fuel.

The results show that the efficiency plateaus around a maximum value within a certain load range, and drops on both sides due to the occurrence of misfire in the lower side and knock in the upper side. The boundaries of efficiency plateau in the current experiments were generally found to correspond to an MPRR of about 7 MPa/ms on the high load limit (HLL) and 3.5% IMEP COV in the low load limit (LLL). This range is also characterized by relatively low CO and HC emissions, as can be seen in Figure 9. However, a reasonable extension of the operating range for each fuel can be achieved by relaxing the constraint on LLL to 5% IMEP COV. This relaxation results in a noticeable, but arguably tolerable, penalty in terms of efficiency and CO emissions. Figure 9 illustrates the definition of the operating range adopted in this work and the impact of relaxing the IMEP COV constraint on the LLL. For the specific example given in the figure, the maximum gross indicated efficiency plateaus at about 43%, but the LLL corresponds to about 35%. The CO emissions amount to about 1.4% at the 5% IMEP COV constraint compared to

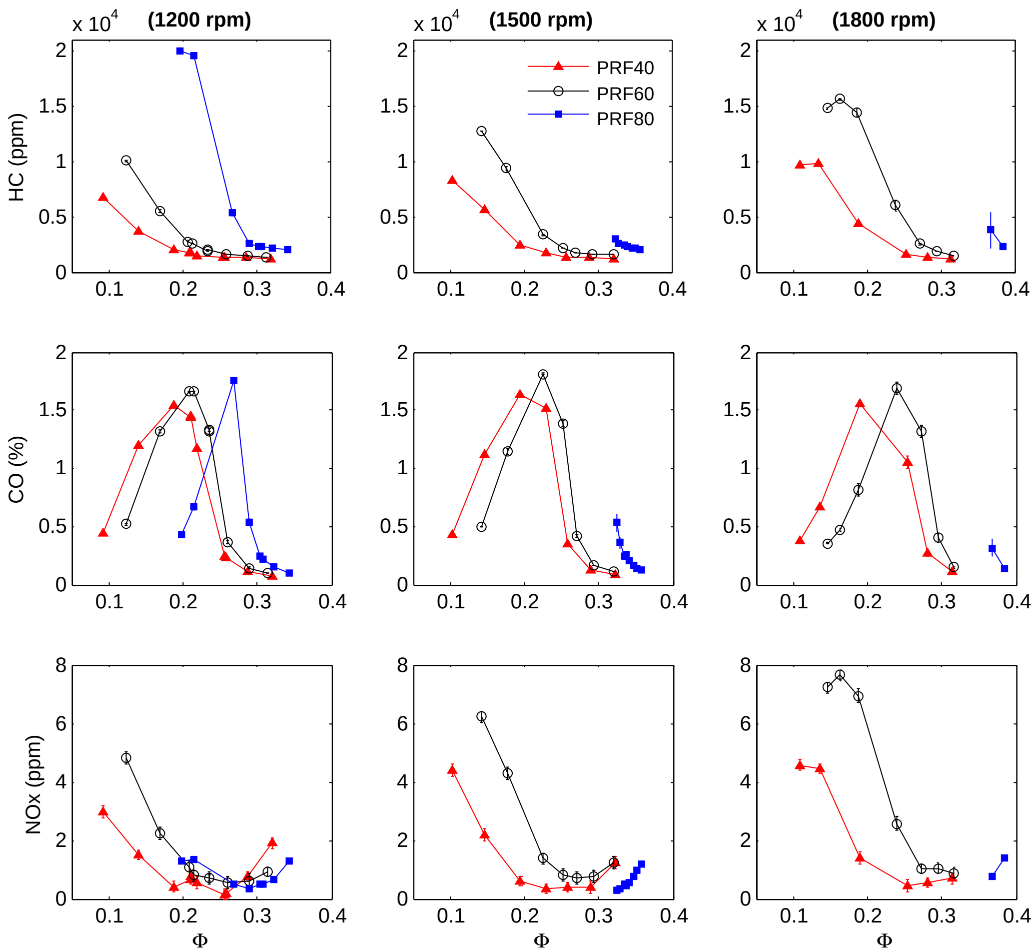

Figure 7: *Exhaust HC, CO, and NOx emissions for the three fuels at 1200 rpm. HC emissions plateau around 1500 ppm through most of the operating range, but start to increase sharply as the engine approaches the misfire region. The CO emissions show very strong sensitivity to operating conditions, ranging from 0.1% near the maximum load to about 1.8% in the misfire region. The extremely low levels of NOx reflect the low temperature combustion regime characterizing the HCCI engine.*

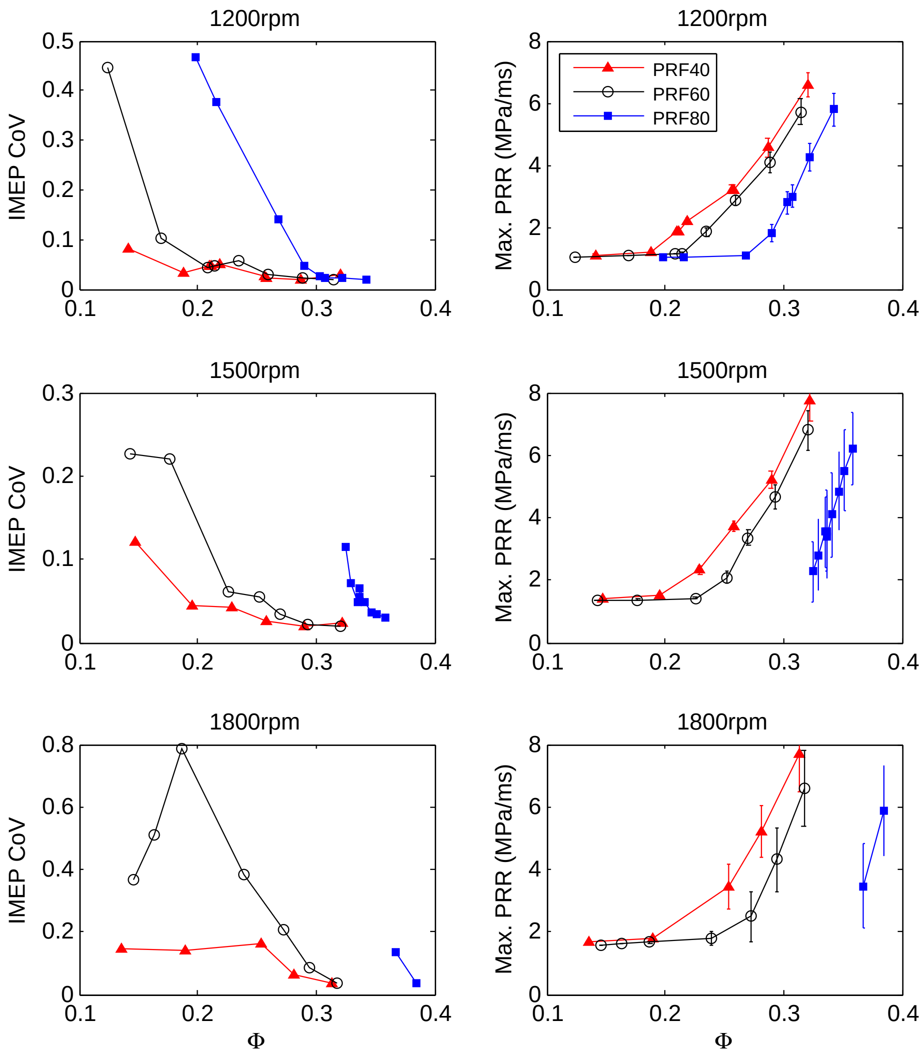

Figure 8: *Maximum pressure rise rate and variation of mean effective pressure with equivalence ratio. Error bars on maximum pressure rise values represent ±1 standard deviation. Increasing the fuel reactivity increases the engine stability at lower loads but on the other hand makes the engine more susceptible to knock at higher loads. The upper boundary of the operating envelope is set at 7 MPa/ms and the lower boundary is set at 5% IMEP COV.*

only 0.4% at 3.5% IMEP COV. Although the gain in the operating range in this example is not significant, reasonable gains are observed for other fuels and conditions.

The CA50 results for all load sweeps are shown in Figure 10. The misfire and knock limits are determined using the conventions described above. For each fuel, the combustion phasing advances with increasing load until reaching the knock limit. The operating window becomes narrower with increasing load and speed.

To illustrate these effects further, the engine operation envelopes for the different PRF fuels are constructed in Figure 11. While the envelopes shrink as the PRF octane rating increases, they also move upward allowing the engine to operate at higher equivalence ratios and to produce more positive work. The constructed envelopes provide a visual demonstration of the potential of the dual fuel approach to extend the HCCI operating window. A fueling system that can dynamically vary the ratio between the two fuel components based on the operating conditions, will allow the HCCI engine to operate at a much wider operating range. This premise was examined further in a previous modeling study in [2].

While these results point out to a clear potential for the variable fuel reactivity to expand the HCCI operating range, they show also that both the LLL and HLL are generally less sensitive to changes in octane rating between 40 and 60 than between 60 and 80, as illustrated in Figure 12. In other words, increasing the reactivity of the PRF fuel becomes increasingly less effective in controlling the HCCI operation as the engine load decreases. This nonlinear effect may necessitate the use of other combustion control means to complement the dual fuel approach during low load operation. One of these possible means is discussed in the following section. Other means like varying the compression ratio could also be considered for the purpose.

4 Intake Temperature Effects

The results presented in the previous section, while indicating a clear potential for the dual fuel to expand the HCCI operating range, show that the approach becomes less effective as the load decreases. For this reason, a set of experiments were carried out to study the potential of intake air heating as a complementary part to the dual fuel strategy during low load conditions.

The experiments reported in this section were carried out at the same conditions described in the previous section, except for the intake air temperature that was raised from 75^{o}C to 90^{o}C. The operating limits conventions described earlier were also applied here. As such, a maximum pressure rise rate of 7 MPa/ms was used to mark the high load limit and IMEP COV of 5% was used to mark the low load limit.

4.1 Results and Discussion

Figure 13 through Figure 15 compare the gross ISFC and indicated efficiency results for the two intake air temperature cases. For both engine speeds of 1200 and 1500 rpm, the

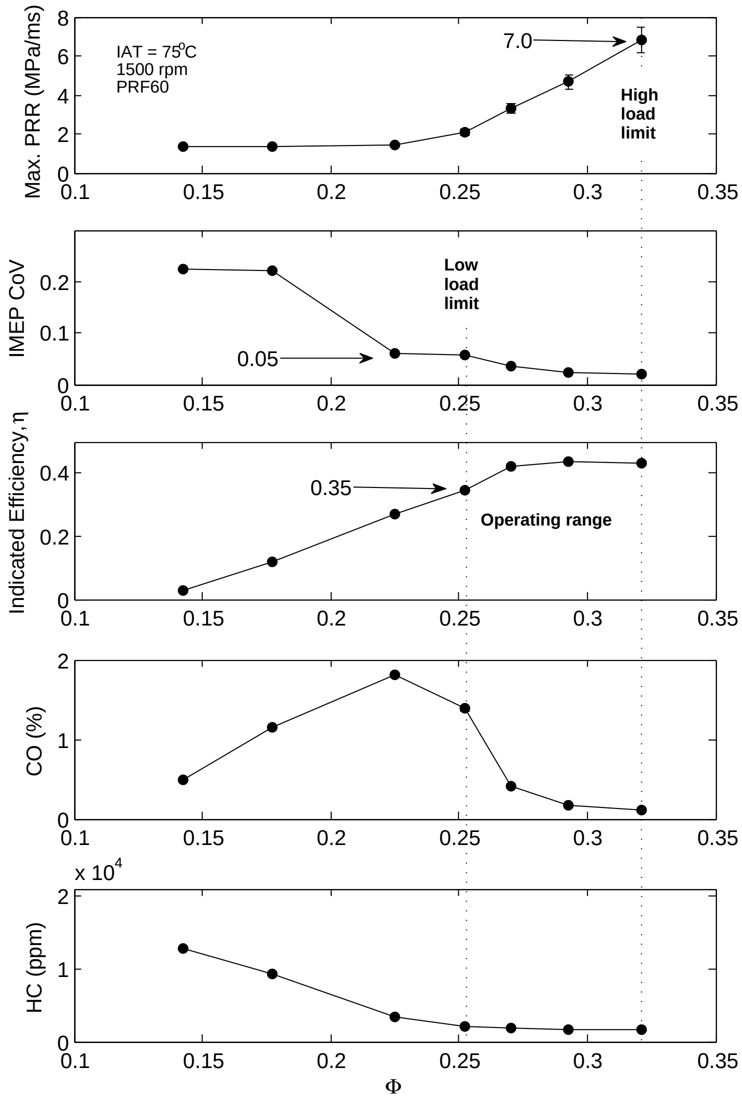

Figure 9: *Definition of the HCCI operating limits. The two limits are set around the edges of the efficiency plateau, with the high limit adhering more to the upper edge to avoid knocking operation. The low limit is characterized by rapid decline in efficiency and rise in CO emissions.*

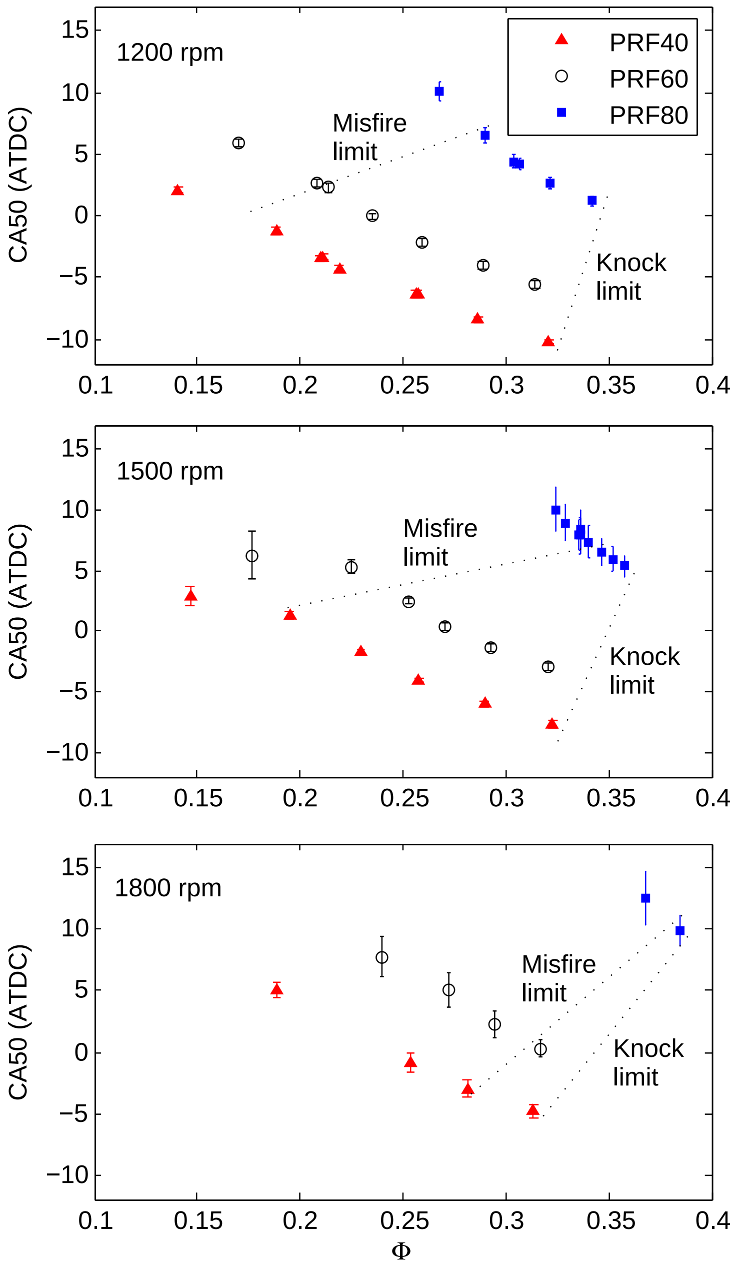

Figure 10: *CA50 results for the different load sweeps. The combustion phasing during the operating window of each fuel advances with increasing load until reaching the knock limit.*

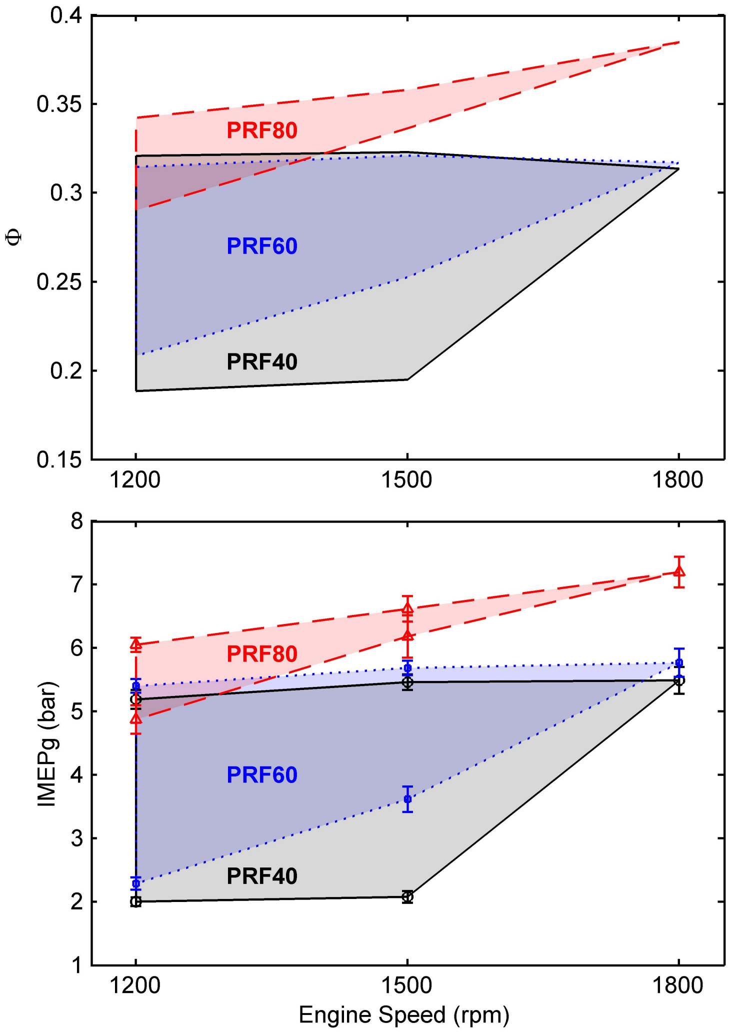

Figure 11: *HCCI operation envelopes for the different PRF ratio fuels. While the envelopes shrink as the PRF octane rating increases, they also move upward allowing the engine to operate at higher equivalence ratios and to produce more work.*

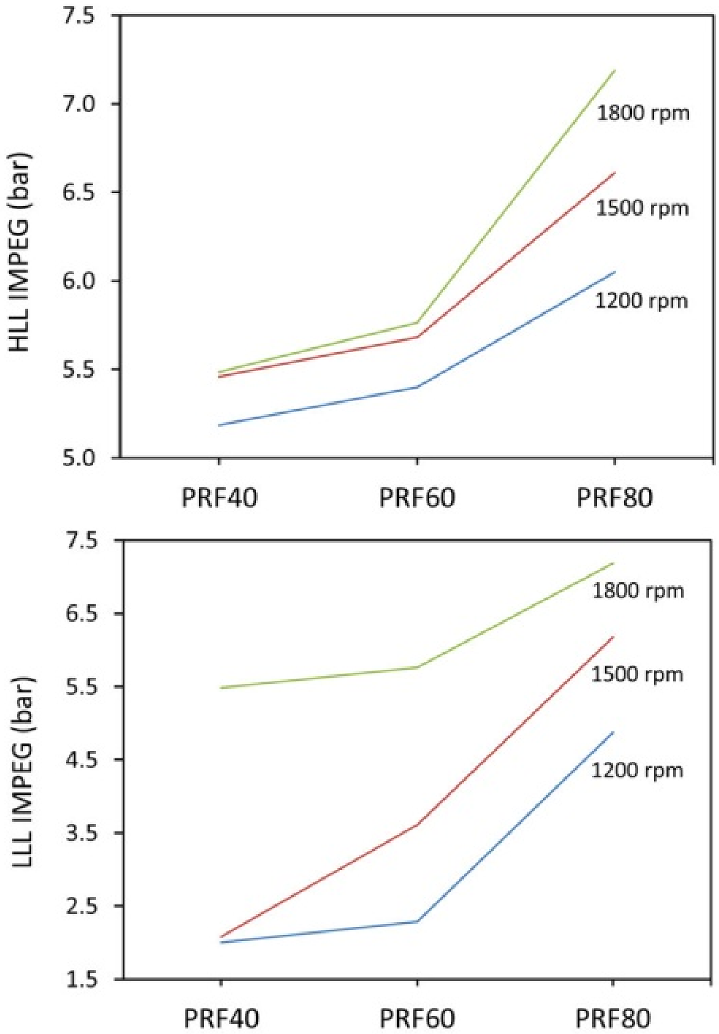

Figure 12: *The LLL and HLL limits are less sensitive to changes in octane rating between 40 and 60 than between 60 and 80. This nonlinear effect renders the variable reactivity approach less effective in the low load region.*

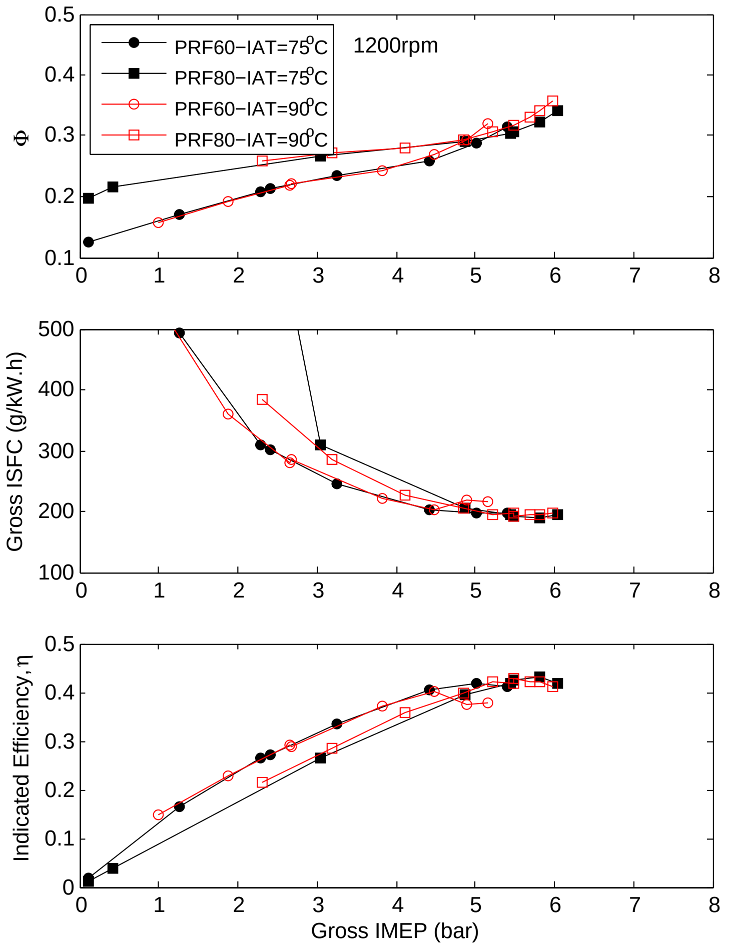

Figure 13: *Gross ISFC and gross indicated efficiency for PRF60 and PRF80 at intake air temperature of 75°C and 90°C, and engine speed of 1200 rpm. The higher intake temperature has hardly any effect on the engine performance at this speed.*

low load range performance of PRF60 was not affected by heating the intake air temperature, but suffered a noticeable drop in indicated efficiency towards the high load end. At 1800 rpm, the higher intake air temperature resulted in a slight deterioration in performance with PRF60 towards the high load end, but slightly improved the performance as the load deceased. With PRF80, the heating of the intake air temperature had no noticeable effect on gross ISFC or efficiency at the three speeds.

The results therefore indicate that heating the intake air temperature has either positive or no effect on the engine performance within the low to medium load region for the two tested fuels. At the high load end, however, higher intake air temperature caused some performance deterioration in the case of PRF60.

Increasing the intake temperature caused a slight advance in combustion phasing, as can be seen in Figure 16. This may explain the slight decrease in efficiency at the higher load end with the higher intake temperature, as the combustion timing becomes too advanced

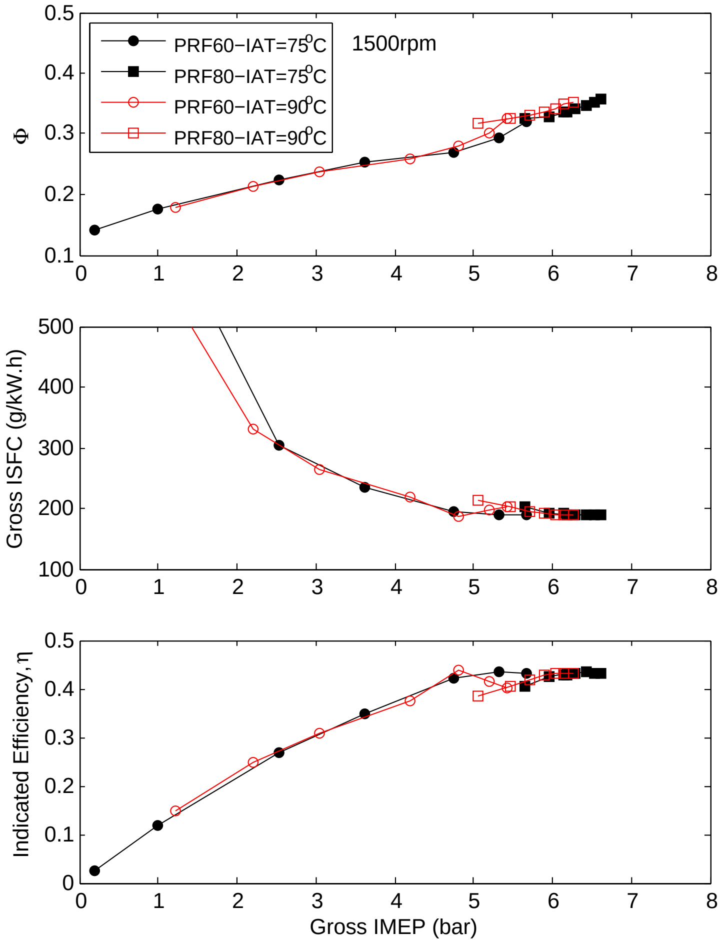

Figure 14: *Gross ISFC and gross indicated efficiency for PRF60 and PRF80 at intake air temperature of 75°C and 90°C, and engine speed of 1500 rpm. Here also, the higher intake temperature has no noticeable effect on the engine performance.*

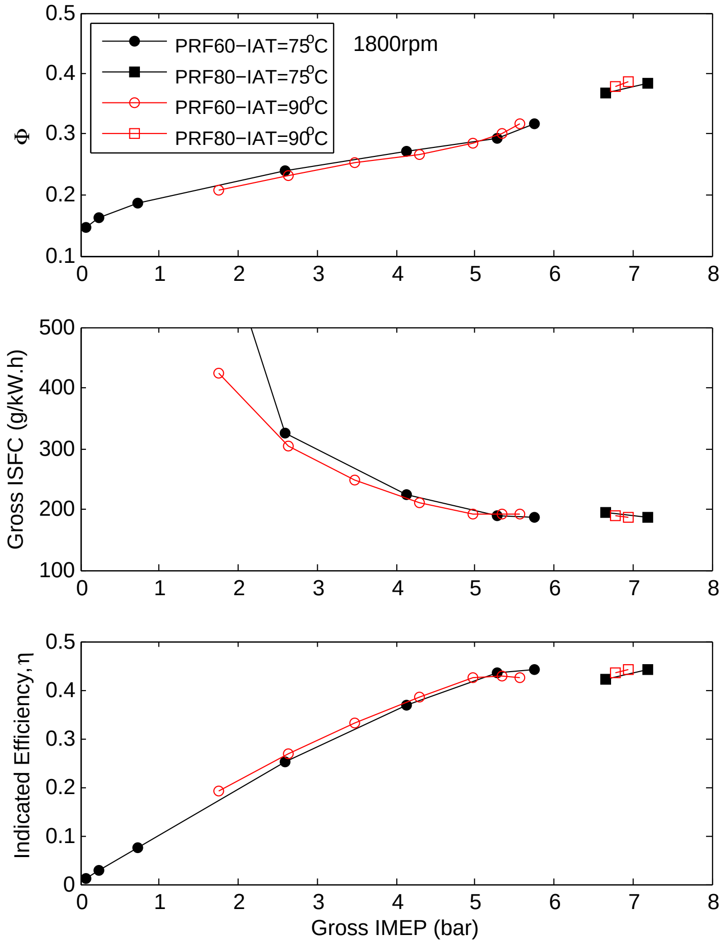

Figure 15: *Gross ISFC and gross indicated efficiency for PRF60 and PRF80 at intake air temperature of 75°C and 90°C, and engine speed of 1800 rpm.*

at these conditions.

The CO, HC, and NOx emission results are shown in Figure 17. Increasing the intake temperature resulted generally in lower CO and HC emissions within the low to intermediate load region in the case of PRF60 and at high load in the case of PRF80. The most noticeable reduction was in CO emissions. This reduction appears to be influenced mainly by the higher temperature history since the difference in equivalence ratio between the two cases is mostly negligible (see Figure 13 through Figure 15).

The values of IMEP COV and MPRR for the two intake temperatures are compared in Figure 18 Heating the intake temperature consistently improved the engine operating stability (as measured by the IMEP COV) in the low and intermediate load regions. On the other hand, the higher intake temperature resulted in a higher pressure rise rate, especially at higher loads. The increase in intake temperature shifts the operating envelope at each speed to the lower load side and enables a more stable operation in general.

A comparison of the operating envelopes for the two intake temperatures are given in Figure 119. In the case of PRF60, increasing the intake temperature to 90^{o}C resulted in a noticeable extension of the lower load range and a slight decrease in the high load limit. The operating envelope, therefore, was expanded more towards the low load region. With PRF80, however, only a slight shift of the operating envelope towards the low load side is observed, especially as the speed increases.

In Figure 20, the expansion in the PRF60 operating envelope due to the heating of the intake air to 90^{o}C is compared to the operating envelope obtained with PRF40 run at 75^{o}C. The results show that both cases resulted in almost similar expansion in the operating envelope with respect to the PRF60 running at 75^{o}C. In other words, increasing the intake temperature from 75^{o}C to 90^{o}C resulted in an operating range equivalent to that obtained by decreasing the PRF octane rating from 60 to 40.

The results indicate that heating the intake air could be used as an alternative to varying fuel reactivity to extend the engine operation in low load region. Significantly more heating will be required, however, to achieve this extension for higher engine speeds. It is also important to note that for the intake air heating approach to be effective, the heating must happen within a short time to meet the requirements of engine transient operation. Achieving this fast response intake air heating in practice could be extremely challenging.

5 Summary and Conclusions

A set of experimental results from a single-cylinder HCCI engine using primary reference fuels has been presented and discussed in this paper. The experiments were designed to scrutinize the effects of varying fuel reactivity (as measured by octane rating) on the HCCI engine operating range, and hence assess the potential of using the dual fuel approach to control the HCCI engine. Engine operation envelopes constructed from the experimental results showed a noticeable fuel effects on HCCI operation. While the envelopes shrunk for mixtures with higher octane rating (i.e. lower reactivity), they nevertheless allowed the engine to operate at higher equivalence ratios and produce more work. These effects became more prominent generally as the speed increased.

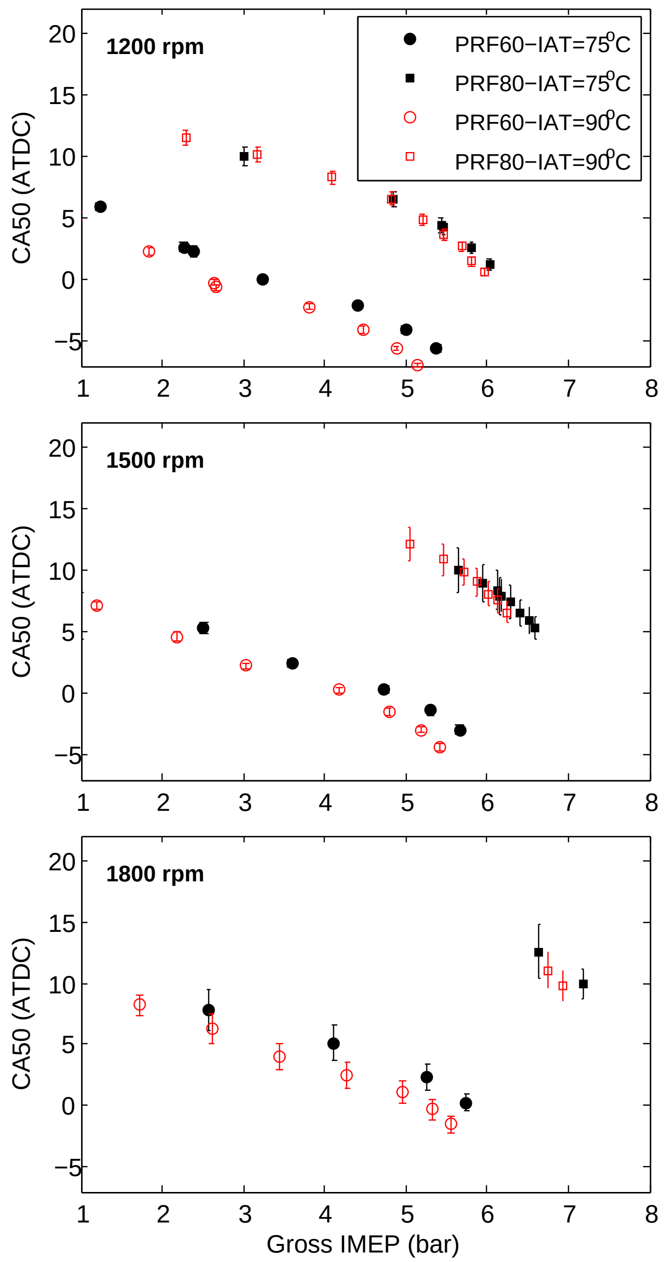

Figure 16: *Comparison of CA50 for the two intake air temperatures. Increasing the intake temperature caused a slight advance in combustion phasing especially in the case of PRF60.*

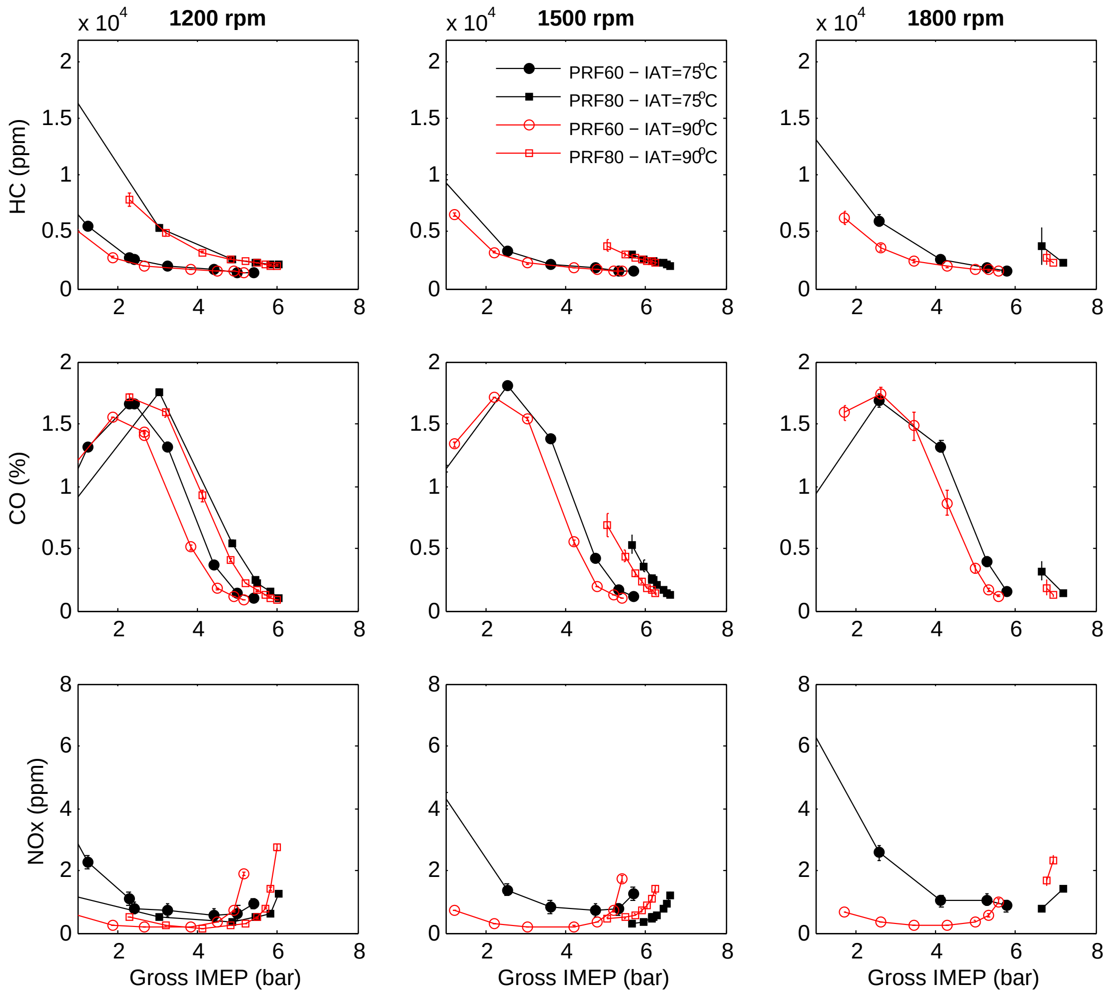

Figure 17: *HC, CO and NOx emissions for PRF60 and PRF80 at intake air temperature of 75°C and 90°C. The error bars represent ±1 standard deviation. The results generally indicate that increasing the intake temperature in the low load region has a positive effect on exhaust emissions.*

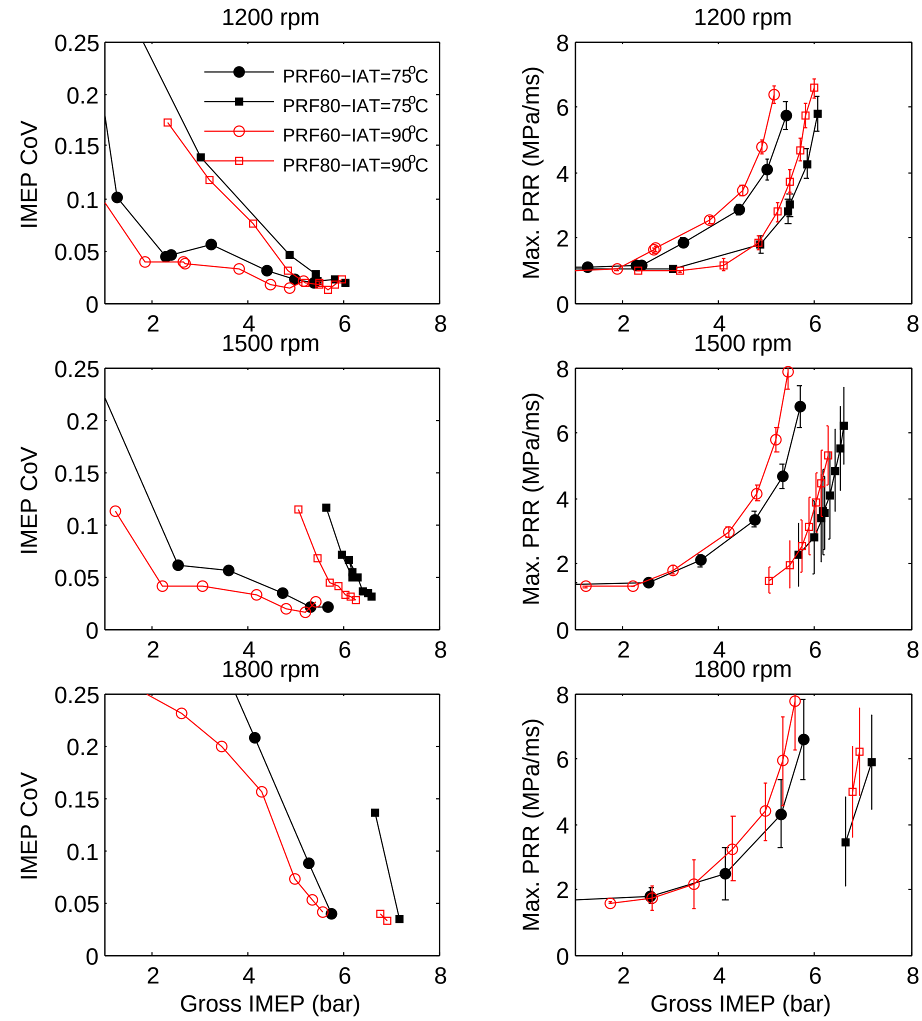

Figure 18: *IMEP COV and MPRR for PRF60 and PRF80 at intake air temperature of 75°C and 90°C. Heating the intake temperature improved the engine operating stability in the low and intermediate load regions but increased the pressure rise rate at higher loads. This shifts the operating envelope at each speed to the lower load side and enables a more stable operation in general.*

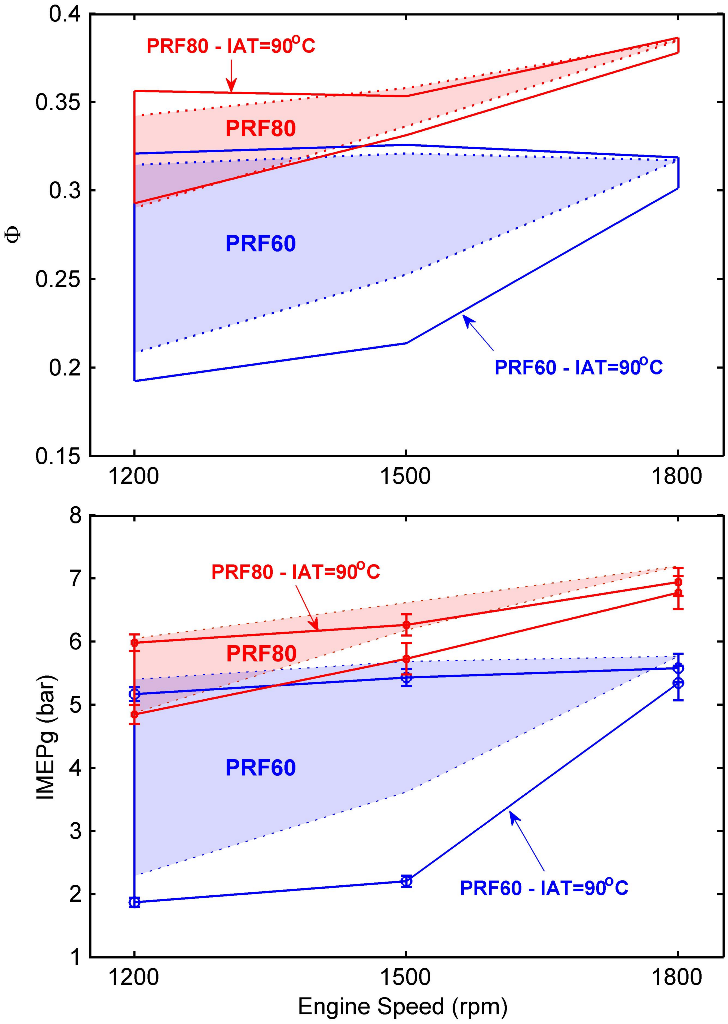

Figure 19: *HCCI operation envelopes for PRF60 and PRF80 at intake air temperature of 75^{o}C and 90^{o}C. In the case of PRF60, increasing the intake temperature resulted in a noticeable extension to the lower load range and a slight decrease in the high load limit. The operating envelope, therefore, was expanded more towards the low load region. With PRF80, only a slight shift of the operating envelope towards the low load side is observed, especially as the speed increases.*

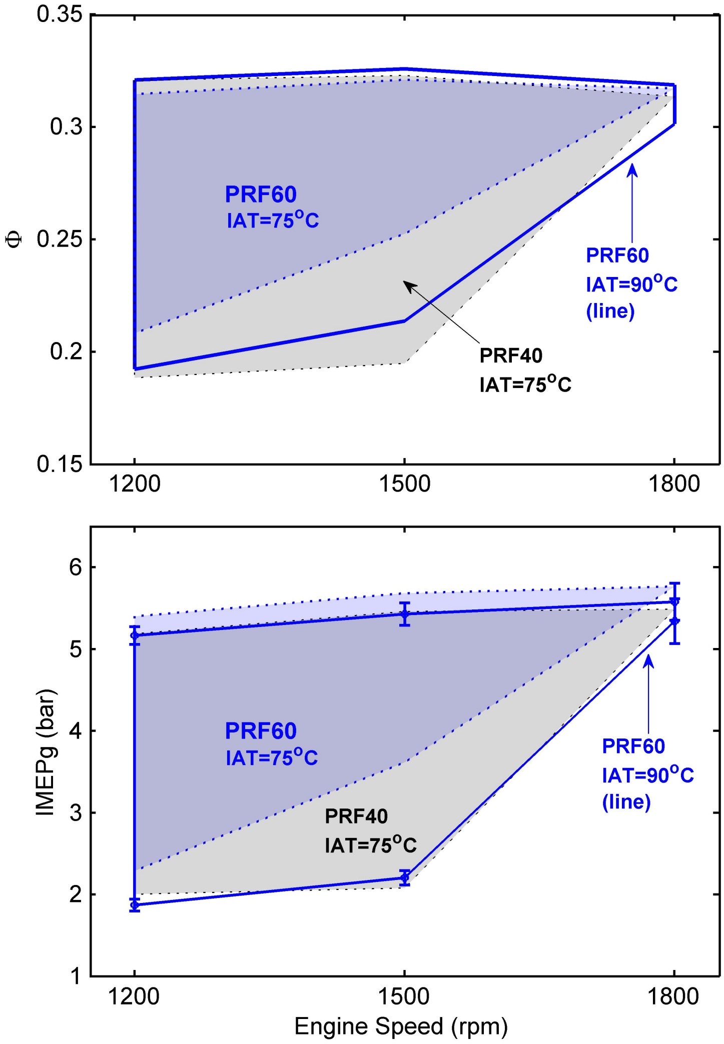

Figure 20: *HCCI operation envelopes for PRF40 and PRF60 at intake air temperature of 75^{o}C compared to the envelope for PRF60 at intake air temperature of 90^{o}C. Increasing the intake temperature from 75^{o}C to 90^{o}C resulted in an operating range almost equivalent to that obtained by decreasing the PRF octane rating from 60 to 40.*

While the results pointed out to a clear potential for the dual fuel approach to expand the HCCI operating range, they also showed that both high and low load limits became generally less sensitive to changes in the fuel as the octane rating decreased. In other words, increasing the reactivity of the PRF fuel became increasingly less effective in controlling the HCCI combustion as the engine load decreased. This could necessitate the use of other combustion control means to complement the dual fuel approach during low load operation.

A set of experiments were performed to examine the feasibility of using the intake air heating as a complementary approach in the low load region. Some positive, but generally not significant, effects on engine performance and exhaust emissions as a result of increasing the intake temperature were observed. The most noticeable effect was the consistent reduction of CO emissions within the low to intermediate load region. The increased intake temperature had hardly any effect on the indicated efficiency.

The results indicated that increasing the intake temperature extended the operating envelope at each speed to the lower load side and enabled a more stable operation in general. A quantification of the extension showed that increasing the intake temperature from 75^{o}C to 90^{o}C resulted in an operating range equivalent to that obtained by varying the PRF octane rating from 60 to 40. This strong effect of intake heating, as a conclusion, allows it to be used to complement the dual fuel approach for expanding the HCCI operating envelope in the low load region.

6 Acknowledgement

The authors would like to thank Yoann Viollet and Hassan Babiker of Saudi Aramco's R&D Center for their help in carrying out the engine experiments.

7 Definitions, Acronyms and Abbreviations

ATDC	After top dead centre
CA50	Crank shaft degree of 50% heat release
COV	Coefficient of variation
HCCI	Homogeneous charge compression ignition
HLL	High load limit
HRR	Net heat release rate
IAT	Intake air temperature
IMEPg	Gross indicated mean effective pressure
ISFC	Indicated specific fuel consumption
LLL	Low load limit
MPRR	Maximum pressure rise rate
PFI	Port fuel injection
PRF	Primary reference fuel
SRM	Stochastic reactor model
TDC	Top dead centre
Φ	Equivalence ratio

References

[1] A. Aldawood, S. Mosbach, M. Kraft, and A. Amer. Multi-objective optimization of a kinetics-based HCCI model using engine data. *SAE Technical Paper 2011-01-1783*, 2011.

[2] A. Aldawood, S. Mosbach, and M. Kraft. HCCI combustion control using dual-fuel approach: Experimental and modeling investigations. *SAE Technical Paper 2012-01-1117*, 2012.

[3] M. Atkins and C. Koch. The effect of fuel octane and dilutent on homogeneous charge compression ignition combustion. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 219, 2005.

[4] J. Cha, S. Kwon, and S. Park. An experimental and modelling study of the combustion and emission characteristics for gasoline-diesel dual-fuel engines. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 225, 2011.

[5] S. Kokjohn, R. Hanson, D. Splitter, and R. Reitz. Experiments and modeling of dual-fuel HCCI and PCCI combustion using in-cylinder fuel blending. *SAE Technical Paper 2009-01-2647*, 2009.

[6] X. Lu, W. Chen, Y. Hou, , and Z. Huang. Study on the ignition, combustion and emissions of HCCI combustion engines fuelled with primary reference fuel. *SAE Technical Paper 2005-01-0155*, 2005.

[7] J. Ma, X. L, L. Ji, and Z. Huang. An experimental study of HCCI-DI combustion and emissions in a diesel engine with dual fuel. *International Journal of Thermal Sciences*, 47, 2008.

[8] T. Ogura, J. Angelos, W. Green, W. Cheng, T. Kenney, and Y. Xu. Primary reference fuel behavior in a HCCI engine near the low-load limit. *SAE Technical Paper 2008-01-1667*, 2008.

[9] J. Olsson, P. Tunestl, and B. Johansson. Closed-loop control of an HCCI engine. *SAE Technical Paper 2001-01-1031*, 2001.

[10] L. Xingcai, H. Yuchun, Z. Linlin, and H. Zhen. Experimental study on the auto-ignition and combustion characteristics in the homogeneous charge compression ignition (HCCI) combustion operation with ethanol/n-heptane blend fuels by port injection. *Fuel*, 85, 2006.