Energy Distributions in a Diesel Engine Using Low Heat Rejection (LHR) Concepts

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Abstract: The energy balance analysis is recognized as a useful method for aiding the characterization of the performance and efficiency of internal combustion (IC) engines. Approximately one-third of the total fuel energy is converted to useful work in a conventional IC engine, whereas the major part of the energy input is rejected to the exhaust gas and the cooling system. The idea of a low heat rejection (LHR) engine (also called “adiabatic engine”) was extensively developed in the 1980s due to its potential in improving engine thermal efficiency via reducing the heat losses. In this study, the LHR operating condition is implemented by increasing the engine coolant temperature (ECT). Experimentally, the engine is overcooled to low ECTs and then increased to 100°C in an effort to get trend-wise behavior without exceeding safe ECTs. The study then uses an engine simulation of the conventional multi-cylinder, four-stroke, 1.9L diesel engine operating at 1500 rpm to examine the five cases having different ECTs. A comparison between experimental and simulation results show the effects of ECT on fuel conversion efficiency. The results demonstrate that increasing ECT yields slight improvements in net indicated fuel conversion efficiency, with larger improvements observed in brake fuel conversion efficiency.

Keywords: Low Heat Rejection; Engine Coolant Temperature; Energy Balance; Fuel Conversion Efficiency

1. Introduction

Energy efficiency is one of the most significant assessment factors concerned with the internal combustion engine. From the perspective of thermodynamics, the IC engine may theoretically approach 100% efficiency since the IC engine converts chemical energy to mechanical energy; chemical energy is fully available to do useful work. The IC engine will still be limited, however, by second law considerations such as irreversible processes [1-2]. Actual IC engines generally convert only approximately one-third of the fuel energy to useful work; the rest is rejected typically in the form of thermal energy to the coolant and exhaust.

The LHR concept has been of interest since the 1980s, when a substantial number of programs investigated the “adiabatic engine” [3-5]. These programs aimed to improve engine efficiency with partial/complete suppression of heat loss through the combustion chamber walls. Thermal barrier coatings have been extensively used in LHR engine designs, which increase the thermal resistance of the combustion chamber walls and consequently increase the level of temperatures inside the cylinder. From the first law of thermodynamics, it can be expected that any retained energy by reducing heat losses through the chamber walls can be converted to useful work and consequently improve the fuel conversion efficiency. The energy balance study on a partially stabilized zirconia (PSZ) coated diesel engine by Modi et al. [6] reported that the heat loss to the cooling system was decreased by 2%, 7%, and 7% for low, medium, and high loads respectively compared to the baseline engine. Praseth et al. [7] claimed 30-40% reduction in coolant load was achieved in coated engine, and a marginal improvement on the brake thermal efficiency was observed. Srithar et al. [8] investigated the effects of thermal barrier coated combustion chamber on a single cylinder diesel engine with dual biodiesel, the specific fuel consumption showed to

be lower than that of the baseline engine by 13.9% and the reduction in coolant losses was found to be around 5%-25%. Panneerselvam et al. [9] reported a review on the application of LHR concept in bio-diesel engines, stating that the energy of bio-diesel can be released more efficiently under LHR operation. Investigations by Kamo et al. [10] and Modi et al. [11] also indicated promising improvements in fuel efficiency at high load conditions using multi-cylinder diesel engines. Jafarmadar et al. [12] used a 3-D computational fluid dynamics code to further study the combustion processes within the chamber of a LHR diesel engine from the perspective of the second law of thermodynamics, which indicated that the exergy efficiency can be improved by LHR concept at all the studied load conditions.

In addition, Reddy et al. [13] summarized that all of the simulation work proved the fuel economy superiority of LHR engines over conventionally cooled engines. However, these facts do not necessarily substantiate a conclusion that LHR engines will outperform the conventional engine. In fact, some of the previous experimental work where the results are mixed show that a number of engine operating parameters are inter-related which can negatively affect the efficiency of a LHR engine. For instance, the higher operating temperature conditions decrease the volumetric efficiency, which in turn adversely influences the energy conversion efficiency [14]. Also, Caton [15-16] and Tunér [17] show that the resulting high gas temperatures in LHR engines yield lower values of the specific heats ratio causing less effective conversion of energy to work energy. Those conflicting results are mostly concluded from studies on ceramic-coated engines; so far, relatively little attention has been devoted to investigating the possibility of LHR engines with the approach of altering coolant temperature, which does not require significant modifications on engine. Reductions in heat losses to the coolant jacket can be supposedly

implemented by raising the engine coolant temperature (ECT), due to the smaller temperature difference between the coolant and the wall. Only a few works, however, have reported the benefits of operating engines at higher ECT conditions from the perspective of heat transfer [18-19]. The main finding was that a higher ECT reduced the heat rejection to the coolant or the net in-cylinder heat transfer rate, the associated gains also include enhancing fuel evaporation and therefore mitigating hydrocarbon (HC) and CO emissions [20-21]. It seems therefore logical to expand on the investigations of LHR concept through varying engine coolant temperature.

Typically, the operation conditions with ECT beyond 110°C are unlikely to be realized due to the needs of avoiding excessive oil temperature [22]. The maximum ECT is also restricted by the occurrence of nucleate boiling causing strong convection when the wall surface temperature goes beyond the coolant saturation temperature. In this case, the simulation method allows for extended studies at ECTs beyond the limited values on an actual engine. The current work is aimed at providing a detailed consideration of the energy balance for a LHR engine using elevated coolant temperatures. As described above, this is an area that has only received limited attention in the past.

This paper describes a study that lays the foundation for simulation-based analysis of LHR engines. The work complements a prior experimental study [23] of conventional and low-temperature combustion with LHR and offers analytical tools to explore more deeply the associated thermodynamics. The current study relies mostly on a one-dimensional engine simulation using a pseudo-predictive combustion approach to predict the energy distributions for a multi-cylinder light-duty diesel engine. This article begins with a brief introduction on the

engine simulation model, which is followed by the theoretical work on the energy balance methodology. Then, the results and discussion section presents the comparative study of energy distributions between simulation and experiment, which is further applied to correlate the improvements in fuel conversion efficiency to the changes in engine coolant temperature.

2. Model Setup and Validation

The diesel engine was modeled using a one-dimensional engine simulation software package (*GT-Power*) [24]. Table 1 lists the main specifications of the engine. In general, such a system-level engine model is developed with built-in templates providing detailed representation of the engine components, such as intake and exhaust manifolds, cylinders, valve train, and engine crank train.

The simulation model was validated using the steady-state measurements from the instrumented engine where five operating conditions were tested [25]. Table 2 lists the settings of engine speed, injection timing, fueling rate, and the primary independent parameter ECT. The engine is maintained at a low load condition of 30 N-m, which is equivalent to approximately 2 bar brake mean effective pressure (BMEP). The engine is operated at low ECTs and increased to 100°C to provide trendwise behavior without exceeding safe operating temperatures. The experimental measurements are collected from 200 consecutive cycles for each cylinder and ensemble averaged.

With given inputs and defined boundary conditions, sub-models including heat transfer model, combustion model, friction model, and nitrogen oxide (NO_x) model were calibrated by

reproducing the experimental data at the five test points. It should be noted that the engine simulation does not include a submodel predicting blow-by fraction and crevice volume.

Table 1: Engine specifications

Engine type	DI Diesel Engine
Displaced volume	1.9 L
Bore	82 mm
Stroke	90.4 mm
Connecting rod	145.5 mm
Compression ratio (nominal)	18:1
Number of valves	16
Number of cylinders	4
Exhaust valve open	112° ATDC
Exhaust valve close	356° ATDC
Intake valve open	361° ATDC
Intake valve close	-132° ATDC

Table 2: LHR test matrix

Case #	Speed (RPM)	Injection timing (°bTDC)	Fueling rate (mg/cycle)	ECT (°C)
1	1500	8	35.0	56.5
2	1500	8	35.6	65
3	1500	8	34.3	75
4	1500	8	34.4	90
5	1500	8	34.6	100

The physics behind these submodels except for emission model were illustrated in details in the previous work [26] and a systematic calibration procedure was developed to generate a single set of model constants, which are capable of predicting the engine performance over multiple operating points. The comparisons of cylinder pressures, flow characteristics, and most engine performance parameters have also been validated in [26] showing that the predictions are generally in agreement with the measurements, whereas disparities in the rates of heat release (ROHR) are observed from the studied cases — one of which (#4) is shown in Figure 1, the validations for the remaining cases are listed in the Appendix A. These discrepancies, however, do not cause a major concern since much of the use of the simulation is to capture

thermodynamic parameters of the engine system where precise match in pressure is the first priority.

Given that NO_x emissions are closely associated with the combustion rate, the combustion model needs to be calibrated with measurements prior to integrating the NO_x model into the engine simulation. The NO_x model used for this simulation study is established based on the extended Zeldovich mechanism, and the kinetic rate constants are taken from the recommended values published in [27]. Figure 2 shows the comparison of NO_x concentration (in units of ppm) and brake specific NO_x (BSNO_x) between the measurements and predictions. The error bars in all data plots are created from different sets of measurements on two separate days. It should be noted that the error bars for Case #1 ($\text{ECT}=56.5^\circ\text{C}$) are zero, since only one set of measurements was attained. In general, the variations of simulation results have similar patterns to that observed from the experimental data. The predicted NO_x concentrations seem to be overestimated by around 2-3% compared to the measurements, but both results show an increase with increasing ECT due to the retained energy causing higher in-cylinder temperatures [23]. The Zeldovich NO formation mechanism is sensitive to both pressure and temperature, but more so temperature than pressure; as the simulated pressures match the validation data well, it is likely that the simulation model's two-zone temperature calculation does not capture the maximum temperature of the burned zone very accurately. Both the simulation and experiment show gradual decreases in BSNO_x due to the improvement in brake power exceeding the increase of NO_x emissions as ECT increases. The underprediction of BSNO_x relative to experimental BSNO_x (compared to the overprediction of NO_x concentration) results from an underprediction in total mass flow rate through the simulated engine. It has been discussed in

[26] that the predicted intake air mass for all the five cases showed to be relatively lower than measurements by less than 5%, where the errors are considered to be mainly caused by the inaccuracy in the intake valve discharge coefficients.

Figure 1 Validation of cylinder pressure and apparent heat release rate

Figure 2 Comparison of NO_x concentration and BSNO_x between experiment and simulation

3. Energy Path Analysis

This study attempts to compare the energy distributions between the simulation and experiment, the lack of information on turbocharger, intercooler and exact engine external structures (e.g., engine block surfaces) prevents the simulation from capturing the unaccounted energy, i.e. the heat rejected to the oil plus convection and radiation dissipated to the environment through the engine's external surfaces. To ensure the experimental energy balance analyses follow a consistent fashion of energy paths that the simulation can provide, a thermodynamic system by taking the combustion chamber as the control volume (CV) was selected to develop the energy balance analyses. Figure 3 depicts the energy flows of this system, and Eq. (1) is the overall energy balance that provides information on the disposition of the input energy. For this control

volume, blowby losses and crevice flows are neglected for both simulation and experimental analyses.

Figure 3 The engine combustion chamber as a thermodynamic open system operating in a steady-state fashion

$$\dot{m}_f h_{fuel} + \dot{m}_{air} h_{air} = \dot{W}_{i,net} + \dot{Q}_{HT} + \dot{m}_{exh} h_{exh} \quad (1)$$

The terms on the left represent input fuel energy and air energy, the energy flows leaving the control volume consist of net indicated power ($\dot{W}_{i,net}$), heat transfer from gases to the chamber walls (net in-cylinder heat transfer rate, $\dot{Q}_{HT}$), and exhaust energy ($\dot{m}_{exh} h_{exh}$). $\dot{m}_f$, $\dot{m}_{air}$, and $\dot{m}_{exh}$ are corresponding to the mass flow rates of fuel, inducted air, and exhaust gas. h_{fuel} stands for the enthalpy of fuel (#2 diesel), which may be approximated with the lower heating value since the sensible enthalpy is almost negligible as compared to the lower heating value. h_{air} and h_{exh} represent the enthalpies of air and exhaust gas, respectively. The engine simulation is capable of providing complete information for these energy flows. The following subsections will briefly review the methodologies used to experimentally determine these terms, i.e. net indicated work, in-cylinder heat transfer, and exhaust energy. Table 3 gives the information regarding how each term involved in Eq. (1) is quantified.

Table 3: Quantification of parameters for experimental energy balance analyses

Parameters	Quantification method
$\dot{m}_f$	Direct measurement
$\dot{m}_{air}$	Direct measurement
$\dot{m}_{exh}$	Direct measurement
h_{fuel}	Approximated to be fuel LHV
h_{air}	Derived from constitutive enthalpies using NASA Glenn polynomials
h_{exh}	Derived from constitutive enthalpies using NASA Glenn polynomials
$\dot{W}_{i,net}$	Calculated from experimental pressure-volume data, see Section 3.1
$\dot{Q}_{HT}$	Experimentally derived from available measurements, see Section 3.2

3.1 Net Indicated Work

The net indicated work is the sum of the gross work and the pumping work, which are evaluated from in-cylinder gas pressures by taking an integral over an engine's mechanical cycle:

$$W_{i,net} = \oint p dV \quad (2)$$

3.2 In-Cylinder Heat Transfer

Directly measuring in-cylinder heat transfer rate is usually complicated and typically requires temperature measurements of multiple positions on the cylinder liner in order to obtain the temperature field. An alternative way to obtain $\dot{Q}_{HT}$ is relating to the engine's overall heat rejection to the environment (e.g., coolant system and other convective and radiative surface losses) [28-29]. Namely, the in-cylinder heat transfer rate can be estimated by subtracting the friction power ($\dot{W}_f$) from the total heat rejection, which is the sum of coolant load ($\dot{Q}_{cool}$) and unaccounted energy ($\dot{Q}_{unac}$).

$$\dot{Q}_{HT} = \dot{Q}_{cool} + \dot{Q}_{unac} - \dot{W}_f \quad (3)$$

The detailed derivation of Eq. (3) is developed in Appendix B. The heat rejection to the coolant is determined from the temperature rise in the coolant ($T_e - T_i$) as it passes through the engine and the mass flow rate of coolant, $\dot{m}_{cool}$.

$$\dot{Q}_{cool} = \dot{m}_{cool} c_p (T_e - T_i) \quad (4)$$

The dominant source of the unaccounted losses is the environmental heat losses [30], i.e. the heat transfer from engine external surfaces to the surroundings by conduction, convection and radiation. The calculations of the surface heat losses have been detailed in [25]. In contrast to the other two energy flows that are directly derived from experimental measurements, using Eq. (3) to calculate $\dot{Q}_{HT}$ may carry errors from the computations of the involved terms ($\dot{Q}_{cool}$, $\dot{Q}_{unac}$, and $\dot{W}_f$) in addition to errors from relevant measurements.

3.3 Exhaust Energy

For steady state, the mass flow rate of the exhaust gas is known in terms of the measured mass flow rates of air and fuel. Therefore, the energy carried by the exhaust gas can be determined from the balance,

$$\dot{m}_{exh} h_{exh} = (\dot{m}_f + \dot{m}_{air}) h_{exh} \quad (5)$$

where the enthalpy of the exhaust gas is referenced to 298 K. The exhaust gas is a known mixture of numerous constitutive species, thus the exhaust enthalpy is namely the mixture enthalpy determined from exhaust temperature and species concentrations. The heat of combustion is aggregated into the exhaust enthalpy for species that have not been completely oxidized, i.e. carbon monoxide (CO) and unburned HC emissions.

From the control volume shown in Figure 3, the target exhaust energy should be the energy corresponding to a pre-turbine temperature, but only the post-turbine gas temperature was attained from the experimental measurements. To better relate the experimentally derived exhaust energy to that calculated by the simulation, it is necessary to analyze the temperature

difference or the difference of energy carried by the exhaust gas between pre-turbine and post-turbine. The difference in the energy between the two states consists of the work output from the turbine and the heat transfer between the turbo surfaces and the environment. Due to the lack of pre-turbine temperature and related gas velocity information, the turbine work cannot be directly calculated from the changes in stagnation enthalpies between the states; solving the work consumed by compressor ($\dot{W}_C$), however, is more straightforward with Eq. (6) and the following assumptions:

- Negligible elevation difference;
- The surface heat losses across the compressor are usually small [30], which allows the compressor to be approximated as an adiabatic control volume.

$$\dot{W}_C = \dot{m}_{air}(h_{02} - h_{01}) \quad (6)$$

Given that the minimum work transfer rate produced by turbine is namely $\dot{W}_C$ when the mechanical efficiency of the turbocharger system is assumed to be 100%, the work output from the turbine can be consequently estimated from Eq. (6). Adding the total energy losses across the turbine, i.e. the sum of turbine work and surface heat losses, onto the exhaust enthalpy calculated from the post-turbine temperature yields the pre-turbine exhaust energy.

4. Results and Discussion

The methods described in the previous section are now applied to the experimental energy balance analyses by using the control volume shown by Figure 3. These results are subject to the constraints of the experimental measurements and to the approximations involved in the methodology. First, this section compares the experimental results to the predictions from simulation and attempts to explain the possible factors causing the disparities in the relevant

energy terms. Then, the ultimate objective is using the energy balance to assess how the variation in ECT impacts the energy conversion efficiency.

4.1 Energy distributions

Figure 4 compares the simulated energy fractions with the experimentally derived results for Case #4 (ECT=90°C). The comparisons for the remaining cases are displayed in the Appendix C, which appear to have similar pattern to that shown in Figure 4. The changes in the three different energy flows with ECT are summarized in Figure 5.

In general, it is found from both simulation and experiment that more than 40% of the input fuel energy is converted into net indicated power. The net indicated power slightly increases as the ECT increases, more discussions on this will be detailed in section 4.2 along with discussions of the fuel conversion efficiency. The simulation results are underestimated by approximately 2-3%, which do not deviate from the experimental results as much as the other two terms do. Typically precise predictions in cylinder pressures can avoid significant inaccuracies of $W_{i,net}$, because the net indicated work is usually derived from the pressure-volume analysis, see section 3.1. Since the simulated pressure is in good agreement with the validation data as indicated in Figure 1, the uncertainties of the flow characteristics are suspected to be the primary reason causing the errors in predicted net indicated power [26].

Figure 4 Energy distributions of experiment and simulation for Case #4

Figure 5 Summarized energy distributions at five different ECTs

Similar to previous studies on LHR engines [10-11], raising ECT also yields gradually decreased net in-cylinder heat transfer. Compared to the term of net indicated power, there are more significant discrepancies in heat transfer existing between the simulation and experimentally derived results. The in-cylinder heat transfer fractions determined from Eq. (3) reveal to be higher than the simulation by about 2-6%. As discussed in section 3.2, using Eq. (3) to estimate the in-cylinder heat transfer will likely result in substantial inaccuracies since the process tends to introduce additional errors from calculating $\dot{Q}_{cool}$ and $\dot{Q}_{unac}$. For example, the $\dot{Q}_{cool}$ is calculated based on the assumption of constant coolant properties, i.e. constant pressure specific heat. In addition, the friction power is obtained by subtracting brake power from net indicated power rather than directly measuring from motoring engine. In this work, a more significant source of errors is likely to be the computation of $\dot{Q}_{unac}$. It is usually difficult to accurately capture the heat losses to the environment, especially when operating the engine under LHR conditions where higher engine surface temperature yields more radiative heat transfer occurring

between external surfaces and the environment; also, the heat transfer coefficients used for the calculations are collected from empirical correlations. Since a direct measurement of heat flux is not available in this study, an alternative way to further examine the simulation results is calculating the cylinder heat transfer from measured pressures by using Hohenberg's correlation [31]. The comparison between engine simulation and Hohenberg model results are detailed in the Appendix D.

On the contrary, experimental exhaust energy fractions tend to be lower than the simulation results by 6-7%, which nearly offsets the disparity found in the heat transfer term. The methodology used to calculate experimental exhaust energy has been introduced in section 3.3. The primary source of errors in the exhaust energy estimation is suspected to be the location of exhaust temperature measurement. Based on the energy systems depicted in Figures 3, the actual exhaust energy should be calculated from the pre-turbine exhaust temperature; more specifically, the target temperature should be measured at a position near the exhaust valve or the upstream end of exhaust manifold, i.e. state 1 shown in Figure 6. As discussed in section 3.3, the post-turbine temperature (temperature at state 3, see Figure 6) is the only available measurement from the instrumented engine. Although the experimental data has been corrected to be the pre-turbine exhaust energy (state 2), the experimentally calculated exhaust energy is still not an exactly "parallel" reference to the simulation-based counterpart; the latter is, therefore, higher than the former due to the lower exhaust gas temperature used for the experimental analyses. As mentioned earlier, such a constraint in temperature measurement consequently causes the heat transfer occurring between exhaust gases and exhaust port walls to be counted toward the in-

cylinder heat transfer in the experimental calculations, which yields a corresponding difference from the simulated heat transfer.

Figure 6 Illustration of the turbocharged engine

1) target state; 2) pre-turbine; 3) post-turbine

A sensitivity study was completed to evaluate how much the exhaust temperature affects the level of exhaust energy. Figure 7 gives the exhaust energy fractions at different exhaust temperatures, where the baseline measurement refers to the result computed from the post-turbine exhaust temperature and the maximum temperature 500 K is set to be the simulated temperature at the target position, i.e. state 1. Changing the exhaust temperature, from post-turbine temperature 418K to 500K, yields different exhaust energy fractions being denoted by the bars in Figure 7, which tend to increase linearly with the temperature. Square symbols represent the relative changes in exhaust energy with respect to the baseline case, which appear to be about 2.5 times faster than the rate of relative change in exhaust temperature, as denoted by the triangle symbols. In general, the exhaust energy is shown to be very sensitive to the temperature; for Case #4, the exhaust energy fraction rises by 48.1% (from 25.1% to 37.2%) as the exhaust temperature relatively increases by 19.6% (from 418K to 500K). Such analysis

confirms the need to have proper exhaust temperature measurement when calculating the exhaust enthalpy. Apart from the uncertainties involved in experimental analyses, there are also some potential inaccuracies existing in the simulation work. For example, generally the combustion models provided by the engine simulation software do not have the capability to predict the hydrocarbon emissions for compression ignition engine models. This means that the simulated exhaust energy does not account for the portion carried by the unburned hydrocarbons.

Figure 7 Sensitivity study on exhaust temperature for ECT=90°C (Case #4)

4.2 Fuel conversion efficiency

Significant improvements in brake fuel conversion efficiency were observed in both simulation and experimental studies, as it is suggested in Figure 8. The simulation results indicate that over 20% relative improvement in the brake fuel conversion efficiency (from 21.1% to 26.4%) is achieved as ECT changes from 56.5°C to 100°C; the experimental analyses also indicate a similar level of improvement (~23.5%). The net indicated fuel conversion efficiency, however, reveals only slight improvement compared to that shown by brake efficiency. Increasing ECT

from 56.5°C to 100 °C causes only 2.7% relative improvement in the experimental net indicated efficiency, such insignificant efficiency gains are also found in the simulation study.

Figure 8 Fuel conversion efficiencies as functions of ECT

In general, the LHR application improves the fuel conversion efficiencies, which may be attributed to several beneficial phenomena caused by elevated ECT:

- Reduced in-cylinder heat transfer resulting in higher gas temperatures promotes the combustion toward completeness. It has been reported in [23] that increasing ECT reduces CO and HC pollutants, they serve as indicators of an increase in combustion efficiency. Such efficiency gains would be observed in both brake and indicated fuel conversion efficiencies; this may partly explain the slight increase in experimental indicated efficiency, which would not appear in the simulation indicated efficiency as the simulation does not capture CO and HC.
- Retained energy from the LHR operation yields shorter ignition delay (ID), which might result in an increased effective expansion ratio depending on the influence of ignition delay on combustion duration. Figure 9 shows how the ignition delay, defined as the interval between start of injection (SOI) and 10% mass fraction burned (MFB), and

combustion duration (10-90% MFB) change with the engine coolant temperature, the ignition delay gradually decreases as ECT increases. It is known that the injection timing for the studied cases is constant at 8°bTDC, the high-temperature condition facilitates the start of combustion to be advanced and consequently the ignition delay tends to become shorter at higher ECTs. A subsequent extension of the combustion duration is due to the shorter ignition delay causing the fraction of burned gases in the premixed combustion phase to reduce, and the fraction of diffusion combustion consequently increases. Generally, diffusion combustion burns at a slower rate such that, for a fixed amount of fuel, increased diffusion burning at the expense of premixed burning will increase combustion duration. The predicted combustion durations deviate from validation data significantly, which likely results from the overestimated burn rates from the engine simulation. Figure 10 compares the experimental burn rates with that calculated by the engine simulation having prescribed pressure history, which indicates that the engine simulation cannot achieve perfect MFB matching even with measured pressures as the input. This is likely because the simulation code is using different methodologies or assumptions to calculate the burn rate. Whereas the simulation results do characterize a variation pattern consistent with the experiment, providing clues to help explain the gains in engine efficiency. In summary, changes to combustion phasing will impact both indicated and brake efficiencies; even though combustion phasing may change slightly at various ECTs, the changes do not appear to be substantial enough to cause significant changes to indicated efficiency.

Figure 9 Ignition delay and combustion duration as functions of ECT

Figure 10 Comparison of burn rates (mass fraction burned), Case #1

- Frictional losses decrease at higher temperature conditions. For the conventional low load condition, the reduction in friction is likely to be the dominant factor causing the noticeable improvement in brake efficiency. Robinson [22] also points out that increasing the coolant temperature of a conventional cooling system usually yields a fuel consumption benefit via reduced crank train friction. This is perhaps due to lowered lubricant viscosity as wall and oil temperatures increase with higher ECTs, decreasing the hydrodynamic friction in bearings, valve trains, along liners, and in the oil pump. Figure 11 indicates the decrease in friction mean effective pressure (FMEP) and increase in mechanical efficiency at higher ECTs, respectively.

Figure 11 FMEP and mechanical efficiency as functions of ECT

5. Summary and Conclusions

The first law of thermodynamics was implemented to examine the strategy of altering engine coolant temperature to devise a version of LHR application in a conventional light-duty diesel engine. With the combustion chamber based control volume, a comparative study between simulation and experiment was carried out to investigate the disposition of initial input energy at five different ECTs. Due to the constraints involved in the measurements, the methodology used for the experimental energy balance analyses was also carefully scrutinized to better analyze the simulation results.

5.1 Research findings

- Both the experimental and simulation results indicate that more than 40% of the input fuel energy is converted to net indicated work, about 1/3 of the energy is rejected to the exhaust gases and the rest is transferred as thermal energy through the combustion chamber walls. Regarding to the net indicated power as a percentage of the total fuel energy, simulation results are generally in agreement with the validation data. Noticeable

disparities were found in exhaust energy fractions due to the constraint of the exhaust temperature measurement, which also partly causes the likely overestimation of experimental in-cylinder heat transfer. Potential uncertainties involved in the engine simulation are also discussed to explain the observed disparities.

- To better understand the proposed LHR concept, the energy balance results are further applied to correlating the fuel conversion efficiency to the variations in ECT. It is found that increasing ECT yields reduction in cylinder heat transfer but almost equivalent increase in the exhaust losses, which explained why varying ECT achieves slight improvements in net indicated fuel conversion efficiency. The brake fuel conversion efficiency, however, shows to be significantly improved at the studied higher ECT conditions where the substantial reductions in frictional losses may occur.

5.2 Future scope of the work

- The second phase of this study is to investigate the gains in engine performance and emissions mitigation through the integration of the proposed LHR concept with low temperature combustion strategy.
- The possible limits of the proposed LHR technique can also be explored through achieving more aggressive reduction in the heat rejection, extended studies at wider operation range, such as high load and high speed conditions, will help to better understand the LHR concept.

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Figure 12 Validation of pressure and ROHR
(Case#1)

Figure 13 Validation of pressure and ROHR
(Case#2)

Figure 14 Validation of pressure and ROHR
(Case#3)

Figure 15 Validation of pressure and ROHR
(Case#4)

Appendix B

In some previous studies on engine energy balance, the engine block has been considered as the steady-state control volume with air and fuel energy entering the open system and four energy flows leaving the system [32-34]. As can be seen from Figure 16, these four energy flows leaving the control volume are brake power ($\dot{W}_b$), heat losses to the coolant ($\dot{Q}_{cool}$), exhaust energy ($\dot{m}_{exh}h_{exh}$), and the unaccounted energy ($\dot{Q}_{unac}$).

$$\dot{m}_f h_{fuel} + \dot{m}_{air} h_{air} = \dot{W}_b + \dot{Q}_{cool} + \dot{m}_{exh} h_{exh} + \dot{Q}_{unac} \quad (7)$$

It is evident that the major difference between Eq. (7) and Eq. (1) is the energy flows leaving the control volume. The unaccounted energy includes the heat rejected to the oil plus convection and radiation dissipated to the environment through the engine's external surfaces.

Figure 16 The engine block as a thermodynamic open system operating in a steady-state fashion

In addition to what shown in Eq. (2), the net indicated work can also be expressed as a relationship of different forms of work:

$$W_{i,net} = W_b + W_f \quad (8)$$

All the work terms discussed in this paper are referred to as the absolute values without accounting for the sign convention, and the relationship of Eq. (8) can also be applied to the corresponding power parameters. Comparing to the different control volume system described in

Figure 3, rearranging Eqs. (1), (7), and (8) yields the net in-cylinder heat transfer rate $\dot{Q}_{HT}$ as given by Eq. (3).

Figure 17 Energy distributions of experiment and simulation for Case #1

Figure 18 Energy distributions of experiment and simulation for Case #2

Figure 19 Energy distributions of experiment and simulation for Case #3

Figure 20 Energy distributions of experiment and simulation for Case #5

Appendix D

Figure 21 compares the instantaneous heat transfer rates calculated from Hohenberg's model and the simulated rates predicted from WoschniGT model, i.e. the simulation data reported in Figure 4. The estimations from Hohenberg's correlation show to be higher than the WoschniGT simulation results during the compression stroke, but which are significantly lower as combustion starts. The average heat flux calculated from Hohenberg's correlation is lower than the WoschniGT simulation results by about 3-5%, see Figure 22. These comparisons are consistent with studies of the heat transfer modeling between Hohenberg's model and the classic Woschni's model [31]. Typically, most models do not account for heat transfer between the hot gases and valve back. The heat transferred from gas through the intake/exhaust port walls is also excluded from both modeling results, which is unavoidably lumped into experimental analyses due to the constraint in exhaust temperature measurements. Therefore, the experimentally derived in-cylinder heat transfer is likely to be overestimated compared to the simulation results.

Figure 21 Heat transfer rates predicted from Hohenberg and WoschniGT models

Figure 22 Average heat transfer predicted from Hohenberg and WoschniGT models