Validation of In-cylinder Heat Flux Estimation Model by Measuring Wall Temperature

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ABSTRACT: A new equation, which was dependent on physical principles, was developed for the study of heat transfer in CI engines which needs turbulence of gas flows to calculate heat flux. Proposed approach was implemented into a 1-D engine simulation, which was used to determine heat flux between in-cylinder gas and wall. Results from the suggested equation were compared to the previous conventional equations; Morel and Hohenberg, and to the engine experiments. The proposed equation showed better accuracy when compared with the conventional equations due to detailed representation of in-cylinder gas flow by dividing the combustion chamber into 6 different regions.

KEY WORDS: heat engine, compression ignition engine, numerical calculation, theory/modeling [A1]

1. Introduction

Along with the rapid industrialization of developing countries such as China, India, and so on, impact on environment, especially global warming, has been on rise since the beginning of the 21st century. According to the report from IEA (International Energy Agency) called “CO₂ Emissions from Fuel Combustion 2017 Overview”, 26.5% of the CO₂ emissions are related to the transportation [1]. Regarding to these data, it is inevitable to see the importance of new innovations on internal combustion engines as they are still the most common means of transportation that are still being used throughout the world. As a result of these, stricter emission regulations started to be adapted since the beginning of 2000s. These regulations have been implemented by various autonomous organizations to every car manufacturer by the means of different types of test methods. The newest method is called Worldwide Harmonized Light Vehicles Test Procedures (WLTP), which is aimed to surpass the conventional New European Driving Cycle (NEDC), in October 2018. The test conditions for WLTP have been risen to 100% for dynamic cycle which aimed to be more representative of real driving conditions thus more realistic emission results from vehicles on the road, where NEDC only had single test cycle. For diesel engines, in order to meet these strict regulations, improvement in thermal efficiency under real driving conditions is a must. One of the ways to achieve higher thermal efficiencies from DI engines is the accurate control of the diesel combustion, which is related to the precise estimation of in-cylinder gas temperature and pressure.

In literature conventional equations such as Woschni, Hohenberg, Morel have been used in number of research for heat transfer representations in order to estimate the heat flux between in-cylinder gas and surrounding wall inside internal combustion engines (ICE). However, these correlations’ applicability to different kinds of ICE are questionable. Thus, most of these empirical equations have been considered to be inaccurate for real driving conditions because of their somewhat simple constructions. Thus, a new equation has been introduced in order to calculate the heat flux from engine cylinder. The suggested equation calculates heat transfer based on the fundamental theories of thermodynamics, which are based on the physical properties of the in-cylinder gas velocity and temperature. Modeling approach was done by dividing the combustion chamber into 6 different regions while taking into account of 4 different gas flows inside the combustion chamber during and after the intake stroke. By its detailed but not too complicate structure, the suggested equation can even be implemented to an engine control unit (ECU), to control the injection timing for reaching the optimum ignition timing for diesel engines. The equation was based on a single zone combustion model which is designed to be simple, yet accurate when compared to previous conventional equations.

For the analysis, actual engine experiments were performed to measure the heat flux and heat loss from the engine. Numerical simulations were done simultaneously in order to compare the results obtained by the conventional correlations; Hohenberg [4]. Morel [12] and suggested empirical equation, Suzuki [13]. Hohenberg and Morel’s correlations can be found as the default equations that are being used to calculate the heat transfer phenomena in commercial engine software, such as GT-Power. The aim of this paper is to develop an equation with low calculation load while obtaining a higher accuracy when compared to the conventional equations such as Hohenberg or Morel. Furthermore, this model will be suggested to be used as a default equation in 1-D engine simulation software.
2. Modeling Approach

2.1. Wall Heat Transfer Model

In order to calculate the wall heat flux from the engine cylinder’s wall to the gas, or vice-versa the following equation was suggested by the authors [1]:

\[
q_w = -\frac{C_s \kappa}{P_0 \kappa - 1} \frac{P T}{\sqrt{T_s}} \left\{ \frac{1}{\sqrt{\pi t}} \left[ \frac{T_w}{T_s} - \frac{1}{\sqrt{t}} \right] - \frac{\psi_k}{4} \frac{P}{P_0} (T_s - T_w) \right\}
\]

\[
\tau = \frac{1}{P_0} \int_0^t \rho dt
\]

(1)

(2)

where \( q_w \) : wall heat flux [W/m²], \( C_s \) : Slope of the thermal conductivity of the in-cylinder gas with respect to temperature [W/m·K²], \( P_0 \) : In-cylinder pressure at TDC during IVO [Pa], \( \kappa \) : Specific heat ratio [-], \( P \) : Instantaneous in-cylinder pressure [Pa], \( T_s \) : Instantaneous in-cylinder gas temperature [K], \( T_w \) : Wall temperature [K], \( t \) : dimensionless time [-], \( \psi_k \) : Karman constant (≈0.4) [-], \( \psi \) : constant pressure specific heat [J/kg·K], \( \dot{u}_i \) : regional gas flow velocity - with turbulence term depending on the region [m/s]. The turbulence term, \( \psi \), is calculated from the 3D-CFD analysis and multiplied with the \( \dot{u}_i \) term to determine \( \dot{u}_i \). \( t \) : elapsed time from TDC during the intake stroke [s]. Subscript \( i \) depicts the 6 different regions inside the engine cylinder (See Figure 1).

The suggested heat transfer equation (1) was based on physical properties and uses the energy and the continuity equations. The model consists two parts. One is the wall surface heat flux due to thermal boundary layer development and gas temperature change, depicted in the first term of the equation. In the second part, wall surface heat flux due to the influence of various gas flows is represented, which was depicted by the \( \dot{u}_i \) term. Regional gas flow representations explained comprehensively in the next section.

Since the suggested equation is based on the physical laws, in addition to being more accurate compared to the previous empirical equations, calculation load was also improved by the reduction of the calculation of the thermal boundary layer \( \left( \psi \right) \) term. Another approach was made by Rakopoulos [5] which took into account of this thermal boundary layer term \( \left( \psi \right) \), however, due to the complexity of the equation, calculation load and time was found to be exceeding the limits for a conventional ECU. This reduction on the \( \psi \) term enabled the suggested equation (1) to do calculations less than 80 microseconds, which is the amount of time per CA for an engine running at 2000 rpm.

Suggested equation (1) was implemented to engine simulation software in a Fortran code by the help of necessary pre-defined subroutine, which was inside the engine cylinder object in order to calculate the heat flux and heat release rate.

2.2. Modeling of the in-cylinder gas flow

In IC engines the mixture of intake air and fuel plays an important role when combustion efficiency is considered. Homogenous the mixture is desired for better the quality of combustion which is a common view on the subject. There are various ways to enable a homogenous mixture and having different gas flows in an engine cylinder is one of them. Our approach consisted of 4 different gas flows inside the engine cylinder, namely; swirl, axial, squish and fuel injection. Swirl flow mainly depends on the intake manifold design, whereas axial and squish flow are more dependent on the engine cylinder and piston designs. Injection flow gets affected by the number of nozzles, nozzle diameter and angle of the fuel injection with injection pressure. In the construction of the in-house built equation, the engine cylinder was divided into 6 different regions, where indicated 4 different gas flows were assumed to be affecting the in-cylinder heat transfer phenomena. This division of the engine cylinder into different regions and their gas flows were also based on the modeling done by Schubert et al. [14]. These 6 different regions are shown in Figure 1 and named as according to their location in the engine cylinder. Gas flow at piston top depicted as \( u_1 \), cavity side as \( u_2 \), cavity bottom as \( u_3 \), liner as \( u_4 \), inner head \( u_5 \) and outer head as \( u_6 \). Equations below were constructed depending on the swirl \( (u_6) \), squish \( (u_{sq1}, u_{sq2}) \), axial \( (u_{axial}) \) and injection \( (u_{inj}) \) flows throughout the working cycle of an IC engine. Injection flow is characterized by Inagaki’s equation [16] which was based on Hiroyasu’s equation [17] depending on the elapsed time, \( t \) [s], between \( t_0 \) : spray breakup time [s] and \( t_{inj} \) : injection period [s] as shown in below:

\[
u_{inj} = 1.48\left((P_{sac} - P_{atm})/\rho_{air}\right)^{0.25} \times \sqrt{d_0 t^{-0.5}} \quad \text{for} \quad t_0 < t < t_{inj}
\]

\[
u_{inj} = \frac{v_{gap,inj}}{\psi(t/t_{inj}) + v_{gap,inj} + 1} \quad \text{for} \quad t \geq t_{inj}
\]

(3)

where \( P_{sac} \) : pressure at injector nozzle [bar], \( P_{atm} \) : pressure inside the engine cylinder at main injection timing [Pa], \( \rho_{air} \) : nozzle inlet density [kg/m³], \( d_0 \) : nozzle diameter [m], \( \psi \) : air drag coefficient on injected fuel [-], \( v_{gap,inj} \) : injection speed at injection end time [m/s]. More details about the equation can be found. Spray breakup time, \( t_0 \), was calculated according to the explanation given by Hiroyasu [17]. Both pressure values of nozzle and cylinder were based on the conservation of energy and continuity equations, which were calculated by the 1-D engine simulation software.

\[
u_1 = \sqrt{\left(\frac{u_{inj}}{2}\right)^2 + u_{\psi,1}^2} \quad \text{Piston Top}
\]

\[
u_2 = \sqrt{\left(2u_{inj} + u_{sq1} + u_{sq2}\right)^2 + u_{\psi,2}^2} \quad \text{Cavity Side}
\]

\[
u_3 = \sqrt{u_{inj}^2 + u_{\psi,3}^2} \quad \text{Cavity Bottom}
\]

\[
u_4 = \sqrt{\left(\frac{u_{inj}}{2}\right)^2 + u_{\psi,4}^2} \quad \text{Liner}
\]

(4)

(5)

(6)

(7)
In total of six different heat flux values were calculated, which were later averaged over three areas; piston ($u_1$, $u_2$, $u_3$), liner ($u_4$), and head ($u_5$, $u_6$), respectively.

2.2. Previous Convective Heat Transfer Correlations

In literature heat transfer equations from Hohenberg [4] and Morel [12] have been used broadly. These conventional empirical equations were pre-defined in the engine simulation software, and could be selected as default heat transfer correlations for heat transfer inside the engine cylinder. These equations were used consecutively for the comparison with the experimental data and suggested in-house equation (1). Heat transfer was simply calculated from the heat flux data multiplied with the area where heat transfer is taking place. In this study, as referred before, three main sections were selected for heat flux calculations; piston, head and liner. From Fourier’s Law in convective heat transfer case heat flux is determined from the following equation:

$$ \dot{q} = \alpha \Delta T $$

where $\dot{q}$ : heat flux [W/m$^2$], $\alpha$ : heat transfer coefficient [W/m$^2$K], $\Delta T$ : temperature difference [K]. The heat transfer coefficient term depends on the used correlation such as Hohenberg, or Morel. In primitive terms Hohenberg’s equation can be considered as the revised version of the Woschni equation [6] which was shown as the following equation:

$$ \alpha = C_1 V_c^{0.06} P^{0.8} T^{-0.4} (\bar{V}_p + C_2)^{0.8} $$

where $C_1$, $C_2$ : constants [-], $V_c$ : Cylinder volume [m$^3$], $P$ : pressure [bar], $T$ : mean gas temperature [K], $\bar{V}_p$ : mean piston speed [m/s]. Depending on Hohenberg’s experiments constants $C_1$ and $C_2$ were chosen as 130 and 1.4, respectively. Hohenberg examined Woschni’s formula, which was based on laws of similarity and changes were made to give better predictions of time-averaged heat fluxes measured with probes in a DI diesel engine with swirl motion. These modifications can be seen in the velocity and the volume terms when compared to the Woschni’s correlation.

Morel offered another approach based on Colburn analogy with an effective velocity representation which took into account of the kinetic energy of squish, intake flow and injection, and swirl gradients at region interfaces [4]. The heat transfer coefficient was formulated as the following equation:

$$ \alpha = 0.5 c_f \rho U_{eff} C_f Pr^{-\frac{2}{3}} $$

where $c_f$ : skin friction coefficient [-], $\rho$ : gas density [kg/m$^3$], $U_{eff}$ : effective gas velocity outside a boundary layer at a particular surface location [m/s], $Pr$ : Prandtl Number [-]. Similar to Suzuki’s approach, engine cylinder was divided into six different regions, where thermal boundary layer thickness is assumed to be spatial variable. The skin friction coefficient term includes previously indicated six sub-regions inside the engine cylinder, which was considered to be representing the thermal boundary layer thickness depending on those six different locations. The assumption was expected to be adequate as boundary layer thickness had a small influence, which was formulated as the following:

$$ c_f = a \frac{\rho U_{eff} \delta}{\mu} $$

where $a$ : constant (=0.046 for flat plate boundary; =0.067 for fully developed pipe flow) [-], $\delta$ : regional thermal boundary layer thickness [m], $\mu$ : dynamic viscosity [Pa-s]. Another difference was made to the gas velocity term, $U_{eff}$, which was formulated as below:

$$ U_{eff} = (U_s^2 + U_i^2 + 2k)^{\frac{1}{2}} $$

where $U_s$, $U_i$ : velocity components parallel to the surface outside of the thermal boundary layer [m/s]. Effective velocity outside the boundary layer for each surface was calculated from swirl, tumble, turbulence and mean piston velocity. $k$ : kinetic energy of turbulence [m$^2$/s$^2$]. Both empirical equations (10) and (11) were focused on having a better gas flow representation. However, both of them lacked the adequacy levels in an actual engine’s working conditions. Suggested equation (1) by Suzuki gives the most detailed version of the gas flow representation in an engine cylinder with the addition of injection flow parameter, thus it is proposed to have the best fit to the experimental results from an actual engine and can be used as an alternative model when heat transfer calculations are concerned in 1-D engine simulations.

### 3. Experimental Apparatus and Conditions

#### 3.1. Engine Specifications and Instrumentation

Experiments were conducted on a single cylinder supercharged diesel engine with a 4 valve direct-injection system, with a variable valve timing mechanism. Details about the engine are given in Table 1. Experimental instrumentation used to measure
the in-cylinder wall heat flux and in-cylinder wall surface temperature were listed in Table 2.

Table 1 Engine specifications.

| Engine                  | Single cylinder engine |
|-------------------------|------------------------|
| Cylinder displacement [cc] | 550                    |
| Bore [mm]               | 85                     |
| Stroke [mm]             | 96.9                   |
| Cylinder offset [mm]    | 6.5                    |
| Piston-pin offset [mm]  | 0.8                    |
| Compression ratio [-]    | 16.3                   |
| Intake valve opening period [deg.] | From 347 to –120 (Comp. TDC is 0 deg.) |
| Exhaust valve opening period [deg.] | From 122 to –330 (Comp. TDC is 0 deg.) |

Table 2 Measurement Instrumentation.

| Instrument                | Manufacturer       | Type                  |
|---------------------------|--------------------|-----------------------|
| Heat flux sensor          | Medtherm corp.     | TCS-K 10702A          |
| Thermocouple              | Medtherm corp.     | TCS-K 10835           |
| Cold junction compensator | Omega eng. Inc.    | MCJ-BATT-B            |
| Amplifier                 | NF corp.           | 5307                  |
| Data logger               | Yokogawa electric corp. | DL850E            |
| In-cylinder pres. sensor  | Kistler Japan      | 6052                  |

Engine head and piston were made of an aluminum alloy, and the liner was made of cast iron. The details of the equipment used are shown in Table 2, and the schematic diagram of the experimental apparatus is shown in Figure 2. Coaxial heat flux sensors and coaxial thermocouples were installed at various parts of the engine where current signals from high-speed response were passed through a DC amplifier and recorded in a data logger together with the piezoelectric in-cylinder pressure sensor. At this time, by synchronizing the phase A of the rotary encoder mounted on the crankshaft and the external clock of the data logger, it was possible to acquire wall heat flux data at every crank angle during a one full cycle of the engine.

3.2. Heat Flux Calculations

Figure 2 illustrates the schematic diagram of the engine used during the heat flux measurement experiments. Figure 3 (d) shows the whole view of the cylinder, and a total of 10 thermocouples which were installed in the engine head, piston and liner. In order to measure the heat flux, inside the engine cylinder, eight sensors were installed (2 at the head, 6 at the liner) and 2 surface thermocouples were installed at piston as shown in Figure 2.

Figure 3 (c) shows the wall temperature measurement position inside the engine cylinder. Figure 3 (a) shows the measurement position of the cylinder head. Coaxial heat flux sensors were installed at 17 mm and 35 mm from the center of the injector. At the cavity area, surface thermocouples were installed on the squish area and piston top. Figure 3 (b) shows the measurement positions of the cylinder liner where 6 heat flux sensors were used on the exhaust and intake sections (3 sensors at each side as shown in schematic diagram 3 (b)), with 43 mm separation between them.

3.3. Experimental conditions

Table 3 shows the experimental conditions used during the heat flux measurement.

Table 3 Experimental conditions.

| Condition               |        |
|-------------------------|--------|
| Engine speed [rpm]      | 1500   |
| Injection pressure [MPa]| 80     |
| Pre-injection timing [deg.] | 347  |
| Pre-injection duration [usec]| 180  |
| Main-injection timing [deg.] | 363  |
| Main-injection duration [usec]| 440  |
| IMEP [kPa]              | 438    |

The lubricating oil temperature and the cooling water temperature were both kept constant at 80 °C. Engine speed was set to 1500 rpm with IMEP at 438 kPa. No exhaust gas recirculation
was used during the experiments. A DC amplifier (NF Corp.) was used in order to analyze the experimental data. Additionally, a data logger (Yokogawa Elec. Corp.) was used to save the necessary data.

4. Experimental Results and Discussion

4.1. Heat Flux

As previously indicated two conventional empirical equations (10) and (11), and newly suggested heat flux equation (1) were implemented into the 1-D engine simulation software and simulations results were compared to the experimental data. Naturally aspirated direct injection diesel engine where engine speed was fixed at 1500 rpm. Crank angle (CA) 360 is the TDC for compression stroke for all figures from here on.

Figure 4 illustrates the heat flux comparison from piston area. In Hohenberg’s equation (10), gas flow was concentrated on mean piston speed along with change in chamber volume and its accuracy to the experimental results was found to be the best at the piston when compared to other sections of the engine cylinder (head or liner). However, its degree of competence was lower when compared to other equations. The suggested equation (1) also showed similar pattern to the Morel’s equation (11) as the turbulence term was mostly caused after the fuel injection and the start of the combustion. Experimental data showed higher heat flux results at engine piston and head when compared to the all indicated equations.

![Heat flux comparison at engine piston.](image)

**Fig. 4 Heat flux comparison at engine piston.**

Figure 5 shows area averaged heat flux data from engine head with the comparison of three equations to the experimental data. Even though all the correlations lacked perfect accuracy, suggested equation (1) showed the best fit to the experimental data. Better representation of the gas flow and its effect on the heat flux from the engine head can be visualized when compared to the conventional correlations (10) and (11). As indicated before the swirl motion effect was included in all correlations, which became somewhat easy to observe before TDC, between 340-360 CA.

After the TDC, fuel injection and start of combustion (SOC), once again all correlations showed an increase in heat flux. However, Hohenberg’s equation could not follow the increasing pattern as other correlations did. For Morel’s equation, addition of the turbulence term seemed to be effective after TDC, which showed best correlation at this region. Suggested equation also showed good correlations as the detailed gas representation seemed to be working.

Liner section of the engine was somewhat the toughest part to measure the heat flux as the area where heat transfer occurred changed simultaneously due to the reciprocating motion of the piston. Measured thermocouples and heat flux sensors were changing due to the reciprocating motion of the engine piston. Once again swirl motion effect is visible for all indicated equations in Figure 6. However, after the SOC Morel’s equation showed an over-prediction by almost twofold when compared to the experimental results. In-housed built equation (1) and Hohenberg’s approach showed very similar results to the experimental values with a slight over-prediction at the peak of the heat flux.

![Heat flux comparison at engine liner.](image)

**Fig. 6 Heat flux comparison at engine liner.**

When all three locations were compared, during the experiments highest heat fluxes were measured on the engine piston. This was a result of hot combustion air directly penetrating towards the engine piston right after SOC. Second highest values were found to be in the engine head and smallest values were calculated at the liner, as expected. Hohenberg showed almost similar results at all sections with biggest difference to the experimental results. Morel showed great consistency at engine head, however at engine piston and liner its accuracy dropped immensely.

In-house built equation (1) showed somewhat similar pattern to the experimental results when heat flux values were compared at three different regions. However, as shown in the figures, there is an overestimation during the second half of the compression stroke and an underestimation during the expansion stroke, for piston and head regions. Similar results were present in Morel’s model which also characterized different gas flows [12]. The reason behind for the overestimation during the compression stroke is due to the swirl flow model used in our study, which needs an improvement to lower its effect during this phase. The underestimation during the
expansion stroke was eliminated partially by introducing a new approach to change the drag coefficient term, \( \psi \). The previous model used the drag coefficient as a constant value, while the present study employed the variable term based on Inagaki et al. [16]. This drag coefficient term represents how the fuel particles are getting affected by the trapped air inside the engine cylinder after the completion of the fuel injection. Further improvement for the expansion stroke can be achieved by remodeling the turbulence term, \( c_1 \), defined in equation (1), which gives the perpendicular velocity term for the gas flow characterization.

Table 4 depicts calculated average heat flux errors values from three different equations to the results obtained throughout the engine experiments at each three sections (piston, head, and liner). Measured and calculated data were focused on compression and expansion (power) strokes, crank angles between 270-450 deg.

Table 4  Are Averaged heat flux error comparison in between experimental results to the equations ( Exp. / Eq. )

|          | Piston | Head | Liner |
|----------|--------|------|-------|
| Eq. (1)  | 12%    | 28%  | 9%    |
| Morel    | 25%    | 22%  | 42%   |
| Hohenberg| 53%    | 29%  | 12%   |

At piston section, in-house built equation (1) showed the best fit to the experimental data with an error of 12%. Morel showed the second best fit with 25%. Hohenberg showed the biggest difference with an error of 53% to the experimental data. For head section, Morel showed the best correlation to the experimental results with an error of 22%, as depicted in Figure 5. In-house built equation also showed good correlation at head section to the experimental data with 28%, where Hohenberg lacked accuracy with an average error of 29%. For the liner section, Hohenberg showed good results with an average error only of 12%, where Morel over-predicted the heat flux by almost twofold as seen in Figure 6. In-house built equation results were the closest to the experimental data at liner section, which resulted with an average error of 9%. When all sections were averaged, Suzuki equation showed the best fit to the experimental results with 16% as the average error. Morel and Hohenberg’s errors to the experimental data were found to be 30% and 32%, respectively.

Similar to Table 4, Table 5 shows the error from indicated equations to the experiments at each three sections for calculated maximum heat flux results.

Table 5  Maximum heat flux error comparison in between experimental results to the equations ( Exp. / Eq. )

|          | Piston | Head | Liner |
|----------|--------|------|-------|
| Eq. (1)  | 15%    | 21%  | 16%   |
| Morel    | 48%    | 28%  | 36%   |
| Hohenberg| 77%    | 67%  | 17%   |

Suggested equation (1) showed the closest fit to the experimental data for the area averaged maximum heat flux values (taking the average of all three section in Table 5 for Eq. (1)) when compared with the other conventional equations, which was calculated to be 17%. For Morel’s equation the error for maximum calculated heat flux values to the experimental results was found to be 37%. Even though at liner section Hohenberg’s results showed a good correlation, when all sections are concerned its error increased to 54% for maximum heat flux comparison. These results once again emphasize the importance of characterization of the in-cylinder gas flows in a detailed manner.

4.2. Total Heat Transfer Rate & Heat Loss Comparison

Figure 7 shows total heat transfer rate calculated from two empirical equations and in-house built equation and their comparison to the experimental data. Similar to the heat flux results, both Morel and equation (1) showed good fit to the experimental data. On the other hand, once again Hohenberg showed the least fit to the experimental data when all sections were concerned. Newly suggested in-house built equation showed the best fit to the experimental results with an accuracy of 85%. This result was considered to be sufficient enough for newly suggested equation to be implemented in an engine simulation software for heat flux and heat loss calculations.

Figure 8 illustrates the comparison of total heat loss during compression and expansion strokes results obtained from experiments and three different equations. Similar to the heat flux results, similar order observed when heat loss calculations were done. Experimental results showed an overall heat loss of 95.87 W, where in-house built equation calculated the heat loss as 81.82 W. From Morel’s equation it was found to be 80.62 W. On the other hand, Hohenberg showed only 59.04 W of heat loss from the engine cylinder. Higher flux values at liner section from Morel’s equation caused affected total higher heat loss calculations, resulting in similar values as calculated from in-house built equation.
5. Conclusion

In order to analyze heat flux and heat release rate values from a direct injection diesel engine, two empirical equations and in-house built equation were implemented to a 1-D engine simulation software in order to compare their accuracy to engine experiments. Purpose of this comparison was to validate the usage of the in-house built equation and its probability of being implemented into a 1-D engine simulation for future usage as a default equation.

1) All indicated equations were focused on a better characterization of the in-cylinder gas flow, as it is one of the most important factors for in-cylinder heat flux calculations in an internal combustion engine. Hohenberg’s equation was one of the first empirical correlations which stressed the importance of the swirl motion. Morel’s equation added more depth into the in-cylinder gas flow and thermal boundary layer effect by dividing the engine cylinder into 6 different sections and having the turbulence term for characterization of the in-cylinder gas flow. However, it was found that Morel’s formula did not represent the gas flow thoroughly. The suggested empirical equation also divided the engine into 6 different sections and added 4 different gas flows; swirl, axial, squish and the effect of the fuel injection.

2) Heat fluxes were analyzed in three different sections inside the engine; piston, head and liner, respectively. Hohenberg equation, the most simply constructed empirical equation of all three, showed the least fit to the experimental data due to its primitive description of the gas flow inside the engine cylinder. Morel’s equation showed averaged best fit at the head section, however, its accuracy at piston and liner sections showed less accuracy when compared to the experimental data, an over-prediction by almost two fold were observed when Morel’s equation was used at the liner section. Newly suggested in-house built equation (1) showed the best overall fit to the experimental results, which was expected since its description of the in-cylinder gas flow has been the most detailed. An improvement of the gas flow model can be achieved by lowering the swirl flow’s effect during the second half of the compression stroke. For the expansion stroke, improvement can be achieved by remodeling the turbulence term, $c_t$, defined in equation (1), which gives the perpendicular velocity term for the gas flow characterization. These improvements on the gas flow representation will increase the efficiency of the heat flux calculations inside the engine cylinder. Average error to the experimental results was found to be 16% for equation (1), 30% for Morel and 32% for Hohenberg, when averaged at each region from Table 4.

3) After the comparison of the results obtained from empirical equations and in-house built equation, even though some errors were found, authors are confident that the suggested equation (1) can be implemented into a 1-D engine simulation software for in-cylinder heat flux calculations for future usage when compared to the currently used models in engine simulation software.

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