Heat Transfer Characteristics of a Diesel Spray Impinging on a Wall*

Jiro Senda**, Hajime Fujimoto**
Masaaki Kobayashi**, Koji Yamamoto***
Yoshiteru Enomoto****

In a high-speed DI diesel engine, fuel sprays impinge surely on a wall of piston cavity. Then, there appears the heat transfer between the impinging sprays and the wall, as a consequence, it has the strong effect on the combustion processes in the engine. The object of this investigation is to obtain the basic information of the phenomenon. In experiments, the fuel was injected once into quiescent atmosphere charged inert gas with high temperature at high pressure. Then, an evaporating single diesel spray was impinging on a flat wall. The temperature distribution on the wall surface in the radial direction was detected by thin film thermocouples of loexconstantan. The heat flux between the impinging spray and the wall surface was calculated by the temperature profile inside the wall due to Fourier's equation using the finite difference method, under the assumption of the 1D heat conduction.

1. Introduction

It is unavoidable that sprays are impinging on the wall of piston cavity in a high-speed DI diesel engine. Fuel droplets in the spray do not evaporate during this process, especially in the case of the short distance between the outlet of injection nozzle and the wall surface of piston cavity. Nevertheless, the atomization is promoted and the turbulence is induced due to the impinging process, consequently, the phenomena have great effects on the combustion processes succeeding the mixture formation. There has been new trials for improvement of combustion by means of the active use of dispersion process of fuel due to the impingement.(1)(2) On the other hand, some components of fuel adhered on the wall surface cause exhaust emissions of soot and unburnt hydrocarbons.

It is significant for designers of combustion chamber to find out the process of heat transfer between the impinging spray and the wall surface. However, there are only a few papers on the measurement of the process. These were carried out in a rapid compression machine(3) and in the combustion chamber in an actual diesel engine.(4)

Authors have been investigating systematically on the phenomenon of the spray impingement on the wall in order to get fundamental mechanism. The first investigation was observation of processes of transformation and dispersion of a minute droplet in order of a several hundred μm impinging with high speed on a wall with high temperature, and measurement of characteristics of heat transfer between the droplet and the wall.(5)(6) The second one was detection of fuel droplets density in a non-evaporating diesel spray injected into quiescent atmosphere with high pressure and impinging on a flat wall with room temperature due to laser light extinction method.(7) The last one was the simul-taneous visualization of liquid and vapor phases at the same section of an evaporating diesel spray injected into quiescent atmosphere with high temperature at high pressure and impinging on a flat wall with high temperature by means of the method of exciplex laser induced fluorescence.(8)

The object of present investigation is elucidation of the process of heat transfer between an evaporating diesel spray and the hot surface of flat wall. Nitridecane as fuel oil was injected into a big-sized constant volume chamber whose charge had quiescent atmosphere with high temperature at high pressure and where a flat wall with relatively high temperature. The formed diesel spray was impinging on the wall at a right angle. The instantaneous temperature distribution was detected by thin film thermocouples(9) set on the surface of hot wall. Then, the heat flux as the representative
data for the characteristics of heat transfer were calculated under some assumptions by the measured temperature distribution. Experimental variables were the ambient temperature, the ambient pressure and the distance between the outlet of nozzle and the wall surface.

2. Experimental Apparatus and Procedure

Fig. 1 is a schematic diagram of experimental apparatus. Dimensions of a constant volume chamber are 140mm in width, 140mm in length and 210mm in height. Two sheets of quartz window are installed in both sides of the chamber. CO gas in a bomb and air in a bomb were charged into the chamber through a Bourdon tube pressure gauge for the shake of control of their pressure. The stoichiometric mixture of both gases was ignited forcibly by a spark plug charged by an ignitor. Then, the mixture was burning within short time, and the atmosphere in the chamber became the state with high pressure at high temperature without production of water. The pressure of the atmosphere decreased gradually due to the heat transfer between the atmosphere and the wall in the chamber. After experiment, burnt gas was exhausted through an exhaust tube, and the atmosphere was scavenged by a vacuum pump. The pressure of the atmosphere was watched by a pressure sensor. The signal of the pressure was amplified by a DC amplifier and was transferred into a control circuit of injection. N-tridecane as used fuel oil was injected only once into the chamber within the short time operation of a magnetic valve, after the pressure in the chamber reached at the appointed value. Fuel oil in a tank was filtrated by a filter and a sediment, thereafter, it was send to an in-line fuel injection pump operated by a motor. The pressurized fuel oil was injected through a sub-fuel injection nozzle, while the experiment did not start. Once the start signal of experiment was given off the magnetic valve was closing and the injection of fuel oil into the chamber through a main fuel injection nozzle and a single diesel spray was impinged at a right angle on a flat wall 21 set in the chamber. The injection pressure was measured by a pressure sensor installed at the injector holder. The dimension of the flat wall was 100mm in width, 100mm in length and 10mm in thickness. The material of the wall was aluminum alloy of loex which is same as that used for the piston of real engine. The temperature of the wall was kept 343K constant by use of a plate typed heater controlled by a voltage regulator. The temperature was corresponding to that at the start of engine. These thin film thermocouples were installed at the surface and the inside of the wall in order to detect the history of temperature. The measuring points were five and were located every 5mm including the center of the wall. The signals of the temperature were transferred into a wave memory and were processed by CPU.

3. Measurement Method of Heat Flux

Fig. 2 is the construction of a thin film thermo-couple used in the experiments. The parent material of thermocouple piece is loex as same that of the wall. The length of the piece is ranged from 3.5 to 4.3mm. The surface of the wall is plated by copper with 7pm in thickness for the sake of high response and high endurance. There is a constantan wire coated by an insulating layer on the center of the piece. The material of the layer has low enough conductivity, as a consequence, there is no problem for the temperature measurement. The constantan wire contacts electrically with thin film copper. The contact point is the hot junction. There is a constantan wire coated with insulating wire also inside the piece, and the cold junction forms between its tip and the inside of piece. Both junctions are touched with the parent material of loex, consequently, the differential thermocouple is made. So, the difference between both temperatures of hot and cold junctions can be measured by the thermocouple. Moreover, the cold junction is made electrical contact with iron wire, and an iron-constantan thermocouple, that is, J type thermocouple of JIS C 1602, is formed. As a consequence, the temperature of the cold junction of the loex-constantan thermocouple can be watched by this J type thermocouple.
thermocouple. The surrounding of the piece is covered by the heat insulator, because the one dimensional calculation of heat transfer between the diesel spray and the wall is made in the vertical direction against the wall. Fig. 3 shows the relation between the thermal electromotive force of the Loex-constantan thermo-couple and the temperature. The thermal electromotive force in the range from 370 K to 570 K which covers the measuring range of the experiment shows the relative large gradient of 0.048 mV/K and the good linearity is given in the range.

The heat flux at the surface of wall can be calculated by use of the temperature difference between hot and cold junctions. It is necessary for this calculation with accuracy to approximate the temperature profile in the piece of the thermocouple, when the phenomenon is unsteady and the wall material has the small thermal diffusivity as the case dealt with in this experiment. It is capable of calculating the temperature profile by means of the finite differential method, that is, FDM, under the assumption of the small distance between both conjunctions. The assumption is satisfied as shown in Fig. 2. The equation of one-dimensional thermal conduction is expressed as follows:

\[ \frac{\partial T}{\partial t} = a \frac{\partial^2 T}{\partial x^2} \]  

(1)

Where \( T \) is the temperature, \( t \) is the time, \( x \) is the distance from the wall surface and \( a \) is the thermal diffusivity of the wall, respectively. \( a \) is defined by \( V \frac{X}{c_p} \) where \( X \) is the thermal conductivity, \( c \) is the specific heat and \( p \) is the density of the wall. The temperature \( T_{x,t+1} \) at the position inside the wall and after the time step \( \Delta t \) is given by the following equation, when eq. (1) is discretized temporally and spatially:

\[ T_{x,t+1} = a \beta T_{x,t-1} + (1 - 2a\beta)T_{x,t} + a\beta T_{x,t+1} \]  

(2)

where \( \beta \) is \( \frac{1}{V} \frac{X}{\Delta x^2} \). Then, eq. (2) can estimate the temperature profile inside the wall by use of the calculative mesh displayed in Fig. 4.

In the case of the implicit differencing, the solution becomes diverging, if the width of mesh does not fulfill the condition of \( 0 < \frac{\Delta x}{\sqrt{a}} < 1/2 \). \( \Delta x \) is 30 \( \mu \)m under this condition, as \( a \) and \( \Delta t \) are given. The initial temperature inside the wall, their profile and the initial surface temperature of the wall must be given in order to solve eq. (2). However, the initial temperature profile can not be measured and it is affected by pre-mixed burning mentioned in section 2. Then, the calculation on the temperature inside the wall is started before the pre-mixed burning and its initial profile at the setting pressure in the chamber is adopted. Owing to this process, the profile shows the linear relation between the surface temperature and the inside temperature of the wall.

The heat flux at the given time \( t \) at the surface of the wall can be estimated by both temperature \( T_{1,1} \) of the wall surface and \( T_{2,1} \) at the small distance from this surface as follows:

\[ q = \lambda(T_{1,1} - T_{2,1}) \Delta x \]  

(3)
The sign of heat flux is set to positive when the heat is transferred from an impinging spray to the wall, namely, the temperature of the spray is larger than that of the wall. Discussion on the heat flux is carried out by the mean value obtained by following three steps: (1) five times measurement of temperature in each experimental condition at five measuring points, (2) computation of each heat flux at each point and (3) calculation of mean heat flux of the five points. Physical properties $p$, $c$ and $X$ of loex as the parent material of the wall are given by function of temperature in the calculation.

**4. Experimental Conditions**

Table 1 summarizes experimental conditions. The injection nozzle used was one hole nozzle and its diameter was 0.2mm. The mean injection pressure was 17.8MPa, the injection quantity was 5.8mg, the injection duration was 1.6msec, and the fuel temperature was 323K, respectively. N-tridecane was used as the representative fuel oil for diesel engines. The wall temperature was kept constant 343K. Experimental variables were the ambient temperature and the impingement distance $Z_w$ between the nozzle exit and the wall surface. $Z_w$ was charged in 20, 24, 30 and 34mm, respectively. There was effect of the momentum flux of spray, as gas-liquid flow near the wall surface. The ambient temperature was given by the equation of state of ideal gas, using the measured back pressure $P_a$. Before pre-mixed burning of CO gas, during and after back pressure drop in the chamber, the ambient density was constant 12.3kg/m$^3$. The effect of the ambient temperature on the evaporation rate of droplets and the temperature of vapor in the spray was recognised, when the ambient temperature was varied.

Fig. 5 is time histories of the ambient pressure $P_a$, the surface temperature $T_w$ of wall and the temperature difference $\Delta T_w$ between the surface and the inside of the wall during and after pre-mixed burning. $R$ in the figure means the radial distance from the spray axis on the wall. The injection timing of fuel oil was set from
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0.3 sec. to 0.9 sec. after ignition of pre-mixed CO gas, as Pa was selected in the range from 3.0N/Pa to 1.5N/Pa as shown in Table I, because the burning ended about 0.2 sec. After ignition. Then, the ambient temperature was ranged from 1000K to 500K as also shown in Table I, corresponding to change in Pa. The pressure drop in Pa was negligible under 3MPa as shown in the enlarged figure in Fig. 5, because the period of the phenomenon dealt with was very short.

5. Experimental Results and Discussion

5.1 Characteristics of Impinging Diesel Spray

Fig. 6 displays examples of photographs of impinging diesel sprays taken by transmitted light under the experimental condition of the ambient temperature Ta equal to 667K, the ambient pressure Pa equal to 2.0N/Pa and the impinging distance Zw equal to 20mm. The diesel spray impinges on the wall at 0.2 msec. after injection start. The mean velocity of spray tip until impingement is about 100m/sec. There is no scattering of droplets in the radial direction after impingement. On the other hand, the scattering can be observed in case of the experiment in the cold atmosphere with high pressure whose ambient density is the same as that under conditions mentioned above. Then, the reason of the tendency in photographs is why fuel droplets are evaporating during their flight in the atmosphere with high temperature.

Fig. 7 shows results of numerical calculation on the impinging diesel spray under same conditions as those in Fig. 6. The calculation was carried out by means of KIVA original code adding the reflection model on droplets proposed by Naber and Reitz. The reflecting velocity of droplet is given by its Weber’s number Wein at impingement. Namely, the perfect reflection occurs, when Wein is smaller than 80, and impinging droplets flow on the wall as a wall jet, when Wein is equal to and larger than 80. The numerical results has pretty well agreement with experimental results.

Fig. 8 is a model of evaporating impinging diesel spray obtained by the numerical and experimental results mentioned above.

5.2 Characteristics of Heat Flux

The effect of the ambient temperature on the heat flux obtained by experimental results and the improved KIVA code is discussed in this section.
Fig. 8 Model of evaporating and impinging diesel spray

Fig. 9 Effect of ambient temperature on heat flux (Z\textsubscript{w} = 20mm)

Fig. 10 Distribution of heat flux in radial direction (Z\textsubscript{w} = 20mm)

Fig. 9 is the history of heat flux as a function of the ambient temperature. The impingement distance Z\textsubscript{w} is 20mm. Case (a) is the results at the impinging point, that is, the radial distance \( r \) equal to 0 mm, and case (b) is those at \( r \) equal to 2.5 mm. And the maximum heat flux is also plotted as an empty circle for each experimental conditions. The rapid increase in heat flux appears just after the impingement of spray on the wall and the heat flux reaches the maximum at about 3msec.

after the impingement. The period of heat transfer is continuing relatively long time, although the duration of injection is only short time of 1.6ms. This is the reason why the heat transfer is succeeding after the end of injection due to the impingement on the wall of the hot gas entrained into the spray and droplets heated by this entrained gas, further, the residence of the droplets or the liquid film formed by these droplets on the wall. The time histories of the heat flux are almost the same for each ambient temperature \( T_a \), and the maximum heat flux \( q_{max} \) is increasing with increase in \( T_a \). However, the rate of change in \( q_{max} \) is relatively small, comparing with the temperature difference between the surface of wall and the ambient. And the change in the heat flux for all conditions of \( T_a \) becomes small when the time is elapsing.

Fig. 10 is the relation between the time history of heat flux and the radial direction \( r \) on the wall, in case of the impingement distance Z\textsubscript{w} equal to 20mm. The ambient temperature \( T_a \) at the ambient pressure \( P_a \) equal to 3.0MPa is 1000K in Case (a) and \( T_a \) at \( P_a \) equal to 1.5MPa is 500K in case (b). The start timing of the
rapid increase in heat flux is delayed, as the measuring points in the radial direction is going farther from the impingement point. Nevertheless, the heat flux shows almost constant in spite of the radial distance in both cases, as the time is passing. The discussion is derived from the experimental results mentioned above. The heat transfer due to the two phase flow of gas and liquid which is composed of the droplets, entrained gas into the spray and fuel vapor is continuing, the heat transfer depends on the ambient temperature and it is different according to radial position, just after the impingement. Therefore, the main cause of the heat transfer is the residual droplets and the residual liquid film flow on the wall with time elapsing, as a result, the heat flux is unrelated to the radial direction farther from the impingement point. From this discussion mentioned above, components of the heat transfer between the impinging diesel spray and the wall are displayed in Fig. 11, referring the author's previous investigation(6).

Generally speaking, the heat transfer between the droplets impinged and the wall is controlled by the mass flux of droplets and the temperature difference between the wall surface and the droplet. Fig. 12 is the relation between the mass flux \( M_{fp} \) of droplets and the time \( t \) from injection start and Fig. 13 shows the difference \( \Delta T_p \) between impinging droplets temperature averaged by their mass and the temperature of wall surface as a function of time \( t \). Both \( M_{fp} \) and \( \Delta T_p \) are calculated by the improved KIVA original code. The results of upper side in both figures are in the case of the ambient temperature \( T_a \) equal to 1000K, those of lower side are in that of \( T_a \) equal to 500K, those of left hand side are in that of the radial distance \( r \) in the range from 0.6mm to 1.3mm. From Fig. 12, the mass flux shows very high value only near the impinging point and the value in case (a) is lower than that in case (b). This is the reason why the decrease in droplet diameter due to high evaporation is promoted by the
higher ambient temperature. The mass flux decreases as the radial distance goes farther from the impinging point and the difference of it in both ambient temperatures becomes smaller. The tendency of $\Delta T_p$ in Fig. 13 shows that the droplets temperature increases as the increase in the ambient temperature and it becomes larger as that in the radial distance. It is distinguished matter that the temperature of droplets is lower than that of wall surface in case of the ambient temperature of 500K.

Fig. 14 is results of the distribution of the vertical velocity component of the entrained gas as function of time from start injection ($Z_e = 20\text{mm}$). The higher the ambient temperature is, the larger the vertical velocity component is. And their distribution is higher near the impinging point. These tendencies are owing to the decrease in the droplet diameter and in its momentum given to the surroundings due to the high ambient temperature. It is appeared that the temperature difference between the entrained gas and the wall surface becomes lower derived from the heat absorption of droplets.

From the calculated results, the following discussion is made: in the case of the ambient temperature $T_a$ equal to 500K, there is little heat flux due to droplets near the impinging point, because the droplet temperature is nearly equal to the temperature of wall surface at this point. However, the heat flux in the experiments in this case is relatively larger than in case of $T_a$ equal to 1000K. As a consequence, almost all the heat flux is brought about by the heat of the entrained gas. On the contrary, in the case that $T_a$ is 1000K, the heat transfer by droplets increases because of the increase in droplet temperature. And the vertical velocity component of droplets decreases due to the increase in $T_a$, although the temperature of entrained gas increases. Then, the heat flux caused by the entrained gas in not balanced with the rate of increase in the temperature difference between the entrained gas and the wall surface. This tendency is explained by the experimental and calculated results.
Fig. 16 displays the relation between the heat flux at the impinging point, that is, at the radial distance \( r \) equal to 0mm, and the time \( t \) from start of injection. The experimental variable is the radial distance \( Z_w \). The heat flux becomes less as \( Z_w \) goes farther from the nozzle outlet. The change in the heat flux becomes smaller and the tendency shows almost the same in spite of \( Z_w \). The reasons are that the main factor of the heat transfer is the heat of entrained gas and the heat flux decreases due to the attenuation of the impinging velocity of entrained gas with increase in \( Z_w \) just after the impingement. When the time from injection start is passing, the heat flux is independent on \( Z_w \) in the case of constant ambient temperature because the residual droplets and the liquid film flow on the wall affect mainly the heat transfer.

The maximum heat flux \( q_{\text{max}} \) at the impinging point shown in Fig. 16 is expressed by the following experimental equation as functions of the difference between the ambient temperature \( T_a \) and the temperature \( T_w \) of wall surface and the impingement distance \( Z_w \):

\[
q_{\text{max}} = 2195 \times (T_a - T_w)^{0.25} Z_w^{-2.52}
\]

\( T_a - T_w : 157 \sim 657\text{K} \)  
\( Z_w : 20 \sim 34\text{mm} \)  

6. Conclusions

The characteristics of heat transfer between the impinging and evaporating diesel spray and the wall with hot surface was investigated by experiments and calculations with improved KIVA original code. And the following conclusions are drawn:

(1) The rapid increase in the heat flux appears just after the impingement of the spray on the wall. The heat flux reaches the maximum at the time ranged from 3msec to 4msec after the injection.

(2) The period of heat transfer is longer than the injection duration due to the entrained gas with high temperature, the adhered droplets and the liquid film flow on the wall.

(3) The maximum heat flux at the impinging point is increasing with increase in the ambient temperature because of the increase in temperature of both impinging droplet and entrained gas.

(4) At the impinging point, the maximum heat flux diminishes with increase in the impingement distance due to the damping of the velocity of the entrained gas.

Discussion and Authors’ Reply

[Question] K. Hirata: Shibaura Institute of Technology

The phenomena dealt with in the paper are too complex. Are they the heat transfer between the spray and the wall with hot surface, or are they that between the spray and the entrained gas? Please discuss the phenomena, separating or simplifying these two types of heat transfer to some extent.

[Authors’ reply]

The evaporating diesel spray experimented in this investigation is the two phase flow of liquid and gas, and it entrains the surrounding gas with high temperature. Then, the phenomena are too complex as pointed out. The heat flux measured is caused by droplets and their vapor in the spray, the entrained gas, the adhered droplets and the liquid film flow on the wall. The arrangement of the heat transfer by use of, for examples, Nusselt No. for each phase with separation of factors mentioned above can not be carried out now. However, it is very difficult to measure the heat transfer for each phase separating these factors. Then, calculation by the improved KIVA original code was carried out in order to obtain the mass flux of impinging droplets, the temperature difference between the droplets and the wall surface and the velocity and the temperature of entrained gas, then, the qualitative discussion was made on the heat transfer process in the phenomena.

References