Effects of LLC Flow Characteristics and Surface Roughnesses on Nucleate Boiling Heat Transfer Mechanisms in IC Engine Cooling System

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Received on September, 14, 2021

ABSTRACT: To improve internal combustion engine cooling systems, it is required to utilize the nucleate boiling heat transfer. Previous studies revealed it is affected by surface roughness, flow velocity, and degree of subcooling. This study investigated the heat transfer mechanisms (latent heat transport mechanism, sensible heat transport mechanism, and bubble agitation mechanism) when changing the above parameters. It was found that the increase in flow velocity and degree of subcooling increased the heat transfer due to the latent and sensible heat transport, while the increase in surface roughness increased the heat transfer due to the sensible heat transport and bubble agitation.

KEY WORDS: heat fluid, refrigerant, engine cooling, thermal management, forced flow nucleate boiling, heat transfer, surface roughness

1. Introduction

Recently, due to the increased risk of global warming, regulations on CO2 emissions of the internal combustion engines (ICE) for automobiles have become stricter, and heat management technology that meets the demands for improved thermal efficiency in ICE cooling systems for automobiles is required. To achieve both heat loss reduction in heat exchange and sufficient cooling of certain high-temperature parts, the nucleate boiling phenomenon with higher heat flux at higher coolant temperature can be utilized. It is necessary to identify the factors that affect nucleate boiling, elucidate the heat transfer mechanism, and develop a highly accurate prediction model of nucleate boiling based on these results.

In the previous studies, Chen1) proposed that the forced flow nucleate boiling heat transfer coefficient is the combination of the forced convection heat transfer coefficient and pool nucleate boiling heat transfer coefficient and suggested a saturated forced flow boiling heat transfer model for water boiling. Steiner et al.3) suggested a heat transfer model at superheated boiling to apply to the ICE cooling system by reconstructing inhibition factors related to nucleate boiling using the bubble dynamics based on Chen's model. They proposed that the water's bubble would show a sliding phenomenon before departure from the heating surface under the forced flow conditions and it would inhibit the heat transfer. The reason for heat transfer decrease can be considered that the heat of bubbles agitation and latent heat decreased because of the sliding phenomenon. Jones et al.3) and Jo et al.4) suggested that the surface roughness and wettability of the heating surface affect pool nucleate boiling heat transfer by using water and FC-77 as the coolant. They observed that significantly more active nucleation sites exist on the surface with larger surface roughness and the heat fluxes were elevated by the surface roughness increase. It can be inferred that the sum of the heat of sensible heat transport, latent heat transport, and bubble agitation was increased by the higher nucleation sites. Chen's model and Steiner et al.'s model have been proven to accurately predict the nucleate boiling heat transfer coefficient under simple coolant operating conditions, however, the effect of the surface properties of the coolant flow path on nucleate boiling heat transfer has not been discussed. Thus, it is necessary to construct the new forced flow nucleate boiling heat transfer model while accounting for the surface properties. According to the above reason, the effect of the surface properties on the nucleate boiling heat transfer needs to be verified by measuring the forced flow nucleate boiling heat flux.

In our previous studies3)(6), the effect of various operating conditions (flow velocity, pressure and degree of subcooling) and heating surface corrosion on heat flux were examined using an experimental apparatus that can simulate the flow of coolant in an actual automotive ICE using water and 50% ethylene glycol aqueous solutions (EG50%) as coolant, and the heat transport of these cases was expounded by referring Steiner et al.'s study2). In order to examine the effect of surface roughness changes on heat flux and heat transfer, we prevented the occurrence of heat surface corrosion by using LLC which contains anti-rust components in it. The heating surface with surface roughness $Sa = 1.5 \, \mu m$, 15$\mu m$, and 30$\mu m$ were manufactured by electric discharge machining (EDM), and surface roughness changed by replacing each heating block, the heat flux and heat transfer coefficient of the heating surface was measured. In addition, measurements in flow velocity change and degree of subcooling change were also performed in the same way.
for each heating block with various surface roughness. The process of bubble growth and departure during boiling was observed using two high-speed cameras installed vertically and horizontally. The heat transfer mechanisms (latent heat transport, sensible heat transport, and bubble agitation) of the nucleate boiling were discussed by measuring the heat fluxes and the bubble generation process when changing in flow velocity, degree of subcooling, and surface roughness.

2. Experimental Setup and Method

2.1. Experimental apparatus

The experimental apparatus is the same as that used in our previous studies(5),(6). The experimental coolant channel schematic diagram is shown in Fig. 1. The schematic view of the observation section and the heating section of the experimental device is shown in Fig. 2. The observation section is designed as a 30 mm × 30 mm rectangular channel and is placed horizontally. In the observation section, there are two glass windows (on top and backside) and one polycarbonate window (on the front side) of size 20 mm × 60 mm. In order to illuminate the heating surface, an LED light source is installed in front of the polycarbonate window. The heating block made of aluminum alloy (A2017) shown in Fig. 2 (a) is installed on the bottom. In the observation flow path, the heat was applied to the heating block using a 6 × 555 W output ceramic heater, the upper surface of which was used as the heating surface, and the coolant in the pipeline was heated to generate boiling. The surface roughness of the heating surface can be changed by using different heating blocks. The bubble growth process on the heating surface was observed with a high-speed camera (MEMRECAM ACS-1, nac Image Technology Inc.) set horizontally, and the departure phenomenon was observed with a high-speed camera (FASTCAM SA-Z, PHOTRON LIMITED.) set vertically. As shown in Fig. 2 (b), to measure the temperature and heat flux in the upstream, midstream, and downstream of the heating surface, nine thermocouples were inserted into the heating block through the observation section and Teflon block. Measurement error by temperature measurement device is 2.3 K. Since the temperature distribution in the heating block was steady-state, the heat flux was assumed to be linear and was calculated by Fourier’s law.

2.2. Experimental condition

In this paper, the experiments simulating the engine operating conditions and the surface properties of the coolant flow path were carried out under the conditions as shown in Table 1. For the 30 mm × 30 mm rectangular channel, LLC was used as the working fluid and heating surface with the surface roughness of $S_{\text{a}} = 1.5 \, \mu\text{m}$, $15 \, \mu\text{m}$, and $30 \, \mu\text{m}$ made of aluminum alloy were used. The operating conditions of the coolant were set to 0.16 MPa for pressure, 0.57 m/s (Reynolds number $(Re) = 2.3 \times 10^4$) and 1.01 m/s $(Re = 4.1 \times 10^4)$ for flow velocity, 11 K and 31 K for the degree of subcooling. The boiling point under the LLC’s pressure 0.16 MPa is 394 K. The degree of superheating, which is the difference between the heating surface temperature and the boiling point, was set at -15 to 25 K. Then, the surface heat flux was measured and the bubble generation was observed.

| Table 1 Experimental condition |
|--------------------------------|
| Channel width [mm] | 30 |
| Channel height [mm] | 30 |
| Coolant component | LLC |
| Surface roughness, $S_{\text{a}}$ [μm] | 1.5, 15, 30 |
| Heating surface material | Aluminum alloy (A2017) |
| Observation section pressure [MPa] | 0.16 |
| Saturation temperature [K] | 394 |
| Observation section flow velocity [m/s] | 0.57, 1.01 |
| Reynolds number $(Re)$ | $2.3 \times 10^4$, $4.1 \times 10^4$ |
| Degree of subcooling [K] | 11, 31 |
| Degree of superheating [K] | -15 to 25 |
2.3. Heating surface roughness

2.3.1. Heating surface roughness manufacture

Aluminum alloy (A2017) was used as the material for the heating surface. To investigate the effect of the surface roughness on nucleate boiling heat transfer, three kinds of heating surfaces roughness were processed by EDM. The surface roughness of the heating surface was measured in the range of 3 mm × 3 mm area using the 3D measurement laser microscope (LEXT OLS4000, OLYMPUS Corp.). The surface micro-geometry of the heating surface at different surface roughness are shown in Fig. 3 (I) to (III). The upper left figure shows the scanned surface profile of the heating surface in the measurement area, the upper right figure shows the height distribution in the upper left figure, and the lower figure shows the roughness curve in the red line in the upper left figure. Further, a Fast Fourier Transformation (FFT) using a roughness curve is carried out, and the result showing the relationship between the spatial frequency and the power spectrum density (PSD) of the roughness curve is shown in Fig. 4. From the result of the height distribution shown in Fig. 3, it can be seen that the height distribution of each surface is even. From the results of Fig. 4, it was shown that the surface shape processed by the EDM was not periodic since some particular frequency components of all the three surface roughness curves were not observed. Furthermore, the measured surface roughness results of the heating surface used in the present study are shown in Table 2.

Arithmetic means roughness $S_a$ is expressed as the average absolute value of the difference in height from the mean surface of each measurement point in the measurement range. Similarly, the maximum height $S_z$ represents the sum of the maximum peak height $S_p$ and the maximum valley depth $S_v$ in the measurement range. $S_{sk}$ means the skewness of roughness in the measurement range, and the value of zero represents the symmetry between the peaks and valleys around the mean line. $R_{lo}$ represents the ratio of the expansion length of the roughness curve to the reference length, and the material ratio $R_{mr}$ represents the proportion of the existence of the real part at a certain height. Schematic diagrams of $R_{lo}$ and $R_{mr}$ are shown in Fig. 5 and they can be calculated using equations (1) and (2),

$$R_{lo} [%] = \frac{L - L_{re}}{L_{re}} \times 100 \quad (1)$$
$$R_{mr} [%] = \frac{L_{rp}}{L_{re}} \times 100 \quad (2)$$

where $L$: expansion length [$\mu m$], $L_{re}$: reference length [$\mu m$] and $L_{rp}$: the length of the real part [$\mu m$]. In this research, the expansion length $L$ was calculated by using equations (3) and (4).

$$l = \sqrt{\Delta x^2 + \Delta y^2} \quad (3)$$
$$L = \sum_{i=1}^{n} l_i = \sum_{i=1}^{n} \sqrt{(\Delta x_i)^2 + (\Delta y_i)^2} \quad (4)$$

where $l$: line segment length between any two measurement points [$\mu m$], $\Delta x$: the change in measured reference length $x = 0.64$ [$\mu m$], $\Delta y$: the change in measured height $y$ [$\mu m$], $n$: the total number of line segments [-], and the cut level for calculating the length of the real part $L_{rp}$ was set at the height of 25% of $R_{z}^{1/2}$. According to these
results in Table 2, as $S_{th}$ increase from 1.5 $\mu$m to 30 $\mu$m, $S_{h}$ and $R_{lo}$ were found tending to increase, which means that the surface with larger $S_{th}$ has a higher height difference and more complex surface profile. All the roughness skewnesses $S_{sk}$ was close to 0, which means that the non-uniformities of the peaks and valleys on these surfaces are nearly symmetrical. This fact is also obvious from the height distributions shown in the upper right of Fig. 3.

![Fig. 5 Schematic of surface roughness curve and definition of parameters for calculation of $R_{mr}$ and $S_{h}$](image)

2.3.2. Heating surface roughness and wettability

It has been shown that wettability (contact angle and direction of surface tension) changes with changes in surface roughness in general. Kubiak et al. proposed a correlation between surface roughness and wettability for several samples with different surface roughness and materials using equation (5)

$$\theta_p = \cos^{-1}[r \times L_S \times (\cos \theta_{th} + 1) - 1] \quad (5)$$

$$r = 1 + \frac{R_{lo}}{100} \quad (6)$$

$$L_S = \frac{R_{mr}}{100} \quad (7)$$

where $\theta_p$: estimated contact angle [deg.], $\theta_{th}$: theoretical contact angle [deg.], $r$: ratio of actual surface area to projected area to the horizontal plane on the surface of an individual in contact with liquid [-] and $L_S$: the ratio of the area in which the liquid is in contact with the individual [-]. $\theta_{th}$ represents the contact angle at the ideal surface when the surface roughness is assumed to be 0. The theoretical contact angle measured using LLC droplets on the aluminum alloy surface used in the present study was 60°. By substituting the surface roughness parameters in Table 2 into equation (5), the estimated contact angles for surfaces with $S_{lo} = 1.5\mu$m, 15\mu m, and 30\mu m were calculated to be 69°, 62°, and 52°, respectively. The surface tension; $\sigma$ (0.048 N/m) and buoyancy forces; $F_{bc}$ acting on the boiling LLC bubbles are shown in Fig. 6, and it can be seen that the smaller the contact angle; $\theta$, the lower the magnitude of the surface tension acting in the vertical direction; $\sigma \sin \theta_p$ with $S_{lo} = 1.5\mu$m, 15\mu m, and 30\mu m were decreased to be 0.045 N/m, 0.042 N/m, and 0.038 N/m, respectively.

![Fig. 6 Schematic of surface tension and buoyancy of bubbles](image)

3. Results and Discussion

3.1. Observation of the bubble growth process and departure phenomenon

In our previous studies, differences in bubble generation and departure phenomena were discussed by observing the boiling of water and EG50% on the heating surface with corrosion. In the case of water, it was observed that the bubbles lifted off from the heating surface immediately as soon as bubbles are generated. However, in the case of EG50%, after the bubbles are generated, the bubbles appeared to slide along the heating surface instead of departing from the surface with a lower departure frequency than water. In the present study, the photographs of the LLC’s boiling phenomenon taken with a high-speed camera are shown in Fig. 7. According to Fig. 7 (I), it was observed that after bubbles are generated, the bubbles stopped growing and stayed on the spot without any sliding. Furthermore, from the photographs shown in Fig. 7 (II), it was also observed that the bubbles departed from the heating surface through the combination of the neighboring bubbles and this departure phenomenon appeared under all the conditions in the present study.

Based on the above phenomena, the heat transfer mechanism governing the bubble growth process and bubble departure in the LLC nucleate boiling phenomenon is shown in Fig. 8. According to Fig. 8 (a), when the bubble does not depart the heating surface, the heat from the heating surface to the bubble always acts as latent heat of vaporization, and the effect of latent heat is dominant. On the other hand, as shown in Fig. 8 (b), when the bubble is larger than the superheated liquid layer, sensible heat transport suppresses the bubble growth. According to Figs. 8 (c) and (d), when the bubble departures the heating surface, the effects of the sensible heat transport mechanism and the bubble agitation mechanism become dominant. Therefore, the magnitude of the effect of the bubble agitation mechanism is directly related to the number of departures. The magnitude of the effect of the sensible heat transport mechanism is considered to depend not only on the number of departures of bubbles, but also on the thickness of the superheated liquid layer near the heating surface at the time of departure (which varies with the mainstream temperature and flow velocity of the working fluid). The heat transfer mechanism of LLC is qualitatively discussed in terms of bubble behavior and heat transfer coefficient.

The number of departures per unit time and unit area (referred to as NOSD) of bubbles and the bubble departure diameter were measured on the heating surface with the degree of superheating $\Delta T_{s0} = 9$ K under the conditions shown in Table 1. The bubble departure diameter was calculated by randomly selecting 10 bubbles in the same area on the heating surface and averaging their diameters at the moment they depart the heating surface. The measurement results for the flow velocity change (0.57 m/s and 1.01 m/s) showed that the NOSD of bubbles was 340 mm²-s⁻¹ and 61 mm²-s⁻¹, respectively, and the departure diameters were 0.044 mm and 0.034 mm, respectively. As the flow velocity increased, the NOSD of bubbles decreased and the departure diameter of bubbles became smaller. Then, the measurement results of the degree of subcooling change (11 K and 31 K) showed that the NOSD of bubbles was 595 mm²-s⁻¹ and 340 mm²-s⁻¹, respectively.
and the departure diameter was 0.086 mm and 0.044 mm, respectively. As the degree of subcooling increased, the NOSD of bubbles decreased and the bubble departure diameter became smaller. The measurement results for the surface roughness change (1.5 μm, 15 μm and 30 μm) showed that the NOSD of bubbles was 340 mm²/s⁻¹, 1881 mm²/s⁻¹, and 1287 mm²/s⁻¹, respectively, and the bubble departure diameter was 0.044 mm, 0.023 mm, and 0.034 mm, respectively. The NOSD of bubbles was more in the order of 1.5 μm, 30 μm and 15 μm, which is opposite to the NOSD. The above results are summarized in Table 3 in Sect. 3.3.

3.2. Surface heat transfer coefficient measurement

The results of the heat flux and heat transfer coefficients of the heating surface with different flow velocities, degree of subcooling, and surface roughness were shown in Figs. 9, 10 and 11, respectively. In these Figures the degree of superheating ΔTsat < 0 represents the forced convection flow region that is not boiling and it is marked using the solid line with no filled circle. When the degree of superheating ΔTsat > 0 represents the boiling region and is marked using the solid line with a filled circle. In this paper, the heat transfer coefficient in the transition phase is not shown.

In Figs. 9 (a) and (b), which show the heat flux and heat transfer coefficient results at flow velocities of 0.57 m/s and 1.01 m/s, it can be observed that these values in the forced convection and boiling regions increased with an increase in flow velocity. After quantitative evaluation, the forced convection heat transfer coefficient at the flow velocity of 0.56 m/s and 1.01 m/s, the average heat transfer coefficients at 1.01 m/s increased by 2.8 kW/(m²-K) than that of 0.56 m/s. In the nucleate boiling region, the average nucleate boiling heat transfer coefficient at the 1.01 m/s increased by 6.6 kW/(m²-K) than that of 0.56 m/s. From the results, the effect of the flow velocity on heat transfer becomes larger in the nucleate boiling region than in the forced convection region.

In Fig. 10 (a), showing the heat flux results, it also can be observed that the heat flux in the forced convection region and the boiling region increases with the increase in the degree of subcooling from 11 K to 31 K, which is the same as in the case of flow velocity. After quantitative evaluation of Fig. 10 (b), the forced convection heat transfer coefficients at the degree of subcooling 11 K and 31 K are almost no changed. In the nucleate boiling region, the average nucleate boiling heat transfer coefficient at 31K increased by 7.4 kW/(m²-K) than that of 11 K. From the results, the effect of degree of subcooling on heat transfer only in the nucleate boiling region.

In Figs. 11 (a) and (b), which show the heat flux and heat transfer coefficients at surface roughness Sa = 1.5 μm, 15 μm, and 30 μm, It can be seen that these values increase in the boiling region while the forced convection region remains unchanged as the surface roughness increases. After quantitative evaluation, the forced convection heat transfer coefficients at surface roughness Sa = 1.5 μm and Sa = 15 μm, the average heat transfer coefficient of Sa = 15 μm increased by 0.20 kW/(m²-K) than that of Sa = 1.5 μm. In the nucleate boiling region, the average nucleate boiling heat transfer coefficient at surface roughness Sa = 15 μm increased by 30.1 kW/(m²-K) than that of Sa = 1.5 μm. The forced convection heat transfer coefficients at the surface roughness Sa = 1.5 μm and Sa = 30 μm increased by 0.88 kW/(m²-K) with the average heat transfer coefficient of Sa = 30 μm than that of Sa = 1.5 μm. In the nucleate boiling region, the average nucleate boiling heat transfer coefficient at surface roughness Sa = 15 μm increased by 30.1 kW/(m²-K) than that of Sa = 1.5 μm. The forced convection heat transfer coefficients at the surface roughness Sa = 1.5 μm and Sa = 30 μm increased by 0.88 kW/(m²-K) with the average heat transfer coefficient of Sa = 30 μm than that of Sa = 1.5 μm. In the nucleate boiling region, the average nucleate boiling heat transfer coefficient at the surface roughness Sa = 30 μm increased by 34.0 kW/(m²-K) than that for Sa = 1.5 μm. This result indicates that the effect of surface roughness on heat transfer is only in the nucleate boiling region. However, the effect of surface roughness on the improvement of heat transfer coefficient was estimated to be limited, because the increase in heat transfer coefficient from Sa = 15 μm to 30 μm of surface roughness was not changed so much. These results are shown in Table 3 in Sect. 3.3.
Fig. 9 Comparison of measured (a) heat flux and (b) heat transfer coefficient when changing flow velocity.

Fig. 10 Comparison of measured (a) heat flux and (b) heat transfer coefficient when changing degree of subcooling.

3.3. Discussion of bubble observation and heat transfer measurement

From Table 3, it was confirmed that the NOSD of bubble decreased and the bubble departure diameter became smaller as the flow velocity increased. As a result, it was confirmed that the nucleate boiling heat transfer coefficient increased. It is inferred that the effect of the bubble agitation mechanism shown in Fig. 8 (d) decreased due to the decrease in the NOSD of bubble. On the other hand, the longer time spent on the heating surface leads to the effect of the latent heat transport, and also the sensible heat transport mechanism is shown in Fig. 8 (b) inferred to increase even in the bubble departure diameter decrease. Therefore, in the increase of the flow velocity, the increase of the effect of the latent heat transport and sensible heat transport mechanism resulted in the increase of the heat transfer coefficient.

From Table 3, it was confirmed that the NOSD of bubble decreased and the bubble departure diameter became smaller as the degree of subcooling increased. As a result, it was confirmed that the nucleate boiling heat transfer coefficient increased. It is inferred that the effect of the bubble agitation mechanism shown in Fig. 8 (d) decreased due to the decrease in the NOSD of bubble. On the other hand, the longer time spent on the heating surface leads to the effect of the latent heat transport, and also the sensible heat transport mechanism shown in Fig. 8 (b) inferred to increase even in the bubble departure diameter decrease. Therefore, in the increase of the degree of subcooling, the increase of the effect of the latent heat transport and sensible heat transport mechanism resulted in the increase of the heat transfer coefficient.
As shown in Sect. 2.3.2, the surface roughnesses become large and the contact angle ($\theta$) becomes small, the magnitude of the surface tension acting in the vertical direction ($\sigma \sin \theta$) becomes small. The decrease in the magnitude of the surface tension acting in the vertical direction decrease the buoyancy force required for detachment. Therefore, the bubble departure diameter also changes. By comparing the surface roughness of $S_a=1.5 \mu m$, $15 \mu m$ and $30 \mu m$, it can be observed that the bubble departure diameter decreased and increased the NOSD of bubble. Therefore, Boiling phenomenon shown in Figs. 8 (c) and (d) become more frequent, and Boiling phenomenon shown in Fig. 8 (b) become less frequent. From the above, it was considered that the nucleate boiling heat transfer coefficient increases with increasing surface roughness because the influence of the bubble disturbance and sensible heat transport mechanisms increases while the influence of the latent heat mechanism decreases, because of the influence of bubble disturbance and sensible heat transport mechanism becomes larger, and the influence of latent heat mechanism becomes smaller.

4. Conclusions

Using LLC as a coolant simulating an actual automotive ICE, the present study measured the heat flux on the heating surface at various flow velocities, degree of subcooling and surface roughness, and observed their effects on the heat transfer coefficients. From these results, the effects on the nucleate boiling heat transfer were studied by observing the bubble growth process and departure phenomena. The important conclusions obtained from the present study are summarized below:

(1) In the LLC boiling phenomenon, as soon as bubbles are generated, they do not depart but stay on the heating surface and stop growing. After that, it departed by merging with neighboring bubbles.

(2) The heat transfer coefficient increases with increasing flow velocity in the forced convection and boiling regions. The increase in the heat transfer coefficient in the boiling region is attributed to the increase in the amount of heat due to the latent or sensible heat transport mechanism as the flow velocity increases.

(3) The nucleate boiling heat transfer coefficient with decreasing degree of subcooling was found to remain unchanged in the forced convection, while it decreased in the boiling region.

The increase in the heat transfer coefficient in the boiling region is attributed to the increase in the amount of heat due to the latent or sensible heat transport mechanism as the degree of subcooling increases.

(4) The heat transfer of nucleate boiling did not change with increasing surface roughness in the forced convection region, but it increased in the boiling region. However, the heat transfer coefficients for surface roughness of $S_a=15 \mu m$ and $30 \mu m$ were almost the same. This confirms that for surface roughness between $S_a=1.5 \mu m$ and $15 \mu m$, the heat transfer increases with increasing surface roughness due to the mechanism of bubble turbulence and sensible heat transport.

This paper is written based on a proceeding presented at 2021 JSAE Congress (Autumn) held at online from 13 October to 15 October 2021.

Acknowledgments

This study was the result of a collaborative research program with the Research Association of Automotive Internal Combustion Engines (AICE) for the fiscal year 2019 - 2021. The authors gratefully acknowledge the concerned personnel.

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