Heat-transfer performance of a cooling tube with an obstacle for protection under frosting conditions

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Abstract
Fundamental experiments were conducted on a cooling tube with an obstacle using liquid nitrogen as the coolant under a frost-formation condition. The background of this study is the development of a heat exchanger for an air-breathing engine, which cools the air using a cryogenic fuel. The obstacle is located in front of the cooling tube. The main purpose of the obstacle is to protect the cooling tube against damages due to foreign objects. The experimental results obtained using an obstacle with a V-shaped cross section show that the obstacle helps in reducing the pressure loss and improving the heat-transfer performance if the distance between the obstacle and the cooling tube is optimally selected. The experiments conducted using obstacles with several types of cross sections show that one of these obstacles with an arrowhead cross section is effective in suppressing the pressure loss, mitigating an adverse effect on the heat-transfer performance, and protecting the cooling tube against damages due to foreign objects.

Keywords: Heat exchanger, Hypersonic aircraft, Turbojet, Cryogenic, Frost

1. Introduction

1.1 Background
A group led by the Japan Aerospace Exploration Agency conducted a developmental study on an air-breathing engine for hypersonic flights (Taguchi et al., 2012a, Sato et al., 2010). Before developing a full-size engine, a small size engine with an air mass flow rate of 1 kg/s has been designed. The design Mach number of this engine is five. Figure 1 shows the outline of this engine. Table 1 lists its specifications. This engine was designed to conduct supersonic flight tests. The flight test was conducted in October 2010 (Taguchi et al., 2012b) using a high altitude balloon at Mach 2. This engine comprises a precooler, which is a heat exchanger used to cool the inlet air using liquid hydrogen fuel. The thrust of the engine decreases in hypersonic flights because of the aerodynamic heating. The air is cooled at the precooler, thus suppressing the decrease in the performance of the engine. Some studies have been conducted on precoolers and are reviewed elsewhere (Wang et al., 2014). Figure 2 shows the image of the precooler installed in the engine. The cooling tubes of the precooler are made of SUS316L. This material is durable for cryogenic and high-pressure liquid hydrogen conditions.

The following two technical problems must be solved to employ precoolers in practice: frost formation and foreign object damage (FOD). With regard to the frost formation problem, a precooler serves as a heat exchanger using cryogenic coolant such as liquid hydrogen. Although the inlet temperature of the engine increases above 1000 °C in hypersonic flights, the temperature on the ground is normal. The frost formation problem occurs on the ground at takeoff. If an aircraft reaches high altitude, the density of the vapor becomes very low. In such a condition, frost does not form on the cooling tubes (Kimura and Sato, 2003). Therefore, if the precooler is not employed until an aircraft reaches the altitude, we can avoid the frost formation. However, the precooler helps in increasing the thrust of the engine, which is another advantage. If the precooler can be used in the takeoff phase, the advantage is considerable for aircrafts. In our previous firing test on the ground, we revealed that the frost formation on the cooling tubes of the precooler is a serious problem affecting the engine performance (Fukiba et al., 2008). We proposed some methods to
solve the frost formation problem. One of the proposed solution involves spraying methanol at the upstream of the precooler (Fukiba et al., 2008; Harada et al., 2001; Sato et al., 2004). However, this method requires methanol and a spraying system including a tank and a nozzle. These additional devices increase the weight of the engine, which is a disadvantage. Finally, we decided not to introduce this system in our engine. Another proposed solution involves defrosting the precooler by impinging on it with high-pressure air leaving the compressor (Fukiba et al., 2009, Sonobe et al., 2015). However, this system was found to be insufficient under some airstream conditions or at certain surface temperatures of the cooling tubes. Accordingly, the precooler installed in our engine was designed to have a large tube pitch to suppress the engine from serious performance degradation. This design led to increase the dimensions of the precooler, which was a disadvantage.

The FOD problem is also serious. Liquid hydrogen is a highly combustible fuel flowing inside the cooling tubes of the precooler. Therefore, foreign objects can cause fatal damage to the engine and aircraft if the tubes break. In February 2014, firing tests were conducted in a supersonic wind tunnel (Kobayashi et al., 2015). In the firing test in an airstream at Mach 4, the cooling tubes of the precooler were damaged by foreign objects. These objects were broken pieces of ceramic, which came from ceramic bricks located upstream of the test section to heat the airstream. Fortunately, liquid nitrogen was used in this test as the coolant, and no explosion occurred. However, FODs such as due to bird strike are unavoidable to engines in practical use. Some countermeasures are necessary to solve this problem.

1.2 Purpose of this work

In this study, a countermeasure is proposed using obstacles placed in front of the cooling tubes of the precooler. The effects of an obstacle on the pressure loss and heat-exchange performance were investigated in this fundamental study. The obstacle was located upstream of a cooling tube. A characteristic of this study is the temperature of the coolant. We used liquid nitrogen in this study; hence, the temperature was cryogenic. It is known that frost on a cryogenic surface has lower density and faster growth speeds than that on higher-temperature surfaces (Ohkubo and Tajima, 1995). On the other hand, previous studies reported that the mass flux on a cryogenic surface is much lower than that on a higher-temperature surface (Barron and Han, 1965; Fukiba et al., 2006). Generally, the heat transfer rate

![Fig. 1 Schematic of small engine for flight test.](image1)

![Fig. 2 Precooler of the engine.](image2)

<table>
<thead>
<tr>
<th>Table 1 Specification of the engine and the precooler.</th>
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<tbody>
<tr>
<td><strong>Engine</strong></td>
</tr>
<tr>
<td>Length</td>
</tr>
<tr>
<td>Cross-sectional area</td>
</tr>
<tr>
<td>Mass</td>
</tr>
<tr>
<td>Thrust (M = 0)</td>
</tr>
<tr>
<td><strong>Precooler</strong></td>
</tr>
<tr>
<td>Overall size</td>
</tr>
<tr>
<td>Number of tubes</td>
</tr>
<tr>
<td>Tube diameter</td>
</tr>
<tr>
<td>Material</td>
</tr>
</tbody>
</table>
decreases if an obstacle is located upstream of a cooling tube. However, this study found that the heat transfer rate increases under certain conditions of frost formation and cryogenic surface. In addition, the pressure loss decreases if the cross-sectional shape and distance between the obstacle and the cooling tube are selected appropriately. Although this study was based on the demand for the development of precoolers for hypersonic aircrafts, the conclusions of this study are useful for all types of cryogenic heat exchangers such as vaporizers of liquid natural gas.

Nomenclature

c specific heat
$c_p$ pressure-loss coefficient
$Q$ heat transfer rate
$q_m$ mass flow rate
$T_{in}$ inlet temperature
$T_{out}$ outlet temperature
$u$ flow velocity
$\Delta p$ pressure loss
$\rho$ density

2. Experimental Setup and Measurements

2.1. Experimental Setup

Figure 3 shows the overall image of the experimental setup. The airflow at a temperature of 20 ± 1 °C and a humidity of 50 ± 3 % is maintained using an air conditioner. The airflow passes the settling chamber and the orifice flow meter. Finally, the airflow reaches the test section. Figure 4 shows the dimensions of the test section. The cross-sectional area of the test section is 40 × 60 mm$^2$. The cooling tube with a diameter of 20 mm is located at a distance of 100 mm downstream of the inlet of the test section. The cooling tube is made of copper. An obstacle made of a resin is placed in front of the cooling tube. The coolant of the cooling tube is liquid nitrogen, which has a boiling point of −196 °C. Because the coolant is not provided for the obstacles, only the cooling tube is cooled. Therefore, frost is formed only on the cooling tube. The pressure measurement orifices are located upstream and downstream of the cooling tube. The pressure difference between these orifices is used to calculate the pressure-loss coefficient.

Table 2 lists the experimental conditions. The flow velocity is obtained by dividing the volumetric flow rate by the cross-sectional area. The flow velocity was decided as follows. The mass flow rate of the precooler for the developed engine is 1.0 kg/s. The flow velocity at the ground is 13 m/s, if we assume that the air at this flow rate passes the inlet cross-sectional area of the precooler at a constant velocity. Note that this flow velocity is much lower than that at an inlet of conventional jet engines. This is because the inlet velocity at the precooler was designed to decelerate to suppress the pressure loss (Fukiba et al., 2009). The diameter of the cooling tubes of the precooler is 2.0 mm. The Reynolds number based on the diameter is 1700. In this study, the flow speed was set at 1.5 m/s. The Reynolds number is 1900, which is similar extent of the flow in the precooler. The test duration was set at 600 s. In 600 s, a hypersonic aircraft can take off and reach the required high altitude where the absolute humidity is sufficiently low to avoid frost formation (Kimura and Sato, 2003).

![Fig. 3 Overview of the test apparatus.](image-url)
Figure 5 shows the cross-sectional shapes of the obstacles used in this study. Five cross-sectional shapes were tested. Among the five shapes, three were conventional shapes, which are V shape (V), circular cylinder (C), and square cylinder (S). The cross sections of the remaining obstacles (A1 and A2) were in the form of arrowheads, which were designated considering the results of the first three conventional shapes. Table 3 lists the symbols and distances from the stagnation point of the cooling tube. The experimental results using obstacles V and C with respect to the distance are reported in sections 3.2 and 3.3, respectively. Through these results, the effect of the distance on the heat-transfer performance is evaluated. Next, the effect of the cross-sectional shape is discussed in section 3.4. The heat-transfer performance of the obstacles S, A1 and A2 are presented in this section. The distance from the obstacles to the stagnation point is a constant at 5.0 mm for the obstacle S and 7.9 mm for A1 and A2.

As mentioned above, a single cooling tube with an obstacle was located between the walls in this fundamental study. This configuration is different from the precooler, which comprises many cooling tubes. However, previous studies reported that the flow field of this configuration is similar to that of a row of tubes. Okamoto et al. (1994) reported that the pressure distribution on the surface of a single cylinder between the walls is largely the same as that in a row of tubes if the ratio of the distance between the walls to the diameter of the single cylinder is equal to the ratio of the pitch of the tubes to the diameter of the tubes. Considering these facts, in this study, the experiment was conducted using a single cooling tube and walls. It should be noted that this study simulated only a row of tubes located along the line perpendicular to the flow. The effect of the obstacles on multiple-row tube banks is not included in this study.
2.2. Measurements

The frost thicknesses at the front stagnation point, side, and rear stagnation point were measured in this study. Moreover, the heat transfer rate and pressure loss were evaluated based on the temperature and pressure measurements. The frost thicknesses were measured using the images captured in the experiment. Two cameras were used, one was located at the side of the tube and the other was 500 mm downstream of the test section exit. Since the distance was far enough, we think the effect on the flow around the tube was negligible. The thicknesses were calculated via the comparison between the frosting and bare tubes. Three experiments were conducted under the same conditions, and the averaged values of these results are reported with error bars based on the dispersions. The uncertainty in the frost-thickness measurements according to a reference (Moffat, 1998) is ± 0.1 mm (with a confidence of 95%).

The definition of the pressure loss coefficient $c_p$ is given in the following equation.

$$c_p = \frac{\Delta p}{\rho u^2}$$  \hspace{1cm} (1)

The heat transfer rate $Q$ is calculated using the equation below.

$$Q = q_m c (T_{in} - T_{out})$$  \hspace{1cm} (2)

It is noteworthy that this equation does not consider latent heat attributed to frost formation. The reason of this is related to the purpose of the precooler. The purpose of the precooler is to cool the air. Therefore, most important point is not the increase in the coolant temperature, but the decrease in the air temperature. In this case, latent heat is not important.

The pressure difference between the upstream and downstream pressure orifices, and the density and flow velocity of the main airflow are used to calculate the pressure-loss coefficient. A differential pressure transducer (Setra Systems Inc., Model 264) is used to measure the pressure difference. The full range of this pressure transducer is 25 Pa. A pressure tap to measure the downstream static pressure is located 150 mm downstream of the cooling tube. Our previous study (Sonobe et al., 2013) showed that the static pressure fluctuation at the point 150 mm downstream of a cylinder was small and negligible. The air density is calculated using the temperature measured using a thermocouple and atmospheric air pressure. The measurement uncertainty of the pressure-loss coefficient is ± 0.6 (with a confidence of 95%).

The heat transfer rate is calculated using the difference between the upstream and downstream temperatures, mass flow rate, and specific heat capacity of the air. These temperatures are measured using K-type thermocouples with the diameter of 0.34 mm. With regard to the downstream temperature, three thermocouples are used considering the spatial variation in the air temperature. The height from the lower wall is 30 mm, which is the center of the flow path in the vertical direction. The distances of the thermocouples from the side wall are 10, 20 and 30 mm, respectively. Before starting the tests, the thermocouples were calibrated as follows: The main air at a constant temperature is provided to the test section without the coolant. After 10 min, the temperature of the test section becomes largely constant. The thermocouples are then calibrated to indicate the same temperature. The uncertainty in the measurement of the heat transfer rate was discussed well in our previous study (Sato et al., 2014) and it was estimated as ± 2.4 W (with a confidence of 95 %). The effect of flow non-uniformity attributed to the wake of the tube on the measurement was also discussed in the literature. The uncertainty caused by this flow non-uniformity was estimated at 0.13 W. The data are recorded using a digital data recorder.

Generally, heat transfer performance is evaluated using Nusselt number. However, in the case of this study, the surface temperature is cryogenic. A previous study (Fukiba et al., 2005) showed that the theory of Nusselt number

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Cross-sectional shape</th>
<th>l [mm]</th>
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<tbody>
<tr>
<td>V</td>
<td>V-gutter</td>
<td>7.9–17.9</td>
</tr>
<tr>
<td>C</td>
<td>Circle</td>
<td>5–20</td>
</tr>
<tr>
<td>S</td>
<td>Square</td>
<td>5</td>
</tr>
<tr>
<td>A1</td>
<td>Arrowhead 1</td>
<td>7.9</td>
</tr>
<tr>
<td>A2</td>
<td>Arrowhead 2</td>
<td>7.9</td>
</tr>
</tbody>
</table>

Table 3 Symbols, shape and distance of the obstacles.
cannot be applied to cryogenic surfaces. This is because latent heat release occurs in the boundary layer of cryogenic surfaces and it varies the temperature profile. Therefore, we use heat transfer rates to evaluate the heat transfer performance in this study.

3. Results

3.1. Preliminary experiments using hot water

Preliminary experiments were conducted wherein hot water was provided to the cooling tube. The aim of these preliminary experiments is to evaluate the heat-transfer performance without the frost formation. The experimental setup was the same as that of the experiments conducted under frosting conditions. The temperature of the hot water was approximately 75 °C. The airflow conditions were also the same as the experiment conducted under frosting conditions; the temperature was 20 °C, the humidity was 50%, and the velocity was 1.5 m/s. The temperature distributions were assessed by traversing a thermocouple mounted on a precision positioning stage. The temperature measurements were conducted every 2 mm on the centerline of 40 mm, which was perpendicular to the axis of the cooling tube. The temperature distribution was stable throughout the experiment. This helped in measuring the temperature distribution. However, in the experiments conducted under the frosting conditions, the temperature distribution is unstable. Hence, the measurement is conducted using three thermocouples located downstream of the cooling tube. The effect of the difference between these temperature measurements on the heat transfer rate was discussed in another study (Sato et al., 2014). The difference in the measured heat transfer rate was approximately 1%.

The results of the five types of obstacles tested in these preliminary experiments were compared with that of the bare cooling tube without any obstacles. Table 4 lists the distance between the obstacles and the cooling tube. The obstacle V is placed at a distance of 7.9 mm, which is the distance along the centerline. The shortest distance between the surfaces of the tube and the obstacle is located at the angle of 45° downstream from the stagnation point of the cooling tube, which is 2.6 mm. The heat transfer rates calculated using the measured temperature distributions are also shown in Table 4. The heat transfer rates were calculated by determining the temperature difference between the upstream and downstream sides of the tube. The downstream temperature was determined using the weighted arithmetic mean of the temperature distribution. The heat transfer rate without employing an obstacle is 9.61. This value is similar to that predicted by the theory of a literature (Grimson, 1938), which is 9.17. The heat transfer rates with obstacles V, A1, and A2 are lower than that without the obstacles. This is because of the decrease in the heat transfer rate around the upstream stagnation point. At the Reynolds number employed in this experiment, the heat transfer in this area is dominant (Welty et al., 2001). The heat transfer rates with the other obstacles are higher than that without the obstacles. The increase in the heat transfer rate is due to the flow interference between the tube and the obstacles. The flow over two cylinders in tandem has been well investigated in previous studies (Sumner, 2010). Based on these studies, the shear layer from the upstream cylinder interferes with the surface of the downstream cylinder. The increase in the heat transfer rate is due to this interference. Table 4 also shows the pressure-loss coefficients. As regards the pressure-loss coefficients, the values are increased by mounting an obstacle.

<table>
<thead>
<tr>
<th>Obstacle</th>
<th>Distance [mm]</th>
<th>Heat transfer rate [W]</th>
<th>Percentage [%]</th>
<th>Pressure-loss coefficient</th>
<th>Percentage [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>W/O</td>
<td>-</td>
<td>9.61</td>
<td>100</td>
<td>2.0</td>
<td>100</td>
</tr>
<tr>
<td>V</td>
<td>7.9</td>
<td>9.28</td>
<td>96.6</td>
<td>2.6</td>
<td>130</td>
</tr>
<tr>
<td>C</td>
<td>5</td>
<td>10.7</td>
<td>111</td>
<td>2.6</td>
<td>130</td>
</tr>
<tr>
<td>S</td>
<td>5</td>
<td>11.1</td>
<td>116</td>
<td>3.7</td>
<td>185</td>
</tr>
<tr>
<td>A1</td>
<td>7.9</td>
<td>7.62</td>
<td>79.3</td>
<td>2.4</td>
<td>120</td>
</tr>
<tr>
<td>A2</td>
<td>7.9</td>
<td>8.27</td>
<td>86.1</td>
<td>2.7</td>
<td>135</td>
</tr>
</tbody>
</table>

3.2. Results with the obstacle V

In previous experiments using ethylene glycol coolant, the heat transfer performance with a V-shaped obstacle was investigated (Sato et al. 2014). The coolant temperatures were approximately −10 °C. When the coolant temperature is
−10 °C, the frost increases even on the stagnation point, where the main airflow does not reach directly. This is because vapor is developed via the diffusion. Consequently, the heat transfer rate and pressure loss did not improve.

Figure 6 (a) shows the frost on the cooling tube using liquid nitrogen coolant. Here, the obstacle V was used, and the thickness of the obstacle was 1.5 mm. The distances between the obstacle and the stagnation point of the cooling tube were 7.9 mm (center) and 27.9 mm (right). The images were taken just after the test finished and the obstacle was removed. We visually observed that the frost was symmetrical across the center line. No frost can be observed on the surface from 90 to 160° downstream from the stagnation point. The frost did not increase on the stagnation point of the cooling tube, which is different from the results observed in case of −10 °C coolant. Figure 6 (b) shows the frost on the cooling tube at the coolant temperature of -10 °C. We can observe frost formation on the front stagnation point in this case. It should be noted that when the distance is 27.9 mm in Fig 6 (a), the frost seems to increase on the stagnation point. However, this frost is originated from the side of the cooling tube. In fact, no frost can be observed on the stagnation point. We found that the location and distribution of the frost formation depends on the variation in the coolant temperature.

![Flow](image)

W/O

\[ l=7.9 \text{ mm} \]

\[ l=27.9 \text{ mm} \]

\[ l=7.2 \text{ mm} \]

(a) Frost at the coolant temperature of -196 °C

(b) Frost at the coolant temperature of -10 °C

Fig. 6 Frost on the cooling tube without an obstacle, with the obstacle V located 7.9 mm upstream from the tube, and with the obstacle located 27.9 mm upstream from the tube at 600 s. Figure 6 (b) shows frost at relatively high coolant temperature. The images are captured using the side camera after removing the obstacle.

Figure 7 (a) shows the frost thickness on the side of the cooling tube with the variation in the distance between the obstacle and the cooling tube. The frost thickness without the obstacle did not increase until approximately 60 s. A previous study (Zdravkovich, 1997) showed that the separation point of the main airflow over a cylinder at the Re number in this case is located approximately between 80 and 90° downstream from the stagnation point. Since the

![Frost Thickness](image)

(a) Side

(b) Downstream stagnation point

Fig. 7 Frost thickness with variation in the distance between the cooling tube and obstacle V.
flow separation decreases the gradient of vapor density, this separation of the flow leads to a decrease in the mass transfer. However, the frost thickness with the obstacles increases from the start of the test. This is because the shear layer generated by the obstacle interferes with the cooling tube. Figure 7 (b) shows the frost thickness on the rear stagnation point. The location of the obstacle affects the thickness significantly. It was visually observed that the surface of the frost roughened regardless of the location of the obstacle. The detachment of the frost on the rear side was also observed. These phenomena lead to an increase in the dispersion of the results.

Figure 8 shows the time variations in the pressure-loss coefficients. The pressure-loss coefficients increase with the increase in the frost. The coefficients at the beginning of the test increase because of the presence of the obstacle when the obstacle is mounted. However, the coefficient for obstacle V with a distance of 7.9 mm falls below that without an obstacle at 80 s from the beginning of the test. After 80 s, the coefficient with the obstacle is smaller than that without the obstacle.

Figure 9 shows the time histories of the heat transfer rates. The heat transfer rates decrease rapidly from the beginning of the test because of frost formation. When the obstacles are mounted, the heat transfer rates increase compared with that without the obstacle. This is because of the combination of the suppression of the frost growth and the change in the flow field due to the obstacles. The time-averaged heat transfer rate throughout the test duration of 600 s with a distance of 7.9 mm is 19.0 W, which is 27% larger than that without an obstacle. The time-averaged heat transfer rate is largest for a distance of 17.9 mm, but in this condition the pressure-loss coefficient is larger than that for 7.9 mm. Considering these results, we found that a distance of 7.9 mm is best in these cases, because this distance can help in decreasing the pressure loss with the increase in the heat transfer rate. However, this distance corresponds to 0.79 mm in real precooler, because the diameter of the tube is one tenth of that in this experiment. In this case, the gap between the tube and obstacle is so small that clogging may occur. This clogging of the gap can change the results. We need to investigate the effect of dimension in future work.

3.3 Effect of the distance with the obstacle C

Next, we conducted the tests with the obstacle C. No frost was attached on the stagnation point in the same manner as the case with the obstacle V even when the distance is 20 mm. The area without the frost around the stagnation point is smaller than that with the obstacle V. Figure 10 (a) shows the frost thickness on the side of the cooling tube with the obstacle C. The frost with the obstacle starts to increase just after starting the test. The thicknesses with the obstacle are larger than that without the obstacle until 180 s. The thicknesses after 180 s are comparable to that without the obstacle. The thicknesses are largely the same at the end of the test regardless of the presence of the obstacle. The distance from the obstacle to the stagnation point of the cooling tube does not affect the thickness of the side frost.

Figure 10 (b) shows the frost thicknesses on the downstream stagnation point. The frost thicknesses are reduced because of the obstacle. In particular, when the distance from the obstacle to the upstream stagnation point of the cooling tube is 10 mm, the rate at which the frost increases is very low until 300 s. In this case, the frost thickness at 300 s is approximately 10% of that without the obstacle. This trend is similar for the case of 5 mm in the distance.
According to a previous study (Sumner, 2010), the two cylinders with a small pitch behave as a single bluff body or “extended-body”. In this case the wake of the downstream cylinder becomes narrower and weaker. This variation of the flow field leads to a decrease in the mass transfer of the vapor to the location. The frost on the rear stagnation point is rough and is detached from the surface frequently. These phenomena increase the dispersion in the results of the frost thickness.

Figure 11 shows the time variations of the pressure-loss coefficients. The coefficient for a distance of 5 mm is similar to that without the obstacle. The coefficients slightly increase with the increase in the distance between the obstacle and the cooling tube. However, the increases are comparable to the uncertainty of the coefficient, which is 0.6.

Figure 12 shows the time variations of the heat transfer rates. The rapid increase in the heat transfer rate, for example at 430 s for a distance of 5 mm, is due to the detachment of a chunk of the frost from the rear surface of the cooling tube. The heat transfer rates increase by mounting the obstacle. However, the difference in the heat transfer rate with and without the obstacle reduces gradually as time proceeds. The heat transfer rates at the end of the test are largely the same. The heat transfer rates for distances of 15 and 20 mm are comparable throughout the test. This result shows that the increase of the distance does not affect any more. Figure 10 (b) showed that the frost on the downstream stagnation point for 10 mm is thinnest. However, the heat transfer rate for 10 mm is also comparable with that for 15 mm or 20 mm. This fact means that the frost on the downstream surface does not affect the heat transfer. This is because the heat transfer at this Re number is mainly attributed to that on the upstream surface of the tube.

3.4. Effects of cross-sectional shapes
The results above show that the obstacle V is more effective in suppressing the pressure loss and increasing the heat transfer rate. However, the strength of the obstacle V is insufficient to protect the cooling tubes against FOD. Table 5 lists the second moment of area of the obstacles. In this study, we started the analysis using the obstacles with three basic cross-sectional shapes: V gutter, circle, and square. Additionally, we designed new obstacles with relatively high second moment of area. These are obstacles A1 and A2, as shown in Figs. 5 (d) and (e). Based on the conclusions of this study, the distance between the obstacle and the cooling tube was set to be short, i.e., 7.9 mm. This section discusses the effect of the cross-sectional shape on the frost growth and the heat transfer performance.

Figure 13 shows the frost on the cooling tube with the various obstacles. The distance between the obstacle and the tube is the same as the distance indicated in Table 4. The frost on the rear side is rough and not uniform because of the detachment of the frost from the rear surface.

Figure 14 shows the side frost thickness of the cooling tube with respect to the obstacles of various shapes. The dispersions in the thicknesses with obstacles S, A1, and A2 are greater than those with the other obstacles. This is because the frost detaches frequently in those cases. The time histories of the thicknesses with obstacles V and C after 180 s are similar to that without the obstacle. The thickness with the obstacle S after 250 s is thinner than that without the obstacle. The thickness with the obstacle A2 after 120 s is the thinnest. At the end of the test, the thickness with the obstacle A1 is found to be the thickest. The thickness with obstacles V, C, and S are comparable with that without the obstacle. The thicknesses with obstacles S and A2 are 10% thinner than that without the obstacle.

The side frost thickness with the obstacle A1 is different from that with the obstacle A2 at the end of the test. This difference is due to the differences in the angle and width at the trailing edge of the obstacles. The trailing edge angle of the obstacle A1 is parallel to the main airflow. The width of the obstacle is 20 mm, which is the same as the diameter of the cooling tube. On the other hand, the trailing edge of the obstacle A2 is inclined at an angle of 15° from the main airflow. The width of the obstacle is 21.2 mm. Therefore, the downstream cooling tube is completely included into the wake generated by the obstacle. This reduces the mass transfer of the vapor to the side of the cooling tube. Consequently, the side frost thickness decreases.
Figure 15 shows the time histories of the pressure-loss coefficients. Generally, the obstacles lead to an increase in the coefficient at the start of the test. In particular, the obstacle S increases the coefficient by almost 2 times at the start compared to the coefficient without the obstacle. The coefficient at the end of the test with the obstacle S is the largest and is 1.5 times larger than that without the obstacle. The coefficient with the obstacle A1 at the start of the test is similar to those of the obstacles V, C, A2, and without the obstacles. However, the gradient of the coefficient with the obstacle A1 is larger than that of the other obstacles. Consequently, the coefficient becomes larger than that in other obstacles and comparable with that of the obstacle S. The gradient of the coefficient with the obstacle A2 is smaller than that with the obstacle A1. The coefficient with the obstacle A2 at the end of the test is smaller than that with the obstacle A1, and comparable with the coefficient without an obstacle.

Figure 16 shows the time histories of the heat transfer rate. The heat transfer rates with the obstacles V and S at the beginning of the test are higher than that without the obstacles. However, the heat transfer rates with the obstacles A1 and A2 at the beginning of the test are lower than that without the obstacles. The heat transfer rates decrease with the growth of the frost. The gradients of the decreases in the heat transfer rates with the obstacles A1 and A2 are lower than those with the other obstacles and those without the obstacles. Consequently, the heat transfer rates with the obstacles A1 and A2 relatively surpass these of the others as time proceeds.

Generally, it is desirable to have a high heat transfer rate and a low pressure-loss coefficient. Here, the performances of the cooling tube with the obstacles are evaluated using the heat transfer rate per unit pressure-loss coefficient. Figure 17 shows the heat transfer rates per unit pressure-loss coefficient. The uncertainty of the heat transfer rate per unit pressure-loss coefficient calculated according to a reference (Moffat, 1998) is mainly affected by the value of the pressure-loss coefficient. The uncertainty is 2.0 at 100 s, and it decreases with increase in the
pressure-loss coefficient. The heat transfer rates with obstacles A1 and S are higher than that without the obstacles. However, these obstacles result in large pressure losses. Consequently, the heat transfer rate per unit pressure-loss coefficient becomes small. The heat transfer rates per unit pressure-loss coefficient with the obstacles V after 100 s are higher than that without the obstacles. From the viewpoint of the heat-exchange performance, the obstacle V leads to the highest performance throughout the test duration. However, the strength of the obstacle A1 is much higher than that of the obstacle V. We need to choose obstacles for practical use considering these characteristics.

4. Conclusions

The effectiveness of the method in suppressing frost growth and protecting the cooling tubes of a precooler using obstacles were evaluated experimentally. Fundamental experiments with a cooling tube and obstacle, which was located upstream of the tube, were conducted. The effects of the distance between the obstacle and the cooling tube, and cross-sectional shape of the obstacle were investigated under frosting conditions with liquid nitrogen as the coolant. The pressure loss and heat transfer rate were measured and evaluated. The following are the conclusions of this study:

(1) The obstacles affect the heat transfer rate and pressure loss. In particular, when a V-shaped obstacle with a distance of 7.9 mm was used, the average heat transfer rate was increased by 27%, with a reduction in the pressure loss.

(2) To reduce the pressure loss, it is desirable to place the obstacle at a shorter distance.

(3) Two arrowhead-shaped obstacles were designed to obtain a high second moment of area. The experimental results show that one of these obstacles is useful in protecting the cooling tube with mitigating the adverse effect on the heat transfer performance.

References


Sato, S., Fukiba, K., Sonobe, N. and Yamada, Y., Suppression of Frost Formation on a Cryogenically Cooled Cylinder


