1. Introduction

Strict emission with the increased appreciation of global warming and energy depletion, complex interactions of nitrogen oxides with reactive hydrocarbons and sunlight produce low-level ozone, acid rain and smog and legislation reflects the view that any emission of NOx is to be avoided or minimized [1].

Survey data has indicated that coastal shipping, excluding fishing vessel, within 100 km of the coast accounts for 17% of the total domestic NOx emissions of Japan [2].

In 1997, a protocol which regulated NOx and other emissions from ships was adopted at the International Maritime Organization (IMO). In the same year, five Japanese gas turbine makers (IHI, KHI, Daihatus Diesel, Niigata Engineering and Yanmar Diesel) jointly established a research group to conduct R&D on a low NOx high efficiency next-generation marine gas turbine, the Super Marine Gas Turbines (SMGTs). The goal of this project is to develop a marine gas turbine with high thermal efficiency (38 – 40%) and low NOx emissions of less than 1 g/kWh [2].

It is generally appreciated that the simple cycle gas turbine loses roughly 70% of the heat energy produced in combustion in its exhaust gas. For the purpose of solving this, SMGT has a recuperator to reduce fuel consumption. However, high reliability, compactness, high heat transfer performance, and low pressure drops are required to the SMGT recuperator. Ryo AKIYOSHI et.al [3] designed and manufactured a plate-fin recuperator for SMGT and they also tested it in practical applications. The fin height of the recuperator is 2.0 mm in the air side and 6.0 mm in the exhaust gas side. However, there are two difficult conditions for the design of reliable compact heat exchanger. First is the condition of starting up and shutting down of gas turbine which would cause strong thermal shocks and thermal stress. Second is the rapid load variation [2]. Both of these two conditions will cause transient heat transfer. And the transient response characteristics during rapid load variation are important in the development of recuperator. For the purpose of obtaining a database for the optimal design of plate-fin recuperator under transient heat transfer conditions it is very important to study heat transfer process from different-size plates in transient conditions under various gas velocities.

In 1960, Siegel [4] analyzed the transient heat transfer for laminar flows in parallel plate and tube for step changes
in wall temperature. Soliman et al. [5] analytically obtained a temperature change in plate by taking into account the turbulent boundary around the plate. However, the solution of heat transfer coefficient for water is 50% higher than their experimental data. Kataoka et al. [6] conducted the transient experiment of water which flows in parallel to a cylinder, and obtained an empirical correlation for the ratios between the transient heat transfer coefficient and steady state one in term of one nondimensional parameter composed of period, velocity, and heater length.

Liu and Fukuda [7-9] obtained experimental data and correlations for parallel flow of helium gas over a horizontal cylinder and a plate. They investigated the diameter-effect of cylinders and shape-effect of plate on transient heat transfer. However, the previous experimental data were obtained under a single plate width of 4 mm, there is no detailed knowledge of the effect of plate-width on transient heat transfer for helium gas.

In this study, heat transfer performance under different heat generation rates for helium gas flowing over a plate with different widths were measured to obtain a fundamental database and guideline for safty assessment and optimal design to plate-fin recuperator or high-efficiency compact heat exchanger which was used in SMGTs (Super Marine Gas Turbines).

2. Experimental Apparatus and Method

Experimental apparatus was reported in previous works [8-10]. Fig. 1 shows the schematic diagram of the experimental apparatus. The experiment apparatus is composed of gas compressor (2), flow meter (5), test section (6), surge tank (3, 8), cooler (7), the heat input control system, and the data measurement and processing system. The vacuum pump was used to degas the loop and test section. The gas was circulated by compressor, and the fluctuations of gas flowing and pressure due to compressor were removed with the surge tanks. Moreover, the gas temperature inside the loop was heated to the desired temperature level by a pre-heater, and cooled by a cooler before the gas flows into the compressor. Flowing rate in the test section was measured with the turbine meter, and the pressure was measured with the pressure transducer. The test heater was made of platinum with widths of 1.5 mm, 4 mm and 6 mm, and thicknesses of 0.2 mm, 0.1 mm and 0.1 mm. It was heated by direct current from a power source. The heat generation rates of the heater were controlled and measured by a heat input control system. The heat generation rate was raised with exponential function, \( \dot{Q} = Q_0 \exp(\tau t) \). Where, \( \dot{Q} \) is heat generation rate, W/m³, \( Q_0 \) is initial heat generation W/m³, \( t \) is time, s, and \( \tau \) is e-fold time or period of heat generation rate, s. A smaller or shorter period means a higher increasing rate of heat generation.

The average temperature, \( T_a \) (K) of the test heater was measured by resistance thermometry using a double bridge circuit including the test heater as a branch. The heat flux on the surface of test heater, \( q \), is calculated by the following equation.

\[
q = \frac{\delta}{2} (\dot{Q} - \rho c_h \frac{d\dot{T_a}}{dt})
\]

Where, \( \rho_h \), \( c_h \) and \( \delta \) are the density, specific heat, and thickness of the test heater, respectively.

Following unsteady heat conduction equation was used to calculate instantaneous surface temperature of the test heater by assuming the surface temperature around the test heater to be uniform.

\[
a \frac{\partial^2 T}{\partial x^2} + \frac{\dot{Q}}{\rho_h c_h} = \frac{\partial T}{\partial t}
\]

Boundary conditions are as follows:

\[
\left. \frac{\partial T}{\partial x} \right|_{x=0} = 0, \quad \left. -\lambda \frac{\partial T}{\partial x} \right|_{x=\delta} = q = \frac{\dot{Q}}{2}
\]

\[
T_a = \int_{\delta}^{\frac{\delta}{2} \dot{T}_0 dx} = \frac{\dot{Q}}{2} \int_{\frac{\delta}{2}}^\delta T_0 dx
\]

Where, \( a \) (m²/s) and \( \lambda \) (W/mK) are the thermal diffusivity and thermal conductivity.

The physical properties of the fluid were calculated

---

Fig. 1 Schematic diagram of experimental apparatus.
based on the following film temperature:

$$T_f = \frac{T_s + T_b}{2}$$

(5)

Where, $T_s$ and $T_b$ are the test heater surface temperature, and the bulk gas temperature, respectively.

The uncertainties of the measurement of the heat generation rate, the heat flux of the test heater, and the heater surface temperature are estimated to be ± 1%, ± 2%, and ± 1 K, respectively. And the uncertainty for the heat transfer coefficient is estimated to be ± 2.5%.

2.1 Test Section

Figure 2 shows a vertical cross-sectional view of test section. The test heater was mounted horizontally along the center part of the circular test channel, which is made of the stainless steel (20 mm in the inside diameter). Platinum plate with an effective length ranging from 3.69 cm to 5.46 cm was used as the test heater.

3. Experimental Results and Discussion

3.1 Experiment Conditions

The transient heat transfer experimental data were measured for the periods of heat generation rate ranged from 49 ms to 17 s at a gas temperature of 303 K and a system pressure of around 500 kPa. The flow velocities ranged from 4 m/s to 10 m/s.

3.2 Experimental Data of Heat Generation Rate, Surface Temperature Difference, and Heat Flux

Figure 3 shows typical experimental data of the time-dependence of heat generation rate, $\dot{Q}$, surface temperature difference, $\Delta T$ ( $\Delta T=T_s-T_b$), and heat flux, $q$, at the gas temperature of 303 K and the heat generation rate increasing periods of 1.68 s, 429.2 ms, and 89.6 ms, respectively. The gas flow velocity is 10 m/s and the width is 6 mm. As shown in this figure, heat generation rate, temperature difference and heat flux increase rapidly as the period is shorter. It is understood that the surface temperature difference and heat flux increase exponentially as the heat generation rate increases with exponential function.

3.3 Transient Heat Transfer Coefficient at Various Periods of Heat Generation Rate

Heat transfer coefficient, $h$, is defined as shown in the following equation.

$$h = q / \Delta T$$

(6)
Figure 4 [8] and Fig. 5 show the heat transfer coefficients at various periods of heat generation rate at a gas temperature of 313 K and 303 K for helium gas with heater width of 4 mm and 6 mm. The heat transfer coefficient, $h$, becomes to approach asymptotic value at every velocity when $\tau$ is longer than about 1 s. The heat transfer process in this region transmits heat as well as usual convective heat transfer through the thermal boundary layer influenced by the flow of helium gas. It is called the quasi-steady-state heat transfer here. On the other hand, when the period, $\tau$ is shorter than about 1 s, $h$ increases as $\tau$ shortens. This shows that the heat transfer process is in the unsteady state. In the region of $\tau$ shorter than 200 ms, the conductive heat transfer near the heater comes to govern the heat transfer process, and the heat transfer coefficient increases greatly with shorter period in this region. It was clarified that the heat transfer phenomenon was divided into a quasi-steady-state heat transfer and a transient heat transfer on the boundary of around 1 s. The coefficient of heat transfer increases with the flow velocity as shown in the Figs. 4 and 5. It can also be found that the heat transfer coefficient of heater with width of 4 mm is much higher than that of heater with width of 6 mm when they are in the same velocity and $\tau$ (e-fold time).

However, the difference of heat transfer coefficient between transient state and the quasi-steady-state at width of 6 mm is higher than that at width of 4 mm. Such difference at width of 6 mm is from 23% to 13% by comparing the heat transfer coefficient at the period of 175 ms and the heat transfer coefficient at quasi-steady-state within the velocity ranging from 4 m/s to 10 m/s. Meanwhile the difference between heat transfer coefficient at 175 ms and the quasi-steady-state is from 9% to 4% for the 4 mm-width heater within the same velocity range. It is considered that the influence of period (e-fold time) to heat transfer coefficient increases with the increase of width.

### 3.4 Correlation for Quasi-Steady-State Heat Transfer

Figures 6 a and b show the relationships between Nusselt numbers and the periods of heat generation rate at velocity of 4 m/s and 6 m/s for plates with 1.5 mm and 6 mm widths. It can be found that the Nusselt numbers are almost constant at the periods from 1.68 s to 17 s and such phenomena can also been found at other velocities (8 m/s and 10 m/s). It was perceived that the heat transfer performance increases with the decrease in the width of the test heater.

Figure 7 shows the relations between the Nusselt numbers ($N_{ust}$) and the Reynolds numbers ($Re_f$) for the widths ranging from 1.5 mm to 6 mm at the velocities ranging from 4 m/s to 10 m/s. As shown in this figure, the Nusselt number increases as the width of test heater decreases. It can be seen the Reynolds number is under the range of laminar flow so the experimental data can be correlated as follows:

$$Nu_{st} = k f^{0.5} Pr f^{1/3}$$  \hspace{1cm} (7)

Where, $Pr_f$ is Prandtl number which is about 0.68 in the range of this experiment and $k$ equals to 1.56, 1.24 [8], and 1.04 for widths of 1.5 mm, 4.0 mm and 6.0 mm, respectively.
and $k$ equals to 0.916 for an infinite plate [11]. The Nusselt number decreases with increasing the width of the heater.

3.5 Comparison of the Experimental Data of Plate with Author’s Data of Horizontal Cylinders

The authors have measured heat flux, surface temperature, and transient heat transfer coefficients for forced convective flow of helium gas over horizontal cylinders under wide experimental conditions [10]. The platinum cylinders with diameters of 1.0 mm, 0.7 mm, 2.0 mm were used as the test heaters.

Figure 8 shows the quasi-steady state and transient heat transfer coefficients of single horizontal cylinders with diameters of 0.7 mm, 1.0 mm and 2.0 mm. The difference of heat transfer coefficient between transient state and quasi-steady state is also influenced by the size of cylinder as the

---

**Fig. 8** Quasi-steady-state and transient heat transfer coefficient at various periods for the cylinders [10].

**Fig. 7** Nusselt numbers at various Reynolds numbers.

**Fig. 9** Quasi-steady-state heat transfer at various sizes of plate and cylinder.
plate appears. The difference increases with the increases of diameter of the cylinder.

Figure 9 shows the experimental data for the plate and cylinders [10] on the Nusselt number versus Re graph. As shown in the figure, the Nusselt number decreases with the increase in diameter.

According to the author’s experimental data, the heat transfer coefficient shows significant dependence on the size and the geometry (shape) of a heater (cylinder and plate). It is well known that convective heat transfer coefficient is related to the flow field developing on the wall of heater, and it is important to develop an expression relating the two dimensional quantities of heat transfer and flow. The difference in results for cylinders and ribbon is physically due to the difference in the thermal boundary layers developed on the cylinder and ribbon surfaces.

4. Conclusions

The forced convection transient heat transfer process for helium flowing over a horizontal plate with different widths was experimentally studied to obtain a fundamental database for the recuperator. Following conclusions were obtained.

(1) Surface temperature difference and heat flux increase exponentially as the heat generation rate increases with the preset exponential function.

(2) It was clarified that the heat transfer coefficient approaches the quasi-steady-state one for the period $\tau$ over about 1 s, and it becomes higher for the period of $\tau$ shorter than about 1 s. The conductive heat transfer becomes predominant for the period less than about 1 s, especially in the region of less than 200 ms.

(3) The heat transfer coefficient was influenced by the widths of the heater: the heat transfer decreased with increasing the width of the test heater.

(4) The difference of heat transfer coefficient between transient state and quasi-steady state is influenced by the width of the heater.

(5) An empirical equation for the quasi-steady-state heat transfer was obtained based on the experimental data. Such equation applies to laminar flow and is also affected by the width of the plates.

Acknowledgments

This work was supported by the Japan Society for the promotion of Science (JSPS) (Grant-in Aid for Scientific Research (C), KAKENHI, No. 24560231)

References