Transient Forced Convection Heat Transfer for Various Gases in the Low Re Region Flowing Across a Horizontal Cylinder*

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ABSTRACT
The knowledge of forced convection transient heat transfer for exponentially increasing heat inputs to a heater with various periods is important as a database for understanding the transient heat transfer process in high temperature gas cooled reactor (HTGR) due to an accident in excess reactivity. In this study, the transient heat transfer coefficients for Helium, Argon, Air, and Nitrogen gases flowing perpendicular to a horizontal cylinder were measured in the low-Reynolds-number region. The platinum heater with a diameter of 1.0 mm was heated by electric current with an exponentially increase of $Q_0 \exp(t/\tau)$. It was clarified that the heat transfer coefficient approaches the quasi-steady-state one for the period $\tau$ over around 1 s, and it becomes higher for the period of $\tau$ shorter than about 1 s. Though the transient heat transfer region is influenced by both convection and conduction heat transfer in the quasi-steady state heat transfer region, the conduction heat transfer becomes predominant as the period becomes shorter, especially in the region of within 200 ms. The transient heat transfer shows independent of the gas flowing velocity when the period becomes very short. Based on the experimental data, the ratio of transient heat transfer to the quasi-steady-state one was correlated as a function of Reynolds number of the gas flow and the non-dimensional period of increasing heat input. For the non-dimensional period larger than about 300, the transient heat transfer approaches the quasi-steady-state one, and shows independent of the Reynolds number.

Key words: Transient Heat Transfer, Forced Convection, HTGR, Exponentially Increasing Heat Input, Period, Low-Reynolds-Number, Helium Gas, Cross-Flow, Horizontal Cylinder
1. INTRODUCTION

Gas turbine has the feature in the higher speed, lightweight, and compact size. It gets attention and is developed as an environmental preservative marine engine \(^{1}\), since the amount of NOx emission is markedly reduced compared with a diesel engine. In the development, a plate fin type regenerator (heat exchanger) is adopted in order to improve thermal efficiency\(^{1}\). When a gas turbine is adopted for marine engine, the transient response at the time of a rapid load change is also an important research subject. For example, gas flow may change with a change of load, and then gas temperature in the heat exchanger entrance may change from the rated one and the thermal efficiency of a gas turbine may become lower. In order to investigate the change of thermal efficiency accompanying a transient change of the temperature of gas, or of the flow rate, the transient heat transfer process of gases needs to be clarified. Furthermore, the understanding of the transient heat transfer process of gas is also important in the research and development of high temperature gas cooled nuclear reactor (HTGR). The HTGR uses helium gas as a cooling medium; it can supply heat with a temperature as high as 1000 °C, and enables achievement of high thermal efficiency and high performance of heat utilization. Moreover, the HTGR has the excellent stability, and has the feature of high burnup of the fuel and easiness of the operation. The heat supplied from the HTGR is applicable to synthetic gas manufacture, steam turbine power generation, etc. in many stages, and it can be used at a very high utilization efficiency of 80%. For example, the heat with a high temperature (high level) is first used for steam reforming to produce hydrogen and methanol, and using the heat of the secondary level, steam is generated, then it generates electricity. If the steam with a temperature of 300 °C from a gas turbine is used for general industry, the total utilization efficiency of heat will exceed 80%\(^{2}\).

An additional relation with a hydrogen energy system is also given briefly here. In the research and development of an international clean energy system using hydrogen (WE-NET)\(^{3}\), the manufacture technology of hydrogen is also important as the same with the transportation, storage, and utilization technology of hydrogen. Utilization of high-temperature gas cooled reactor (HTGR) will also be expected to perform a great contribution in the future to the construction of the clean secondary energy system using hydrogen. However, some problems have not been solved in the development of HTGR, and the accumulation of fundamental research is expected. The knowledge of forced convection transient heat transfer at various periods of exponentially increasing heat input to a heater is important as a database for safety assessment of the transient heat transfer process in a HTGR due to an accident in excess reactivity.

The transient heat transfer has not been solved though many analytical solutions and experimental results were reported concerning the steady state heat transfer. The following theoretical researches were carried out on transient convection heat transfer when the wall temperature of test heater rises in the shape of a step. In 1960, Siegel\(^{4}\) analyzed the transient heat transfer for laminar flows in parallel plate and tube when the surface temperature of wall varied in step changes. Then, Sparrow et al. \(^{5}\) and Goodman \(^{6}\) analyzed the transient heat transfer for turbulent flow in a tube. In these works,
the transient heat transfer was analyzed for step changes in wall temperature; there was no verification with experiment data.

Concerning the problem of transient heat transfer with exponentially increasing heat generation rate \( (\dot{Q} = Q_0 \exp(t/\tau)) \), where \( \dot{Q} \) is heat generation rate, \( Q_0 \) is initial heat generation rate, \( t \) is time, and \( \tau \) is period of heat generation rate), there are only a few analytical and experimental works as far as the authors know. Soliman et al.\(^7\) 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.\(^8\) conducted the transient experiment of water which flows in parallel with 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.

The above previous researches have not resulted in a correlation with reliability based on physical model. Moreover, there is almost no experimental research on transient forced convection heat transfer process for helium gas flowing over a test heater with exponentially increasing heat generation rate, and there is no detailed knowledge on the effects of the period of heat generation rate and the flow velocity on the transient heat transfer for helium gas.

In this research, to obtain the fundamental data for the thermal analysis of transient response of a heat exchanger, and for the safety analysis of a high-temperature gas cooled reactor, and also to clarify the transient heat transfer problem, the transient natural convection heat transfer and the transient forced convection heat transfer for various gases (Helium, Argon, Air and Nitrogen) flowing perpendicular to a horizontal cylinder are experimentally studied in the low Reynolds number region when the heat input to the test heater is increased exponentially with various rates or periods. The effects of flow velocity and the period of heat input on the transient heat transfer are clarified, and empirical correlations for the transient heat transfer are obtained.

2. EXPERIMENTAL APPARATUS AND METHOD

2.1 Forced Convection Heat Transfer Experiment Apparatus

Figure 1 shows the schematic diagram of the experiment 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 preheater, 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 temperature of
the turbine meter exit and the temperate near test section heater were measured by chromel-
alumel thermocouples with precision of ± 1 K. Helium, Argon, Air and Nitrogen gases were used as the working fluids.

2.2 Test Section

Figure 2 shows a vertical section in test section (6), the heat generation control system (16) composed of an analogue computer, and the measurement and the data processing system which consists of A/D converter (17), personal computer (18), and D/A converter (19). The test section is a rectangular channel made of the acrylic fiber (100mm in width, 50mm in height, and 1100mm in total length). Straightening screens are installed in the entrance and outlet of the test channel to make the flow velocity distribution as uniform as possible.

A platinum cylinder of 1.0mm in diameter, 60mm in length, and 40mm in effective length was used as the test heater. It was installed horizontally in cross-flow state at the center part of the cross-section of the channel, and the position of the test heater was 600mm downstream from the entrance of channel.

2.3 Experimental Method and Procedure

The experiments were carried out according to the following procedure. The working fluid such as helium etc. was first filled to the test loop after the test loop was degassed by a vacuum pump. The working fluid was circulated by driving compressor. Flowing rate was sequentially lowered from maximum stream flow in stages. The regulation of the flowing rate was carried out by using the by-pass valves of the test section and the by-pass valve of the compressor. After the pressure was confirmed to be stable at each flow velocity in the loop, the electric current was supplied to the test
heater, and the heat generation rate $\dot{Q}(=Q_0 e^{t/\tau})$ was raised exponentially, then the test heater surface temperature and the heat flux accompanying the passage of the time were measured.

The platinum test heater 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 average temperature of test heater was measured by resistance thermometry using a double bridge circuit including the test heater as a branch. The test heater was annealed and its electrical resistance versus temperature relation was calibrated in water, and washed with a trichloroethylene liquid before using it in the experiment.

The heat flux of the heater is calculated by the following equation.

$$q = \frac{D}{4} \hat{Q} - \rho_h c_h \frac{D}{4} \frac{dT_a}{dt}$$

Here, $\rho_h$, $c_h$, $D$, and $T_a$ are the density, specific heat, diameter, and the average temperature of the test heater, respectively. The test heater surface temperature can be calculated from unsteady heat conduction equation of the next expression by assuming the surface temperature around the test heater to be uniform.

For the cylindrical test heater, we have,

$$\frac{\partial T}{\partial t} = a \left( \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} \right) + \frac{\dot{Q}}{\rho c}$$

Boundary conditions are as follows,

$$\left. \frac{\partial T}{\partial r} \right|_{r=0} = 0, \quad -\lambda \left. \frac{\partial T}{\partial r} \right|_{r=R} = q$$

$$T_a = \frac{\int_0^R T(2\pi)dr}{\int_0^R (2\pi)dr} = \frac{2}{R^2} \int_0^R T_{tr}dr$$

Where, $\dot{Q}$ (W/m$^3$) is the internal heat generation rate (measurement value), $T_a$ (K) is the average temperature of test heater (measurement value), $q$ (W/m$^2$) is the heat flux on surface of test heater obtained by Eq.(1) (measurement value), $a$ (m$^2$/s) is thermal diffusivity, and $\lambda$ (W/mK) is thermal conductivity.

The errors of the measurement of surface temperature of the test heater and the heat flux are estimated to be ±1 K and ±2% respectively.

When the experimental data were processed, the physical properties of the fluid were calculated based on the film temperature (the average temperature of surface temperature $T_s$ and flowing gas
temperature $T_\infty$, which was used as a temperature of the representative.

In this experiment, the mean flow velocity was not used as the mean velocity around the test heater due to a velocity distribution along the heater. The velocity distribution along the heater was calculated as follows.

$$u_p = \frac{3}{2}u_a \left[1 - \left(\frac{x}{b}\right)^2\right]$$  \hspace{1cm} (5)

Where, $x$, $b$, and $u_a$ are distance from the center between two plates, half of the distance between the plates, and the mean flow velocity. Then the mean velocity around the test heater was calculated as the average value of central velocity (maximum velocity, $u_{\text{max}} = \frac{3u_a}{2}$) for fully developed laminar flowing and the mean velocity, $u_a$.

3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Experiment Conditions

Table 1 shows the experimental conditions carried out in this study. The heat generation rate was raised with exponential function ($\dot{Q} = Q_0 \exp(t/\tau)$). Here, $\tau$ is the period of heat generation rate. A smaller or shorter period means a higher heat generation rate.

3.2 Time-dependence of Heat Generation Rate, Surface Superheat, and Heat Flux

Figure 3 shows typical experimental data of the time-dependence of heat generation rate, surface superheat ($\Delta T$), and heat flux at the heat generation rate increasing period of 500 ms under flow velocity of 0.6 m/s (Reynolds number 12) and gas temperature of 23.7 °C. It is understood that surface superheat and heat flux increase exponentially as the heat generation rate increases with the exponential function.
3.3 Effect of the Period of Heat Generation Rate on Transient Convective Heat Transfer

Coefficient of heat transfer \( h \) is defined by 
\[
h = \frac{q}{\Delta T}.
\]
\( \Delta T \) is a difference between the surface temperature of the test heater and the temperature of the fluid. It is called surface superheat here. Figure 4 shows the coefficient of heat transfer versus time at the periods of 9.63 s and 318 ms for a flow velocity of 0.6 m/s under the pressure of 249 kPa. The heat transfer coefficients approach constant values from higher initial values when the time passes over a certain time of about 5 times of the period \( (t/\tau > 5) \). It was confirmed that the heat transfer coefficients approach asymptotic values similarly at all periods and velocities regardless of the length at the period. These asymptotic values will be used as the transient heat transfer coefficients. \(^8\)

Figures 5(a) and (b) show the relation between transient heat transfer coefficient \( h_t \) and period \( \tau \) at surface superheats of 80 °C and 300 °C under the natural convection \((U=0)\) and the forced convection \((U=0.6 \text{ m/s})\), respectively. When \( \tau \) is longer than about 1s, \( h_t \) becomes a constant asymptotic value. The heat transfer process in this region transmits heat as well as a usual convective heat transfer through the thermal boundary layer influenced by the flow of the gas. It is called the quasi-steady-state here. On the other hand, \( h_t \) increases as \( \tau \) shortens in the region where period \( \tau \) is shorter than about 1s. This shows that the heat transfer process is in the unsteady state, and the heat transfer in this region has received greatly the influence of the temperature gradient.
within the thermal boundary layer around the test heater. Especially, in the region of \( \tau \) shorter than 200 ms, the thermal boundary layer becomes very thin, then the conduction 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. The phenomenon which divides into a quasi-steady-state heat transfer and the transient heat transfer process on the boundary of about 1s was clarified.

When the data at the surface superheats of 80 °C and 300 °C are compared, \( h_r \) varies somewhat with \( \Delta T \) in the region where \( \tau \) is longer than about 1s at the flow velocity of 0, but the changes in \( h_r \) are hardly seen when there is a flow velocity.

Figure 6 shows the typical experimental data of Argon, Nitrogen and Air at the velocity of around 0.25 m/s. These data show that they are divided into quasi-steady-state heat transfer region and the transient heat transfer region on the boundary of around 1 s as well as the case of the helium gas.

### 3.4 Quasi-Steady-State Natural Convection Heat Transfer

Figure 7 shows heat flux versus superheat for the quasi-steady state natural convection heat transfer at a period of 20s. The gas temperature is 23.0 °C, and the pressure is 200kPa. The curve obtained from the laminar natural convection heat transfer correlation for horizontal cylinder by Takeuchi et al. \(^9\) is shown in this figure for the comparison. It is understood that the experimental data agree well with Takeuchi et al.'s natural convection heat transfer correlation.
3.5 Quasi-Steady-State Forced Convection Heat Transfer

Figure 8 shows the relation between the quasi-steady-state Nusselt number $N_u_f$ and Reynolds number $Re_f$ at the case of the helium gas. They are shown on $N_u_f/Pr_f^{0.4}$ vs. $Re_f$ graph. The average heat transfer correlation in the uniform flow by Hilpert \(^{10}\) is shown with the short dashes line in this figure for the comparison. The solid line is the following correlation (Eq.(6)) obtained from the experiment data. Higher value than the correlation of Hilpert is indicated in the region where $Re_f$ is very small though author’s experimental data becomes to agree with the correlation by Hilpert as $Re_f$ increases. The difference may be arising from the difference in the kind of the fluid.

$$N_{uf} = 1.59Re_f^{0.22}Pr_f^{0.4} \text{ for Helium (6)}$$

The number of $Pr_f$ of helium gas is about 0.68 in the range of this experiment.

On the other hand, the following quasi-steady-state heat transfer correlation were similarly obtained for argon, nitrogen, and air gas.

$$N_{uf} = 2.85Re_f^{0.22}Pr_f^{0.4} \text{ for Argon (7)}$$
$$N_{uf} = 2.40Re_f^{0.22}Pr_f^{0.4} \text{ for Nitrogen (8)}$$
$$N_{uf} = 2.35Re_f^{0.22}Pr_f^{0.4} \text{ for Air (9)}$$

The gradient of the Reynolds number is a small value of 0.22. The reason for this is that the Reynolds number of this research is in the range of the low Reynolds number.

3.6 Transient Forced Convection Heat Transfer

Figure 9 shows the relation between the Nusselt number and the Reynolds number in the transient heat transfer of the helium gas. The data and the correlation for the periods larger than about 1s are shown in the same figure for comparison. The Nusselt number is influenced by the period and flow
velocity. It approaches asymptotic value in the quasi-steady state heat transfer as the period increases. The effect of flow velocity becomes weak for the smaller period by decreasing the gradient of the data in the graph.

Figure 10(a)-(d) shows the Nusselt numbers $N_{u_{tr}}$ of the gases (He, Ar, N$_2$, Air) at various periods with the Reynolds number $Re_b$ (evaluated at bulk temperature) as a parameter. When period $\tau$ is longer than about 1s, Nusselt number $N_{u_{tr}}$ approaches asymptotic value, that is, a quasi-steady-state value. $N_{u_{tr}}$ is influenced and increases with the flow velocity when comparing at different Reynolds numbers at a certain period. On the other hand, when period $\tau$ is shorter than about 1s, $N_{u_{tr}}$ increases as the period $\tau$ shortens and shows little dependence on the Reynolds number.
Kataoka et al. carried out an experiment on the transient heat transfer from a platinum cylinder to water flowing along the test heater. They reported an empirical correlation for the ratio of Nusselt number of the transient to quasi-steady-state Nusselt number using a dimensionless period, $\tau^* (=\tau U/l$, $U$: flow velocity, and $l$: characteristic length). Figure 11(a)-(d) shows the ratios of transient Nusselt numbers, $\text{Nu}_t$, of the gases, to quasi-steady state Nusselt numbers, $\text{Nu}_s$, at various periods and Reynolds numbers. Here, the characteristic length used is the diameter of test heater, then the dimensionless period, $\tau^*$, is expressed as $\tau U/d$. As can be seen from the figure, the transient heat transfer in all Reynolds numbers approaches orderly the quasi-steady-state heat transfer from the low Reynolds number with the increase of the dimensionless period. The heat transfer shifts to the quasi-steady-state heat transfer for longer period and shifts to the transient heat transfer for shorter period at the same flow velocity. The transient heat transfer approaches the quasi-steady state one for higher flow velocity at the same period. It can be seen from Fig.11, the ratios of $\text{Nu}_t$ to $\text{Nu}_s$ decrease to unit as the nondimensional period increases. The transient heat transfer approaches steady state one for the nondimensional period larger than about 300. The following empirical equation was obtained based on the experimental data. It was shown as the solid and dashed lines in the figures.

\[
\frac{\text{Nu}_t}{\text{Nu}_s} = (1 + K\tau^{*1.25}) \\
K = CRe_b^{0.25}
\]
Where, $\text{Nu}_t$, $\text{Nu}_{st}$, $\tau$, and $\tau^* (=\tau U/d)$ are Nusselt number of the transient, quasi-steady-state Nusselt number, period of heat generation rate, and dimensionless period. Coefficient $K$ was obtained as a function of the Reynolds number evaluated by a free stream temperature. As shown in Fig.12, it is proportional to Reynolds number in any gas, and differs according to the gas. The proportion coefficient $C$ is obtained as 7.68 for helium and 2.05 for Argon, Nitrogen, and Air gases, respectively. Most of the proportion coefficients are constant values excluding helium. The physical properties value that the helium gas is greatly different from other gases is thermal conductivities. It is about 6-8 times as large as other gases. It is considered that the reason why the value of coefficient $C$ of helium gas is different from those of other gases is due to the thermal conductivity of helium gas being greatly higher than other gases.

4. CONCLUSIONS

Transient heat transfer coefficients from a horizontal cylinder in the cross-flow of helium, argon, nitrogen, and air gases at low-Reynolds-number region were experimentally measured using an exponentially increase of heat input added to the horizontal cylinder. The following results were obtained.

(1) The transient heat transfer coefficient under natural or forced convection condition approaches asymptotic value in quasi-steady-state for the period longer than about 1s, and increases as the period shortens for the period shorter than about 1 s.

(2) Though the transient heat transfer region is influenced by both convection and the conduction heat transfer in the quasi-steady state heat transfer region, the conduction heat transfer becomes predominant as the period shortens, especially in the region of within 200 ms.

(3) The empirical correlation for the transient forced convection heat transfer was obtained by the quasi-steady-state heat transfer correlation, the dimensionless period of heat generation rate, and the Reynolds number of the flow based on the experiment data. It was clarified that, in the correlation, $\text{Nu}_t/\text{Nu}_{st} = (1 + CRe_b \tau^{-1.25})$, the value of coefficient $C$ of helium gas is different from those of other gases.
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