Experimental and Theoretical Study on Transient Heat Transfer for Forced Convection Flow of Helium Gas over a Twisted Plate with Different Length

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Forced convection transient heat transfer for helium gas flowing over a twisted plate with different length was experimentally and theoretically studied. The heat generation rate of the twisted plate was increased with a function of \( Q = Q_0 \exp(t / \tau) \) (where \( t \) is time, \( \tau \) is period). Experiment was carried out for three kinds of effective length of 26.8, 67.8 and 106.4 mm at various periods ranged from 35 ms to 13 s and at gas temperature of 303 K under 500 kPa. The flow velocities ranged from 4 m/s to 10 m/s. It was found that average heat transfer coefficient becomes higher with the increase of flow velocity and decreases with the increase of plate length under the same period of heat generation rate. Numerical simulation results were obtained for average surface temperature difference, heat flux and average heat transfer coefficient of twisted plate and showed reasonable agreement with experimental data.

Key Words: Transient heat transfer, Twisted plate, Numerical solution, Heat transfer coefficient, Length effect

1. Introduction

The VHTR (Very High Temperature Reactor) has become the most competitive candidate for the Next Generation Nuclear Plant (NGNP) prototype concept according to the US Energy Policy Act of 2005. However, some thermal hydraulic problems are still under study in the development of VHTR like transient heat transfer problem which may occur due to some accidents e.g. power burst, rapid depressurization and withdraw of control rods. For safety assessment, it is necessary to study the transient forced convection heat transfer process accompanying exponentially increasing heat input to a heater.

In this study, a series of twisted plate with different length have been experimentally and numerically studied to clarify the effect of length on transient heat transfer. Three-dimensional transient simulation were carried out by ANSYS FLUENT 14.0 code to obtain the local heat transfer coefficient and clarify the mechanism of length effect.

2. Experimental study

2.1 Experimental apparatus

![Schematic diagram of experimental apparatus](image)


Experiment apparatus was reported in previous papers [1-3], as shown in Fig.1. 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 plates with thickness of 0.1 mm, were used as the test heaters. They were twisted with the same helical pitch of 20mm, and length of 26.8mm, 67.8mm and 106.4mm (pitch numbers of 1, 3 and 5), respectively.

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

\[
q = \frac{\delta}{2} (Q - \rho_h c_h \frac{dT_{w}}{dt})
\]  (1)

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

The instantaneous surface temperature of test heater was calculated by the following equation assuming the surface temperature of test heater to be uniform.

\[
\alpha \frac{\partial^2 T}{\partial x^2} + \frac{\partial}{\partial t} \left( \frac{\partial T}{\partial t} \right) = \frac{T_{w}}{\rho_h c_h}
\]  (2)

Physical properties of the fluid were calculated based on the film temperature \( T_f \). Where, \( T_f = (T_s + T_h)/2 \), \( T_s \) and \( T_h \) are the test heater surface temperature and the bulk temperature of flowing gas, respectively.

The heat generation rate was raised with exponential function, \( \dot{Q} = Q_0 \exp(t / \tau) \). Where, \( \dot{Q} \) is heat generation rate, W/m², \( Q_0 \) is initial heat generation rate, W/m², \( \tau \) is time, \( s \) and \( \tau \) is period of heat generation rate, s.

2.2 Experimental results and discussion

Figure 2 shows typical experimental data of the time-dependence of heat generation rate \( \dot{Q} \), surface temperature difference \( \Delta T \), and heat flux q. It can be seen that the surface temperature difference and heat flux increases exponentially as
the heat generation rate increases exponentially. Surface temperature difference is the difference between the average surface temperature of the twisted plate \(T_{\text{av}}\) and the inlet gas temperature \(T_{\text{w}}\), expressed as:

\[
\Delta T = T_{\text{av}} - T_{\text{w}}
\]

(3)

The heat transfer coefficient, \(h\), is defined as shown in the next equation.

\[
h = \frac{q}{\Delta T}
\]

(4)

![Fig.2 Time-dependence of Q, q, \(\Delta T\) at 10 m/s.](image)

The heat transfer coefficients at various length were obtained from the values of heat flux and temperature difference. Based on the obtained heat transfer coefficients, it can be found that the heat transfer coefficient decreases with the increase of heater length. The quasi-steady-state heat transfer coefficient for the heater with effective length of 26.8 mm is about 700 W/m²K, about 24% higher than that of the 67.8 mm heater. While the quasi-steady-state heat transfer coefficient improved only about 5% by comparing the 67.8 mm heater to the 106.4 mm heater. So, it can be concluded that the heat transfer coefficient along the length direction of a twisted plate (heater) is not constant and the distribution is nonlinear.

3. Numerical simulation results

The Reynolds Stress Model was used to model the turbulent flow regime. The convection term was discretized using the second order upwind scheme, and the linkage between the velocity and pressure was computed using the SIMPLE algorithm. The Enhanced Wall Treatment model was chosen for the near-wall modeling method because it combines the use of a blended law-of-the-wall and a two-layer zonal model and generally requires a fine near-wall mesh that is capable of solving the viscous sub-layer. The dimensionless distance \(y^+ (= \mu_\text{w} y/\nu)\) was considered [4]. To ensure the mesh quality, the first near-wall node is placed at \(y^+ \approx 1\) and over 20 cells were placed in boundary layer. In addition, a convergence criterion of \(10^{-3}\) was used for all of the calculated parameters and the simulation result converged in each time step.

The mesh independency was tested for three different mesh size. For the case of \(U = 10\) m/s, \(\tau = 1.4\) s, helical pitch size = 20 mm, mesh dependency was examined by solving the flow field with total mesh cells of about 100,000, 400,000 and 1,160,000 respectively. For each grid size, the \(y^+\) is satisfied firstly (\(y^+ \approx 1\)). The temperature difference \(\Delta T\) was compared and less than 0.6% difference exists between the two finer meshes. It is indicated that the finer mesh resulted in mesh-independency solution. The solution mesh used in this study is the finer mesh which is built of hexahedrons by ANSYS ICEM CFD 14.

Figure 3 shows the comparison of numerical simulation result with the experimental result. The symbols represent experimental data and the lines represent simulation results. It can be seen that the heat transfer coefficient increases with the decrease of effective length. The numerical solution results agree well with experimental data within 4.0%.

![Fig.3 Comparison of numerical simulation result with experimental data.](image)

4. Conclusions

Forced convection transient heat transfer for helium gas flowing over a twisted plate with different length was experimentally and theoretically studied. Following conclusions were obtained.

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

(2) It is obtained that there exists a distribution of heat transfer coefficient along the twisted plate, it becomes higher for a small length of heater.

References


