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Experiment of heat transfer in oscillating turbulent flow in a pipe with constant heat flux

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Abstract: Heat transfer characteristics of oscillating turbulent air flow in a pipe heated with constant heat flux were experimentally investigated. The experiments were performed over a range of 245.7 to 902 of the kinetic Reynolds number and 25 to 175 of the dimensionless oscillation amplitude. The effects of these two dimensionless parameters were analyzed. The results show that the cycle-averaged local Nusselt number increases with both the kinetic Reynolds number and the dimensionless oscillation amplitude. The space-cycle averaged Nusselt number also effectively increases with the kinetic Reynolds number and the dimensionless oscillation amplitude. Based on the experimental data, a correlation equation of the space-cycle averaged Nusselt number for air in terms of these two dimensionless parameters has been obtained.

Key words: oscillating turbulent flow; heat transfer; kinetic Reynolds number; Nusselt number; constant heat flux

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Oscillating flows are common physical phenomena frequently encountered in such engineering applications as Stirling engines, cryocoolers and other periodical processes in thermal and chemical systems. Understanding the process of heat and mass transport in oscillating flow is a topic of interest for these applications.

Previous studies of oscillating flows mainly concentrated on flow structures^[1-9]. And a great deal of attention has been devoted to the study of transition from laminar flow to turbulence flow. For example, Hino et al.^[3] studied transition by hot-wire measurements. Their measurements showed a laminar-like flow during the acceleration phase of the half cycle and a turbulent-like flow during the deceleration phase. Akhavan et al.^[7] investigated experimentally the transition of oscillating flows in circular pipes. Seume et al.^[8] experimentally studied the transition in an oscillatory pipe flow with the parameters range covered in the heat exchangers of Stirling engines

and cryocoolers.

Recently, more studies have been concentrated on heat transfer of oscillating pipe flows. To more adequately predict the Stirling engine performance, it is necessary to have models of local heat transfer between oscillatory flow and the wall of heat exchanger. Fluid flow and heat transfer of laminar forced convection have been widely studied numerically by Zhao and Cheng^[10-12]. Experimental results show that the oscillating flows can enhance heat transfer. Chai et al.^[13] showed that the heat exchange capability of the oscillating heat pipe heat exchanger is about 3 times higher than that of a common tube heat exchanger. However, their measurement is in the laminar flow region. In addition, they did not establish the relationship of the enhancement of heat transfer with the governing parameters such as the Reynolds number. YAN et al.^[14] used the Reynolds-averaged Navier-Stokes (RANS) method with Saffmann's turbulence

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flow model to solve the time-dependent turbulent Navier-Stokes and energy equations for oscillating pipe flows.

In this paper, the heat transfer of the oscillating turbulent flow in a pipe subjected to a constant wall heat flux was experimentally investigated. The effects of displacement and frequency on heat transfer in oscillating flow in a pipe were analyzed. Based on the experimental data, a correlation equation of the space-cycle averaged Nusselt number with the dimensionless parameters of A_0 and Re_0 was obtained. The correlation equation will be useful for the design of heat exchangers in Stirling engines and cryocoolers as well as the estimate of the machines' performance.

1 Experimental setup and instrumentation

1.1 Experimental facility

Fig. 1 shows the schematic of the oscillating flow facility and the experimental setup. The facility consists of three major parts: oscillating flow generator, test section and coolers. The oscillating flow generator is a mechanism that generates a sinusoidal oscillating flow. The mechanism consists of a scotch-yoke with adjustable stroke lengths and compression cylinder (AIR-TAC SCD100 $\times$ 225, 100 mm in diameter and 225 mm in length). The scotch-yoke was driven

by an alternating current motor (1.5 kW) with variable speed capabilities to drive the piston forward and backward sinusoidally. By changing the motor speed and the center off-set of the scotch-yoke, oscillating flows with different frequencies and amplitudes were generated. In the present experiments, oscillating frequencies from 1 Hz to 10 Hz were employed. The locations of 10 mm to 70 mm with every 10 mm distance from the center of the plate were selected and the oscillating flows with the maximum displacement of 20 mm to 140 mm with every 20 mm distance were provided. The test section was a long copper tube, 1000 mm in length and 20 mm in diameter. The test section was connected by two identical coolers having the same inside diameter as the test section and 600 mm in length. The test section was heated constantly by a wrapped-around insulated flexible heater, which was connected to a transformer (TDGC-2/0.5) supplying with 100, 150 V and 200 V. A layer of glass wool insulation of 50 mm thickness was applied over the flexible heater, followed by a thin sheet of aluminum foil. The two coolers were fabricated out of two concentric tubes with water as the coolant.

Because both the geometry and the imposed hydrodynamic and thermal boundary conditions were symmetric with respect to the mid-section of the heated tube, the sensors were installed

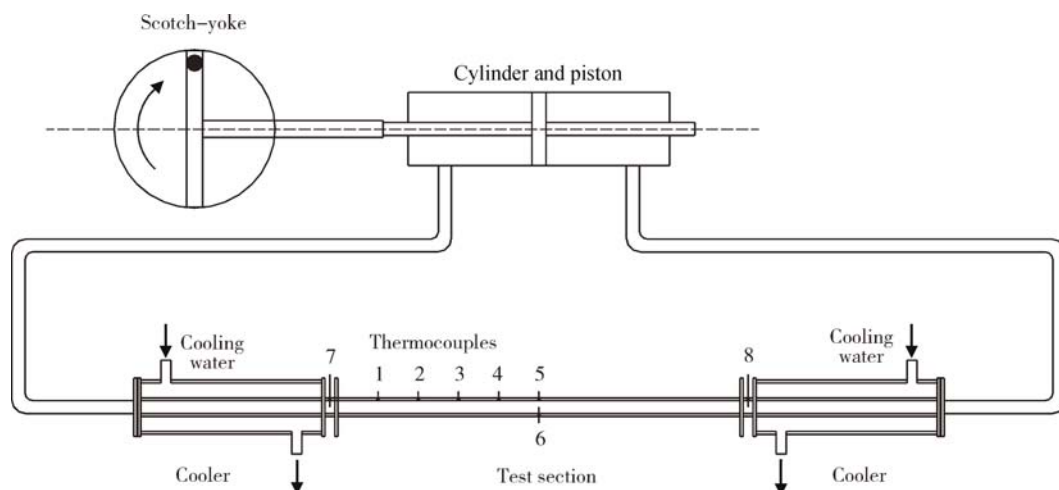


Fig. 1 Schematic of oscillating flow facility and experimental setup

only in the left-hand side of the test rig. Five T-type thermocouples were used to measure the test section wall temperatures. Two fine thermocouples were carefully installed inside the two mixing chambers to measure inlet and outlet fluid temperatures. Another thermocouple was used to measure the centerline fluid temperature in the middle of the test section. The cooling water inlet and outlet temperatures were measured by the thermometers.

1.2 Experimental procedures and instrumentation

Before the commencement of the experiments, all thermocouples were carefully calibrated. By looping the test section to the oscillating flow regenerator, heat transfer experiments of oscillating flow could be conducted. Cooling water with room temperature was forced through the coolers installed to cool the dissipated heat. The frequencies of oscillating flow were adjusted by controlling the motor speed. For fixed amplitude, the experiments were carried out by increasing the oscillating frequency while keeping the power input unchanged. To obtain a cyclic steady state, the temperatures on the pipe wall were monitored by the data acquisition system. More than 200 cycles of data were obtained under sampling rate 200 Hz for different oscillating frequencies. All the signals were collected by a data acquisition system consisting of voltage amplifier (Beijing Sailing STWB-XT), 12-bit A/D card (National Instruments USB-6008), control software (Labview) and computer.

2 Data collection and uncertainty analysis

In oscillating flow heat transfer, the dimensionless parameters kinetic Reynolds number Re_ω (i. e. dimensionless oscillatory frequency) and dimensionless oscillation amplitude A_0 are playing very important roles. The kinetic Reynolds number of air flowing in a pipe of diameter D is given by

$$Re_\omega = \frac{\omega D^2}{\nu} \quad (1)$$

Where, ω is the angular oscillation frequency, and ν is the kinematic viscosity.

The dimensionless oscillation amplitude A_0 is defined as

$$A_0 = \frac{x_{\max}}{D} \quad (2)$$

Where, $x_{\max}$ is the maximum fluid displacement in the pipe.

The critical parameter^[9] by which flow transition occurs in oscillating flow is

$$\beta_c = (A_0 \sqrt{Re_\omega})_{\text{cri}} \quad (3)$$

Critical values β_c obtained by various investigators^[3-9] are near 750 to 800. In present experiments, the kinetic Reynolds number is from 245.7 to 902, and the dimensionless oscillation amplitude is from 25 to 175. And most of experimental cases are in the turbulent region.

To minimize the data reduction uncertainty, the cycle-averaged method was employed to reduce the present experimental data over 200 complete cycles. The cycle-averaged local Nusselt number was calculated by

$$Nu_x = \frac{h_x D}{\kappa} = \frac{q_w}{T_{w,x} - T_{in}} \frac{D}{\kappa} \quad (4)$$

Where, q_w is the wall heat flux; κ is the thermal conductivity; h_x , $T_{w,x}$ and T_{in} are the heat transfer coefficient, the cycle-averaged local wall temperature and the bulk air inlet temperature, respectively.

And the space-cycle averaged Nusselt number was calculated by

$$\overline{Nu} = \frac{q_w D}{\kappa (\overline{T}_w - \overline{T}_{in})} \quad (5)$$

Where, $\overline{T}_w$ and $\overline{T}_{in}$ are the space-cycle averaged wall temperature and space-cycle averaged temperature at the mixing chamber, respectively.

The uncertainties of the measured data can be classified into two groups: random uncertainties, which can be treated statistically, and systematic uncertainties, which can not be treated in the same way. With careful experimentation, systematic uncertainty can be minimized. The accuracies of the thermocouple temperature readings are within $\pm 0.1^\circ\text{C}$. After the cycle-averaging process, uncertainty of temperature is

3.0%. The uncertainty of wall heat flux q_w is estimated to be 3.0%. The uncertainty Nu_x is determined by the method described by^[15]

$$\delta_{Nu} = \sqrt{\left(\frac{\partial Nu}{\partial q_w} \delta q\right)^2 + \left(\frac{\partial Nu}{\partial T_w} \delta T_w\right)^2 + \left(\frac{\partial Nu}{\partial T_{in}} \delta T_{in}\right)^2} \quad (6)$$

In the present experiments, the uncertainties of the measured data were assumed to be independent and random with normal distribution. Using Eq. (6), the uncertainty of Nu_x is calculated to be 5.5%.

3 Results and discussions

Fig. 2 presents the cycle-averaged temperature distribution on the wall along the axial direction in oscillating flow with different frequencies and wall heat fluxes. The wall temperatures located at the entrance is lower than that at the center of the test section. The local wall temperature distribution curves are convex with the maximum point at the center of the test section.

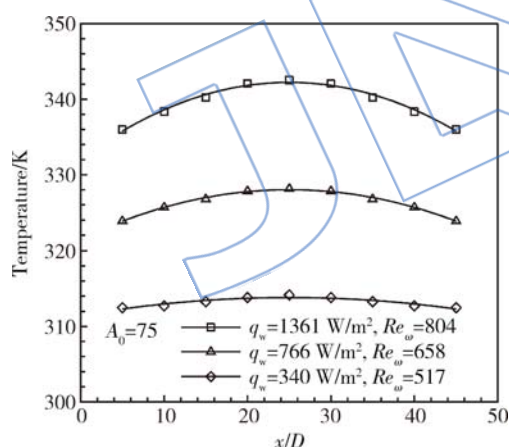


Fig. 2 Cycle-averaged temperature distribution on wall in oscillating flow

Fig. 3 shows the cycle-averaged local Nusselt number distribution along the dimensionless axial distance in oscillating flow with the different frequencies and wall heat fluxes. It can be seen that the cycle-averaged local Nusselt number decreases as dimensionless location x/D approaches the center of the test section. The local Nusselt number distribution curves are concave with the minimum value around the center of the

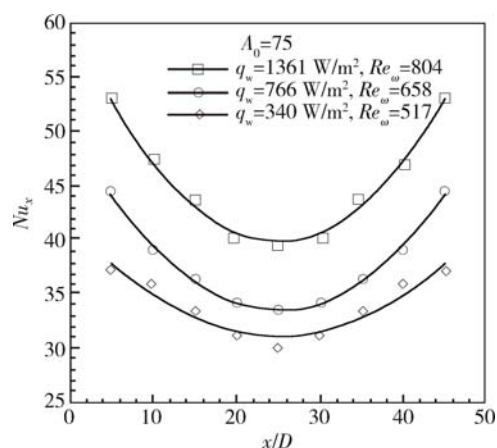


Fig. 3 Cycle-averaged local Nusselt number distribution in oscillating flow

test section.

The effect of kinetic Reynolds number on the cycle-averaged wall temperature for dimensionless amplitude $A_0 = 75$ with the wall heat flux $q_w = 766 \text{ W/m}^2$ is presented in Fig. 4. It can be observed that the cycle-averaged wall temperature profile decreases with the increase of the kinetic Reynolds number. It implies that high oscillatory frequency corresponds to low cycle-averaged wall temperature.

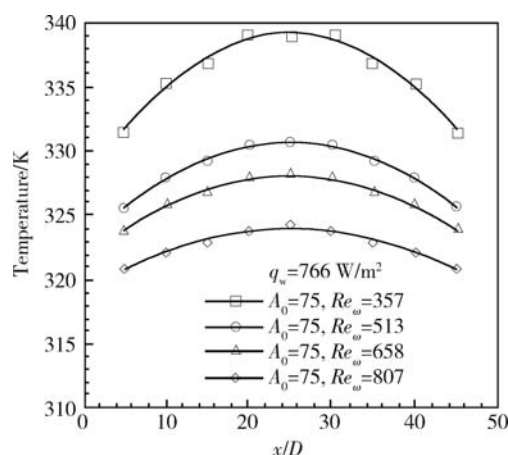


Fig. 4 Effect of kinetic Reynolds number on heat transfer of oscillating flow: cycle-averaged wall temperature distribution

Fig. 5 shows the effect of kinetic Reynolds number on the cycle-averaged local Nusselt number of oscillating flow. It is shown that the cycle-averaged local Nusselt number increases with the increase of kinetic Reynolds number. It is also seen that for varying oscillatory frequencies,

the distribution curves of cycle-averaged local Nusselt number are concave with the center of the test section.

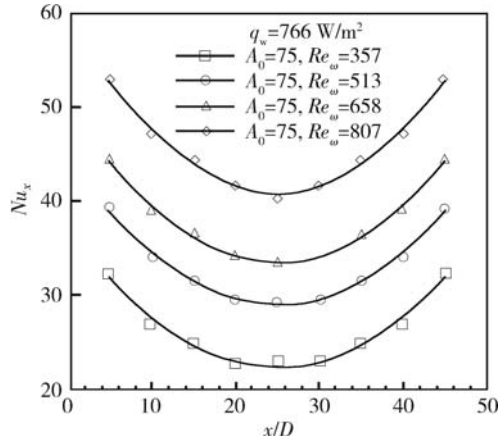


Fig. 5 Effect of kinetic Reynolds number on heat transfer of oscillating flow; cycle-averaged local Nusselt number distribution

Fig. 6 shows the effect of dimensionless oscillation amplitude of flow displacement A_0 on heat transfer of oscillating flow with different kinetic Reynolds numbers. It can be seen that the temperature profile decreases with the increase of the dimensionless oscillation amplitude of flow displacement for various kinetic Reynolds numbers. For different displacements, the same shape of temperature distribution curve was obtained as shown in Fig. 4.

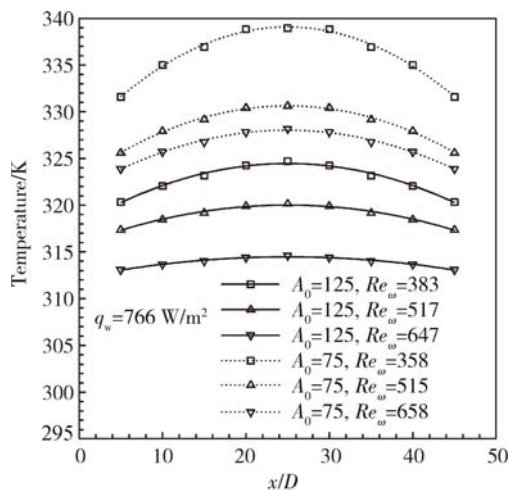


Fig. 6 Effect of dimensionless oscillation amplitude on heat transfer of oscillating flow; cycle-averaged wall temperature distribution

Fig. 7 presents the effect of dimensionless oscillation amplitude of flow displacement A_0 on the distribution of cycle-averaged Nusselt number at different kinetic Reynolds numbers. It is clear that the profile of cycle-averaged local Nusselt number for large A_0 is higher than that for small A_0 . It can be deduced that higher heat transfer rates can be obtained by larger displacement oscillating flows with high kinetic Reynolds number.

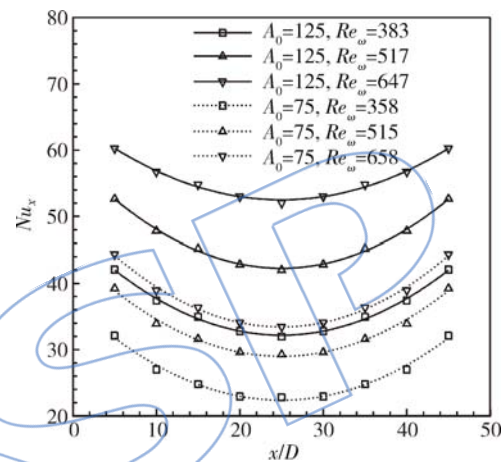


Fig. 7 Effect of dimensionless oscillation amplitude on heat transfer of oscillating flow; cycle-averaged local Nusselt number distribution

In order to evaluate the effects of dimensionless oscillation amplitude A_0 and kinetic Reynolds number Re_w on the total heat dissipation rate in a pipe subjected to oscillating flow, the space-cycle averaged Nusselt number was used to calculate the averaged Nusselt number of the whole length of the section. Fig. 8 presents the space-cycle averaged Nusselt number versus kinetic Reynolds number with different dimensionless amplitudes. Generally, the space-cycle averaged Nusselt number increases with dimensionless parameters of both A_0 and Re_w .

Using the least squares method, the data in Fig. 8 can be collapsed to a fitting line as shown in Fig. 9. The resulting correlation equation of the space-cycle averaged Nusselt number with the dimensionless oscillation amplitude and kinetic Reynolds number for oscillating flow with Re_w of 250 to 900 and A_0 of 75 to 150 in a circular

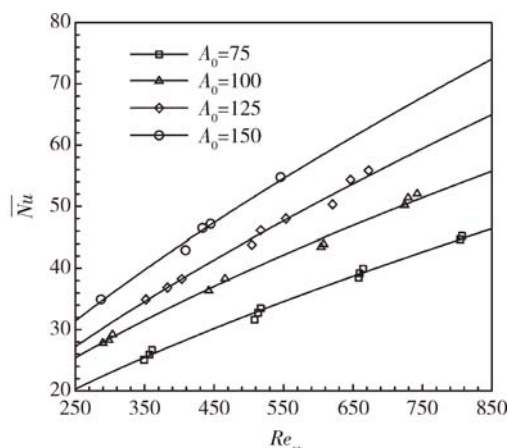


Fig. 8 Effect of A_0 and Re_w on space-cycle averaged Nusselt number

pipe is

$$\overline{Nu} = 0.0306 A_0^{0.6381} Re_w^{0.6778} \quad (7)$$

Eq. (7) indicates that higher heat transfer performance can be obtained by oscillating flow with relatively low frequency and high displacement.

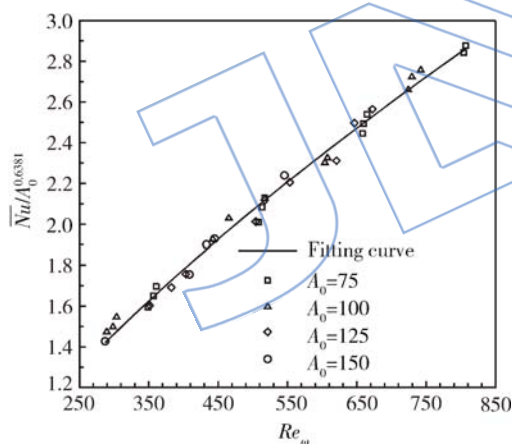


Fig. 9 Correlation equation of space-cycle averaged Nusselt number with A_0 and Re_w

4 Conclusions

The effects of frequency and oscillation amplitude on heat transfer characteristics of turbulent oscillating pipe flow are experimentally investigated. The pipe was subjected to a constant heat flux and the dissipated heat was cooled by the coolers to ensure the steady inlet condition. From the obtained results, the following may be

concluded:

1) The cycle-averaged local Nusselt number decreases as dimensionless location x/D approaches the center of the test section. The local Nusselt number distribution curves are concave with the minimum value around the center of the test section.

2) The cycle-averaged local Nusselt number increases with the increase of both kinetic Reynolds number and dimensionless oscillation amplitude. It implies that higher heat transfer rates can be obtained by larger displacement oscillating flows with high kinetic Reynolds number.

3) A correlation equation of the space-cycle averaged Nusselt number with the dimensionless oscillation amplitude and kinetic Reynolds number for oscillating flow is obtained. The correlation equation will be useful for the design of heat exchangers in Stirling engines and cryocoolers as well as the estimate of the machines' performance.

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