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Effect and heat transfer correlations of finned tube heat exchanger under unsteady pulsating flows

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ABSTRACT

The enhancement of heat transfer for finned tube heat exchangers by using a pulsating flow was studied. The amplitude, force frequency, Reynolds number and blockage ratio of heat exchangers in pulsating air flows were investigated. The experimental setup of the unsteady pulsating flow was designed and constructed to experiment with such parameters. The air was used as a working fluid. Experiments on unsteady pulsating flows were carried out in the range of $10 \leqslant f_p \leqslant 50$ Hz and $13.33 \leqslant A \leqslant 15.35\%$. The results indicate that both the force frequency and amplitude of the pulsating flow play important roles in heat transfers. The heat transfer coefficient of the fin heat exchanger was enhanced to be higher than that of the steady flow condition. The new empirical correlation can predict the experimental applicability of the data within 12.3%.

1. Introduction

The characteristics of unsteady pulsating flows have potential to enhance the heat transfer coefficient in applications such as electronics cooling, compact heat exchanger, air conditions, and heat transfer of food as well as several engineering applications. A pulsating flow is one kind of oscillating flow. Oscillating flows have a reciprocating flow, such as gas flowing inside a pulse tube refrigerator or thermoacoustic systems [1,2]; that is, systems in which the flow of the gas path reverses periodically. In other words, the direction of flow never reverses in a pulsating flow. A pulsation flow affects the heat transfer in a device. Suksangpanomrung et al. [3] showed that the heat transfer rate of the rectangular bluff plate in a channel is enhanced by using the pulsating flow and indicated that the heat transfer rate higher than uniform steady flow. However, both the amplitude and force frequency factors of a pulsating flow must be optimal too.

Several researchers have analyzed inline pulsation flows. Moon et al. [4] carried out experimental studies on heat transfers of several heated rectangular block arrays inside a channel and indicated that air-side heat transfer coefficients increased by flow pulsation effects. Ji et al. [5] investigated the relation between the heat transfer rate and the lock-on phenomenon; they showed that under the lock-on regime, the air-side heat transfer coefficient is increased. In the case of inline flow pulsation, the lock-on phenomenon occurs when the force frequency of flow pulsation is twice the natural frequency [6]. In numerical work, the simulation of flow pulsation clearly revealed that under the influence of pulsation, the flow and thermal field phenomenon will instability becomes cause unsteady flows [7,8]. In addition, the influence of the pulsating flow can be applied by combination with the nanofluid technique to enhance the heat transfer of devices. Rahgoshay et al. [9] performed two-dimensional simulations in a circular tube with an isothermal wall. They concluded that the pulsating flow inside the tube influenced heat transfer enhancement slightly more than it increased the Reynolds number and volume fraction of nanofluid. Akdag et al. [10] reported that using a low pulsating frequency while increasing both pulsating amplitude and nanoparticle volume fraction leads to an increase in the Nusselt number of a wavy channel.

This study focused on the case of the finned tube heat exchanger type, which is used in a wide variety of applications of thermal-fluid systems. In general, the heat transfer coefficient on such a device is enhanced by using a fin configuration, such as a wavy fin and louver fin configuration [11–13], or a high fluid velocity per unit time to increase the particles of gas that make contact with the fins. The straight fin configuration, which has a uniform cross-sectional area, is one of the most popular surfaces, but it is still being developed because it has lower performance. The purpose of this work is to experimentally examine the enhancement of heat transfer under a pulsating flow through several straight fins of a heat exchanger; the amplitude (A), frequency (f_p), Reynolds

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Nomenclature

A	oscillating amplitude of axial inlet velocity
Br	blockage ratio
c_p	specific heat of fluid (J·kg^{-1}·K^{-1})
D_h	hydraulic diameter (m)
f_p	dimensional forcing frequency (Hz)
h_o	local heat transfer coefficient (W·m^{-2}·K^{-1})
$\bar{h}_o$	average heat transfer coefficient for steady flow (W·m^{-2}·K^{-1})
H	height of test section (m)
j	Colburn j-factor
k_f	thermal conductivity of fin (W·m^{-1}·K^{-1})
k	thermal conductivity of fluid (W·m^{-1}·K^{-1})
L	length of fin (m)
L_o	length of heat exchanger (m)
Nu_p	average (time-mean) Nusselt number for pulsating flow
Nu_s	average Nusselt number for steady flow
$Nu\,(x, t)$	local instantaneous Nusselt number
P	perimeter (m)
P_f	fin pitch (m)
Pr	Prandtl number
q	heat transfer rate (Watt)
Re	Reynolds number
S	length of the heat exchanger from inlet of the test section (m)
t	time (s)
T	temperature (K)
T_b	base temperature of the fin (K)
T_∞	air temperature (K)
U_i	inlet velocity (m·s^{-1})
U_o	time-averaged velocity of inlet flow (m·s^{-1})
U_{rms}	RMS velocity (m·s^{-1})
W	fin width (m)

Greek symbols

δ	fin thickness (m)
ρ	density (kg·m^{-3})
v	kinematic viscosity (m^2·s^{-1})
μ	dynamic viscosity, (kg·m^{-1}·s^{-1})
τ	cycle time (rad)
ω	angular frequency (rad·s^{-1})

Subscripts

D_h	hydraulic diameter
p	pulsating component
s	steady-state component
W	fin width

number and blockage ratio of finned tube heat exchangers were the principal factors studied. Based on the experimental data, a new empirical heat transfer correlation was developed to predict the coefficient of heat transfer in a pulsating flow.

2. Experimental apparatus

Fig. 1 shows a schematic diagram of the experimental setup. The test setup consisted of a test section and a data acquisition system. In the test section of an open wind tunnel was installed a heat exchanger fabricated with a combination of Plexiglas and metal sheets. The Plexiglas plates were 3 mm thick, 210 mm wide and 990 mm long. Two sheets of Plexiglas were used to observe the flow characteristic inside the test section. The top and bottom of the channel test section were metal sheets 2 mm thick, 210 mm wide and 990 mm long. The test section was attached to a construction cone, which was designed and fabricated to take in a large volume of main airflow, reduce it and introduce it to the test section as a small-volume, high-velocity airflow with low turbulence. An axial AC fan, model TA 20060, with a maximum airflow volume of 8.07 m^3/s, was used to draw air from the surroundings through a honeycomb into the test section and reject it back to the surroundings. The honeycomb was used to create a uniform axial direction. An AC power controller was connected to the axial fan to adjust the air velocity at the inlet of the test section. The range of turbulence intensity of the air inlet within the steady flow condition test was less than 0.8%. A hot-wire anemometer was used to measure the air velocity distribution inside the test section. An oscillating pulsation flow was made by using a woofer loudspeaker 381 mm in diameter. A loudspeaker was placed on a cone-shaped chamber and attached to the bottom surface of the metal sheet, which was drilled with a 190 mm hole (see Fig. 1). A function generator, which was used to provide the required sound wave signal in the test section, was connected to a signal amplifier to amplify the signal and then send it to the loudspeaker. The water loop consisted of a 200-liter storage tank, an electric heater and a DC water pump to control the water flow rate. The storage tank was covered well by microfiber insulation 75 mm in thickness, and the water pipe was insulated by rubber insulation with a thickness of 9.5 mm. Two heat exchangers were built to test the effect of flows over the plain fins. Fig. 2 shows a schematic of the finned tube heat exchanger used in this work. The first type of heat exchanger, called the HX-01, and the second, the HX-02, were constructed from copper. The dimensions of the two heat exchangers are listed in Table 1. Fig. 3(a) shows the position and detail of the heat exchanger inside the test section and includes

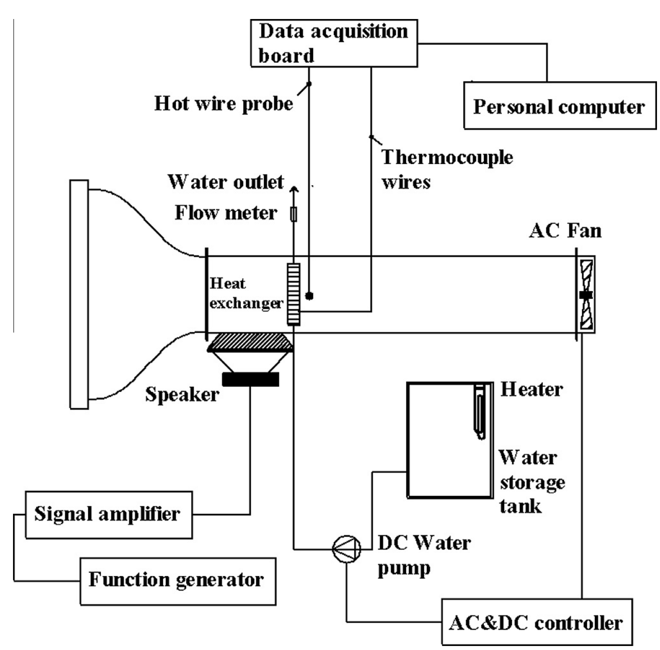

Fig. 1. Schematic diagram of the experimental setup.