2006

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ABSTRACT

Offset-strip fin heat exchangers take advantage of boundary layer restarting to enhance heat transfer over that of plain-fin heat exchangers. Typically, these heat exchangers are operated at low Reynolds numbers (Re < 1,000). At higher Reynolds numbers, vortex shedding and turbulent flow in the array may cause further enhancement of heat transfer, but with an increase in pressure drop. The additional pumping power required at higher Reynolds numbers is unsuitable for conventional air-conditioning and automotive applications. However, the compactness and low weight afforded by the high heat transfer coefficients is extremely attractive for some emerging aerospace applications, such as advanced air-breathing propulsion systems for space launch. Experiments have been undertaken to measure the heat transfer and friction factor of an offset-strip fin array at Reynolds numbers up to 100,000, and the results are discussed in this paper.

1. INTRODUCTION

Offset-strip fin heat exchangers have been in use for decades. This type of compact heat exchanger takes advantage of boundary layer restarting to enhance heat transfer over that of plain-fin heat exchangers. Since the average boundary-layer thickness decreases significantly when offset-strip fins are used, the convection coefficient increases. At moderate Reynolds numbers ($Re > 700$), vortex shedding can occur in the array, causing further heat transfer enhancement. Both of these enhancement mechanisms also increase pressure drop across the array. Heat transfer is typically proportional to the flow velocity, but the power required to move the flow (pumping power, or fan power) is proportional to the velocity cubed; therefore, offset-strip fin heat exchangers are usually operated at Reynolds numbers much below 1,000 in order to manage heat-duty-pumping-power design tradeoffs. At these low Reynolds numbers, the air-side flow is steady and laminar. The thermal-hydraulic performance of offset-set strip fin heat exchangers has been studied extensively for flows in this regime. A thorough literature review is provided by Manglik and Bergles (1995). This review includes the development of the most widely used correlations for friction factor and Colburn *j* factor in offset-strip fin arrays. These correlations were developed using data in the range $120 < Re < 10{,}000$.

The research undertaken to date to study the flow and heat transfer properties of the offset-strip fin heat exchanger has been performed almost exclusively in the range *Re* < 20,000, with a few additional data points up to approximately *Re* = 30,000. The available literature does, however, show that operating offset-strip fin heat exchangers at higher Reynolds numbers may lead to a sizeable enhancement of heat transfer. Joshi and Webb (1987) performed flow visualization experiments in this type of fin array. They observed four distinct flow patterns. At the lowest Reynolds numbers, the wake from the upstream fin was smooth and laminar. Next, oscillations occurred where the wake impinged on the downstream fin. The third regime was characterized by unsteadiness in the entire region between fins. At the highest Reynolds numbers, vortices were shed in the array. Mochizuki *et al.* (1988) also performed flow visualization experiments in this type of fin array. They found that steady laminar flow was present throughout the array at lower Reynolds numbers. Vortex shedding was observed in the downstream section of the array at *Re* ~ 1,000 and moved upstream as Reynolds number increased. Turbulent flow was seen in the downstream part of the array at *Re* ~ 2,200 and also moved upstream as Reynolds number increased. DeJong and Jacobi (1997) performed flow visualization and mass transfer experiments in offset-strip fin arrays. They observed steady, laminar flow in the range *Re* < 500. Vortex shedding was seen at higher Reynolds numbers, starting in the downstream fins and moving upstream as Reynolds number increased. Turbulence was observed in the downstream fins for *Re* < 1,000, moving upstream as Reynolds number increased. They compared the mass transfer results to the theoretical interrupted-plate solution. The mass transfer was lower than expected in the steady regime due to thermal wake effects. In the vortex shedding regime, however, mass transfer was significantly higher than expected from the theoretical interrupted-plate solution.

By operating offset-strip fin heat exchangers in a vortex-shedding, turbulent regime, much higher convection coefficients than observed at low Reynolds numbers (*Re* < 1,000) are expected. In the research reported in this paper, experiments were performed to measure the friction factor and heat transfer coefficients of an offset-strip fin array at Reynolds numbers in the range 10,000 < *Re* < 100,000, which is up to an order of magnitude greater in Reynolds number than that reported in previous work. The significant heat transfer enhancements expected at these air-side flow rates could be of use in applications where compactness and low weight are of extreme importance and where the high air-side pressure drop is tolerable. The results of mass transfer experiments undertaken using the same array are discussed in an earlier paper (Michna *et al.*, 2005).

2. METHOD

Since no information on the performance of the offset-strip fin array is available for Reynolds numbers greater than approximately *Re* = 20,000, pressure drop and heat transfer experiments were undertaken to characterize the thermal-hydraulic performance of this array at Reynolds numbers up to approximately *Re* = 100,000.

2.1 Apparatus

The pressure drop and heat transfer experiments were performed in an open loop wind tunnel. The tunnel was connected to a compressor system capable of delivering dry air indefinitely at 1 kg/s and approximately 300 K, at pressures up to 6.8 atm. The air from this system entered the laboratory through a pressure regulator, which was used to control flow rate, and routed to the tunnel through a 76 mm diameter pipe. The air exited the pipe radially into a settling chamber before passing through a number of wire screens and a hexagonal honeycomb section. The flow finally passed through an 11:1 area ratio contraction section before entering the test section. Freestream velocities ranging from 2 to 50 m/s could be achieved in the test section. Turbulence intensity in the test section was measured with a hot-wire anemometer to be 4% or less for all velocities tested.

The test section was 152 mm x 146 mm, and was constructed out of steel. The fins in the array had a height of 146 mm, a length of 25.4 mm, and a width of 3.18 mm. The fin pitch was equal to the length of the fin. The hydraulic diameter of the fin was calculated to be 34.8 mm. The geometry of the array used in the experiments is shown in Figure 1. Four static pressure taps were located in the center of each side of the test section 25.4 mm upstream and downstream of the array. Each set of four taps was connected with tubing, and the tubing was connected to an inclined manometer. A Pitot-static tube was inserted into the flow upstream of the array in the center of the test section. It was connected to a second inclined manometer to measure the dynamic pressure of the air flow.

Solid aluminum fins were inserted into the test section for the pressure drop experiments. For the heat transfer experiments, a set of steel fins consisting of two fin halves with a thin-film heater in between was inserted into the

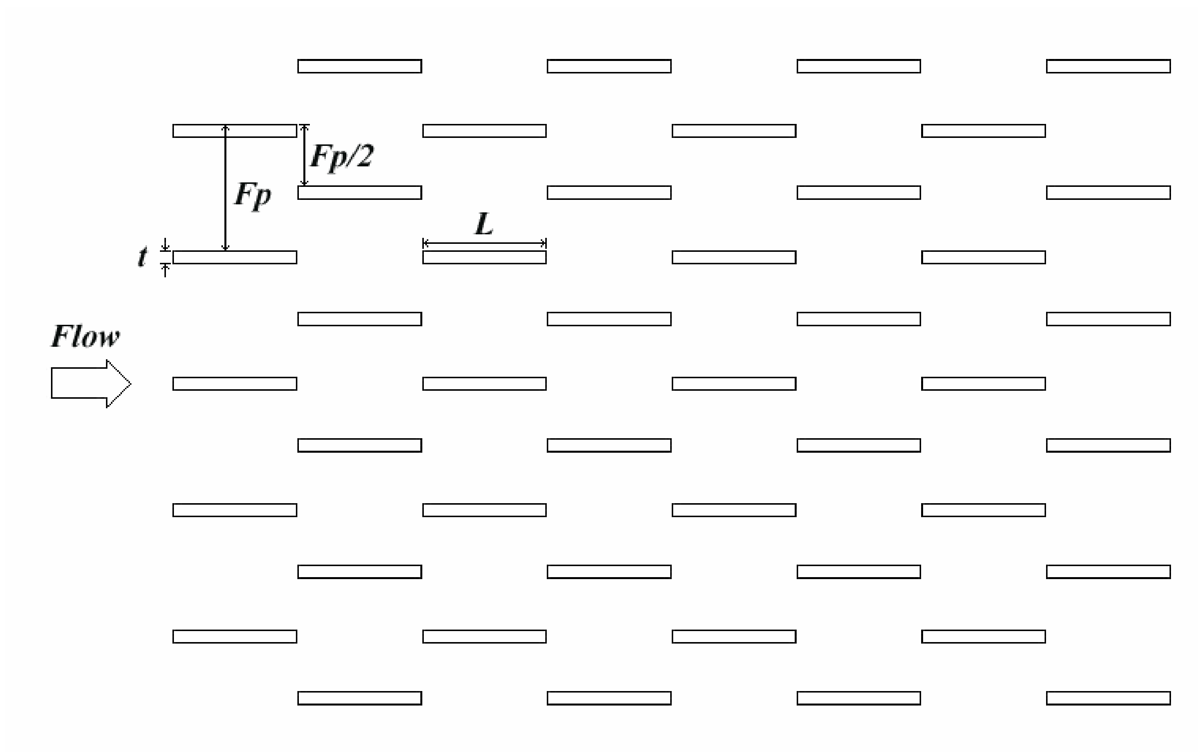

Figure 1. The geometry of the offset-strip fin array under investigation. The array height (into the page) was 146 mm, F_p = 25.4 mm, L = 25.4 mm, and t = 3.18 mm.

test section. The thin film heaters had a nominal resistance of 9 ohms and were connected in series. A 320 volt power supply was used to supply the power to the heaters. Although all of the fins contained thin-film heaters, two of the heated fins had five thermocouples imbedded in them to measure the surface temperature of the fin. Detailed geometry of these fins is shown in Figure 2. These fins were placed in the center fin location of two adjacent rows so the effects of the side walls were mitigated. These fins were then moved so that the heat transfer coefficient from each of the eight rows in the array could be measured. The upstream temperature of the flow was measured by

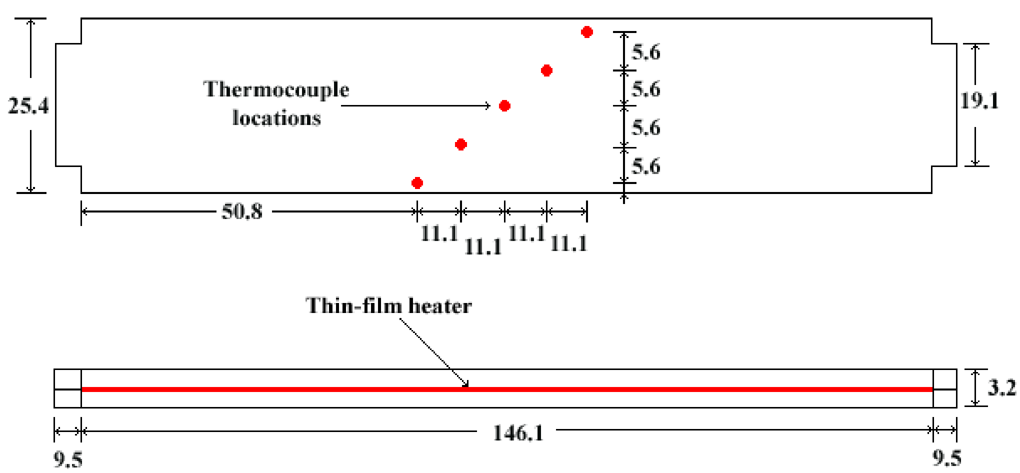

Figure 2. Geometry of the measurement fins. All dimensions are in mm.

another set of five thermocouples. The voltage and current being applied to each of the measurement fins was recorded so the power of the heaters could be determined.

2.2 Experimental Procedure

For the pressure drop experiments, the rate of air flow through the test section was controlled with the pressure regulator. The dynamic pressure and temperature of the flow, the pressure drop across the test section, and the room pressure were recorded for flows with Reynolds numbers in the range $5{,}000 < Re < 120{,}000$.

For the heat transfer experiments, the test section was very well insulated, and the rate of air flow through the test section was controlled with the pressure regulator. The dynamic pressure of the flow and the room pressure were recorded. The voltage and current applied to the whole set of fins and the voltage across each of the heaters in the measurement fins was recorded. The tunnel was run long enough so that steady state could be reached. Finally, the thermocouple temperatures were recorded using a National Instruments data acquisition card and a Labview Virtual Instrument program at a rate of 500 Hz for 2 minutes. The average over those 2 minutes was then calculated. A total of 15 temperatures (5 upstream and 5 for each of the two measurement fins) were recorded.

2.3 Data Reduction

The properties of air were calculated from the temperature of the upstream air flow and the average pressure of the air in the array. The frontal velocity of the air, U_{fr}, was determined from the dynamic pressure measured by the Pitot-static tube, and the velocity in the test section, U_c, was calculated from the frontal area, A_{fr}, and minimum flow area, A_c, using Equation (1). The hydraulic diameter, D_h, was calculated using Equation (2).

$$U_c = U_{fr}\left[\frac{A_{fr}}{A_c}\right] \tag{1}$$

$$D_h = \frac{4A_c}{\left(A_t / L_{array}\right)} = \frac{2wL\left(F_p - t\right)}{w\left(L+t\right)+L\left(F_p - t\right)} \tag{2}$$

The fanning friction factor, f, was calculated from the pressure drop across the array, ΔP_{array}, using Equation (3). This equation accounts for the effects of entrance and exit losses, as well as the acceleration of the flow.

$$\Delta p_{array} = \frac{\rho_1 U_c^{\,2}}{2}\left[\left(K_c + 1 - \sigma^2\right) + 2\left(\frac{\rho_1}{\rho_2} - 1\right) + f\left(\frac{4L_{array}}{D_h}\right)\frac{\rho_1}{\rho_m} - \left(1 - \sigma^2 - K_e\right)\frac{\rho_1}{\rho_2}\right] \tag{3}$$

The power of the thin-film heater within the fin, P_{heater}, was simply calculated from the voltage, V, and current, I, of the heater using Equation (4). The heat transfer coefficient was calculated from the local air temperature, $T_{air,local}$, the fin temperature, T_{fin}, the area of the fin, A_{fin}, and the power of the thin-film heater in the fin using Equation (5).

$$P_{heater} = V \cdot I \tag{4}$$

$$P_{heater} = h \cdot A_{fin} \cdot \left(T_{fin} - T_{air,local}\right) \tag{5}$$

Next, the modified Colburn j factor was calculated from the heat transfer coefficient using Equations (6) and (7).

$$Nu = \frac{h \cdot D_h}{k} \tag{6}$$

$$j = \frac{Nu}{RePr^{0.4}} \tag{7}$$

2.4 Uncertainties

Uncertainties were calculated using standard methods (Kline and McClintock, 1953). The uncertainty in the value of *Re* was less than 5% for *Re* > 10,000. The uncertainty in the value of friction factor was less than 10% for *Re* > 10,000 and improved significantly as the Reynolds number increased. The uncertainty in the value of the modified Colburn *j* factor was less than 8% for all measurements.

3. RESULTS AND DISCUSSION

3.1 Pressure Drop

The friction factor results are shown in Figure 3. The measured data are shown with the correlation of Manglik and Bergles (1995). In the range *Re* < 20,000, the data agrees with 10% of the predicted values. The friction factor decreases monotonically as Reynolds number increases in this range. However, the friction factor stops decreasing at this point and exhibits a kind of oscillatory pattern as it increases to $f \sim 0.06$ as the Reynolds number increases to 100,000. This oscillatory pattern is unexpected. However, it does show that the physics of the flow at high Reynolds numbers is quite different than that at low Reynolds numbers. More research needs to be performed to identify the cause of these oscillations.

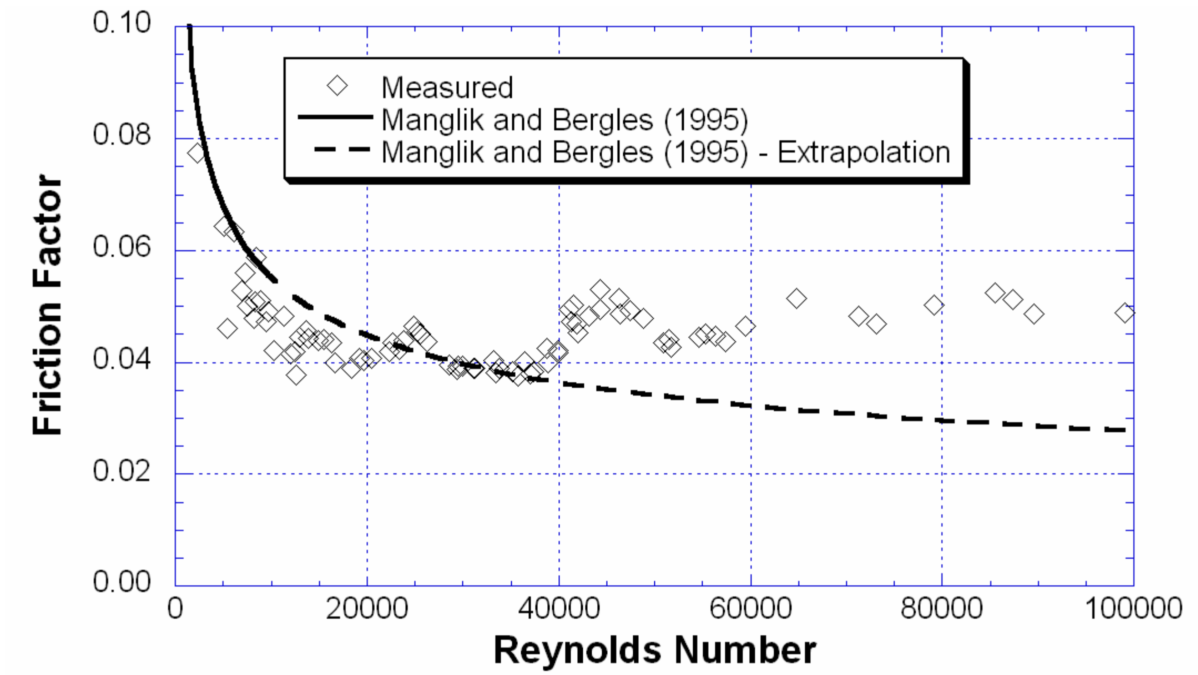

Figure 3. Fanning friction factor, *f*, of the offset-strip fin array as a function of Reynolds number. The correlation of Manglik and Bergles (1995) is included for comparison.

3.2 Heat Transfer

The average heat transfer coefficient in the array is shown in Figure 4. The measured data are shown with the correlation of Manglik and Bergles (1995). It can be seen that the heat transfer coefficient is greater than that predicted by the correlation over the entire range of the experiments. It should be noted, however, that the correlation was developed using data in the range 120 < *Re* < 10,000, and the measured data shown in Figure 4 is in the range *Re* > 11,000. Some of the deviation may also be caused by the fact that the fin thickness to fin length ratio in these experiments is outside of the range of the strip-fin geometries used to develop the correlation. This array perhaps shows more bluff-body behavior as a result, which might explain the 30% deviation at low Reynolds numbers. However, as the Reynolds number increases, the deviation becomes much greater, until the measured heat transfer coefficient is more than twice that predicted by the correlation at *Re* ~ 100,000. The larger deviation at high

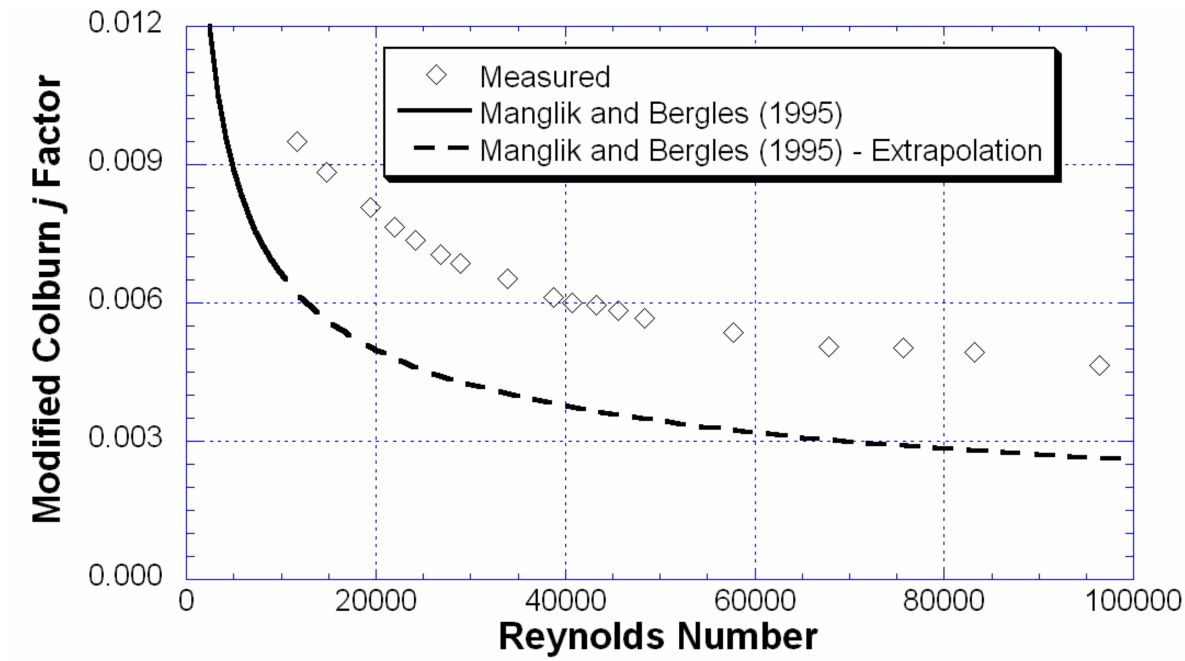

Figure 4. Modified Colburn *j* factor of the offset-strip fin array as a function of Reynolds number. The correlation of Manglik and Bergles (1995) is included for comparison.

Reynolds numbers suggests that a different type of flow is present in the array at these Reynolds numbers – one that is likely turbulent and vortex shedding rather than steady laminar.

Row-by-row heat transfer results are shown in Figure 5. The first row of fins consistently exhibits the lowest heat transfer coefficient, while the second row of fins shows the highest. It is likely that at least the front of first fin is unaffected by any vortex shedding, and that the boundary layer is initially laminar. This would result in the lower heat transfer coefficient at this location. One possible explanation for the second fin having the highest heat transfer

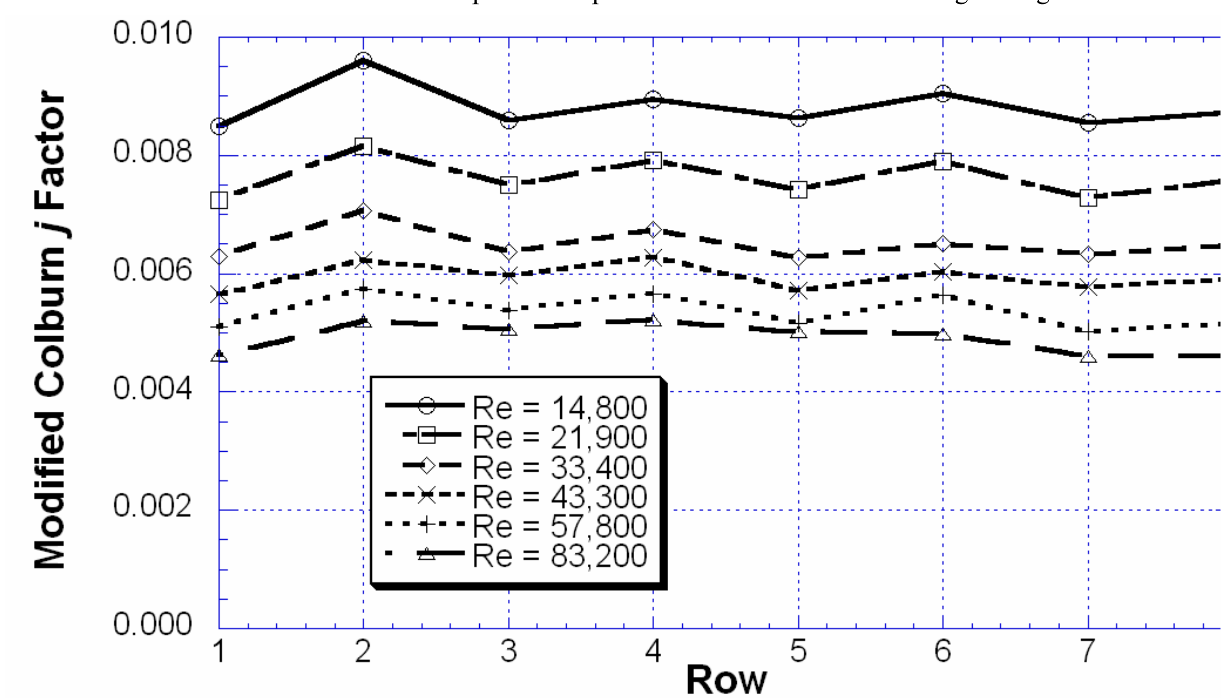

Figure 5. Row-by-row modified Colburn *j* factors at 6 different Reynolds numbers.

coefficient is that it benefits from vortex shedding from the first row and turbulent flow, but unlike the remaining fins downstream, there is no thermal wake from upstream fins to degrade the heat transfer performance.

4. CONCLUSION

In the experiments undertaken to characterize the thermal-hydraulic performance of the offset-strip fin array operating at very high Reynolds numbers, both the pressure drop and heat transfer coefficients were measured to be approximately twice that predicted by the best available correlation, which was developed using data from low-Reynolds number conditions. This may be a result of vortex shedding and turbulent flow at these high Reynolds numbers.

Operation of an offset-strip fin heat exchanger at high Reynolds numbers may be useful in systems where minimizing heat exchanger size or mass is more important than minimizing fan power.

NOMENCLATURE

A_c	minimum free flow area	(m^2)
A_{fin}	heat transfer area of fin	(m^2)
A_{fr}	frontal area	(m^2)
A_T	total heat transfer area	(m^2)
D_h	hydraulic diameter	(m)
f	Fanning friction factor	(–)
F_p	fin pitch	(m)
h	heat transfer coefficient	(W/m^2-K)
I	current	(A)
j	modified Colburn j factor	(–)
k	conductivity of air	(W/m-K)
K_c	entrance loss coefficient	(–)
K_e	exit loss coefficient	(–)
L	length of fin	(m)
L_{array}	length of array	(m)
Nu	Nusselt number	(–)
P_{heater}	heater power	(W)
Pr	Prandtl number	(–)
Re	Reynolds number	(–)
t	fin thickness	(m)
$T_{air,local}$	local air temperature	(K)
T_{fin}	fin temperature	(K)
U_c	average velocity in test section	(m/s)
U_{fr}	frontal velocity	(m/s)
V	voltage	(V)
w	fin height	(m)
ΔP_{array}	array pressure drop	(Pa)
ρ_1	density of air at entrance	(kg/m^3)
ρ_2	density of air at exit	(kg/m^3)
ρ_m	average density of air	(kg/m^3)
σ	area ratio	(–)

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ACKNOWLEDGEMENT

The authors thank the Critical Research Initiative of the University of Illinois at Urbana-Champaign and the National Science Foundation for financial support.