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AIR-SIDE THERMAL-HYDRAULIC PERFORMANCE OF AN OFFSET-STRIP FIN ARRAY AT REYNOLDS NUMBERS UP TO 120 000

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ABSTRACT

Offset-strip fin heat exchangers are used in a variety of applications. These heat exchangers are usually operated at low air-side Reynolds numbers, $Re < 1\,000$, to avoid high pressure drops. However, vortex shedding and turbulent flow present at higher Reynolds numbers can serve as heat transfer enhancement mechanisms. A review of the literature reveals that little is known about the thermal-hydraulic performance of these arrays at $Re > 10\,000$. Pressure drop and mass transfer experiments are performed on a scaled-up offset-strip fin array at air-side Reynolds numbers not previously studied, $5\,000 < Re < 120\,000$. The resulting friction factors and modified Colburn *j* factors are shown to be twice that expected by extrapolating correlations developed using data from low air-side Reynolds number flows. Offset-strip fin heat exchangers operating in this regime are used in applications where the cost of additional pressure drop is acceptable or in liquid to liquid applications.

NOMENCLATURE

A_c = minimum flow area, m^2
A_{nap} = area of naphthalene on fin, m^2
A_T = total heat transfer area, m^2
D_h = hydraulic diameter, m
D_{na} = diffusivity of naphthalene in air, m^2/s
$\overline{(dm/dt)}_i$ = average mass transfer rate of ith fin row, kg/s
f = Fanning friction factor, dimensionless
h = height of fin, m
$\overline{h_m}$ = average mass transfer coefficient, m/s
j = modified Colburn j factor, dimensionless
L = length of fin, m
L_{core} = length of core, m
m_f = final mass of fin, kg
m_i = initial mass of fin, kg
Re = Reynolds number, dimensionless
Sc = Schmidt number, dimensionless
$\overline{Sh}$ = average Sherwood number, dimensionless
t = thickness of fin, m
U_c = velocity at minimum flow area, m/s
U_{fr} = velocity at face of heat exchanger, m/s
$\dot{V}_{air}$ = volumetric flow rate of air, m^3/s

Δm_{exp} = mass sublimed during install/removal, kg
ΔP_{core} = pressure drop across the core, Pa
Δt = time of exposure, s
ρ = density of air, kg/m^3
$\rho_{nap,v}$ = density of naphthalene vapor, kg/m^3
$\rho_{nap,inf}$ = density of naphthalene in the core flow, kg/m^3
υ = kinematic viscosity, m^2/s

INTRODUCTION

Compact heat exchangers are used in a wide variety of applications, including air-conditioning condensers and evaporators and automotive radiators, among others. The offset-strip fin geometry is commonly used for enhancement of the air-side performance of these compact heat exchangers. Under normal operating conditions, the heat transfer enhancement stems from the restarting of the thermal boundary layers as the air flows through the fin array. Since the average boundary layer thickness in the array decreases significantly when offset-strip fins are used, the convection coefficient increases. The thermal-hydraulic performance of offset-strip fin heat exchangers has been studied extensively for low-Reynolds number applications. A thorough review of the literature is provided by Manglik and Bergles (1995). This review includes the development of correlations for friction factor and modified Colburn j factor of offset-strip fin arrays using data taken in the flow range $120 < Re < 10\,000$.

Since these heat exchangers are generally operated at low air-side flow Reynolds numbers to keep the pressure drop low, vortex shedding and turbulence—and the attendant enhancements of heat transfer—are not present. Joshi and Webb (1987) investigated the flow structure in offset-strip fin heat exchangers and observed four different flow regimes. The flow progressed from steady and laminar throughout the array, to having small oscillations at the upstream portion of the fins, to having the entire streamwise region between fins oscillating, and finally to vortex shedding. Mochizuki *et al.* (1988) observed the flow pattern in offset strip-fin arrays to be steady and laminar at low Reynolds numbers, with vortex shedding and finally turbulence appearing as the Reynolds number increased. In flow visualization experiments by DeJong and Jacobi (1997), vortex shedding began to occur in offset-strip fin arrays at air-side flow Reynolds numbers greater than $Re = 700$, and flow in the downstream portion of the array entered the turbulent regime at Reynolds numbers above about $Re = 1\,000$.

By operating offset-strip fin heat exchangers in a vortex-shedding, turbulent regime, much higher convection coefficients are expected. In the research reported in this paper, experiments were performed to measure the heat transfer and pressure drop characteristics of an offset-strip fin array at much higher Reynolds numbers than have been previously investigated. These experiments were performed in the range of $5\,000 < Re < 120\,000$, 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 flow rates could be beneficial in applications where compactness is of extreme importance or the high air-side pressure drop is tolerable.

Initial results from these experiments and possible applications of an offset-strip fin array for use in advanced air-breathing propulsion systems, such as liquid air cycle engines, are discussed in an earlier paper (Michna et al., 2004).

METHOD

Since no information on the performance of the offset-strip fin array at Reynolds numbers higher than

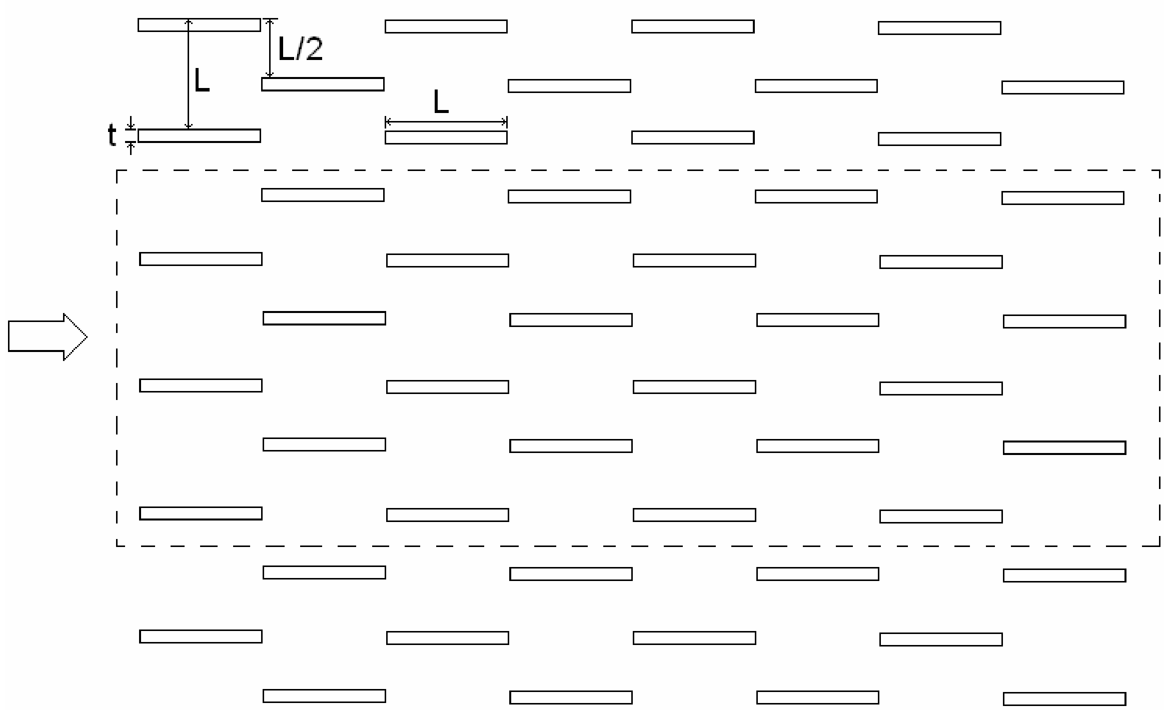

Figure 1. The scaled-up offset strip fin array under investigation. The array height (into the page) was 152 mm, L=25.4 mm, and t = 3.18 mm. The fins within the dotted box were used for measurements in the naphthalene sublimation experiments.

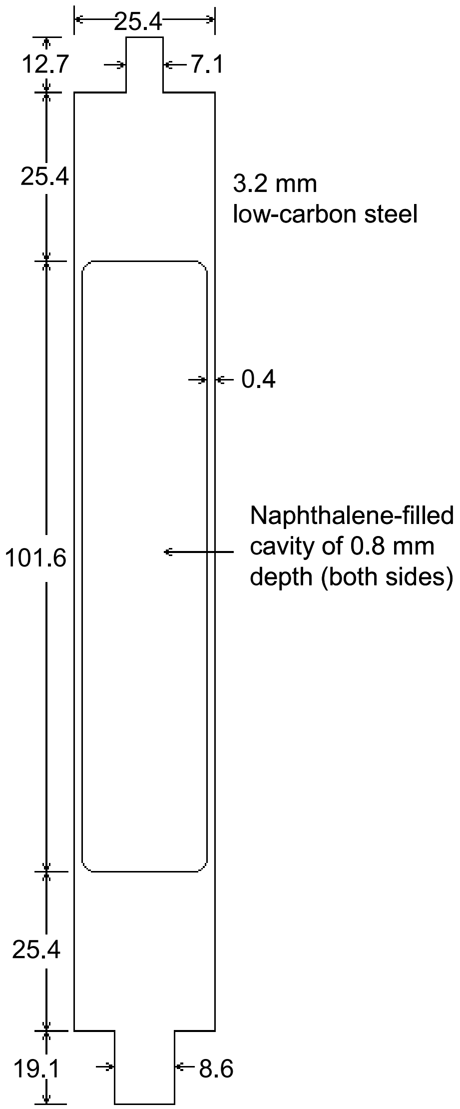

Figure 2. Fins used in naphthalene sublimation experiments. All dimensions in mm.

approximately Re = 10 000 is available, pressure drop and mass transfer experiments were undertaken to characterize the thermal-hydraulic performance of this array at Reynolds numbers up to approximately Re = 120 000.

Apparatus

The pressure drop and mass transfer experiments were performed in an open loop wind tunnel. This tunnel was connected to a plenum/compressor system capable of delivering dry air indefinitely at 1 kg/s and 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 though a 76 mm diameter pipe. The air exited the pipe radially into a large settling chamber, and the flow was conditioned by 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 between 1 and 50 m/s could be achieved in the test section.

The test section had a cross section of 152 x 152 mm, and was made from clear acrylic. The fins in the array had a height of 152 mm, a length of 25.4 mm, and a width of 3.18 mm. The array was square, with the fin pitch equal to the length of the fin. The hydraulic diameter of the array was calculated to be 35.0 mm. The geometry of the array used in the experiments is shown in Figure 1. The fins inside the dotted line in Figure 1 were used for measurements in the mass transfer experiments. The top plate of the acrylic test section was partitioned to allow for very quick installation and removal of the fins. Four static pressure taps, placed at the center of each side of the test section, were located just upstream and downstream of the test section. Each set of four taps was connected with tubing, and the tubing was connected to an inclined manometer to give an average reading of the pressure drop. A Pitot-static tube was inserted into the flow upstream of the test section. It was connected to a second inclined manometer and was used to measure the dynamic pressure of the air flow. A thermocouple was inserted into the large settling chamber to measure the temperature of the flow.

Solid aluminum fins were inserted into the test section for the pressure drop experiments, while different fins, made from low-carbon steel, were used in the mass transfer experiments. A cavity 0.8 mm deep was machined into each side of the fins into which naphthalene was cast. Detailed geometry of these fins is shown in Figure 2.

Experimental Procedure

Pressure drop experiments. Solid aluminum fins were used in the pressure drop experiments. These fins were used to ensure that the geometry of the test array did not change due to naphthalene sublimation during testing. 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.

Mass transfer experiments. The naphthalene sublimation method was used in these experiments. This method has been thoroughly reviewed in previous work (Goldstein and Cho, 1995) and has been shown to be an accurate method for determining convective properties of many geometries. In preparation for the mass transfer experiments, the specimens were made by casting 99.9% pure naphthalene into the cavities of the fins. The naphthalene was melted in a beaker, and the molten liquid was poured into the cavity until it was slightly overfull. This ensured that the naphthalene surface was at least as high as the fin surface after the shrinkage associated with solidification and cooling had occurred. After the naphthalene returned to lab temperature, the excess was shaved with a razorblade using the cavity edges as a guide. The naphthalene surface was polished with fine-grit sandpaper for smoothness and visually inspected. The fins were then stored in a sealed, naphthalene-saturated box until needed for the experiment.

Since a very well-mixed flow was expected in the array, all 48 of the fins in the test section contained naphthalene-filled cavities on both sides. Only the middle three rows, as shown in Figure 1, were used for the mass transfer measurements. These rows were located furthest away from the test section walls in order to mitigate wall effects.

The 24 fins that were not to be used for measurements were installed in the test section by removing the entire top plate of the test section. The top plate was then replaced, and the 24 fins that were to be used for mass transfer measurements were individually weighed on a precision balance. These fins were placed into the test section as quickly as possible through the opening of the top plate. The pieces covering the opening were replaced, room pressure was measured, and the wind tunnel was started.

The flow rate was controlled by the pressure regulator. In order to maintain a constant Reynolds number, the dynamic pressure, as measured by the Pitot-static tube, was held constant. The temperature of the flow was recorded once per minute throughout the test. An average of this temperature was used to determine thermophysical properties of the air and naphthalene.

The exposure time of the naphthalene fins was determined by estimating the mass transfer rate and calculating a sufficient exposure period to ensure that the amount of naphthalene sublimed would result in a small measurement error in the change of mass of the fins. The exposure period, however, had to be short enough that the geometry of the fins did not significantly change during the test. Exposure times ranged from 60 minutes to 8 minutes. The range of Reynolds numbers in the mass transfer tests was approximately 5 000 < *Re* < 120 000, the same range as in the pressure drop experiments..

Data Reduction

Pressure drop experiments. The properties of air were determined using the temperature recorded in the settling chamber of the wind tunnel 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 and minimum flow areas using Eq. (1). The hydraulic diameter was calculated using Eq. (2), and the Reynolds number (based on hydraulic diameter) was calculated using Eq. (3).

$$U_c = U_{fr}\left[\frac{L}{L-t}\right] \qquad (1)$$

$$D_h = \frac{4A_c}{\left(A_T/L_{core}\right)} = \frac{2wL(L-t)}{w(L+t)+L(L-t)} \qquad (2)$$

$$Re = \frac{U_c D_h}{\upsilon} \qquad (3)$$

The Fanning friction factor, *f*, was calculated from the pressure drop measured across the array, ΔP_{array}, using equation (4), which neglects entrance and exit effects.

$$f = \frac{2\Delta P_{core}}{\rho U_c^{\,2}}\left[\frac{D_h}{4L_{core}}\right] \qquad (4)$$

Mass transfer experiments. Before calculating the mass transfer rate of the fins in the array, an experiment was performed to estimate the amount of naphthalene that sublimed each time the fins were weighed, installed, and removed in the experiments. All of the procedures for performing a mass transfer experiment as described above were performed, with the exception of operating the wind tunnel. The change in mass of each of the 24 fins was recorded, and the average was used in the data reduction procedure. The freestream naphthalene vapor density, $\rho_{nap,\infty}$, was calculated for each row of fins using the rate of sublimation from the upstream fins as shown in Eq. (5). In this equation, the index i represents the row number.

$$\rho_{nap,\inf,n} = \sum_{i=1}^{n-1}\frac{6\overline{(dm/dt)_i}}{\dot{V}_{air}} \qquad (5)$$

The modified Colburn *j* factor was calculated using Eqs. (6) - (8).

$$\overline{h_m} = \frac{(m_i - m_f) - \Delta m_{\exp}}{A_{nap}(\rho_{nap,v} - \rho_{nap,\inf})\Delta t} \qquad (6)$$

$$\overline{Sh} = \frac{D_h \overline{h_m}}{D_{na}} \qquad (7)$$

$$j = \frac{\overline{Sh}}{ReSc^{0.4}} \qquad (8)$$

Uncertainties

Uncertainties were calculated using standard methods (Kline and McClintock, 1953). The uncertainty in 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 5% for all measurements.

RESULTS AND DISCUSSION

Pressure Drop

Friction factor results are shown in Figure 3. The data measured in this work are shown in the figure with the results of previous work studying this array geometry (Ge, 2002). At low Reynolds numbers (Re < 10 000), the new data do not agree with the previous work. It should be noted, however, that the friction factor obtained in this work is within 10% of the most appropriate available correlation, developed by Manglik and Bergles (1995) from a large database of performance of offset-strip fins at low Reynolds numbers, in the range $Re < 20\ 000$. The values obtained by Ge (2002) deviate much more from the values expected based on this correlation. It can be seen that the friction factor stops decreasing at approximately $Re = 20\ 000$, and exhibits a kind of oscillatory pattern as it increases to $f \sim 0.06$ as the Reynolds number increases to $Re \sim 100\ 000$. This behavior may be evidence of the presence of regimes in which vortex shedding does and does not occur. The friction factor at high Reynolds numbers is approximately twice that predicted by the correlation. This suggests that the physics of the flow is quite different at high Reynolds numbers.

Mass Transfer

The mass transfer results are shown in Figure 4; again, the data of Ge (2002) are included for comparison. It can be seen in the figure that the mass transfer results obtained in this work follow the trend observed in the previous work. At low Reynolds numbers, the data agree within 30% of the most appropriate correlation (1995), but as the Reynolds number increases, the measured modified Colburn j factor increasingly deviates from this correlation, until it is approximately twice that predicted at Re ~ 100 000.

The 30% deviation from the correlation even at low Reynolds numbers is probably caused by this array showing more bluff-body behavior than those used to develop the correlation. The fin thickness to fin length ratio in this experiment is approximately 4 times that of the strip-fin geometries used to develop the correlation, and the correlation itself is only weakly dependent on this ratio. The much larger deviation at high Reynolds numbers, however, is evidence of a different type of flow in the array – one that is most likely turbulent and vortex shedding rather than steady laminar.

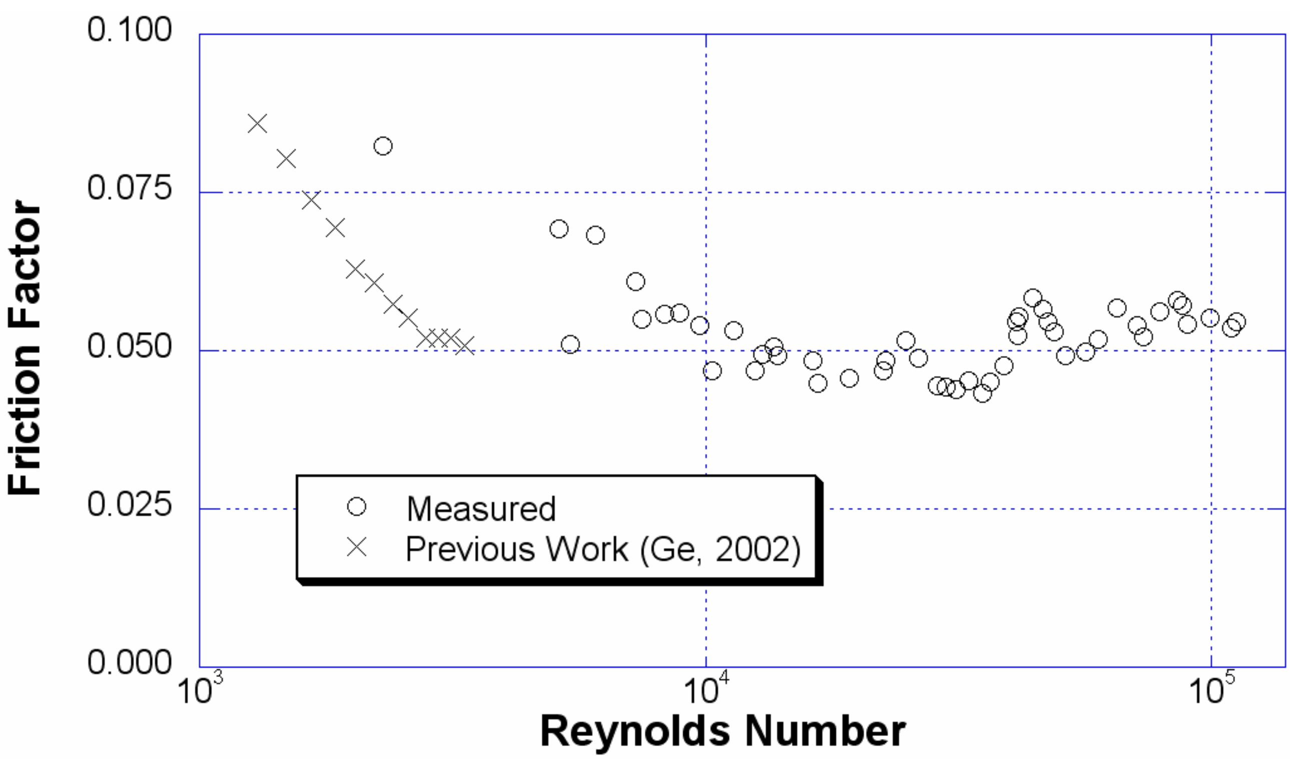

Figure 3. Friction factor, *f*, of the offset-strip fin array as a function of Reynolds number. Low Reynolds number data from previous work is included for comparison.

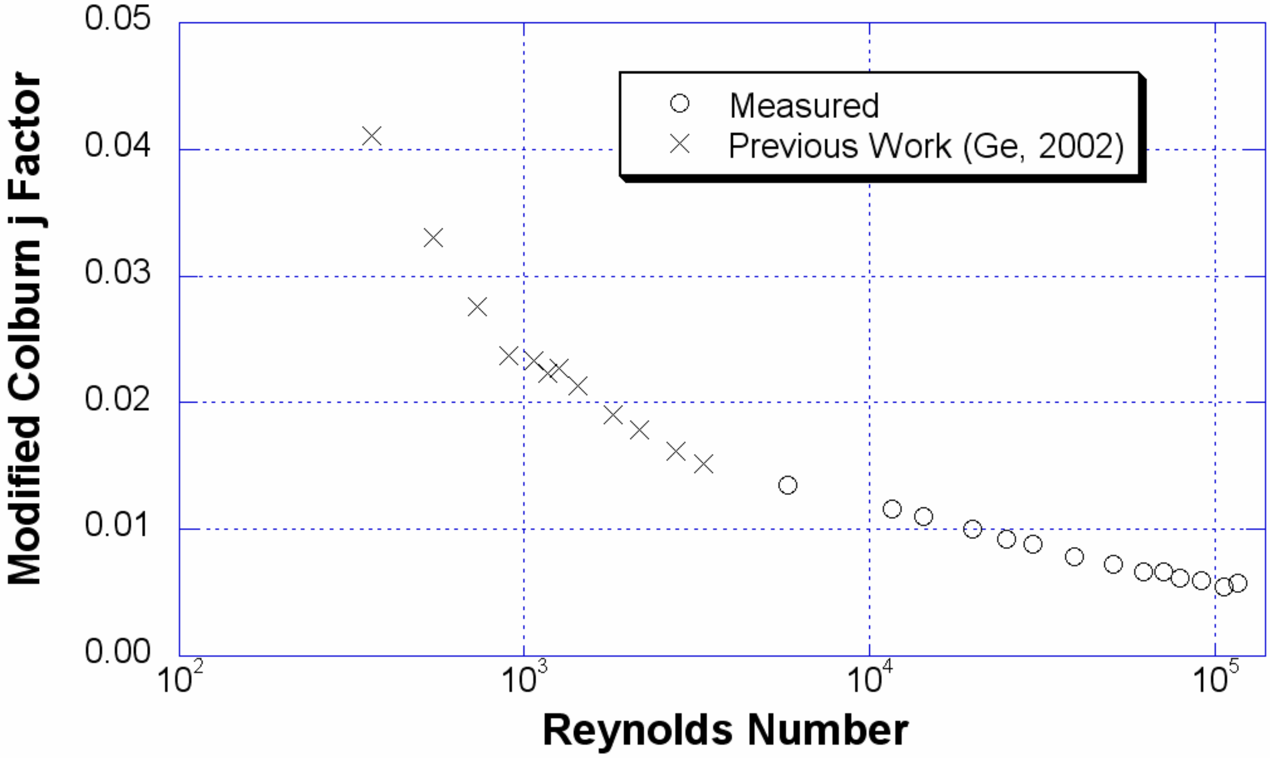

Figure 4. Modified Colburn *j* factor of the offset-strip fin array as a function of Reynolds number. Low Reynolds number data from previous work is included for comparison.

Applications

There are a number of possible applications for an offset-strip fin heat exchanger operating at high fin-side Reynolds numbers. One application is that of a liquid-to-liquid heat exchanger for aerospace applications. In these applications, compactness and light weight are usually the most important aspects of the design. Consider an offset-strip fin heat exchanger with D_h = 1.5 mm, a compactness of 2000 m^2/m^3, and water flowing on the finned side at Re = 30 000. If the tube-side thermal resistance was neglected and the fin efficiency assumed to be unity, the heat transfer in this example would be over 10 GW/m^3. Accounting for the tube side resistance and actual fin efficiency would decrease this heat duty, but the heat transfer density would still be very high. A very compact and lightweight heat exchanger could be designed in this way.

Another application for this type of heat exchanger is inlet cooling of gas turbine inlet air. Power output and efficiency of a gas turbine engine can be improved through cooling of the inlet air. The commonly used method of cooling the gas turbine inlet air is evaporative cooling. However, that method has limitations, especially in locations where the relative humidity is consistently high (Gareta *et al.*, 2004). A vapor-compression cycle with compact heat exchangers operating at high Reynolds numbers might eliminate this problem.

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 convection coefficients were determined to be approximately twice those predicted by correlations developed using data from low-Reynolds number conditions. It is suggested that this is a result of a vortex shedding and the turbulent flow at these high Reynolds numbers.

Operation of an offset-strip fin heat exchanger under these conditions may be useful in systems where minimizing heat exchanger size or maximizing overall thermal performance is more important than minimizing fan power, when large air-side pressure drops may be otherwise unimportant or in liquid-to-liquid applications.

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