averaged Nusselt number for these cases is still less than that for an annulus with no baffles.

Although the relation between the Nusselt number and baffle length is complicated by the orientation angle, it is clear that for most cases, baffles with a length of $H/D = \frac{1}{2}$ produce the best result. In addition, it is found that inner baffles generally work better than outer baffles.

To reveal the nature of the oscillatory flows the variation of normalized Nusselt numbers with time is shown in Fig. 5 for an annulus with three outer baffles. The corresponding cases of full baffles are also included for comparison. For $\gamma = 20$ and 30 deg, the variation curves not only have a similar pattern but they also have a same period. However, there is a time lag between them. For $\gamma = 40$ deg, no similarity can be found in their patterns. In comparison to that of the full baffles, the period for the outer baffles is shorter and the averaged Nusselt number is also smaller.

Conclusions

In this study the feasibility of using radial baffles to reduce convective heat losses from pipe insulation has been successfully demonstrated. From the results obtained, it shows that the effectiveness of a baffle is not directly proportional to its length. It is found that a baffle with a length of $H/D = \frac{1}{2}$ gives the best result in heat loss reduction. For a given baffle the efficiency is also dependent on its orientation. Therefore, when applying radial baffles to pipe insulation, it is of equal importance to adjust the baffle orientation than to simply emphasize the baffle length.

Acknowledgment

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Heat Transfer in Turbulent Boundary Layers with a Short Strip of Roughness

Robert P. Taylor* and Walid M. Chakroun†
Mississippi State University, Mississippi State, Mississippi 39762

Introduction

THIS note presents the results of an experimental investigation of the influence of a short strip of roughness on the heat transfer and fluid flow in the zero pressure gradient flat-plate turbulent boundary layer. The primary motivation for this work is to gain insight into the heat transfer and fluid flow in a turbulent boundary layer with a wide variation in surface roughness. Of particular interest are inservice gas turbine blades which have been shown to have a wide variation of both the magnitude and statistical character of the roughness from point-to-point around the blades.[1] The average roughness ranged from about 10–1.5 μm. This is very rough considering that the thickness of the boundary layer is on the order of 1 mm. A particular feature of this data was the large variation in the character of the roughness around the blade.

A good review of the literature on surfaces that contain an interface between rough and smooth surfaces is given by Smits and Wood.[2] Andreopoulas and Wood[3] reported extensive measurements of velocity profiles, turbulence quantities, and skin friction distribution for flow over a smooth surface which was roughened in one narrow strip at about midplate using sandpaper. References 4 and 5 reported heat transfer data for the case where the first one-third of their flat plat was rough and the remainder was smooth. They also presented predictions using the discrete element method roughness model.

In this note, the experimental results are presented for Stanton number distributions, mean velocity profiles, and turbulence intensity profiles for a turbulent boundary layer with a short strip of surface roughness. These results are compared with the results of the rough-to-smooth experiments of Taylor et al.[5] and the all-smooth experiments of Coleman et al.[6]

Experimental Apparatus and Measurements

The experiments were performed in the turbulent heat transfer test facility which is a closed loop wind tunnel designed for boundary-layer heat transfer experiments. The description and qualification of this facility are given in the open literature (Coleman et al.[7]), and are not repeated here because of the space limitations of a technical note. In Ref. 7, the smooth-wall experiments were shown to be in excellent agreement with the accepted classical experiments and correlations, the flow was shown to be a proper two-dimensional turbulent boundary layer, and the uncertainties associated with the data collection and reduction were discussed in detail.

Figure 1a shows a schematic diagram for the test surface used in these experiments. The first 0.7 m of the test surface is smooth, the next 0.2 m is roughened with 1.27-mm-diam hemispheres, and the remaining 1.5 m is smooth. This is accomplished using 7 smooth plates, 2 rough plates, and 15

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*Professor, Thermal and Fluid Dynamics Laboratory, Mechanical and Nuclear Engineering Department. Member AIAA.

†Instructor, Thermal and Fluid Dynamics Laboratory, Mechanical and Nuclear Engineering Department. Member AIAA.

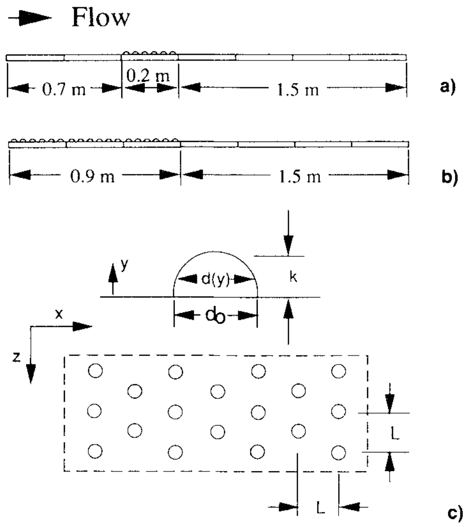

Fig. 1 Schematic diagram of the test surfaces and the roughness element layout.

smooth plates, with each plate being 0.1-m wide in the flow direction. This case is referred to as the rough-strip case. Figure 1b shows the test surface used in Refs. 4 and 5 with the first 0.9 m roughened with the hemispheres and the remaining 1.5 m smooth. This case is referred to as the rough-to-smooth case. The roughness elements are spaced 2-base diam apart in a staggered array as shown in Pt. c of the figure.

The boundary layer is tripped at the exit of the 19:1 area ratio nozzle with a 1 × 12-mm wooden strip. This trip location is immediately in front of the heated surface.

Stanton Number Determination

The Stanton number was determined using an energy balance on each of the individually heated plates. The Stanton number was computed from the definition using measured values of the plate heater power, plate temperature, freestream total temperature, freestream velocity, and experimentally calibrated models for the radiation loss and conduction to the support system. To miminize conduction losses the support rails were heated to approximately the same temperature as the plates. Conduction and radiation losses were typically 0.5–1.0%. For the Stanton number data in this note, the overall uncertainty as discussed by Taylor et al.[5] and Coleman et al.,[7] ranged from ±2 to ±5% for the 0.1-m-wide plates depending on flow conditions. For these experiments the nominal wall temperature was 45°C and the nominal air temperature was 28°C.

Profile Measurements

The profiles of mean velocity and longitudinal velocity fluctuation $\overline{u'^2}$ were taken with a horizontal hot-wire under isothermal flow conditions. At each measurement position, 1000 instantaneous anemometer output voltage readings 0.01 s apart were taken and converted into velocities using a fourth-order least-squares calibration equation. The mean of the 1000 velocities was used as the mean velocity at that location and the variance was taken as $\overline{u'^2}$. According to Coleman et al.[6] the overall uncertainties were ±2% for u and ±5% for $\overline{u'^2}$. All profile measurements were taken at levels above the roughness. For this distributed roughness the vigorous mixing at these levels made the distinction between crests and valleys on the surface negligible.

Experimental Results

Experimental heat transfer and fluid flow results have been obtained for zero pressure gradient, constant wall temperature, incompressible, turbulent boundary-layer flow of air with nominal freestream velocities, U_∞, of 12, 28, 43, and 58 m/s. The corresponding x-Reynolds numbers, $Re_x = U_\infty x/\nu$, ranged from 100,000 to 8,000,000. The x-Reynolds number immediately upstream of the rough strip ranged from 500,000 to 2,000,000, and those immediately downstream of the rough strip ranged from 700,000 to 3,500,000. A more detailed presentation is given in the paper by Taylor and Chakroun.[8] A representative subset of this data is presented in this note.

Figure 2 shows plots of Stanton number St vs x-Reynolds number for the rough-strip, rough-to-smooth, and all-smooth cases for freestream velocities of 12 and 58 m/s. The rough-wall portions of these cases are considered to be in the fully rough regime based on the results of Hosni et al.[9] The rough-to-smooth data are taken from Taylor et al.,[5] and the all-smooth data are taken from Coleman et al.[6] All of these data sets were collected in the same facility using the same instrumentation and procedures. Therefore, all of the data contain many of the same bias errors and can be compared with a high degree of confidence.

The figures show that the rough strip has greatly enhanced the heat transfer. The Stanton number over the rough strip is roughly twice that of the equivalent smooth-wall case. For all of the freestream velocities considered, the first rough plate has a Stanton number that is slightly larger than the equivalent all-rough Stanton number. However, the second rough plate has essentially the same Stanton number as the all-rough data. Downstream of the roughness, the rough-strip cases behave much like the rough-to-smooth cases. The Stanton number drops abruptly to a value at or below the equivalent all-smooth case. With a very short distance all three Stanton number distributions merge into the smooth-wall Stanton number distribution.

Figure 3 shows a plot of the mean velocity profiles for the rough-strip case with U_∞ = 12 m/s. Profiles were taken over

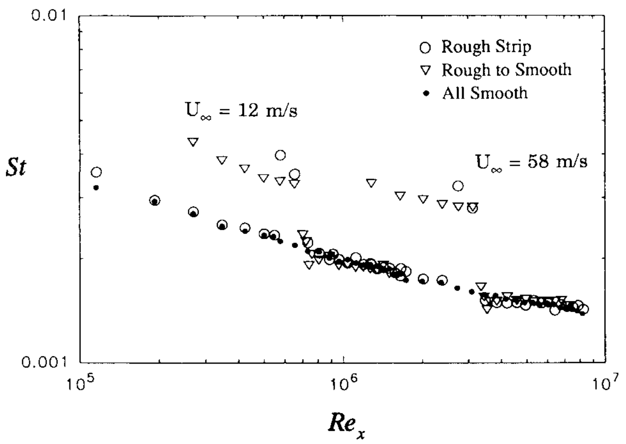

Fig. 2 Comparison of the Stanton number distributions for the rough-strip, rough-to-smooth[5] and all-smooth[6] cases for U_∞ = 12 m/s and U_∞ = 58 m/s.

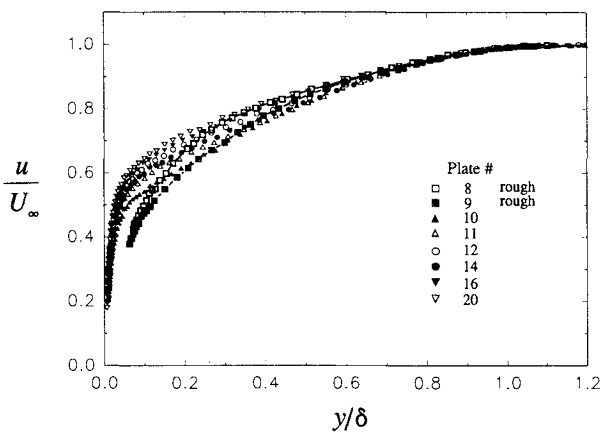

Fig. 3 Mean velocity profiles over and downstream of the rough strip for U_∞ = 12 m/s.

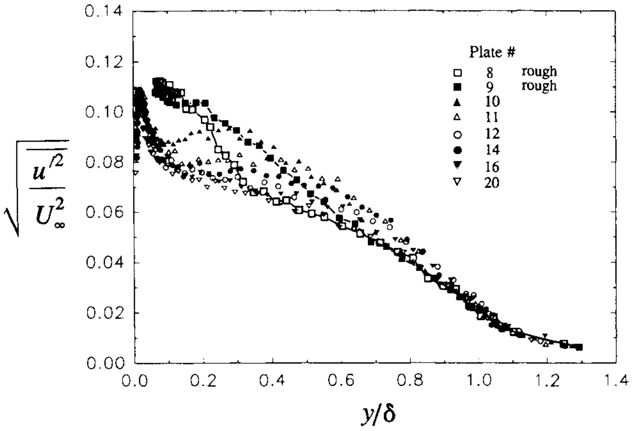

Fig. 4 Turbulence intensity profiles over and downstream of the rough strip for U_∞ = 12 m/s.

each of the two rough plates and over selected smooth plates downstream of the roughness. The velocity profiles over the rough plates are indicated by the open and filled square symbols. These symbols are connected with line segments to aid in finding these profiles. The influence of the roughness is seen to be large. The velocity profile over the first rough plate (no. 8) is strongly retarded in the lower portion of the boundary layer relative to the smooth-wall profiles. The roughness influence extends to approximately $y/\delta = 0.4$ where δ is the 99% boundary-layer thickness. Over the second rough plate (no. 9), the roughness influenced region extends to approximately $y/\delta = 0.7$. Downstream of the roughness the velocity profiles quickly take on smooth-wall like characteristics near the wall, but an intermediate roughness influenced region is still apparent downstream of the interface.

Figure 4 shows the turbulence intensity, $\sqrt{u'^2/U_\infty^2}$, profiles for the same conditions as the mean velocity profiles in Fig. 3. The evolution of the turbulent boundary layer is shown more clearly by this plot. Over plate no. 8 the near-wall turbulence intensity is much larger than equivalent smooth-wall conditions. The roughness influence clearly extends to about $y/\delta = 0.4$. Over the other rough plate (plate no. 9) there is no additional increase in turbulence intensity very near the wall; however, the roughness influenced region extends much farther into the boundary layer to about $y/\delta = 0.7$. After the roughness the turbulence intensity quickly takes on smooth-wall characteristics very near the wall. In the intermediate region the roughness influence continues to spread but decreases in magnitude. Downstream of the roughness there are three easily identifiable layers. Far from the wall the turbulent boundary layer continues to evolve as a smooth-wall boundary layer. In the intermediate region there is a roughness influenced internal layer. Very near the wall there is a smooth-wall-dominated internal layer which spreads to eventually envelop the entire layer. After some distance these three layers mix together to form a new smooth-wall boundary layer.

Acknowledgment

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Determination of the Thermal Conductivity of Iron Aluminide as a Function of Temperature

Rick D. Wilson*
U.S. Bureau of Mines, Abany, Oregon 97321
and
Jack H. Devletian†
Oregon Graduate Institute of Science and Technology, Beaverton, Oregon 97006

Introduction

THE thermal conductivity of several iron aluminide alloys was determined as a function of temperature. Four-point electrical resistivity data were used in conjunction with the Wiedemann-Franz-Lorenz equation[1] to determine the thermal conductivity of Fe_3Al. Rectangular sample bars were brought into equilibrium at several temperatures up to 1473 K before electrical resistivity measurements were made. A sample of iron with identical configuration, along with literature data, was used to verify the accuracy of the method. The results indicate that the thermal conductivity of Fe_3Al alloys is independent of composition and increases linearly as a function of temperature.

One of the principal requirements for a heat transfer model is the accuracy of the values for the thermal conductivity as a function of temperature. Unfortunately, values for intermetallic iron aluminide alloys near the Fe_3Al stoichiometric composition are unavailable in the literature.

Thus, the objective of this research was to determine the thermal conductivities of near-Fe_3Al alloys over a wide range

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*Materials Engineer, 1450 SW Queen Avenue.

†Professor of Materials Science and Engineering, 19600 NW von Neumann Drive.