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AN EXPERIMENTAL INVESTIGATION OF THE RIB SURFACE-AVERAGED HEAT TRANSFER COEFFICIENT IN A RIB-ROUGHENED SQUARE PASSAGE

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ABSTRACT

Turbine blade cooling, a common practice in modern aircraft engines, is accomplished, among other methods, by passing the cooling air through an often serpentine passage in the core of the blade. Furthermore, to enhance the heat transfer coefficient, these passages are roughened with rib-shaped turbulence promoters (turbulators). Considerable data are available on the heat transfer coefficient on the passage surface between the ribs. However, the heat transfer coefficients on the surface of the ribs themselves have not been investigated to the same extent. In small aircraft engines with small cooling passages and relatively large ribs, the rib surfaces comprise a large portion of the passage heat transfer area. Therefore, an accurate account of the heat transfer coefficient on the rib surfaces is critical in the overall design of the blade cooling system.

The objective of this experimental investigation was to conduct a series of thirteen tests to measure the rib surface-averaged heat transfer coefficient, h_{rib}, in a square duct roughened with staggered 90° ribs. To investigate the effects that blockage ratio, e/D_h, and pitch-to-height ratio, S/e, have on h_{rib} and passage friction factor, three rib geometries corresponding to blockage ratios of 0.133, 0.167 and 0.25 were tested for pitch-to-height ratios of 5, 7, 8.5 and 10. Comparisons were made between the rib average heat transfer coefficient and that on the wall surface between two ribs, h_{floor}, reported previously. Heat transfer coefficients of the upstream-most rib and that of a typical rib located in the middle of the rib-roughened region of the passage wall were also compared.

It is concluded that:

1) the rib average heat transfer coefficient is much higher than that for the area between the ribs,

2) similar to the heat transfer coefficient on the surface between the ribs, the average rib heat transfer coefficient increases with the blockage ratio,

3) a pitch-to-height ratios of 8.5 consistently produced the highest rib average heat transfer coefficients amongst all tested,

4) under otherwise identical conditions, ribs in upstream-most position produced lower heat transfer coefficients than the mid-channel positions,

5) the upstream-most rib average heat transfer coefficients decreased with the blockage ratio, and

6) thermal performance decreased with increased blockage ratio. While a pitch-to-height ratio of 8.5 and 10 had the highest thermal performance for the smallest rib geometry, thermal performance of high blockage ribs did not change significantly with the pitch-to-height ratio.

NOMENCLATURE

a	channel height (Figure 1)
b	channel width (Figure 1)
A	non-turbulated channel cross-sectional area (ab)
A_{floor}	wall heat transfer area between two ribs
A_{rib}	rib total heat transfer area (three sides)
A_{total}	total heat transfer area ($A_{rib}+A_{floor}$)
AR	channel aspect ratio (b/a)
AR_r	rib aspect ratio (e/w)
D_h	hydraulic diameter based on non-turbulated cross-section (4A/P=a)
e	rib height
$\bar{f}$	Darcy friction factor $\frac{\Delta P(D_h/L)}{1/2\rho U_m^2}$
$\bar{f}_s$	Darcy friction factor in an all-smooth-wall channel
h_{floor}	average heat transfer coefficient on the wall surface between a pair of ribs
$h_{overall}$	overall average heat transfer coefficient on a rib and the wall surface between a pair of ribs
h_{rib}	rib average heat transfer coefficient

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k	air thermal conductivity
L	length of the rib-roughened portion of the test section
m	air mass flow rate
$\overline{Nu}$	rib average Nusselt number ($h_{rib}D_h/k$)
$\overline{Nu}_s$	average Nusselt number in a smooth channel
P	unturbulated passage perimeter
Pr	Prandtl number
Re	Reynolds number
S	rib pitch (center-to-center)
T_f	film temperature ($0.5(T_s + T_m)$)
T_m	air mixed mean temperature
T_s	surface temperature
U_m	air mean velocity
w	turbulator width
X	distance between the instrumented rib and test section entrance (Figure 1)
α	angle of attack
μ	air dynamic viscosity
ΔP	pressure drop across the rib-roughened portion of the test section
ρ	air density

INTRODUCTION

Various cooling methods have been developed over the years to ensure that the turbine blade metal temperatures are maintained at a level consistent with airfoil design life. The objective in turbine blade cooling is to achieve maximum internal heat transfer coefficients while minimizing the coolant flow rate. One such method is to route coolant air through rib-roughened serpentine channels within the airfoil and convectively remove heat from the blade. The coolant is then ejected either at the tip of the blade, through the cooling slots along the trailing edge or film holes along the airfoil surface.

Geometric parameters such as channel aspect ratio (AR), rib height-to-passage hydraulic diameter (e/D_h) or blockage ratio , rib angle of attack (α), the manner in which the ribs are positioned relative to one another (in-line, staggered, criss-cross, etc.), rib pitch-to-height ratio (S/e) and rib shape (round versus sharp corners, fillets, rib aspect ratio (AR_t), and skewness towards the flow direction) have pronounced effects on both local and overall heat transfer coefficients. Some of these effects were studied by different investigators such as Abuaf et al. (1986), Burggraf (1970), Chandra et al. (1987, 1989), Han et al. (1978, 1984, 1985, 1992), Metzger et al. (1983, 1990), Taslim et al. (1988, 1991), Webb et al. (1971). A great majority of these studies, however, measure the heat transfer coefficients on the surface between the ribs and a few measured the overall average heat transfer coefficient for the combined ribs and area between them.

The objective of this investigation was to, while isolating the ribs from the wall surface thermally, measure the overall heat transfer coefficient on the rib surface and to study the effects pitch-to-height ratio (S/e), blockage ratio (e/D_h) and rib orientation (upstream-most position or in the middle of channel rib-roughened portion) have on the rib surface heat transfer coefficient for a channel aspect ratio (AR) of one.

Among those investigations dealing with the measurement of heat transfer coefficients on the ribs are the following. Solntsev et al. (1973) experimentally investigated heat transfer in the vicinity of sudden two- and three-dimensional steps of circular and square cross-sectional areas mounted on a flat surface in an open channel. Enhancement in heat transfer coefficient is reported for a range of Reynolds number between 10^4 to 10^5.

Berger and Hau (1979) did an experimental study of flow over square ribs in a pipe using an electrochemical analogue technique to measure mass/heat transfer on the ribs as well as on the wall surface between them. For a blockage ratio (e/d) of 0.0364, they varied the rib pitch-to-height ratio from 3 to 10 for a range of Reynolds number between 10000 and 25000. At a Reynolds number of 10^4, they showed enhancements, compared to smooth channels, in mass (heat) transfer on the ribs in the order of 4.4 and 5.2 for pitch-to-height ratios of 10 and 7 respectively.

Metzger et al. (1988) used a thermal transient technique to examine the contribution of the rib heat transfer to the overall heat transfer of a rib-roughened wall with variations in rib angle of attack and pitch. Square ribs representing a blockage ratio of 0.14 were mounted on only one wider side of a 0.154 aspect ratio rectangular channel. Main conclusions were that heat transfer on the rib surface significantly contributed to the overall rib-roughened wall heat transfer and this contribution was mainly dependent on rib pitch-to-height spacing, with very little effect of rib angle.

Lockett and Collins (1990) used a holographic interferometry technique to measure heat transfer coefficient in a 0.25 aspect ratio rectangular channel. Square ribs with sharp as well as round top corners representing a blockage ratio, e/D_h, of 0.067 and a pitch-to-height ratio of 7.2 were mounted on one of the wider sides of the channel perpendicular to the flow direction. They reported overall enhancements in heat transfer of up to 2.24 at a Reynolds number of 7400.

Liou et al. (1991) performed numerical as well as experimental investigation of turbulent flow in a 4:1 aspect ratio rectangular channel roughened on two opposite wider sides with square ribs in an in-line arrangement perpendicular to the flow direction. The rib blockage ratio, e/D_h, was 0.081 and four pitch-to-height ratios of 5, 10, 15 and 20 were examined at a fixed Reynolds number of 33000. Two-dimensional Navier-Stokes equations in elliptic form in conjunction with the k-ε turbulence model were solved numerically and holographic interferometry technique was used in the experimental part. They reported an enhancement in heat transfer on the rib surface of 3.1.

Sato et al. (1992) investigated the flow characteristics and heat transfer in a rectangular channel with a total of twenty square ribs on two opposite walls in staggered, in-line and quarter-pitch-shift arrangements. The channel aspect ratio was 0.2 and the ribs, mounted on the two wider sides of the channel, had a blockage ratio of 0.12. Details of the flow and heat transfer over a typical rib-roughened section (including the rib surface) well downstream of the first rib were presented. They concluded that the staggered arrangement had a better heat transfer performance than the other two arrangements.

Dawes (1992) solved the three dimensional Navier-Stokes equations in a rotating serpentine coolant passage of cylindrical geometry roughened with square ribs. Rib blockage, *e/d*, and pitch-to-height ratios were 0.2 and 10, respectively. Results of this work were compared with other numerical and experimental works.

Liou and Hwang (1993) also used the holographic interferometry technique to measure heat transfer coefficient and friction factor in a 0.25 aspect ratio rectangular channel with three rib shapes including a square rib geometry mounted on two opposite wider sides in an in-line arrangement perpendicular to the flow direction. For one blockage ratio of 0.08 , they tested four rib pitch-to-height ratios of 8, 10, 15 and 20 in a range of Reynolds number between 7800 and 50000. Heat transfer coefficient was measured over the ribs as well as the wall surfaces between the ribs. They reported overall enhancements in heat transfer of 2.2 and 2.7 for semi-cylindrical and square ribs for the range of Reynolds number tested, respectively.

TEST SECTIONS

Figure 1 shows schematically the layout and cross-sectional area of a typical test section and rib geometry details are shown in Table 1. All test sections, with a length of 116.84 *cm*, had a square 3.81 *cm* by 3.81 *cm* cross-sectional area. Three walls of these channels were made of 1.27 *cm* thick clear acrylic plastic. The fourth wall, on which the surface heaters and instrumented copper rib were mounted and all measurements were taken, was made of a 10.16 *cm* thick machineable polyurethane slab. Eighteen ribs of square cross-section with sharp corners were symmetrically staggered on the polyurethane and opposite acrylic walls (nine on each wall) at 90° angle of attack to the air flow. The entrance region of all test sections was left unturbulated to simulate the cooling passage in the dovetail region of a gas turbine blade. All ribs but one were machined out of acrylic plastic and were mounted on the walls using a special double-stick 0.05 *mm* thick tape with minimal temperature deformation characteristics. The instrumented rib on which all measurements were taken was machined out of copper. Inside this copper rib, as centrally as possible, was installed a 60-Ohm cylindrical electric heater running the full length using a highly conductive silver glue. Also installed in the copper rib were three calibrated thermocouples to measure the surface temperature. These three thermocouples were equally spaced over the length of the rib with their beads close to the rib surface. Their temperature readings were found to be the same within a fraction of a degree. For data reduction, the average of the three temperatures was used. Copper rib surfaces were polished to minimize the radiational heat losses from the copper rib to the unheated wall. Rib heat transfer coefficient measurements were performed for two distinct rib locations. First, the copper rib was mounted in the middle of the rib-roughened portion of the channel (fifth rib) and other eight acrylic plastic ribs were arranged on each side with the desired rib pitch-to-height ratio. Second, the copper rib was moved to the upstream-most position and the other eight ribs were mounted downstream of it. Table 1 shows the rib location from the channel entrance, *X*, for each geometry.

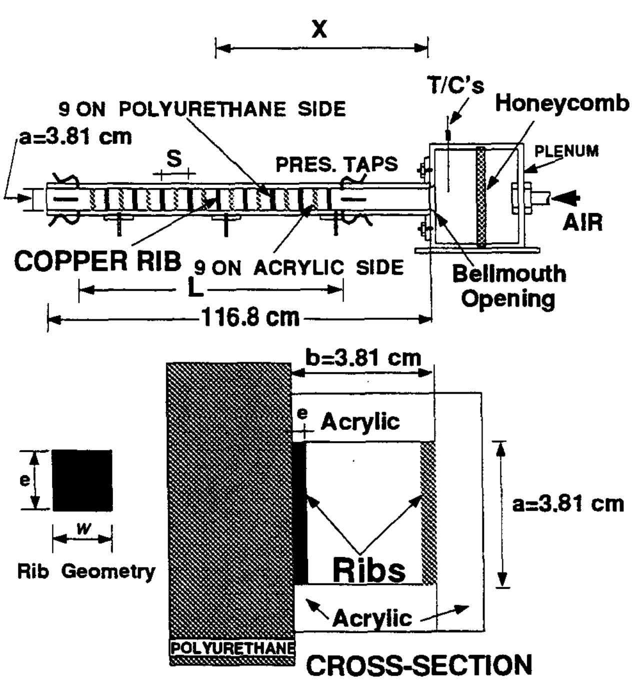

Figure 1 Schematic of a Typical Test Section

Two 3.81 *cm* by 27.94 *cm* custom-made etched-foil heaters with a thickness of 0.15 *mm* were placed on the polyurethane wall abutting both sides of the copper rib using the same special double-stick tape. The test sections were covered on all sides by 5 *cm* thick styrofoam sheets to minimize heat losses to the environment. Surface heat flux in the test section was generated by the heaters through a custom-designed power supply unit. Each heater was individually controlled by a variable transformer.

Before testing, thermocouples were calibrated using ice water and boiling water reference points and calibration curves were constructed for minor deviations (within a fraction of a degree). For a typical test run, the Reynolds number was set by precisely fixing the mass flow rate. The heat flux was induced by adjusting heater power until the copper rib reached the desired temperature. Enough time was given so that the system came to thermal equilibrium at which time data was recorded. Power to the copper rib was then

TESTS	e (mm)	e/D_h	S/e	X (cm)	X/D_h	Remarks
1	9.525	0.25	10	72.39	19.	Middle Position
2	9.525	0.25	8.5	72.39	19.	Middle Position
3	9.525	0.25	7	72.39	19.	Middle Position
4	9.525	0.25	5	72.39	19.	Middle Position
5	9.525	0.167	8.5	40.	10.5	Upstream-most Position
6	6.35	0.167	10	57.15	15.	Middle Position
7	6.35	0.167	8.5	57.31	15.04	Middle Position
8	6.35	0.167	5	57.31	15.04	Middle Position
9	6.35	0.167	8.5	35.72	9.375	Upstream-most Position
10	5.08	0.133	10	57.15	15.	Middle Position
11	5.08	0.133	8.5	57.15	15.	Middle Position
12	5.08	0.133	5	57.15	15.	Middle Position
13	5.08	0.133	8.5	39.88	10.47	Upstream-most Position

AR=1, AR_t = e/w = 1, $\alpha = 90^o$ Staggered for all geometries

Table 1 Specifications

increased to gather data at a higher surface temperature. This procedure was repeated for all copper temperatures and flow rates.

Static pressure taps were mounted on all three acrylic plastic walls of each test section to measure the pressure drop across the rib-roughened portion of the test section. A contact micromanometer with an accuracy of 0.025 *mm* of water column measured the pressure differences between the static pressure taps. A critical venturimeter, with choked flow for all cases tested, measured the total mass flow rate entering the test section. The reported friction factor is the overall passage average, $\overline{f}$ and not just the rib-roughened surfaces. Details of the experimental apparatus and test procedures are reported by Wadsworth (1994).

The radiational heat loss from the heated rib (and wall) to the unheated walls as well as losses to ambient air were taken into consideration when heat transfer coefficients were calculated. The reported heat transfer coefficients are the averages over the rib surfaces and not that of wall surfaces between the ribs. The heat transfer coefficients on the roughened walls for various geometries are reported by those investigators mentioned in the **Introduction** section. Experimental uncertainties, following the method of Kline and McClintock (1953), were determined to be ±6% and ±8% for the heat transfer coefficient and friction factor respectively.

RESULTS AND DISCUSSION

Average rib heat transfer results for the thirteen rib geometries are compared with all-smooth-wall channel Dittus-Boelter (1930) correlation ($\overline{Nu}_s=0.023Re^{0.8}\,Pr^{0.4}$) in Figures 2, 3, 5, 7, 8, 10, 11, 13 and 15. With this correlation, the enhancement (relative to smooth walls) in rib-roughened heat transfer coefficients is readily evaluated. The thermal performance based on the same pumping power is given by $(\overline{Nu}/\overline{Nu}_s)/(\overline{f}/\overline{f}_s)^{1/3}$ (Han et al., 1985), where $\overline{f}_s$ is the all-smooth-wall friction factor from Moody (1944). Air properties for Nusselt and Reynolds number calculations are based on the local film temperature, T_f for all cases.

Figure 2 shows the Nusselt versus Reynolds numbers for the first rib geometry corresponding to a blockage ratio, e/D_h, of 0.25 and pitch-to-height ratio, S/e, of 10. Copper rib temperature is varied from 43.3°C to 82.2°C with no change in the measured heat transfer coefficient. This lack of effect of copper surface temperature on heat transfer coefficient continued for all geometries and temperatures examined. Furthermore, this insensitivity of the measured heat transfer coefficient to the rib surface temperature supports the accuracy of our accounting for the heat losses to the ambient air and radiational losses from the heated copper rib to the unheated surrounding walls.

To investigate the effects pitch-to-height ratio have on rib heat transfer and channel overall friction factor, the same rib geometry was tested for four pitch-to-height ratios of 5, 7, 8.5 and 10 (geometries 1 through 4 in Table 1) results of which are shown in Figure 3. Also shown in Figure 3 are the rib heat transfer results for an S/e of 8.5 when the instrumented copper rib was mounted in upstream-most position (geometry 5 in Table 1). The middle position Nusselt numbers did not change significantly with the pitch-to-height ratio, but were consistently higher for an S/e of 8.5. The heat transfer enhancement for the upstream-most rib was considerably lower than for those in the middle of the rib-roughened region, indicating that upstream ribs and those staggered on the opposite wall contribute significantly to the very high level of heat transfer enhancement of downstream ribs by interrupting the flow and diverting its direction thus promoting high levels of mixing. Friction factors for these geometries are shown in Figure 4. Higher friction factors for S/e of 10 and 8.5, compared to 5, are in line with the Colburn (1933) analogy between heat transfer coefficient and friction factor.

Also shown in Figure 3 are the heat transfer results for the area between the ribs (called floor heat transfer by some investigators) reported by Taslim et al. (1991b). It can be seen that, for the mid-stream ribs, the rib average heat transfer coefficients are much higher than those for the area between the ribs (h_{floor}). Therefore, the contribution of the ribs to the overall heat transfer in a rib-roughened passage is significant. A simple area-weighted averaging analysis leads to the following relation.

$$\frac{h_{rib}A_{rib}}{h_{overall}A_{overall}} = \left\{1 + \frac{h_{floor}}{h_{rib}}\left[\frac{1}{3}\left(\frac{S}{e}-1\right)\right]\right\}^{-1}$$

where $h_{overall} = (h_{rib}A_{rib} + h_{floor}A_{floor})/(A_{rib}+A_{floor})$ and $A_{total} = A_{rib} + A_{floor}$. For example, at a typical value of

$$\frac{h_{floor}}{h_{rib}} = \frac{2}{3}$$

and at $S/e=10$ and 5, the $h_{rib}A_{rib}$ can be as high as 33% to 53% of $h_{overall}A_{total}$, respectively.

The next series of four tests, shown in Figure 5, correspond to a rib blockage ratio of 0.167 three of which (geometries 6 through 8 in Table 1) were performed for pitch-to-height ratios of 10, 8.5 and 5 with the instrumented copper rib mounted in the middle of

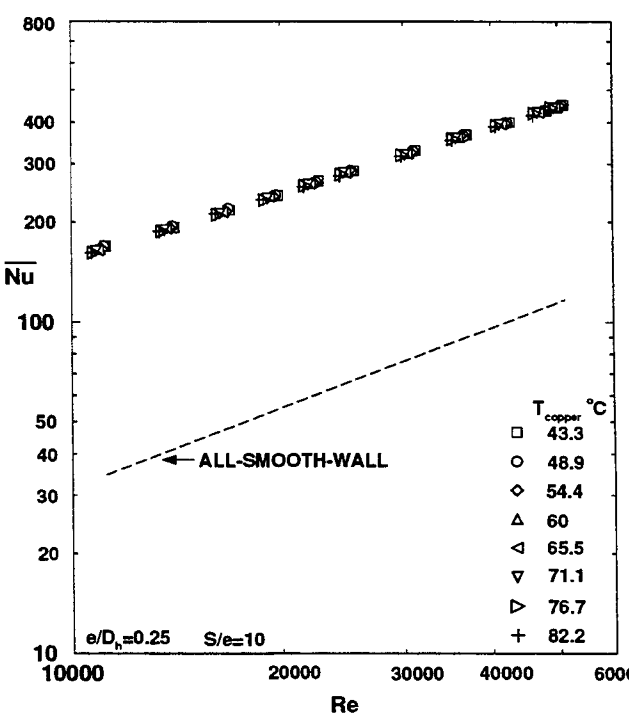

Figure 2 Rib Average Nusselt Number for a Range of Rib Surface Temperatures

the rib-roughened region. The fourth test represents the heat transfer results when copper rib was mounted in upstream-most position (geometry 9 in Table 1). Again, the rib average heat transfer results for the pitch-to-height ratio of 8.5 were slightly higher than those of 10 and significantly higher than those of S/e=5. Also, heat transfer enhancement for the upstream-most rib was lower than that for midpoint ribs, although not as much lower as it was for the blockage ratio of 0.25 as seen in Figure 3 (we will see that this difference further reduces for a still lower blockage ratio test). Note also that the rib average heat transfer coefficients are much higher than those on the floor, represented by the solid line (Taslim et al., 1994). The friction factors for these tests are shown in Figure 6, and the trend is similar to that of Figure 4, i.e., S/e=8.5 showing a slightly higher friction factor.

Figure 7 shows the results of two tests of identical geometries for which the foil heaters were on and off, respectively. No difference, beyond experimental uncertainties, was observed between the two sets of results, indicating that the thermal boundary layer, being interrupted repeatedly by the ribs, did not affect the heat transfer. It would appear that the mixing phenomenon was the dominant driving force for the high levels of heat transfer coefficient.

The next four tests, shown in Figure 8, correspond to a yet smaller rib blockage ratio of 0.133. The first three tests (geometries 10 through 12 in Table 1) were performed for pitch-to-height ratios of 10, 8.5 and 5 with the instrumented copper rib mounted in the middle of the rib-roughened region. The fourth test represents the heat transfer results for the copper rib mounted in the upstream-most position (geometry 13 in Table 1). Floor heat transfer results (Taslim et al. (1991b)) are also shown for comparisons. Again, heat transfer results for the pitch-to-height ratio of 8.5 were slightly higher than those for 10 and significantly higher than those for S/e =5. In contrast to the above-mentioned cases of higher blockage ratios, the heat transfer enhancements for the upstream-most rib were comparable to those of midpoint ribs at low Reynolds numbers, and only at higher Reynolds numbers did they start to deviate. This is an indication that the contribution of the staggered ribs on the opposite wall to the copper rib heat transfer coefficient, caused by the diversion of flow towards the ribs on opposite wall, is more significant for higher blockage ratios and at higher Reynolds number. The two closest geometries found in open literature are also shown in that figure. The data point from Metzger et al. (1988) is for a one-side-roughened channel, and the data point from Sato et al. (1992) is for a staggered arrangement both in a very low aspect-ratio channel. The differences between those blockage ratios (e/b =0.243, one-wall and $2e/b$ =0.4, respectively) and present geometries ($2e/b$ =0.266) would appear to explain the differences in heat transfer. The friction factors for the present tests are shown in Figure 9. The pitch-to-height ratio shows significantly less effect on the friction factor than the other two higher blockage ratios tested.

Figure 10 combines the results of all three geometries of the rib in the upstream-most position at S/e of 8.5. It is seen that smaller ribs produce higher heat transfer coefficients. Not benefiting from effects of ribs on the opposite wall, it is speculated that this behavior is due to the change of flow pattern over different ribs. In other words, in the extreme case, recirculating bubbles may form on both the back and top of the big rib reducing the contribution of the rib

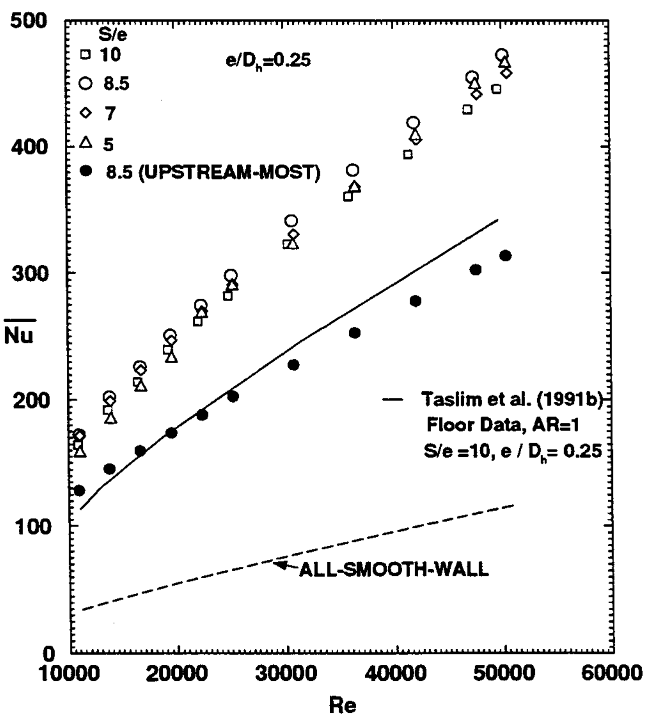

Figure 3 Rib Average Nusselt Number for a Range of Pitch-to-Height Ratios, e/D_h =0.25

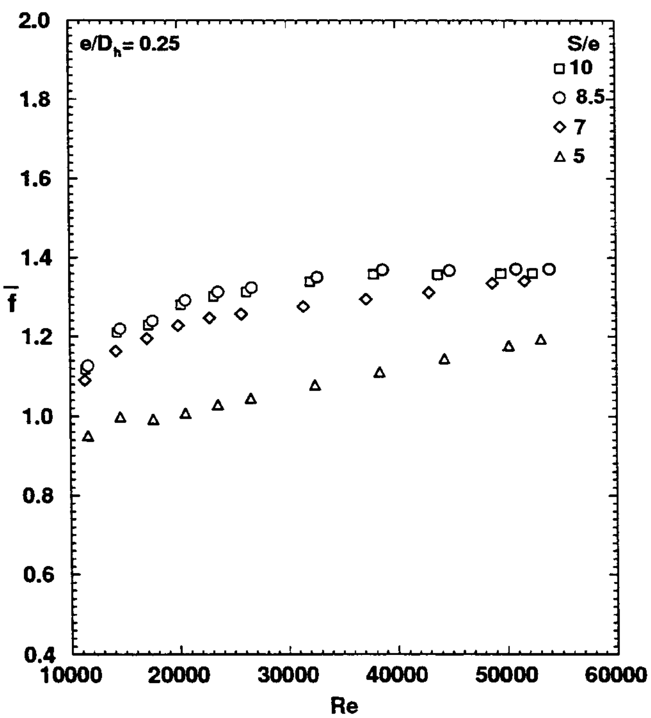

Figure 4 Channel Average Friction Factor for a Range of Pitch-to-Height Ratios, e/D_h =0.25

top surface to heat transfer, known to be major for square ribs with sharp corners by all investigators mentioned above. For the smaller ribs, however, the top surface is in contact with core air, thus higher heat transfer coefficients are produced.

Figures 11 and 12 compare the mid-channel rib heat transfer coefficient and channel friction factors for three blockage ratios at one pitch-to-height ratio of 10. In contrast with the heat transfer coefficient on the area between a pair of ribs, which is highly affected by the blockage ratio, the rib heat transfer coefficient does not show as strong of a dependence on blockage ratio for S/e =10. However, as ribs are brought closer to each other by reducing pitch-to-height ratios, rib heat transfer coefficient is more and more affected by the blockage ratio (Figures 13 and 15). The corresponding friction factors for S/e of 8.5 and 5 are shown in Figures 14 and 16.

Finally, the thermal performances of all geometries tested are compared in Figure 17. It is seen that as the blockage ratio increases the rib thermal performance decreases and thermal performance of high blockage ribs does not change significantly with the pitch-to-height ratio. For each rib geometry, the upstream-most rib has the lowest thermal performance. This was expected since those ribs had lower heat transfer coefficient than those in the middle of the rib-roughened region.

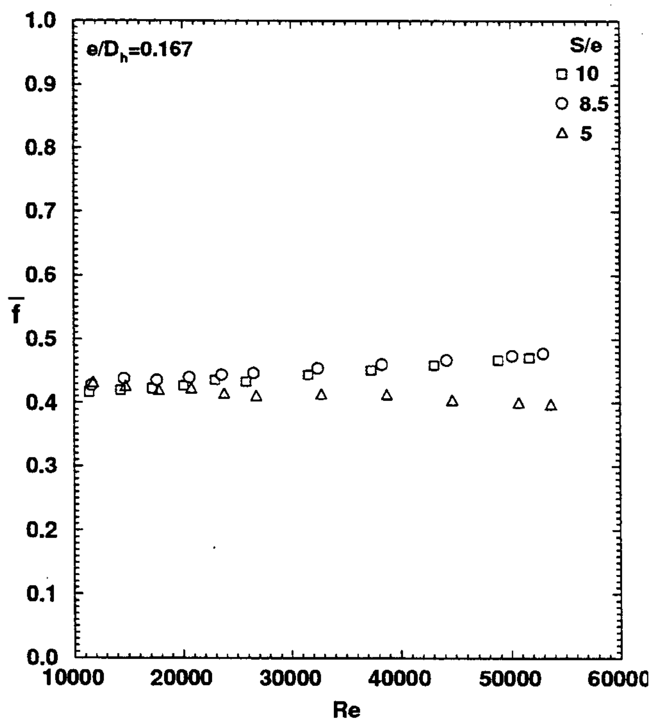

Figure 6 Channel Average Friction Factor for a Range of Pitch-to-Height Ratios, e/D_h = 0.167

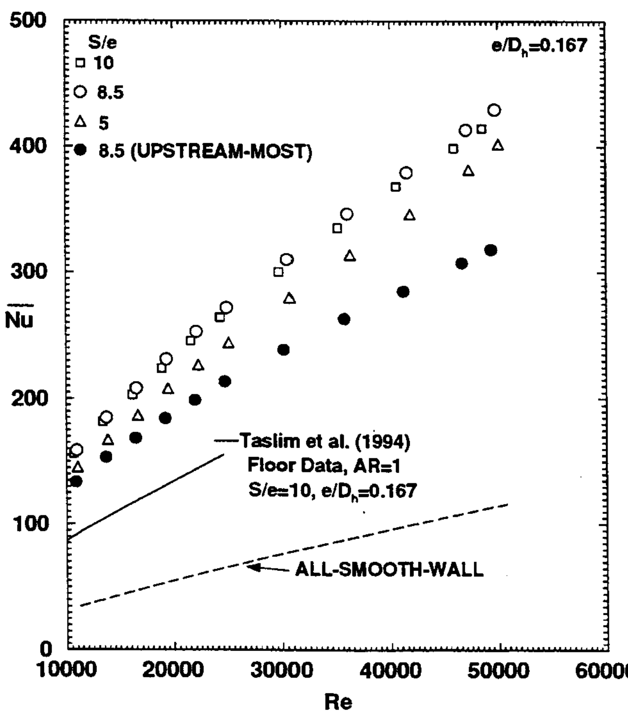

Figure 5 Rib Average Nusselt Number for a Range of Pitch-to-Height Ratios, e/D_h=0.167

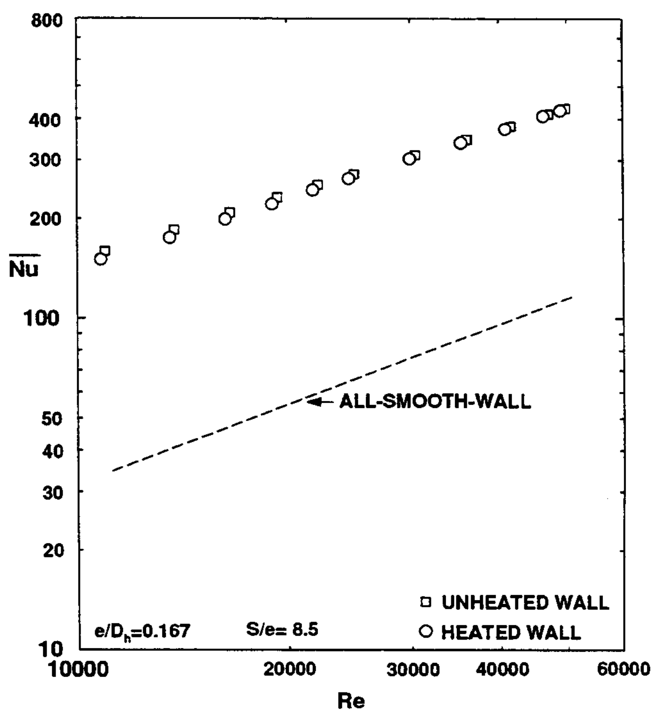

Figure 7 Comparison of Rib Average Nusselt Number for Heated and Unheated Walls

CONCLUSIONS

A total of thirteen rib geometries representing three blockage ratios in a practical range for small aircraft engines and at three pitch-to-height ratios were tested for heat transfer and pressure loss variations. From this study, it is concluded that:

1) For the geometries tested, the rib average heat transfer coefficient is much higher than that for the area between the ribs. For high blockage ribs with large heat transfer areas, commonly used in small gas turbines, the rib heat transfer is a significant portion of the overall heat transfer in the cooling passages. As for the area between the ribs, rib average heat transfer coefficient increases with blockage ratio.

2) Among all tested pitch-to-height ratios, a ratio of 8.5 consistently produced higher heat transfer coefficients.

3) Under otherwise identical conditions, ribs in the upstream-most position produced lower heat transfer coefficients. In that position, for the three rib geometries tested, the rib average heat transfer coefficients decreased with the blockage ratio.

4) Thermal performance decreased with increased blockage ratio. While a pitch-to-height ratio of 8.5 and 10 had the highest thermal performance for the smallest rib geometry, thermal performance of high blockage ribs did not change significantly with the pitch-to-height ratio.

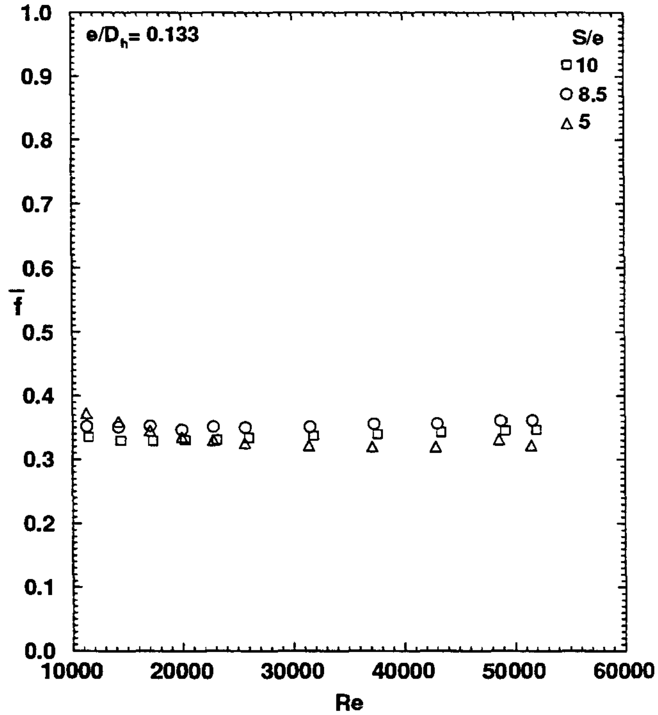

Figure 9 Channel Average Friction Factor for a Range of Pitch-to-Height Ratios, $e/D_h = 0.133$

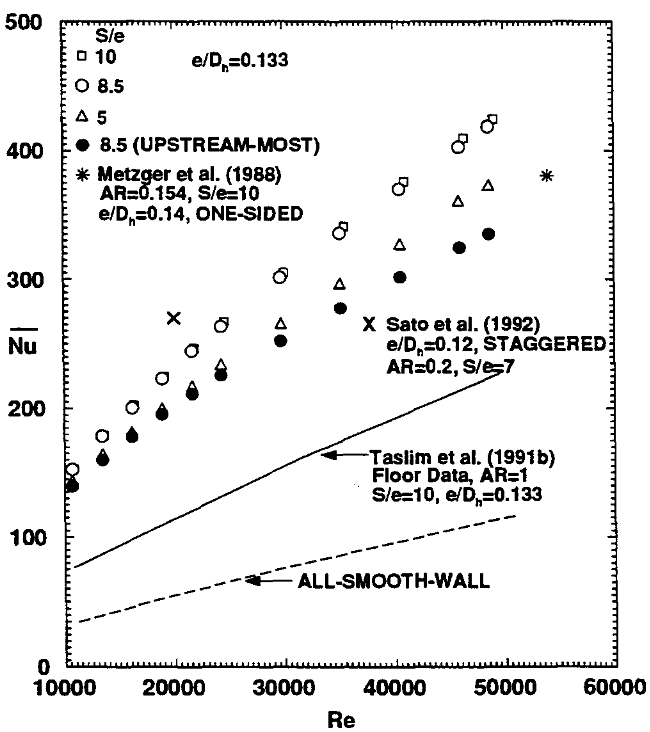

Figure 8 Rib Average Nusselt Number for a Range of Pitch-to-Height Ratios, $e/D_h = 0.133$

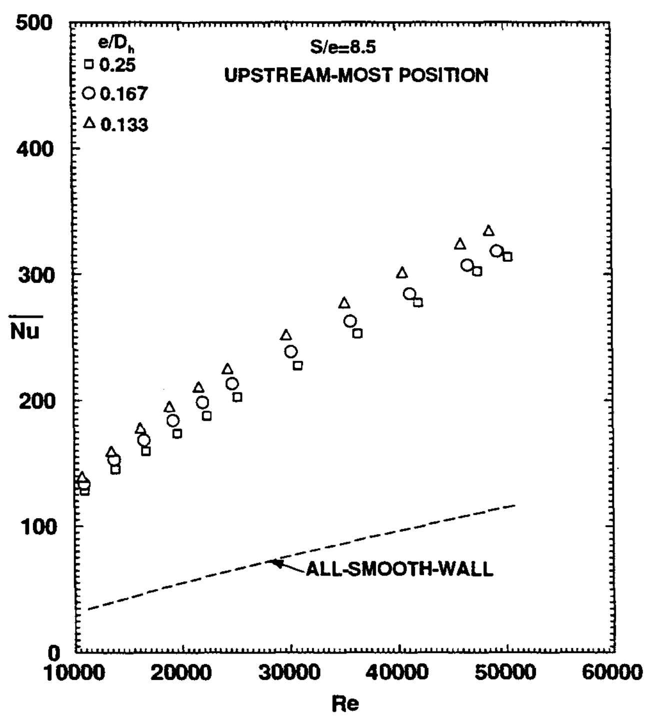

Figure 10 Average Nusselt Number of the Upstream-Most Ribs for a Range of Blockage Ratios

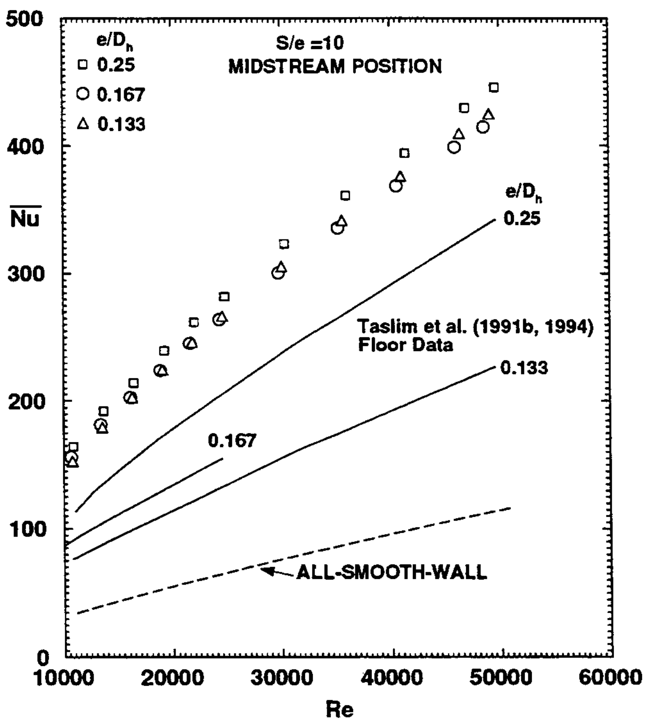

Figure 11 Rib Average Nusselt Number for a Range of Blockage Ratios, S/e = 10

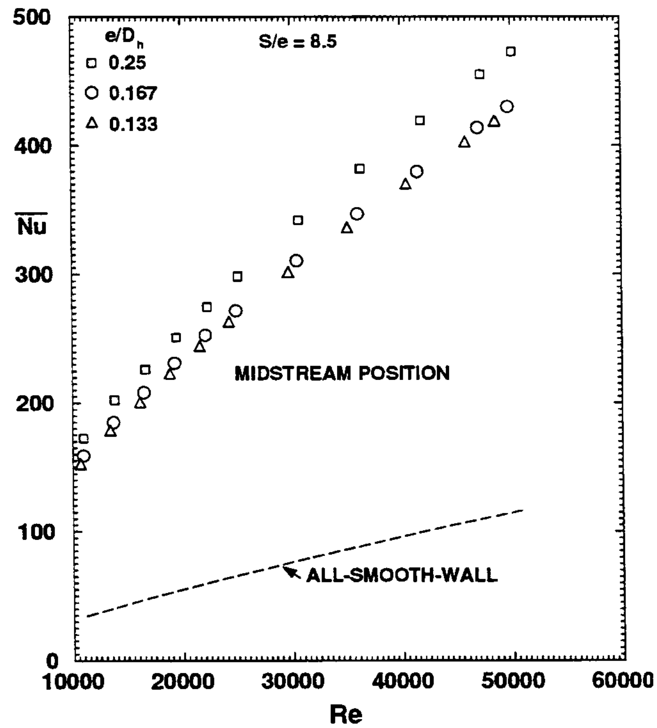

Figure 13 Rib Average Nusselt Number for a Range of Blockage Ratios, S/e = 8.5

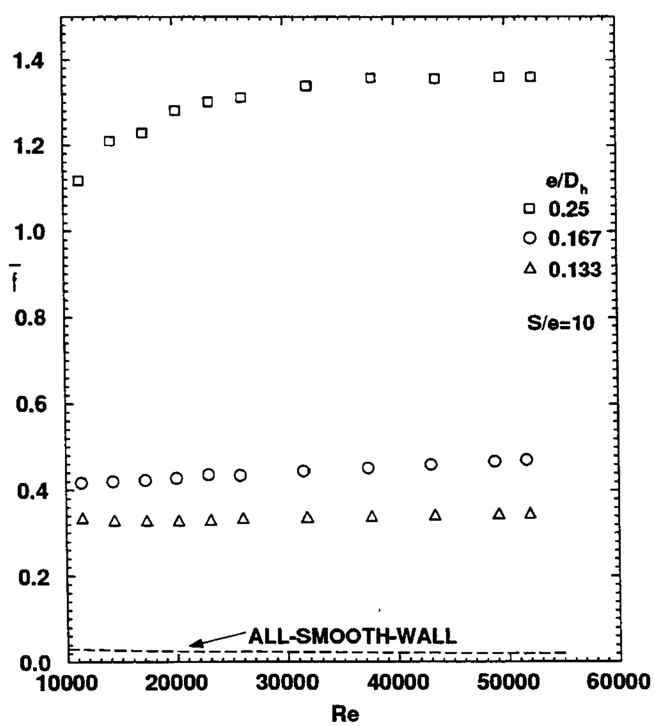

Figure 12 Channel Average Friction Factor for a Range of Blockage Ratios, S/e = 10

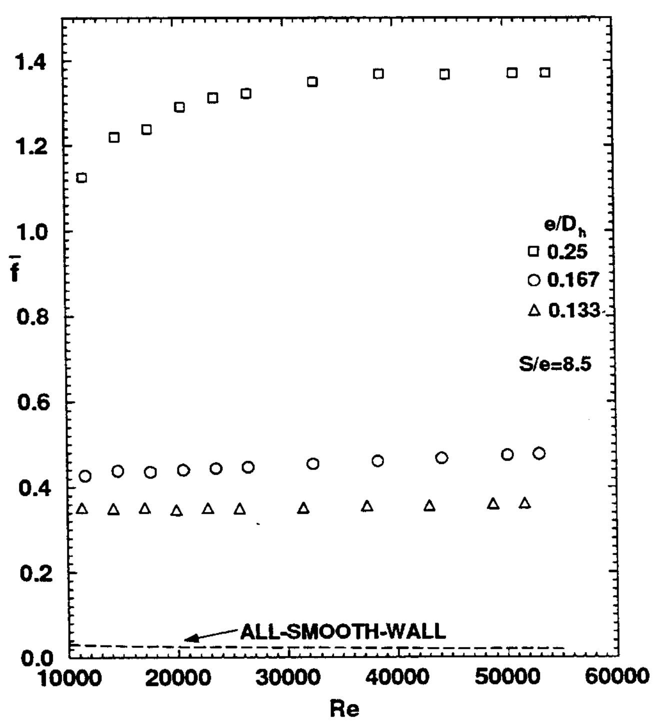

Figure 14 Channel Average Friction Factor for a Range of Blockage Ratios, S/e=8.5

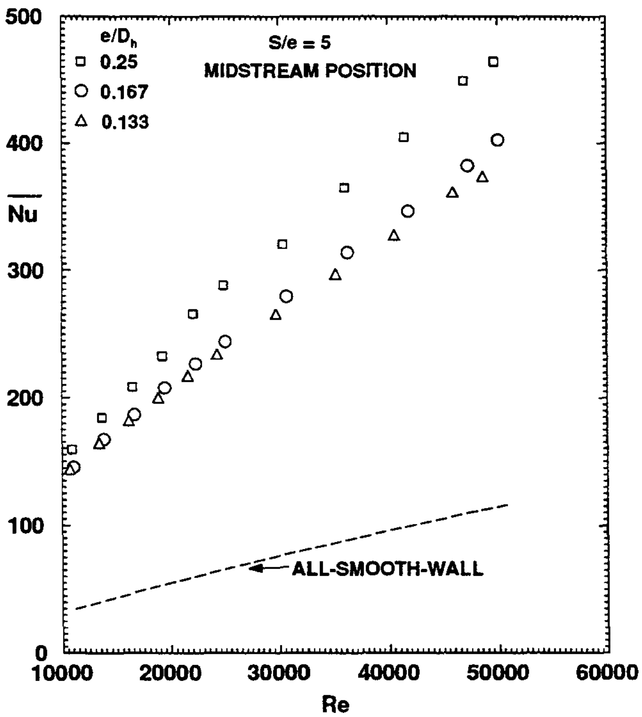

Figure 15 Rib Average Nusselt Number for a Range of Blockage Ratios, S/e = 5

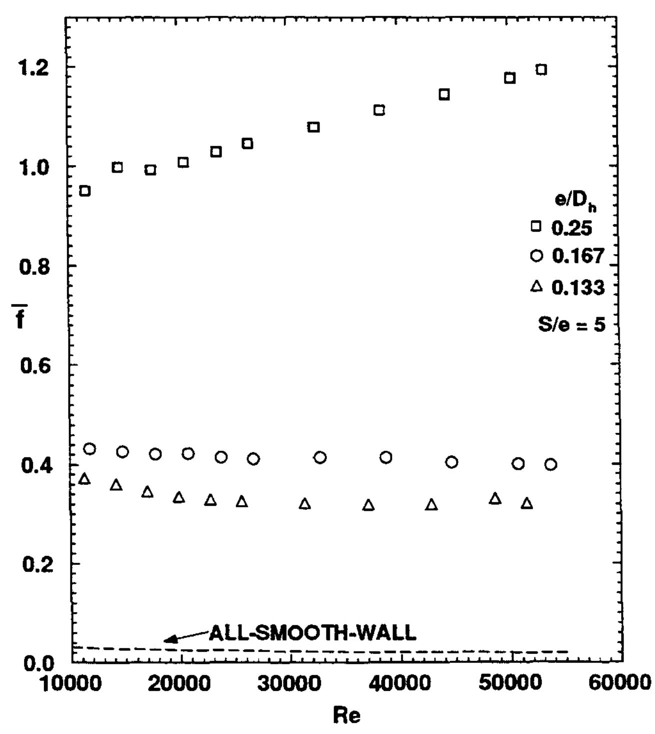

Figure 16 Channel Average Friction Factor for a Range of Blockage Ratios, S/e = 5

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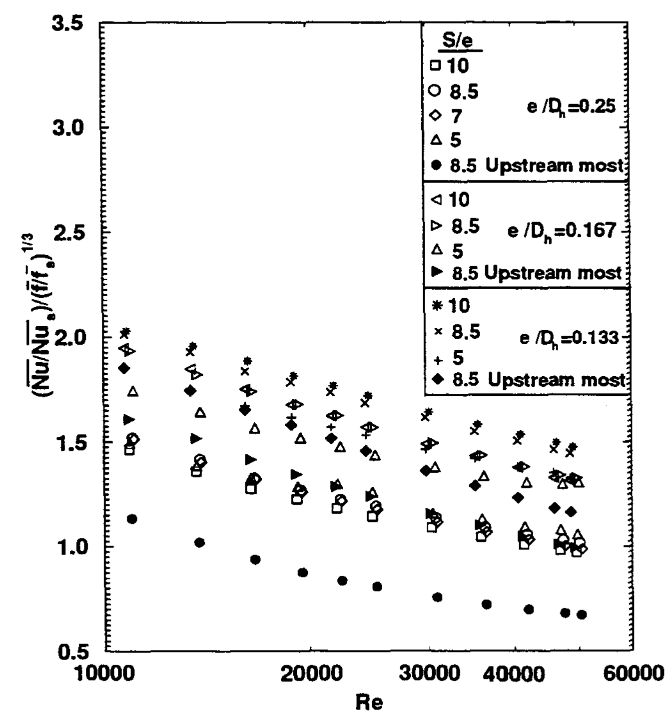

Figure 17 Thermal Performances of the Thirteen Geometries

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