EFFECT OF PERMEABLE RIBS ON HEAT TRANSFER AND FRICTION IN A RECTANGULAR CHANNEL

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ABSTRACT
Heat transfer and friction characteristics in a rectangular channel with perforated ribs arranged in-line on two opposite walls are investigated experimentally. Five perforated rib open-area-ratios (0, 10%, 22%, 38%, and 44%) and three rib pitch-to-height ratios (10, 15, and 20) are examined. The Reynolds number ranges from 5000 to 50000. The rib height-to-channel hydraulic diameter ratio and the channel aspect ratio are 0.081 and 4, respectively. Laser holographic interferometry is employed not only to measure the heat transfer coefficients of the ribbed wall but also to determine the rib apparent permeability. It is found that ribs with appropriately high open-area-ratio and high Reynolds number are permeable, and the critical Reynolds number for evidence of flow permeability decreases with increasing the rib open-area-ratio. Results of local heat transfer coefficients further show that the permeable ribs have an advantage of obviating the possibility of the hot-spots. Moreover, the duct with permeable ribs gives a higher thermal performance than that with solid-type ribs.

NOMENCLATURE
A width of channel
AR channel aspect ratio, A/B
B height of channel
cp specific heat at constant pressure
De hydraulic diameter, 2B(1+B/A)
f friction factor, equation (5)
H rib height
kf film temperature, air thermal conductivity
Lw wetted length in one pitch
ΔL channel length for fully developed pressure drop
m mass flow rate
n number of holes in a rib
Nu local Nusselt number
NuP periodic fully developed (average)Nusselt number for the ribbed duct
NuS average Nusselt number for the smooth duct (at the same mass flow rate)
NuS* average Nusselt number for the smooth duct (at the same pumping power)
Pi rib pitch
Pr Prandtl number
ΔP pressure drop across the fully developed test section
cconv convective heat flux from the wall
Re Reynolds number, UDe/v
Reb Reynolds number based on the boundary layer thickness, U/W
T temperature of air
Tb local bulk mean temperature of air
TbP average bulk mean temperature of air, \(\int_{0}^{L} T_b dX\)/L
Tf film temperature, \(T_w + T_b/2\)
Tm air temperature at duct inlet (i.e. room temperature)
Tw local wall temperature
Twall average wall temperature
dT/dY air temperature gradient
U average channel velocity without ribs
W rib width
Xp axial coordinate (Xp=0 at inlet reference, Fig.2)
Xn axial coordinate (Xn=0 at rib rear edge, Fig.2)
Y transverse coordinate, Fig.2
Z spanwise coordinate, Fig.2

Greek Symbols
α angle of attack
β open-area-ratio of the perforated rib, equation (1)
δ boundary layer thickness
ρ air density
φ radius of the hole distributed over the perforated rib

Subscripts
b bulk mean
N rib index
n number of the holes drilled through the perforated ribs
s smooth
w wall

INTRODUCTION

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To increase the specific thrust and to reduce the specific fuel consumption (SFC), high turbine entry gas temperature (1400 – 1600 deg C) has become the trend in advanced aero-engine design. Such a high gas temperature is far above the allowable metal temperature; therefore, turbine blades must be cooled in order to operate in the high gas temperature environment. Turbulence promoters inside cooling passages enhance the overall heat transfer to the cooling air. Researchers have modeled these ribbed cooling passages as straight rectangular channels with two opposite ribbed walls and two smooth walls. Earlier evident works include: Burggraf (1970) reported the results of turbulent airflow in a square duct (AR=1) with transverse solid-type ribs (β=90 deg, α=0) on two opposite walls for the Reynolds number ranged from 1.3x10^6 to 1.3x10^7. The wall temperature distributions were measured by thermocouples. With a hydrodynamically fully developed condition at the heated duct entrance, the average Nusselt number of the ribbed side wall and the friction factor were approximately 2.38 times and 8.6 times the corresponding values for fully developed smooth duct flows. The average Nusselt number of the smooth side wall was 19% over that of the duct with four smooth walls. Similar trends were obtained for three channel entrance geometries (long duct, short duct, and bended entrance). Tanasawa et al. (1983) employed the resistance heating method and thermocouple technique to determine the heat transfer coefficients in a channel with turbulence promoters. Three types of turbulence promoters, namey, fence-type, perforated plate-type, and slitted plate-type, were tested in their work. Results showed that the surfaces with perforated plate-type turbulence promoters gave the excellent performance under the constant pumping power conditions. Note that the thin plates were insulated; and therefore the additional surfaces caused by the existence of plates have no contribution to the heat transfer enhancement. Han (1984) conducted experimental works to examine the effects of the rib pitch-to-height ratio (Pi/H=10, 20, and 40) and the rib height-to-hydraulic diameter ratio (H/De=0.0221, 0.0422, and 0.0630) on the heat transfer coefficient and friction of the fully developed airflow in a square duct with two opposite ribbed walls. The temperature distribution were measured by thermocouple. The results showed that the Stanton number and friction factor of the ribbed duct were about 1.5 to 2.2 times and 2.1 to 6 times, respectively, those of the smooth duct for the range of the test data. Later, Han (1988) investigated the effect of the channel aspect ratio (AR=1/4, 1/2, 1, 2, 4) on the distributions of local heat transfer coefficients in channels with two opposite ribbed walls. Both the local and average Nusselt numbers were measured by the thermocouple technique and resistance (stainless steel foil) heating method. The results were obtained for the solid-type rib with angle-of-attack 90 deg. It was found that the increased ribbed-side-wall heat transfer in a smaller aspect ratio channel was higher than that in a large aspect ratio channel for a constant pumping power; however, the increased average heat transfer was slightly lower. Lockett and Collins (1990) conducted the double-exposure holographic interferometry measurement in a fully developed channel flow with square and rounded rib—roughness on one wall. The ribs were solid. One rib pitch—to—height ratio (Pi/H=7.2) and one rib— to—channel height ratio (H/De=0.106) were investigated in their work. It was found that the heat transfer distribution was Reynolds number dependent for the rounded rib, but independent for the square rib. Lau et al. (1991) conducted experiments to study the effects of replacing the aligned 90 deg full ribs on two opposite walls of a square channel with angled discrete ribs (five equal segments of the angled full ribs staggered, that is, the compound array of three and two ribs) of turbulent heat transfer and friction for fully developed airflow. The temperature distributions were measured by thermocouple. Results showed that parallel 60 deg discrete ribs had the highest ribbed wall heat transfer, parallel 30 deg discrete ribs caused the lowest pressure drop, and crossed arrays of angled ribs had poor thermal performance and were not recommended. Liu and Hwang (1993) experimentally studied the effect of the rib shapes on the heat transfer and friction characteristics in periodic fully developed duct flows. Three rib shapes (square, triangular, and semicircular) with the same rib height (H/De=0.081) were investigated in their work. The local as well as the average Nusselt numbers were determined by a real-time laser holographic interferometry (LHI). It was found that the three shaped ribs had comparable thermal performances under the constant pumping power constraint. As indicated in the above discussion, the relevant geometric parameters involved in the above investigations are passage aspect ratio, AR; rib angle-of-attack, α; rib pitch-to—height ratio, Pi/H; blockage ratio, H/De; rib shapes; and the manner by which ribs are positioned with respect to each other. However, a serious problem still remains and must be overcome: Previous measurements (Lockett and Collins, 1990; Liou and Hwang, 1992a, 1992b, 1993) and calculations (Liu et al., 1992, 1993) of detailed local Nusselt number distributions show that the hot—spots (Nu/Nu<1) exist in the recirulating region behind the solid—type rib because the flow is nearly stagnant relative to the main stream in this region. The hot—spots will deteriorate blade materials. It is important to search for an efficacious rib configuration for improving heat transfer in the recirulating region. To fulfill this motivation, the effect of replacing the solid—type ribs by the perforated ribs on the local heat transfer characteristics will be examined in this paper. The main objective of this paper is to study the effect of replacing the solid—type rib by the perforated ribs on the local heat transfer characteristics. First, flow visualization is conducted to determine the permeability of the perforated rib. A criteria of the rib permeability is proposed as a function of the rib open—area—ratio and the Reynolds number. Then, laser holographic interferometry is employed to measure the local heat transfer coefficient distribution of the ribbed wall. Since the airflow partly passes through the permeable rib and directly impinges the recirulating cell behind the rib, it is of interest whether the channel with permeable ribs can improve the heat transfer rate in this region. Finally, the thermal performances of the perforated ribbed ducts are quantified with respect to a corresponding smooth duct. It is questionable whether the perforated ribs can provide a better thermal performance than the solid—type ribs. The parameters investigated in this work are the rib open—area—ratio, β=0, 10%, 20%, 22%, 38%, and 44%; the rib pitch—to—height ratio, Pi/H=10, 15, and 20; and the Reynolds number, 5000<Re<50000. The rib height—to—channel hydraulic diameter ratio and the channel aspect ratio are fixed at values of 0.081 and 4, respectively. The ribs are square with sharp—edged corners.

EXPERIMENTAL APPARATUS

Instrumentation

In this work, the temperature distribution of airflow in the ribbed duct is measured by a real—time holographic interferometry (single exposure method, Liou and Hwang, 1992a). The overall arrangement of the holographic interferometer is illustrated in detail in Fig. 1. The coherent source used is a high—power, argon—ion laser, Spectra—Physics Model 2000. The photographic emulsion BES5 made by Agfa—Gevaert Ltd. is found to be suitable for recording material for combining a good compromise between light sensitivity and resolution. The instantaneous interference field is digitized by a CCD camera (COHU, Model 6400) which allows 512x512 pixel resolution with 256 grey levels per pixel and recorded on a VHS videocassette recorder for storage and further image processing.

While the flow field temperature is measured by LHI, the wall temperature of the test section is measured by thermocouples. Copper—constantan thermocouples (i.e. T type) are used to measure the local wall temperature of the ribbed duct. The junction—bead of the thermocouple is about 0.15—mm in diameter. The temperature signals are transferred to a hybrid recorder (Yokogawa, DA—2500) with 30 channels. All of the data are then sent to a PC—AT via GPIB interface. The pre—processing of the raw data can be carried out by...
using a built-in BASIC program by which the non-dimensional parameter can be calculated.

**Test Model**

Figure 2 shows the coordinate system, configuration, and dimension of the test duct. The test duct is 1200-mm long and has a rectangular cross section of 160mm by 40mm (YZ plane). As shown in Fig.2, the perforated ribs are attached symmetrically to the top and bottom walls (aluminum plates, 3-mm in thickness) of the test ducts. The rib angle-of-attack is 90 degree. Aluminum plates and ribs are adopted in this work for their high conductivity and machinability. Thermofoils of thickness 0.18mm are adhered uniformly between the aluminum plate and a 6-mm-thick fiberglass board to insure good contact. In addition, two pieces of balsa wood (20-mm in thickness) are used to prevent the heat loss from the upper and lower sides of the heated plates. The thermal resistance of the glue (0.13-mm thick or less) used at each of the above-mentioned interfaces is negligible (less than 2%). The region of optical view is instrumented with 28 thermocouples distributed along the spanwise centerline (Z=0) of the heated plate and ribs for wall temperature measurements, as shown in Fig.2. Two pressure taps are used to measure the static pressure for the fully developed duct flows.

Figure 3 shows a photograph of the perforated ribs investigated in this work. The rib open-area-ratio (\( \beta \)) is defined as

\[
\beta = \left( \frac{n \pi r^2}{4A_H} \right) \tag{1}
\]

where \( n \) is the number of the holes drilled through on the perforated rib, \( r \) the radius of the hole, \( A \) the width of the channel (i.e., the length of the perforated rib), and \( H \), the rib height. In this work, the rib open-area-ratios investigated are 0, 10%, 22%, 38%, and 44%; the rib pitch-to-height ratios are 10, 15, and 20; the Reynolds number, based on the duct hydraulic diameter and bulk mean velocity, extends from \( 5.0 \times 10^3 \) to \( 5.0 \times 10^4 \); and the rib-to-channel height ratio (or the ratio of the rib height-to-channel hydraulic diameter) is 0.13 (0.081).

**EXPERIMENTAL CONDITIONS AND DATA ANALYSIS**

Two-dimensionality of the actual temperature field, thermal boundary conditions of the test section, and analysis of the interference fringe had been described in detail in Liou and Hwang (1992a), and is not elaborated on in this paper.

**Heat Transfer Coefficient**

In this study the entire temperature field is revealed by the infinite-fringe interferometry and subsequently enables the calculations of local and average heat transfer coefficients of the heated surface. The convection heat transfer coefficient can be presented in terms of the local Nusselt number \( Nu \), which is defined as

\[
Nu = \frac{-\left( \frac{dT}{dY} \right)_{w} \cdot De}{\left( \frac{T_w - T_0}{T_0} \right)} \tag{2}
\]

where the air temperature gradient \( \left( \frac{dT}{dY} \right)_{w} \) is determined by curve fitting, based on a least-squares method through the near wall values for temperature and fringe shift; \( T_w \) is read from the thermocouple output; and \( T_0 \) is calculated from an energy balance, \( T_0 = T_w + \frac{Q}{mc_p} \), where \( Q \) is the quantity of heat given to air from entrance to the considered cross section of the duct and can be obtained by the integrated form of \( \int_{Z}^{A} \left[ (k_r \cdot \left( \frac{dT}{dY} \right)_{w} \cdot A \right] \cdot dX \). The maximum uncertainty of local Nusselt number is estimated to be less than 6.5% by the uncertainty estimation method of Kline and McClintock (1953). The average Nusselt number is evaluated by the following equation

\[
\bar{Nu}_p = q_{conv} \cdot De \cdot (k_r \cdot (T_w - T_0)) \tag{3}
\]

where \( q_{conv} \) is the convective heat flux from the ribbed wall and is estimated by subtracting the heat loss from the supplied electrical input (Liou and Hwang, 1992a). The maximum uncertainty of \( \bar{Nu}_p \) was estimated to be less than 9.8%. The local and average Nusselt
numbers of the present study are normalized by the Nusselt number for fully developed turbulent flow in smooth circular tubes correlated by Dittus-Boelter as:

\[ \text{Nu110u} = \text{Nu}/(0.023 \cdot \text{Re}^{0.8} \cdot \text{Pr}^{0.4}) \]  

(4)

**Friction Factor**

The friction factor of the periodic fully-developed flow is expressed as:

\[ f = \frac{(-\Delta P/\Delta L)}{\rho U_0^2} \cdot \frac{D_e}{L} \]  

(5)

where the pressure gradient, \( \Delta P/\Delta L \), is evaluated by taking the ratio of the pressure difference and the distance of two successive pressure taps. The maximum uncertainty of \( f \) is estimated to be less than 7.3%.

**RESULTS AND DISCUSSION**

**Interference Patterns**

Typical examples of the interferograms taken from the temperature fields of the perforated-rib and the solid-type-rib geometries are shown in Figs.4(a)–(g). Figures 4(a)–(f) show the finite-fringe(also called wedge-fringe) interferograms. If there are no disturbances in the field, parallel, equally spaced, and alternately dark and bright fringes will appear on the interferogram, as shown in Fig.4(a). When a disturbance is present with the test section, the optical path is no longer uniform. The fringes then are no longer straight, but curved. The disturbed finite-fringe interferograms for the different rib open-area-ratios are shown in Figs. 4(b)–(f) at \( \text{Re}=20000 \). For the solid-type rib(\( \beta=0 \)) in Fig.4(b), there is no fluid passing through the rib, and the total fluid has to turn from the duct wall into the contraction between the two opposite ribs. It can be observed that the fringes are highly distorted in the regions of the flow over and behind the rib top. This indicates that the flow introduces a strong shear layer from the rib top, which drives the recirculating flow behind the rib (Fig.5(a)). At the lowest rib open-area-ratio, \( \beta=10\% \) (Fig.4(c)), the distorted fringes seem to have no difference from those of the solid-type rib, and the fluid still passes through the rib contraction only. Basically, it is impermeable. As the rib open-area-ratio is larger than 22\% (Figs.4(d)–(f)), the saw-shaped fringes found behind the rib reveal that a part of fluid passes through the rib and the separation bubble behind the rib is thus broken up. This can be supported by a comparison of the flow visualization between Fig.5(a) and Fig.5(b), which shows that the separation bubble behind the solid-type rib disappears for the perforated rib geometry. Moreover, the distorted region on the top of the perforated rib becomes thinner than that on the solid-type rib. This indicates that the high convective heat transfer from the rib-top surface is accompanied with the solid-type rib. The reason is that for the perforated rib a large amount of heat has been convected by the fluid that passes through the rib, which is conducted from the rib base, hence a reduction of the heat transfer rate on the rib top. This is
reflected by the lower local Nusselt number distribution, and will be shown later. Figure 4(g) is a typical isotherm-pattern interferogram (infinite fringe set) for the perforated rib geometry. From the information of the whole-field air temperature distributions given by the interferograms (infinite fringe set), the local heat transfer coefficient of the perforated-ribbed walls can be calculated.

**Permeability Limit**

In accordance with the above flow visualization results, the permeability limit of the perforated rib is plotted as a function of the Reynolds number and the rib open-area-ratio, as shown in Fig. 6. The half-solid symbols are the actual values obtained from the experiment, and the error bounds are caused by the unsteady or intermittent appearance of the separation bubble (or saw-shaped fringes). The solid-curve passing through these symbols is a curve-fitting result. The permeability limit is a criterion of the change of the flow patterns. When the ribs are permeable (above the solid curve), the flow pattern of the multi-mixing-layers appears behind the rib, which is caused by the multi-jets emitting from the rear face of the rib. For data lying below the solid curve, the ribs are impermeable, and typical flow patterns of separation, reattachment, and recirculation are found. It can be seen from this figure that the impermeable zone is found to be in the region where the values of flow Reynolds number and the rib open area ratio are lower. The critical Reynolds number of initiation of flow permeability decreases with increasing the value of the rib open-area-ratio. Note that for the range of the Reynolds number investigated, the ribs with $\beta = 10\%$ are impermeable. A critical value of $\beta$ exists between $10\%$ and $22\%$, which makes the rib permeable for the range of Reynolds number investigated. The solid and open circular points in Fig. 6 are the results obtained by using smoke-wire flow visualization (Yomada and Osaka, 1992). In their work, the authors used a single perforated rectangular plate to stand against a flat wall where turbulent boundary layer is developing. The Reynolds number based on the boundary layer thickness is 3150. It was concluded that the critical value of $\beta$ is between $32.5\%$ and $48.5\%$, and below which there is a recirculation cell behind the plate. As seen in Fig. 6, the results obtained in the previous work are very satisfactory for the solid-curve obtained in this work. Note that the boundary thickness in this work is half of the channel height because the investigated rib-pairs are located at the fully developed region of the channel (Liou and Hwang, 1992b).

**Local Nusselt Number**

Figure 7 shows the distributions of the local Nusselt number ratio

![FIGURE 6 DEPENDENCE OF PERMEABILITY LIMIT AS A FUNCTION OF REYNOLDS NUMBER AND RIB OPEN-AREA-RATIO.](http://proceedings.asmedigitalcollection.asme.org/)

![FIGURE 7 LOCAL NUSSELT NUMBER DISTRIBUTIONS ALONG THE PERFORATED RIBBED SURFACES.](http://proceedings.asmedigitalcollection.asme.org/)

for various $\beta(10, 22, 38, \text{ and } 44\%)$ and at a fixed $Re(20000)$. The dotted line is the results of the solid-type ribbed wall. Similar to the solid-type rib results, the $Nu/Nu_0$ distributions for the perforated rib investigated show the presence of the local maximum of $Nu/Nu_0$ on the upstream tip of the rib due to forced convection augmented by flow acceleration. However, the value is lower than that of the solid-type rib and decreases with an increase of $\beta$. This is reasonable because for the ribbed-wall with a large value of $\beta$, a large amount of heat conducted from the rib base has been convected by the airflow through the rib, and therefore, the convective heat to the rib top (or convective heat from the rib-top surface to the test section) is reduced. In addition, the augmented forced convection between the two opposite ribs decreases as $\beta$ increases due to the reduction of the blockage ratio. Concerning the results between the ribs, it can be observed that the difference between the permeable ($\beta \geq 22\%$) and impermeable ($\beta \leq 10\%$) rib geometries is evident. The results of the duct with solid-type ribs have been described in detail in Liou and Hwang (1992a). For the duct with permeable ribs, there are two-peak values of $Nu/Nu_0$ within the streamwise distance of the two successive ribs. The first local peak value is located at the duct wall just downstream of the rib, and it increases with an increase of $\beta$. Although the level of turbulence intensity is not measured quantitatively in this work, it is believed to be caused by the effect of the intense jet turbulence generating from the rear of the rib by the qualitative observation of the highly fluctuating fringes in this region. The second peak located in the middle of the successive ribs is considered to be the reason of an approach of the shear layer from the rib top to the duct wall. As $\beta$ increases, the second peak of $Nu/Nu_0$ moves downward because the flow rate through the perforated walls increases with an increase of $\beta$. Note that the hot-spots ($Nu/Nu_0 = 1$) around the concave corner behind the solid-type rib do not arise in the permeable-ribbed wall. In this work the augmentation of the heat transfer is due to the combined effects of the extended surfaces and the enhanced turbulence (Liou et al., 1992). Considering the effect of the enhanced turbulence only, one may integrate the local heat transfer coefficient along the duct wall, ie., $X_w/H=0.9$ (Liou and Hwang, 1992a). As shown in Fig. 7, the value of the Nusselt number averaged over the duct wall between two successive ribs (i.e., $Nu_{avg+0.9}$) for the permeable rib geometry is higher than that for the solid-type rib geometry (Fig. 7, dotted line), typically $Nu_{avg+0.9} = 2.3$ and $2.1 Nu_0$ for $\beta = 44\%$ and 0, respectively. This result reveals that the heat transfer augmentation...
due to the enhanced turbulence for the permeable rib geometry is higher than that for the solid-type rib geometry. Such a trend agrees with that in Tanasawa et al. (1983), in which thin, insulated perforated plates were used as turbulence promoters, i.e., the effect of the extended area of the plate is not considered.

**Average Nusselt Number and Friction Factor**

Ribs with a relatively high open-area-ratio, coolant flow passing through the ribs, are accompanied with a higher heat transfer area as compared with the solid-type ribs. To place the results on a common basis, the averaged Nusselt number (equation 4) is based on the projected area of the corresponding ribless wall. Thus the magnitude of Nu can reflect the combined effects of the extended surfaces provided by the ribs and the enhanced turbulence by distortion the velocity and temperature fields caused by the presence of ribs (Liou et al., 1992). Figure 8 gives the average Nusselt number of the perforated-ribbed walls as a function of Reynolds number. The solid line (Liou and Hwang, 1992a) is the results of the solid-type ribbed wall. It can be observed from this figure that for all of the rib open area ratios investigated the heat transfer augmentation ($\text{Nu}_{\text{p}}$/$\text{Nu}_{\text{s}}$) is achieved. The increments of the average heat transfer coefficients for the permeable-ribbed wall are about the same as those for the solid-type-ribbed wall, typically 120-180% as compared with the smooth-duct results for the range of the investigated Reynolds number. However, for the impermeable perforated-ribbed-wall, i.e., $\beta=10\%$ for all $Re$ investigated and $\beta=22\%$ at $Re=10000$, the heat transfer coefficients are slightly lower than those of the permeable or solid-type ribbed wall. The explanation of this fact is as follows: As shown in Fig.7, the solid-type and impermeable perforated ribbed geometries have the comparable heat transfer enhancements in the interrib region ($X_\text{p}/H=0-9$). However, the heat transfer rate for the top wall ($X_\text{p}/H=1-0$) of the impermeable perforated rib ($\beta=10\%$) is slightly lower than that of the solid-type rib. The lower heat transfer rate for the impermeable perforated ribs may be due to a reduction of the effective conductivity of the rib caused by the presence of the stagnant air in the holes. Thus, the averaged rib and duct-wall heat transfer coefficient for the impermeable perforated ribbed geometry is slightly lower as compared to that for the solid-type rib one. In fact, channels with solid-type ribs and impermeable perforated ribs may be considered simply as channels with relatively high and low conductivity ribs, respectively. The higher heat transfer enhancement associated with the channel with higher conductivity ribs is physically

![FIGURE 8 AVERAGE NUSSELT NUMBER VersUS REYNOLDS NUMBER.](image)

**FIGURE 9 FRICTION FACTOR VS. REYNOLDS NUMBER.**

reasonable. (Liou and Hwang, 1993).

The effect of the open-area-ratio of the rib on the fully developed friction factor is shown in Fig.9. The pressure drops across the test channel are measured by the unheated ow conditions. The friction factor for the perforated rib is higher than its counterpart for the smooth channel (dashed line, Blasius correlation), but lower than that for the solid-type rib (dotted line, Liou and Hwang, 1992b). It is almost constant regardless of the values of $Re$. In the case of $\beta=10\%$, the value of $f$ is found to be almost the same as that of the solid-type rib. This reflects the fact of impermeability for $\beta=10\%$ concluded before. In comparison with the results of the duct flows with solid-type ribs, the values of $f$ are approximately 100, 85, 75, and 60% of that of the solid-type rib for $\beta=10$, 22, 38, and 44%, respectively, in the range of the Reynolds number investigated. As expected, for a given Reynolds number the friction factor decreases with increasing $\beta$ because of the less cross-sectional blockage for ribs with the larger $\beta$. The effects of Reynolds number and rib open-area-ratio on the friction factor can be correlated as follows:

$$f = 0.1735 \beta^{-0.163}$$

Note that the above correlation is valid only for the permeable rib.

**Performance Comparison**

General tendency found in the previous discussion of the solid-type rib is that the value of $f$ is large when $\text{Nu}_{\text{p}}$ is large. From the results of the moderate heat transfer enhancement and lower pressure drop penalty achieved by the large value of $\beta$, a high thermal performance under the constant pumping power constraint, may be expected to be accompanied with a large value of $\beta$. The pumping power required to feed the fluid through the duct is proportional to $fRe^3$. Thus in Fig.10 the performance shown by the ratio of $\text{Nu}_{\text{p}}$/$\text{Nu}_{\text{s}}$ is plotted against $fRe^3$ (Tanasawa et al., 1983; Liou and Hwang, 1993). Figure 10 shows that the improvement in Nusselt number ratio of the permeable ribbed duct is more pronounced than that of the solid-type ribbed duct (dotted line). Note that at lower Reynolds number both the perforated and solid-type ribbed geometries give the higher thermal performance than those at higher Reynolds number. Therefore, the usage of the perforated ribs in the large rib open-area-ratio and the low Reynolds number range
CONCLUDING REMARKS
Manufacturing of blades with permeable turbulators may be technically difficult up to now. However, it is important to understand the characteristics of heat transfer and fluid flow in channels with permeable ribs before the exact application. Moreover, information of flow over permeable ribbed walls is useful for the general area of heat exchangers. In this work, the turbulent heat transfer and friction in a channel with perforated rib on two opposite walls have been studied experimentally. The main findings are: A permeability criteria of the perforated rib is presented as a function of the rib open area ratio and the Reynolds number. The critical Reynolds number of initiation of flow permeability decreases with increasing the rib open-area-ratio. The high rib open-area-ratio and Reynolds number allow the ribs to be permeable. Results of the local heat transfer coefficient distributions reveal that the hot-spots occurring in the region around the concave corner behind the solid-type rib do not arise in the corresponding region of the permeable-rib geometry. As compared with the conventional solid-type rib results, the average Nusselt number/friction factor of the perforated rib geometry are about 75%/100%, 100%/85%, 100%/75%, and 105%/60%, respectively, for $\beta$=10%, 22%, 38%, and 44%. The moderate heat transfer coefficient and lower pressure drop accompanied with the large value of $\beta$ reflect a higher thermal performance.

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