Heat/mass transfer and pressure drop in a triangular-rib-roughened rectangular channel

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In this paper, the heat/mass transfer analogy was used to investigate the heat transfer and pressure drop in a square channel with triangular ribs on its two opposite walls. Reynolds number varied from 1×10^4 to 7×10^4; the dimensionless heights of the triangular ribs H/W were 0.04, 0.07, and 0.1; and their dimensionless pitches S/W were 0.45, 0.63, 1.0, 1.37, 1.55, and 2.1. Experimental results showed that the heat transfer coefficients of the wall with triangular rib were about 1 to 2.3 times larger than those of a smooth-channel wall, and the pressure drops along this roughened channel were about 1 to 10 times larger than those for a smooth channel. Correlations of heat transfer and pressure drop were obtained, which are useful for practical designs.

Keywords: heat transfer; pressure drop; roughened surface

Introduction

The effect of rib-roughness geometry on heat/mass transfer and friction characteristics for a rib-roughened channel is studied in this paper. Because of this geometry's broad range of engineering applications, many studies have been performed in this area. For example, Sparrow et al. (1983) investigated the effect of a cylindrical rough rib on the local heat transfer of a roughened wall and obtained the relations of Reynolds number, dimensionless rib heights, and dimensionless pitches to heat transfer and friction characteristics. More attention has been paid to studies of ribs having a rectangular or square cross section. For example, Burggraf (1970) studied the heat transfer and friction in a square channel with rectangular rough ribs attached on two opposite walls. Han et al. (1984, 1988a, 1988b) systematically investigated the effects of the ratio of rib pitch to height and the ratio of height to hydraulic diameter on the friction factors and heat transfer coefficients for fully developed turbulent air flow in a square channel with two opposite square-rib-roughened walls. When the performances of ribs with crossed oblique angles were compared with those of transverse ribs by Zhang et al. (1984), the results showed that the heat transfer of transverse ribs is better. To further enhance the heat transfer, Zhang et al. (1987) used a rib-slot-type rough rib, and the results of their study showed that the heat transfer performance of this kind of arrangement is even better, but the pressure drop is very large. Ichimiya et al. (1987) investigated the effects that ribs on an insulated wall with different shapes or orientations had on heat transfer to the opposite, smooth, heated wall in a parallel plate duct. The purpose of using an unheated rib-roughened wall is to examine the effects of roughened elements on heat transfer performance alone, i.e., to remove the interactive effects of the extended surface and the turbulence. Therefore, in the Ichimiya et al.'s (1987) studies, only the local heat transfer performance on the opposite smooth heated wall was measured. The variations in performance were induced solely by the different shapes or orientations of the roughened elements. It is evident that their study is very fundamental. The shapes of ribs on the insulated plate are square or triangular, and the authors concluded that when the oblique plane of the triangular rib is positioned downstream to the flow, as shown in Figure 1, the heat transfer performance will be the best. But in practical applications, both opposite walls should be roughened so as to obtain better heat transfer performance. This conclusion was the motivation for the present study.

In this paper, the investigation focused on the heat/mass transfer and friction characteristics of triangular ribs that were transversely installed on two opposite walls of a square channel, as shown in Figure 1. The goal of this study was to enhance the heat transfer in a rectangular or square channel and also in an internal cooling passage for gas turbine blades. The heat transfer test was performed using the naphthalene sublimation technique. The Reynolds number varied from 1×10^4 to 7×10^4. The ratios of rib height to hydraulic diameter (H/W) were 0.04, 0.07, and 0.1. The ratios of rib pitch to hydraulic diameter (S/W) were 0.45, 0.63, 1.0, 1.37, 1.55, and 2.1. The roughened ribs had an isosceles-triangle shape. The triangular ribs were attached on both the top wall and the bottom wall in a symmetrical fashion.

Experimental apparatus and data reduction

The entire test rig and the instrumentation were located in an air-conditioned laboratory, which was maintained at a constant temperature over the entire period of the test.

A schematic sketch of the experimental apparatus is shown in Figure 2. The test system mainly consists of a test section, a data acquisition system, and a centrifugal blower. The suction-mode wind tunnel is used to avoid the disturbance and the temperature rise of the incoming air due to compression;

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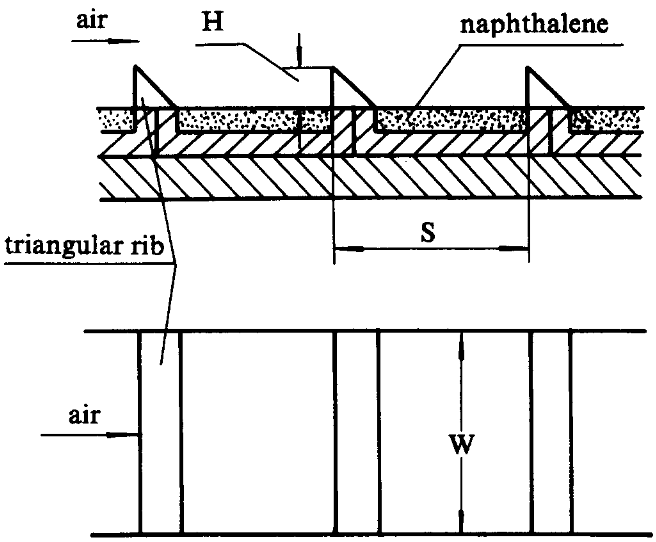

Figure 1 Schematic diagram of the triangular-rib-roughened channel (dimensions of rib are exaggerated)

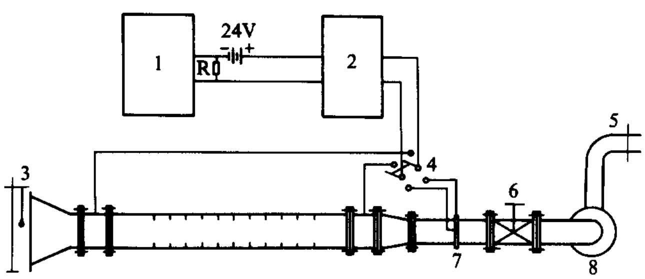

Figure 2 Outline of the testing system. 1: Data acquisition system; 2: 1151 differential pressure transducer; 3: thermometer; 4: selecting switch; 5: atmosphere; 6: regulator; 7: flute tube; 8: blower

otherwise, the temperature of the test specimen will differ from that of the air. A flute tube is used to measure the air-flow velocity. Signals from the velocity head, detected by the flute tube and the pressure drop of the test section, were transformed by an A/D card and then input into the data acquisition system. The test specimen pictured in Figure 1 was divided into several units so as to facilitate the determination of the quasi-local heat transfer coefficient which will be explained in the next section.

A precise optical balance was used to measure the total mass sublimated, ΔM. Then the average mass transfer coefficient β can be calculated as follows:

$$\beta = \frac{\Delta M}{\Delta\tau A(\Delta\rho_n)_{lm}} = \frac{\dot{M}}{A(\Delta\rho_n)_{lm}} \tag{1}$$

where

$$(\Delta\rho_n)_{lm} = \frac{(\rho_{nw} - \rho_{ni}) - (\rho_{nw} - \rho_{ne})}{\ln\dfrac{(\rho_{nw} - \rho_{ni})}{(\rho_{nw} - \rho_{ne})}} \tag{2}$$

Here, ρ_{nw} was calculated from the perfect gas law and the naphthalene vapor pressure–temperature relation, which was originally suggested by Sogin (1958), but is here converted into the SI system:

$$\log P_{nw} = 13.564 - 3729.4/T \tag{3}$$

$$\rho_{nw} = P_{nw}/(RT) \tag{4}$$

$$\rho_{ne} = \rho_{ni} + \frac{\dot{M}}{G} \tag{5}$$

For the first test sample, $\rho_{ni} = 0$, because the air in the laboratory room is naphthalene free. For the following test samples, the exit of the former is the entrance of the next. The Sherwood number can be calculated as

$$\mathrm{Sh} = \frac{\beta W}{D} = \frac{\beta W}{\nu}\mathrm{Sc} = \frac{W\cdot\mathrm{Sc}}{A}\cdot\frac{\dot{M}}{\nu(\Delta\rho_n)_{lm}} \tag{6}$$

According to the suggestion of Sogin (1958), the value of Sc is taken as 2.5, and ν, strictly speaking, stands for the

Notation

A	Mass transfer area
D	Naphthalene–air diffusion coefficient
f	Friction factor
G	Air flow rate
H	Rib height
L	Length of the test section
$\dot{M}$	Mass transfer rate
ΔM	Mass of the sublimated naphthalene during a test run
P	Partial pressure
ΔP	Overall static pressure drop
R	Universal gas constant
S	Rib pitch
T	Thermodynamic temperature of the naphthalene surface
U	Free flow velocity
W	Hydraulic diameter of the channel, which equals the width of the channel

Greek symbols

β	Quasi-local mass transfer coefficient
ρ	Density
$\Delta\rho$	Density difference
ν	Kinematic viscosity of air
$\Delta\tau$	Duration of the test run
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number based on channel hydraulic diameter
Sc	Schmidt number for naphthalene vapor–air
Sh	Sherwood number

Subscripts

a	Air
e	Exit
i	Entrance
lm	Logarithmic mean
n	Naphthalene vapor
w	Wall
0	Smooth channel

kinematic viscosity of the air–naphthalene-vapor mixture, but can be replaced by the value for pure air, since the concentration of naphthalene is extremely low.

The Nusselt number now can be calculated according to the principle of the heat/mass transfer analogy:

$$\mathrm{Nu} = \mathrm{Sh}(\mathrm{Pr}/\mathrm{Sc})^{0.4} \tag{7}$$

The friction factor is defined as

$$f = \Delta P \cdot W/(2L\rho_a U^2) \tag{8}$$

and the friction factor for the fully developed turbulent flow in a smooth square channel f_0 was calculated according to the revised Karman–Prandtl equation (Brundrett 1978) for square channels:

$$\frac{1}{f_0^{1/2}} = 4.0\log(\mathrm{Re}f_0^{1/2}) - 0.15 \tag{9}$$

It should be noted that since the triangular ribs were made of Plexiglass, they do not participate in the mass transfer process. This paper investigates the effect of ribs on heat transfer on the wall surface, and the nonparticipation of the ribs does not significantly alter the heat transfer performance of the wall.

It is obvious from Equation 6 that the maximum uncertainty in calculating Sh was determined by the maximum uncertainties in determining $\dot{M}$, v, and $(\Delta\rho_n)_{lm}$. Relatively speaking, the uncertainty was mainly determined by the accuracy of $(\Delta\rho_n)_{lm}$. Among ρ_{nw}, ρ_{ne}, and ρ_{ni}, ρ_{nw} plays the most important role. Therefore, the room temperature should be kept constant during the experiment, because ρ_{nw} will be changed by about 5 percent for a variation of 0.5 K in naphthalene surface temperature. During our experiment, the temperature variation was controlled to within 0.3 K. Uncertainty analyses showed that the maximum uncertainties in calculating $\dot{M}$, v, and $(\Delta\rho_n)_{lm}$ were 1.2 percent, 1.4 percent, and 2.9 percent, respectively. Therefore, the maximum total uncertainty of the Sherwood number was about 5.5 percent.

Results and discussion

Before conducting experiments with triangular-rib-roughened walls, the mass transfer coefficients for a fully developed air flow in a channel with rectangular-rib-roughened walls were

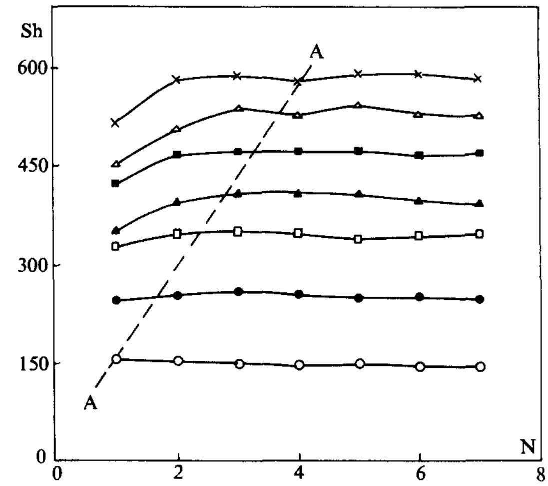

Figure 4 Quasi-local value of Sherwood number. ○, Re = 1 × 10^4; ●, Re = 2 × 10^4; □, Re = 3 × 10^4; ▲, Re = 4 × 10^4; ■, Re = 5 × 10^4; △, Re = 6 × 10^4; ×, Re = 7 × 10^4. $H/W = 0.1$; $S/W = 1.0$. Sherwood numbers to the right of the A–A dashed line remain almost constant along the flow direction

measured. A comparison between the present results and those given by Han et al. (1988b) is shown in Figure 3. The discrepancy is less than 3 percent, which gives confidence in the present experimental results.

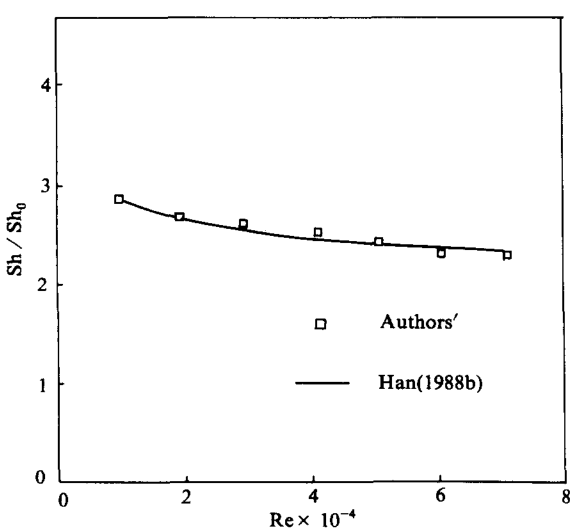

Figure 3 Results of verifying experiment

Mass transfer in the entrance region

In this part of experiment, the mass transfer coefficient of every roughened unit (the wall surface between two neighboring ribs) was separately measured and calculated according to Equation 1. In fact, the calculated value of β for each unit is an average value of that unit, but it is called the quasi-local value of that unit when we consider all the units as a whole. Figure 4 shows the test results of the quasi-local value for Sherwood number in the entrance region, in which N is the ordinal number of the unit. The quasi-local Sherwood numbers on the right of the dashed line A–A remain almost constant along the flow direction. Therefore, it can be concluded that the flow has become fully developed. The results of Figure 4 show that the lower the Reynolds number, the earlier the flow will become fully developed. This phenomenon can be explained as follows: in order to simulate the real flow situation in the turbine blade, there is no turbulence eliminator in the present experiment; therefore, the higher the Reynolds number of the incoming flow, the larger the turbulence intensity of the flow.

Experimental results in the fully developed region

Because the entering flow will be fully developed after flowing across only a few ribs, in engineering practice, the whole flow can be considered as fully developed. For this reason, attention will now be focused on the performance of the fully developed flow in the channel with triangular-rib-roughened walls.

The experimental results of Sherwood number vs. Reynolds number and friction factor vs. Reynolds number for $H/W = 0.1$ and $S/W = 1.0$ are given in Figure 5 and Figure 6. For comparison, the performance of a smooth channel quoted from Zhang (1984) and our experimental results for the rectangular-rib-roughened walls are also presented.

Comparisons show that the mass transfer coefficients for the triangular-rib-roughened walls with $H/W = 0.1$ and $S/W = 1.0$

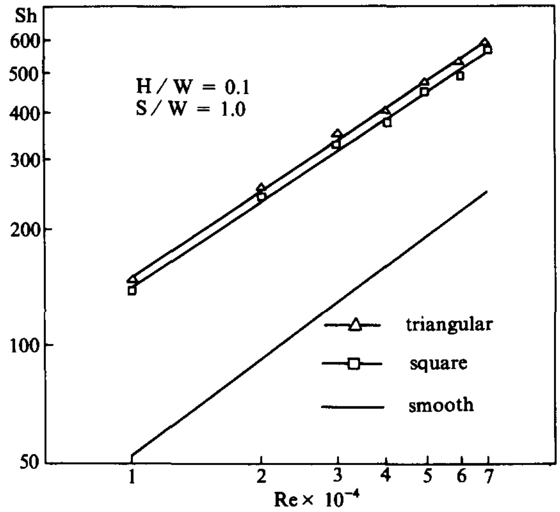

Figure 5 Sherwood number vs. Reynolds number

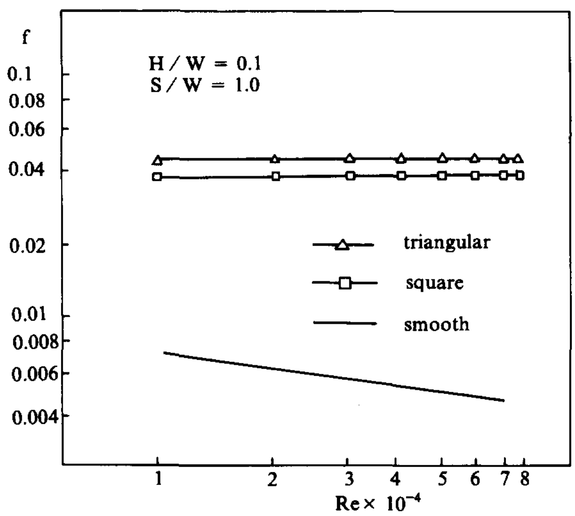

Figure 6 Friction factor vs. Reynolds number

are about 1.5 times higher than those for the smooth channel, and the friction factors are around 6 to 10 times larger. The mass transfer performances for the triangular-rib-roughened walls are 5.7 to 7.4 percent greater than those for the rectangular-rib-roughened walls, and the friction losses increase by about 18 percent. It is also noticeable from the figures that the effect of the Reynolds number on transfer performance is significant, while the friction factor is only a weak function of the Reynolds number. The variation of the Sherwood number and the friction factor for the triangular-rib-roughened walls with different H/W and different Reynolds numbers is shown in Figures 7 and 8; the results show that both the Sherwood number and the friction factor increase as H/W increases. It is known that the dimensionless rib height H/W is the quantity governing the turbulence intensity induced by the rib; an increase in H/W means an enhancement of turbulence and thus leads to an increase in mass transfer and resistance. The role of the dimensionless pitch S/W is also worth discussing. If S/W is too large, the induced turbulence will die out before the flow reaches the next rib, thus lowering the mass transfer. On the other hand, small S/W prevents the disturbed flow behind the rib from reattaching; therefore, a wake is formed in the whole downstream region, and the effect of the induced turbulence on the wall surface will also be lessened. Consequently, an optimum S/W will exist. The test results for different S/W are shown in Figures 9 and 10. It can be seen from Figure 9, for $\mathrm{Re} = (3\text{–}5) \times 10^4$, that there are optimum values for S/W, at which the Sherwood numbers have their maxima. As the Reynolds number becomes larger, the maximum is not so evident.

After a series of experiments, the following correlations were obtained:

$$\mathrm{Sh} = 0.31\mathrm{Re}^{0.717}(H/W)^{0.148}[1.0 + 0.225(S/W) - 0.308(S/W)^2 + 0.082(S/W)^3] \tag{10}$$

$$f = 0.578\ \mathrm{Re}^{0.0114}(H/W)^{1.04} [1.0 - 0.254(S/W) - 0.0017(S/W)^2 + 0.023(S/W)^3] \tag{11}$$

Maximum deviation between the corresponding values as calculated from the above correlations and the experimental

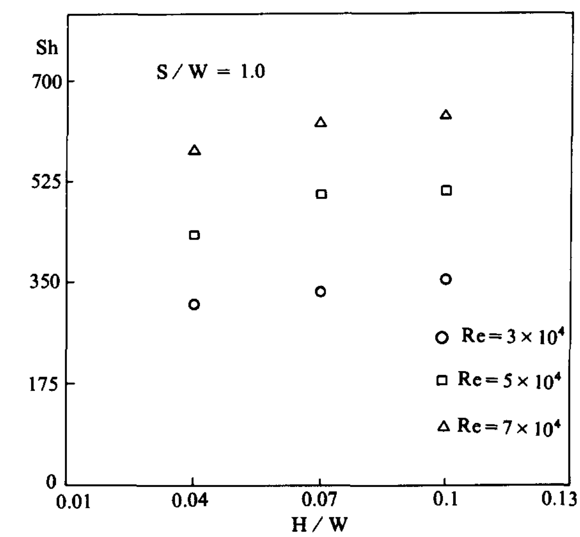

Figure 7 Effect of H/W on Sh

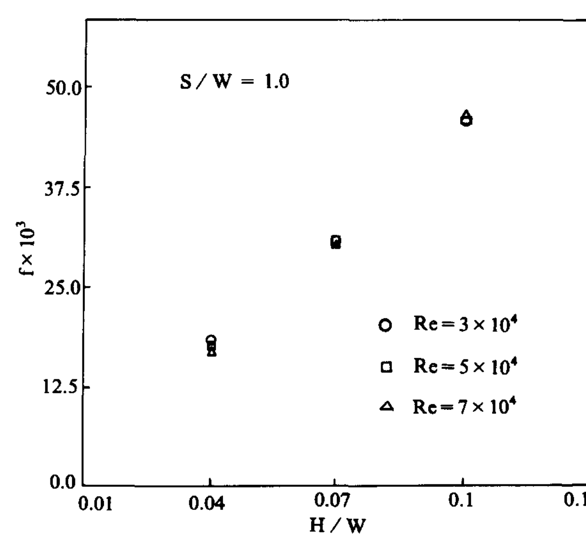

Figure 8 Effect of H/W on f

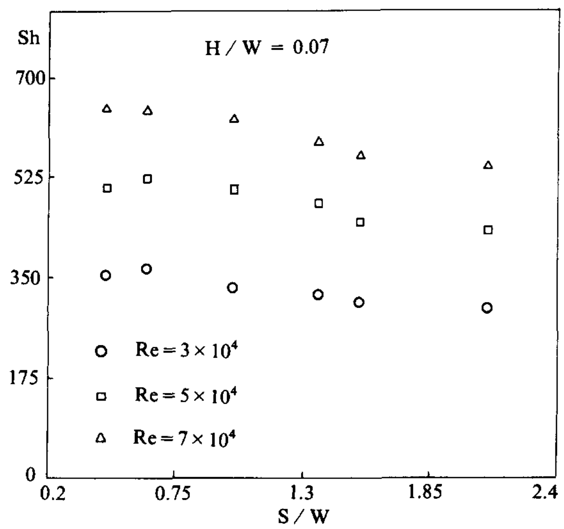

Figure 9 Effect of S/W on Sh

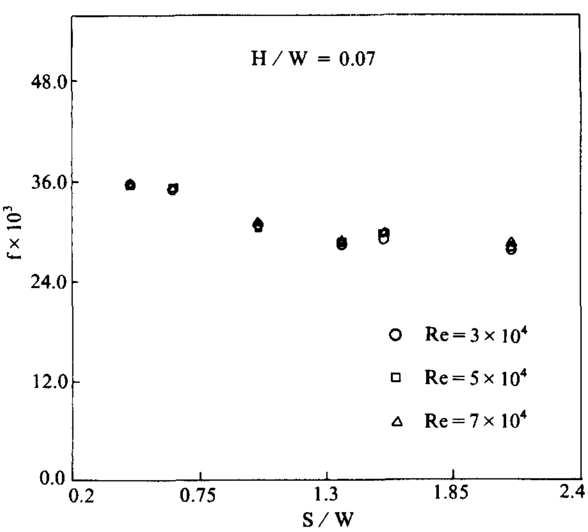

Figure 10 Effect of S/W on f

data, is 7.2 percent. We recalculated all the Sherwood numbers and friction factors under experimental conditions according to Equations 10 and 11, and then normalized them with the corresponding values of the smooth-surface channel. The results are shown in Figure 11. It can be seen from that the normalized Sherwood numbers are in the range of 2.0 to 3.3, while the values of f/f_0 are from 2.0 to 11.0. In the low-Reynolds-number region, the increases of the normalized Sherwood number and the values of f/f_0 are well matched. But the values of f/f_0 increase much more quickly than the normalized Sherwood number as the Reynolds number increases.

Conclusions

In the present study, the main conclusions about the heat/mass transfer and resistance performances of turbulent flow over

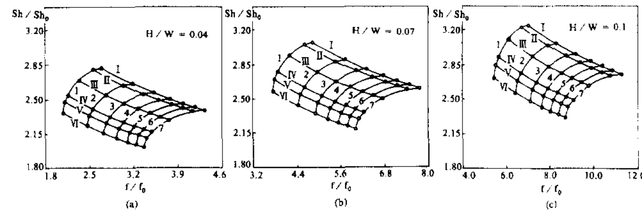

Figure 11 Normalized Sherwood number vs. normalized friction factor. (a) $H/W = 0.04$; (b) $H/W = 0.07$; (c) $H/W = 0.1$. 1: Re $= 1 \times 10^4$; 2: Re $= 2 \times 10^4$; 3: Re $= 3 \times 10^4$; 4: Re $= 4 \times 10^4$; 5: Re $= 5 \times 10^4$; 6: Re $= 6 \times 10^4$; 7: Re $= 7 \times 10^4$; I: $S/W = 0.45$; II: $S/W = 0.63$; III: $S/W = 1.00$; IV: $S/W = 1.37$; V: $S/W = 1.55$; VI: $S/W = 2.10$

triangular ribs attached on two opposite walls in a square channel are as follows.

(1) In the case of $H/W = 0.1$ and $S/W = 1.0$, experimental results show that the larger the Reynolds number, the longer the entrance length. Generally speaking, the heat/mass transfer will be fully developed after the fourth rib;
(2) the fully developed heat/mass transfer rate is 1 to 2.3 times greater than that for a smooth-channel flow, and friction characteristics reach about 2 to 11 times those along a smooth channel;
(3) the heat/mass transfer increases as the Reynolds number and H/W increase. As to S/W, there exists an optimum value; and
(4) correlations of the Sherwood number and the friction factor, which are useful for engineering calculation, were presented.

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