Characteristics of Heat Transfer in the Channel with Twisted Tape

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ABSTRACT: Heat transfer distributions and friction factors in square channels (3.0 × 3.0 cm) with twisted tape inserts and with twisted tape inserts plus interrupted ribs are respectively investigated. The rib height-to-channel hydraulic diameter ratio, e/D_h, is kept at 0.067 and test section length-to-hydraulic diameter ratio, L/D_h is 30. The square ribs are arranged to follow the trace of the twisted tape and along the flow direction defined as axial interrupted ribs. The twisted tape is 0.1 mm thick carbon steel sheet with diameter of 2.8 cm, length of 90 cm, and 2.5 turns. Two heating conditions are investigated for test channels with twisted tape inserts and rib turbulators : (1) electric heat uniformly applied to four side walls of the square duct, and (2) electric heat uniformly applied to two opposite ribbed walls of the square channel. Results show that the twisted tape with interrupted ribs provides a higher overall heat transfer performance over the twisted tape with no ribs.

Nomenclature

A	: heat transfer area [m^2]
D_h	: hydraulic diameter [m]
e	: rib [m]
f	: friction factor
G	: Mass flux [kg/m^{2}s]
h	: heat transfer coefficient [W/m^2 ℃]
k	: thermal conductivity [W/m ℃]
$\dot{m}$	: mass flow rate [kg/s]
Nu	: Nusselt number, hD_h/k
$\dot{Q}$	: rib pitch [m]
p	: heat transfer rate [W]
Re	: Reynolds number, $\rho D_h u_b/\mu$
x	: distance from entrance [m]

Subscripts

b	: bulk average
r	: rough
s	: smooth
w	: wall

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1. Introduction

It is well known that rib turbulators enhance the convective heat transfer rate in internal flow passages. Webb et al.[1] and Gee et al.[2] showed that heat augmentation and pressure drop in a tube with rib turbulators depend on configurations and the flow Reynolds number. Han[3], Han et al.[4], Ahn et al.[5] investigated heat transfer and friction characteristics in rectangular channels with rib turbulators. They concluded that heat transfer enhancement and pressure drop depend on rib height, spacing, and angle of attack, and heating condition.

Considering the increasing effects of both twisted tape and rib turbulators on heat transfer, Zhang et al.[6] used a compound technique(twisted tape inserts plus rib turbulators) to enhance heat transfer in tube flows. Their results show that heat transfer is enhanced by combining twisted tape and interrupted ribs rather than twisted-tape or interrupted ribs only.

Several questions still remain unsolved that must be analyzed systematically : The previous results are for flow in circular tubes. In reality, square channels are more common than circular tubes in most applications(e. g., turbine blade internal cooling passage). It is questionable whether the twisted tape can provide the same heat transfer performance in a square channel as in a circular tube. The question arise because, the secondary flow generated by the swirling motion due to the twisted tape is different in square channels than circular tubes. Heat transfer performance from the combined use of a twisted tape and interrupted ribs in a square must be also investigated.

The objective of the present study is to investigate the effect of the twisted tape and the combined effect of the twisted tape and interrupted ribs on heat transfer distribution and friction in square channels. The surface heating effect on the heat transfer coefficient is also investigated.

2. Experimental apparatus and data reduction

The experimental apparatus is shown in Fig. 1. The main components of the facility consist of a blower, an orifice flow meter, an entrance channel, and a test section. A 0.86 kW blower forces air into the pipe and orifice meter and then through the straightener, entrance channel, and test section. The air is exhausted into the atmosphere from the end of test channel. The valve next to the blower is used to adjust the mass flow rate through the orifice meter. The square duct test section used with only twisted tape inserts and twisted tape plus interrupted ribs have a cross section of 3.0 cm by 3.0 cm.

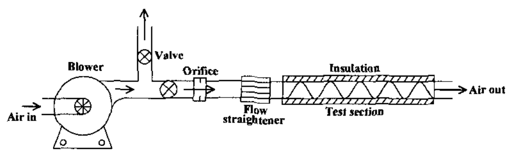

Fig. 1 Schematic of experimental setup.

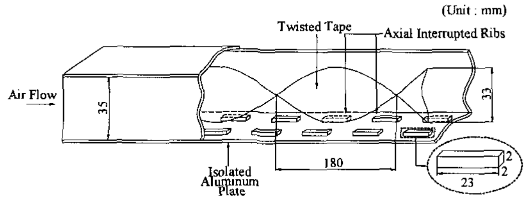

Fig. 2 Square test channel with twisted tape insert plus axial ribs.

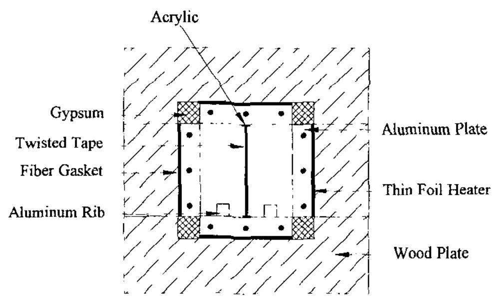

Fig. 3 Details of cross test section.

The twisted tape inserts and axial interrupted ribs are placed inside the test section square channels with length $L = 90$ cm as shown in Fig. 2. The twisted tape is 0.1mm thick carbon steel with the diameter of $D = 2.8$ cm and length of 90 cm. It has 2.5 turns throughout the entire duct length. The twisted tape is thermally insulated(thin insulation between the tape and channel wall) from the four side walls of the square duct to reduce heat conduction between tape rims and the aluminum duct.

The dimension of an axial interrupted rib is 2 mm × 2 mm × 23 mm as shown in Fig. 2. The gaps between the ribs are 23 mm in axial direction and 15 mm in traverse direction. Test channels with the twisted tape plus interrupted ribs have aluminum ribs glued periodically on the inner bottom wall of the test section. The thin wood strip (0.2 mm thick) are placed between the aluminum plates in the axial direction. The thin stainless foil heater (0.1 mm thick) is installed at the backside of the aluminum plates. The foil connects to a transformer to provide controllable uniform heating to the test section. Two heating conditions are investigated for test channels with twisted tape inserts and rib turbulators : (1) electric heat uniformly applied to four side walls of the square duct, and (2) electric heat uniformly applied to two opposite ribbed walls of the square channel. The four sided heating is used as a reference. However, from application point of view, two sided heating is important. The internal serpentine cooling passage of turbine blade can be a good example. The pressure and suction side of an airfoil receives most of the heat, whereas, the other two sides can be considered unheated compared to the pressure and suction side. Therefore, the internal cooling passage in a turbine airfoil can be stimulated as two side heated and two side unheated case. An unheated smooth entrance duct made of plexiglas and having the same cross section is connected to the test section. This unheated portion serves as entrance duct and establishes a hydrodynamically fully developed flow at the entrance to the heated section.

The local surface temperatures of the test section are measured by copper-constantan thermocouples distributed along the length placed at each aluminum plate of each wall. These thermocouples are embedded into the pre-drilled holes on the outer surface of 10 positions of each aluminum plate. Therefore, 40 copper constantan thermocouples are used on the entire channel walls. A 48-channel Hybrid Data Logger and a computer are used for data(temperature) acquisition and data reduction.

The regional heat transfer coefficient is calculated from regional heat transfer rate per unit surface area from the inner wall to the cooling air, the local wall temperature on each aluminum plate, and the local bulk mean air temperature as:

$$h = \frac{q - q_{loss}}{A(T_w - T_b)} \tag{1}$$

The regional total heat transfer rate (q) generated from the stainless thin heaters is determined from the measured resistance and current($q = I^2 R$) on each side of the test channel. The heat loss (q_{loss}) is determined experimentally by supplying electrical power to the test section until a steady condition is achieved for a no flow condition. This is done for several different power inputs to obtain a relation between the total heat loss from each surface and the corresponding surface temperature using Fourier's law. The heat loss is 5% of the power inputs for Reynolds number of 10,000.

It is found that the foil provides nearly uniformly heat flux on the entire test channel. The bulk mean air temperature used in Eq. (1) is determined assuming a linear air temperature rise along the test duct. The local bulk mean air temperature in Eq. (1) is also calculated by energy conservation as :

$$T_{b,i} = T_{in} + \sum (q - q_{loss}) \dot{m} c_p \tag{2}$$

With the measured inlet air temperature (T_{in}) and the accumulated net heat input from the test duct inlet to the i th position. Eq. (2) calculated the local bulk mean air temperature (T_b) at i th position. The calculated outlet bulk mean air temperature agrees with the measured values within 5%. The inlet bulk mean temperature is about 21~28 ℃ and wall temperature is around 50~60 ℃, depending on the test conditions. To reduce the effect of changing Reynolds number on the heat transfer enhancement, the calculated regionally averaged Nusselt number is normalized by the Nusselt number for fully developed turbulent flow in smooth circular tubes correlated by Dittus-Boelter[7] as :

$$\frac{Nu_r}{Nu_s} = \frac{hD_h/k}{0.023Re^{0.8}\mathrm{Pr}^{0.4}} \tag{3}$$

A manometer measures the pressure drop across the square channel. The average friction factor in fully developed flow is calculated from the measured pressure drop across the test channel and the mass flow rate of the air as :

$$f = \frac{\Delta P}{(4L/D_h)(G^2/2\rho g_c)} \tag{4}$$

The uncertainties associated with the length scale, temperature, and differential pressure used in the data reduction was ±1.0 mm, ±0.℃, ±1.0 Pa. The thermophysical properties of the air were assigned an uncertainty of ±3.0%, based on the observed variations in the reported values in the literature. The standard deviation in the air bulk velocities was found to be within ± 4%, and the maximum uncertainty in the heat transfer rate ($\dot{Q}$) was estimated to be ± 6.2%. These uncertainties would result in the maximum uncertainty of the convective heat transfer coefficient of about ±8.9% at Re =19,100. The experimental uncertainties were estimated using the procedure outlined by Kline and McClintock.[8] It was found that the uncertainties for Reynolds number and friction factor were about ± 5.9% and ± 9.7%, respectively.

3. Experimental results and discussion

Fig. 4 represents the local Nusselt numbers in the smooth channel with twisted tape inserts in the two-sided heating and four-sided heating conditions, respectively.

The results show that the local Nusselt numbers in the two-sided heated case are slightly

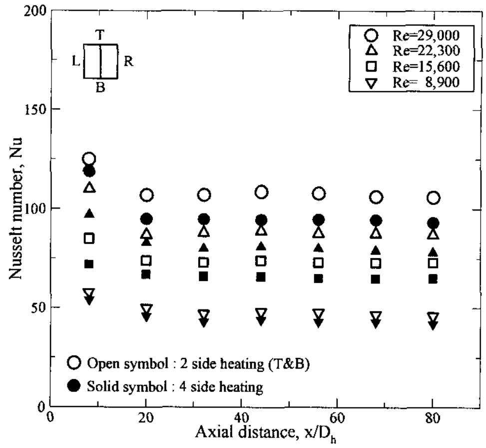

Fig. 4 Local Nusselt numbers.

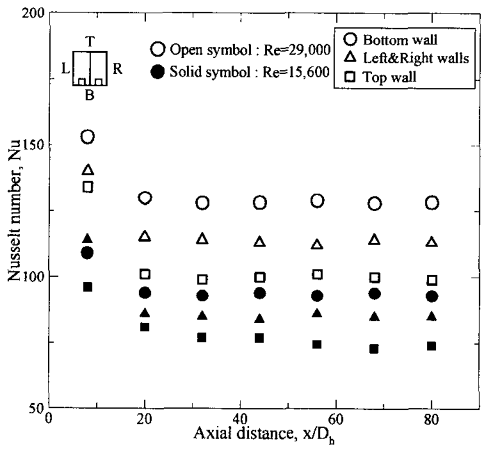

Fig. 5 Local Nusselt numbers in the 4 side heating channel.

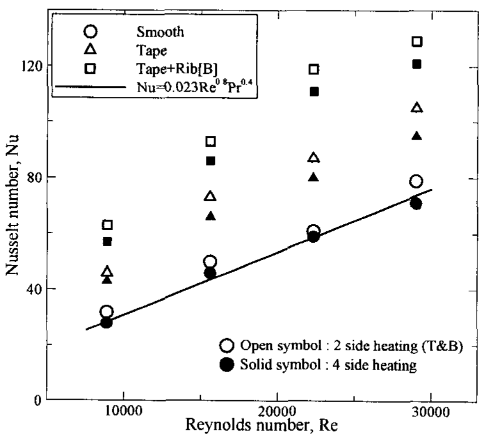

Fig. 6 Average Nusselt numbers

higher than the four-sided heated case.

This occurs because the colder fluid moves from the two unheated walls toward the two heated walls, which results in a higher heat transfer coefficient. Values of the fully developed Nusselt numbers with heating applied to two opposite walls were 1.08 to 1.19 times greater than those obtained with heating applied all four walls at same Reynolds number.

Thus the effect of Reynolds number was more prominent in two-sided heating condition than in the four-sided heating condition. The local Nusselt number decrease as the x/D_h increases, and maintains a nearly constant value $x/D_h = 7$. This is because the region up to $x/D_h = 7$ are supposed to be thermally developed region.

Fig. 5 shows the streamwise Nusselt number distributions based on top wall temperatures, left or right wall temperatures, and bottom wall temperature for the test section with addition of ribs with twisted tape, respectively. The Nusselt numbers based on bottom wall were 16 and 27% greater than those on the adjacent smooth sides and opposite smooth side at Reynolds number of 29,000. The higher Nusselt numbers on the ribbed bottom wall were due to the increased level of turbulence generated by the ribs, which broke up the growth of the thermal boundary layer.

Fig. 6 shows the average Nusselt numbers for the fully developed flow in the smooth channel only, in the smooth channel with the twisted tape, and in the bottom ribbed channel with the twisted tape, respectively. The test section with the twisted tape plus the ribbed wall has the greatest Nusselt number.

The twisted tape creates a swirling motion in the square channel. Thus, a centrifugal force is superimposed over the main longitudinal flow that produces a secondary motion in the channel. The net effect of this change in the flow field increased pressure drop and heat transfer enhancement. Interrupted ribs also act as turbulence promoters in the main flow field. Thus,

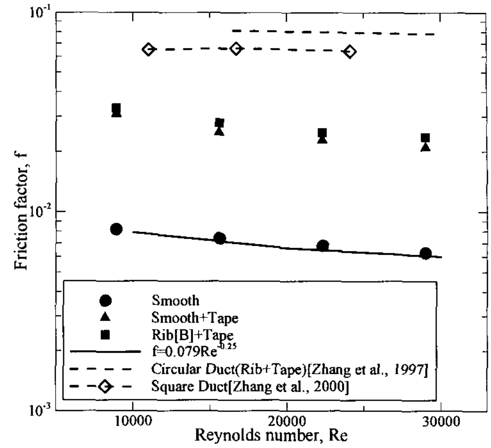

Fig. 7 Friction factors.

the additional of ribs with twisted tape increases local turbulence and secondary motion.

The empirical correlation by Dittus and Boelter[7] for a smooth channel is also plotted for a comparison. It is evident from Fig. 6 that there is excellent agreement between the existing correlation and our results on the condition that the entire channel walls are heated.

Fig. 7 shows the average channel friction factors by obtaining from experimental data for the smooth channel, the channel with the twisted tape, and the channel with the twisted tape and ribs on the bottom wall, respectively. The empirical equation by Blasius for a smooth circular tube is included for a comparison. The present results for a smooth channel agreed well with the Blasius correlation within 2.5%.

The result also showed that the friction factor decreased with increasing Reynolds number since the relative increase in the magnitude of the fluid velocity squared was greater than the increase in the wall shear stress with increasing Reynolds number.

The channel with the twisted tape and ribs on the bottom wall has the maximum friction factor in the present work. This was due to the greater flow resistance experienced with additional turbulators, leading to higher friction factor values.

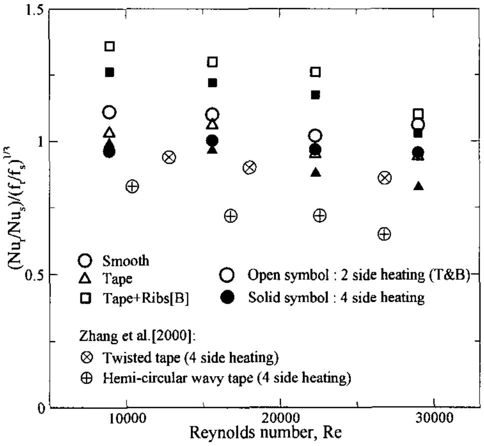

Fig. 8 Heat transfer performance under a constant pumping power.

The experimental data by Zhang et al.[6] for the smooth tube with the twisted tape is included for a comparison. The friction factor by Zhang et al., was nearly 3.6 times greater than our present work for the smooth square with the twisted tape at Reynolds number of 29,000. The results may occur from the edge or corner effect in square channels. The flow field inside a square channel with twisted tape is more complicated than that in a circular tube because the secondary flow entrains in the four corner and creates corner vortices. This reduces a greater pressure drop in square channels compared to circular tubes. Interrupted ribs produce a higher heat transfer coefficient and friction factor.

The interrupted ribs induce flow separation and reattachment, resulting in a secondary flow relative to the swirl flow generated by the twisted tape. This combined effect of swirl flow and turbulence secondary flow produces a higher pressure drop penalty. And Zhang et al.[9] for the ribbed square channel with the twisted tape in Fig. 7 has much higher values than our work indicating solid square symbols. This is attributed to the fact that Zhang et al.[9] covered the square channel with the ribs on the entire four walls. Fig. 8 shows the performance curve, $(Nu_r/Nu_s)/(f_r/f_s)^{1/3}$ as a function of the Reynolds number. This curve reflects the overall heat transfer performance of a channel taking the friction factor effect into account. Results show for the smooth channel only, the smooth channel with the twisted tape, and the ribbed channel with twisted tape, respectively. Wall heating conditions are incorporated into the results. The results show that the two-sided heating provides better overall heat transfer performance than the four-sided heating. And the twisted tape with interrupted ribs provides a higher overall heat transfer performance over the twisted tape with no ribs. It is because the ribs give a better increment in heat transfer than in friction factor. For a comparison, the results obtained by Zhang et al.[9] in a 4-side heated square channel with the twisted tape only was included. The present data has a good agreement with Zhang et al.[9]

The twisted tape insert in a channel produces a higher pressure drop. Due to this high pressure drop, the use of tape insert will be limited and worse than some existing technology for turbine blade cooling. However, we presented the results and it will be available in the literature. The turbine cooling system designer can decide whether they can use the results for any specific needs.

4. Conclusions

1) In the smooth channel with twisted tape inserts, values of the fully developed Nusselt numbers with heating applied to two opposite walls were 1.08 to 1.19 times greater than those obtained with heating applied all four walls at same Reynolds number.

2) For the test section with addition of ribs with twisted tape, the Nusselt numbers based on bottom wall were 16 and 27% greater than those on the adjacent smooth sides and opposite smooth side at Reynolds number of 29,000.

3) The friction factor in the smooth circular

tube with twisted tape was nearly 3.6 times greater than in the smooth square channel with twisted tape at Reynolds number of 29,000.

4) The twisted tape with interrupted ribs provides a higher overall heat transfer performance over the twisted tape with no ribs.

Acknowledgement

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