

Effect of Channel Orientation on Heat Transfer in Rotating Rectangular Channel (AR=4) with 90° Rib Turbulators

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Abstract

An experimental investigation to determine the effect of the channel orientation on heat transfer enhancement in stationary and rotating rectangular channel of constant aspect ratio (AR=4) with facing 90° rib turbulators. A Reynolds number range tested in this study was 8000-13500. The effect of channel orientations of $\beta=90$ deg and 135 deg with respect to the plane of rotation is investigated and the rotation effect is also studied, where the rotational velocity is fixed at 500 rpm. The results show that the higher heat transfer enhancement for channel at 135° by 2.7 to 2.2 times while was 2.5 to 2 times for $\beta=90$ ° for stationary case. But for rotational cases the heat transfer enhancement was 3.4 to 2.6 and 2.8 to 2.3 for $\beta=135$ ° and $\beta=90$ °, respectively for Reynolds number range 8000 to 13500, comparing with smooth non rotating pipe adopted by Dittus-Boelter/McAdams. Also the results show that the average percentage of heat transfer enhancement of duct at $\beta=135$ ° was 3% higher than duct at $\beta=90$ ° for stationary case while for rotating case was 12% for the range of Reynolds number tested in this work.

Keywords: Enhancement, Heat transfer, Rotating, channel orientation, Turbines cooling

NOMENCLATURE			
A	Surface area of the plate (m ²)	Re	Reynolds number ($U_{av}\rho D_h/\mu$)
AR	Aspect ratio	T	Temperature (°C)
D_h	Hydraulic diameter (m)	T_b	Bulk Temperature (°C)
h	Heat transfer coefficient (W/ m ² °C)	T_{in}	Inlet Temperature (°C)
k	thermal conductivity (W/mK)	μ	Laminar viscosity (kg/m.s)
Nu	Nusselt number (hD_h/k)	β	Angle of channel orientation
Nu_o	Nusselt number through a smooth non-rotating pipe	ρ	Density (kg/m ³)
Pr	Prandtl number	e	Rib height (mm)

1. Introduction

Whenever turned the world into the industry more, the energy demand is increasing more and more. Therefore, many research efforts have focused recently on finding ways to contribute to the reduction of energy consumption. One area effort focused on trying to develop and raise the efficiency of the turbine engine. Reduce the associated maintenance costs of gas turbines as well as increasing the fuel efficiency costs represent a common goal for many researches. With the widespread of the energy generation applications and the aircrafts, reduce the costs can be enormous. One way of raising the efficiency of the turbine as well as the thrust of an aero-turbine through increased combustion temperature. This will lead to a major problem related to the sensitivity of the turbine components for high temperatures, particularly the turbine blades. So must deal with high temperatures entering the turbine (1600-1800 K), through study the physics of turbulent heat transfer in the cooling models. The turbine blades contain internal passages that used to cool the blades, through the absorption of excess heat that coming with hot gases. This will lead to increase the blades life, as well as helping to increase temperatures entering the turbine, leading consequently to improve the efficiency of the turbine. Because of the curvature of the turbine blades, the cooling blade passages can take different angles of orientation relative to the plane of rotation, Fig. (1) [1].

There are many previous studies on turbulent flow and heat transfer in gas turbine blade cooling channels. But limited researchers focused on the effect of channels orientation on the distribution of heat transfer. W.-L.Fu [1] conducted experimental work to study the heat transfer in rotating two-pass rectangular channels with two channels orientation (90° and 135°). He found that the 135° channel orientation creates less heat transfer difference between the leading and trailing walls than the 90° channel orientation for both smooth and ribbed channels. Johnson et al. [2, 3] published experimental investigations of the heat transfer phenomenon in a multi-pass, heat transfer model with both radially inward and outward flow. The model orientation with respect to the rotational centerline for 0° and 45° . Trip strips, skewed at 45° to the flow direction, were machined on the leading and trailing surfaces of the radial coolant passages to determine the effects of buoyancy and Coriolis forces on heat transfer distribution. These studies showed that the skewed trip strips rather than normal trip strips are recommended and geometry-specific tests will be required for accurate design information also the heat transfer in the turn regions and immediately downstream of the turns in the second passage with flow inward and in the third passage with flow outward was also a function of model orientation with differences as large as 40 to 50 percent occurring between the model orientations with forward flow and rearward flow of coolant. Parsons et al. [4] conducted an experimental study on the influences of channel orientation and wall heating condition on the local surface heat transfer coefficient in a rotating, two-pass, square channel with 60° and 90° ribs on the leading and trailing walls. The results showed that the effects of the Coriolis force and cross-stream flow reduce as the model orientation changes from untwisted to twisted case. Dutta and Han [5] experimentally studied the local heat transfer in rotating smooth and ribbed two-pass square channels with three channel orientations (0° , 45° , and 90°). The results showed that the change in the channel orientation about the rotating frame causes a change in the secondary flow structures and associated flow and turbulence distribution. Griffith et al. [6] determined that the model orientation with respect to the rotation plane duct significantly affects the leading and side surfaces, yet does not have much effect on the trailing surfaces for both smooth and ribbed surfaces. Al-Hadhrami and Han [7] investigated the effect of rotation on heat transfer in two-pass square channels with two channel orientation (90° and 135°). They found that the orientation of the channel with respect to the plane of rotation affected the heat transfer distribution. More specifically, they determined that the 90° channel orientation with respect to the axis of rotation produces greater rotating effect on heat transfer over the 135° channel orientation. Al-Hadhrami et al. [8] published experimental work to studying the heat transfer in two-pass rotating rectangular channels with two channel orientations (90° and 135°). It was discovered that a 90° channel orientation produces greater rotating effect on heat transfer than a 135° orientation. Agarwal [9] experimentally study the effect of channel orientation on heat/mass transfer in rectangular smooth and ribbed serpentine passages with and without rotation. He concluded that the channel orientation have an influence on the rotating channels heat transfer distribution. Liou et al. [10] investigated the heat transfer of a 45° rib-roughened rectangular duct with two channel orientations. They found that the channel orientations have a significant effects on the rotating channels heat transfer distribution.

In this work, the study of effect of facing orthogonal ribbed two opposite walls in non-rotating and rotating ducts with two channel orientations (90° and 135°) on heat transfer will be accomplished. Therefore, it is of very interest to experimentally detect the developing zones averaged heat transfer distribution in a rotating, rectangular channel of aspect ratio 4:1. The truth that no literature on such study exists today on this subject leads to the following questions:

- 1.How does the channel orientation will effect on the heat transfer distribution along the duct within Orthogonal ribbed rectangular channel of aspect ratio (AR=4)?
- 2.How does the channel orientation will effect on the zones heat transfer distribution of duct within Orthogonal ribbed rectangular channel of aspect ratio (AR=4)?

Answers of these questions are presented in this paper.

2. Experimental Facility

The rotative test system is shown in Figure (2). A changing frequency motor (speed controlled by an inverter model (SV022iC5) is linked by means of a belt to a rotating hollow shaft. This shaft rotate from the base of the test system to the work platform and is attached orthogonally to a hollow, rotary arm. The test section is incorporated (united) into the hollow rotary arm, which rotates in a plane perpendicular to the rotary shaft. Heater and thermocouple cables are linked to slip-ring collectivity mounted to the rotary shaft (36 –channels slip-ring was locally manufactured, especially for present work). The output of the thermocouples is transferred to a data selector switch. Heaters power input

coming from the variable transformers also passes through the slip ring collectivity. Compressed air received from an air compressor passes through a pressure regulator. The air (coolant) after leaving the pressure regulator will be entering the cooling system (to cool and dehumidifier the air) and enter through orifice flow meter [11] and then passes through mechanical seal (rotary union fixed below rotary hollow shaft, locally manufactured, especially for present work) and then passes in a hollow rotating shaft and turn 90 deg. to a hollow arm (fixed on the hollow shaft) and then to test section (fixed at the end of the hollow arm) to cooling internal walls of the test section and finally expelled to atmosphere.

3. Test Section

This experiment includes one-pass rectangular test section: AR=4:1, Figure (3), with facing Orthogonal-type ribbed wall. To enhance heat transfer, ribs are fixed on two opposite sides, leading and trailing walls. Each of the upper, lower, leading and trailing wall has one copper plates in the Y-direction (for upper and lower wall) and Z-direction (for leading and trailing wall). But in the X-direction (flow direction) all four walls has six plates. The plates on leading, trailing, upper and lower sides has a width dimension of 25 mm and has a height dimensions of 50 mm and 12.5 mm for leading and trailing and upper and lower respectively [6]. Beneath each six copper plates in the same side in the X-direction there is only one heater strip, i.e. there is four heaters one on every wall. The 2.5 mm deep hole from the surface of the plate is drilled in the back of each 3.0 mm thick copper plates to install the thermocouple. The cross section dimensions of ribs (made of copper) is 2.5 mm for width and 2 mm for height and the rib pitch-to-height ratio (p/e) is 13.5 are fixed (glued) on each of leading and trailing walls. The tests were accomplished for both non-rotating and rotating duct, and Reynolds numbers of 8000, 10000, 12000 and 13500. The speed of rotation is fixed at 500 rpm and the channel orientation angle is 90° and 135°.

4. Data Reduction

This work examines the heat transfer coefficient (regionally averaged) within the rotating channel at various locations. The heat transfer coefficient is calculated by using the following formula [1]:

$$h = \frac{Q_{net}/A}{(T_w - T_{b,x})} \quad (1)$$

The net heat flux is obtained by using (only) the voltage and current supplied to the four heaters from the variable transformers with neglect the heat transfer loss from the duct walls to the outside, due to small differences between the ambient and the outside test section surface temperature.

The area used in all cases in this work is the projected area of the smooth duct (i.e. neglecting the increase in area due to the ribs) [1, 6, 12, 13, and 14]. The wall temperature (T_w) is measured directly by using the thermocouples fixed in small holes at the back of copper plates segments. Due to a high thermal conductivity of plate (copper material), assume the temperature of each plate segment is homogeneous [1]. The air bulk temperature (T_b) is measured by using five thermocouples. As long as, the linear interpolation is used to find the bulk temperature at any other locations.

The Nusselt number in this study is obtained from the following formula:

$$Nu = \frac{hD_h}{K} \quad (2)$$

The fully developed turbulent flow Nusselt number through a smooth non-rotating pipe (Nu_o) is obtained by equation (3) and it is famous as Dittus-Boelter/McAdams correlation for heating ($T_w \geq T_b$) [5].

$$Nu_o = 0.023 . Re^{0.8} . Pr^{0.4} \quad (3)$$

Therefore, from equations (2 and 3), the Nusselt number ratio is giving as:

$$\frac{Nu}{Nu_o} = \left(\frac{hD_h}{K} \right) \left(\frac{1}{0.023 . Re^{0.8} . Pr^{0.4}} \right) \quad (4)$$

5. Results and Discussion

Figure (4) contain the stationary and rotational duct data for two different channel orientation angles 90° and 135° . Each case is subdivided into four experiments: (a) $Re=8000$, (b) $Re=10000$, (c) $Re=12000$, and (d) $Re=13500$. The results, in general, show that the angle of channel orientation at 135° give higher heat transfer comparing with channel orientation at 90° in both stationary and rotational cases. Figure (5) shows the results of average Nusselt number ratio for both channel orientation angles in stationary and rotation cases. The results show that higher heat transfer enhancement for channel at 135° by 2.7 to 2.2 times while was 2.5 to 2 times for $\beta=90^\circ$ for stationary case. But for rotational cases the heat transfer enhancement was 3.4 to 2.6 and 2.8 to 2.3 for $\beta=135^\circ$ and $\beta=90^\circ$, respectively for Reynolds number range 8000 to 13500, comparing with smooth non rotating pipe adopted by Dittus-Boelter/McAdams. For zones (regional) Nusselt number distribution, in general, case of $\beta=135^\circ$ have a higher heat transfer comparing with $\beta=90^\circ$ except at zone five, Figure (6). The average percentage of heat transfer enhancement of duct at $\beta=135^\circ$ was 3% higher than duct at $\beta=90^\circ$ for stationary case while for rotating case was 12% for the range of Reynolds number tested in this work.

6. Conclusions

This study experimentally investigate the heat transfer in stationary and rotating rectangular duct with constant aspect ratio ($AR=4$) with two channel orientation angles $\beta=90^\circ$ and $\beta=135^\circ$ with orthogonal ribs on two opposite walls (leading and trailing). The Reynolds number tested were 8000, 10000, 12000, and 13500. The results, in general, revealed that the average percentage of heat transfer enhancement of duct at $\beta=135^\circ$ was 3% higher than duct at $\beta=90^\circ$ for stationary case while for rotating case was 12% for the range of Reynolds number tested in this work.

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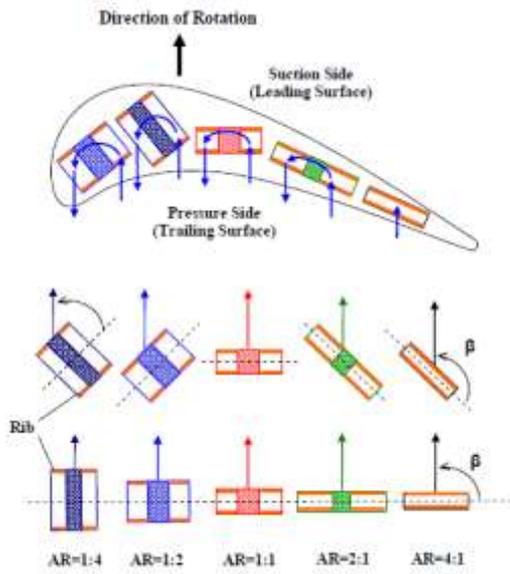


Fig. 1 Channel orientation angles [1]

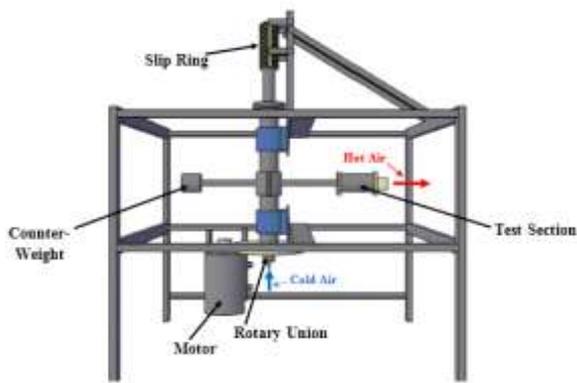


Fig. 2 Experimental test rig assembly.

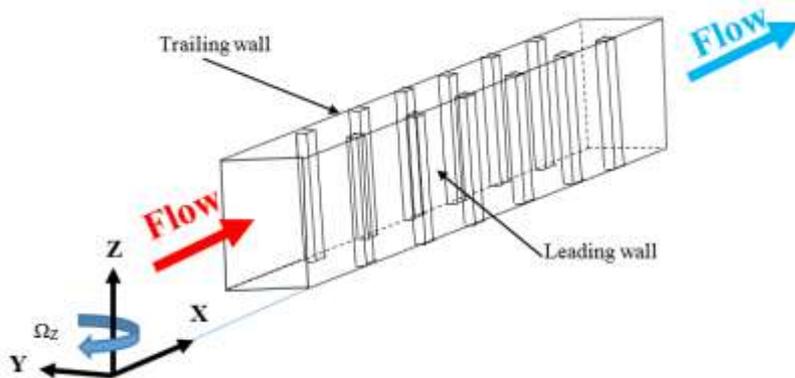


Fig. 3 Rectangular duct (rotating) with Orthogonal-type ribs on opposite walls.

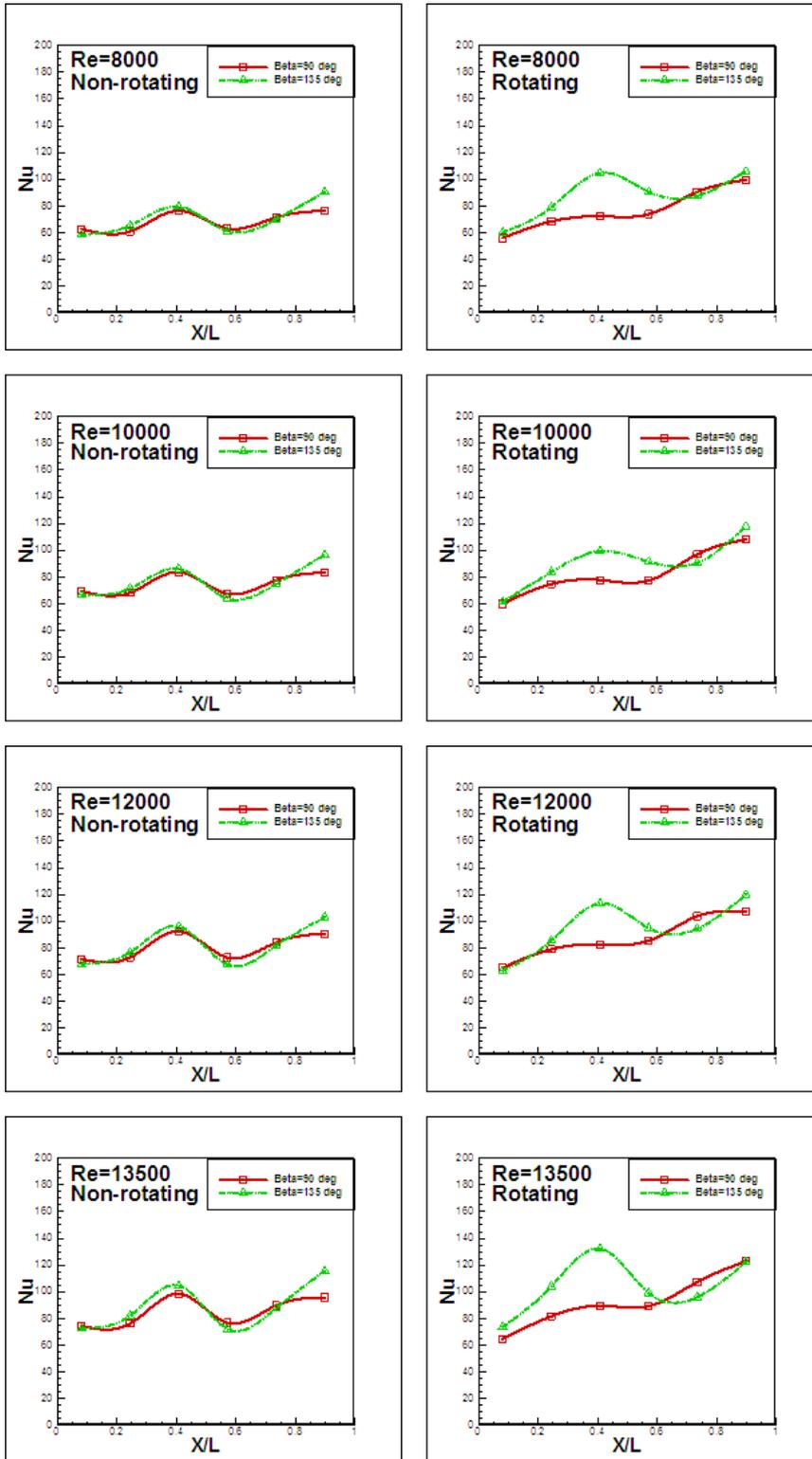


Fig. 4 Comparison of Nusselt number distribution along duct.

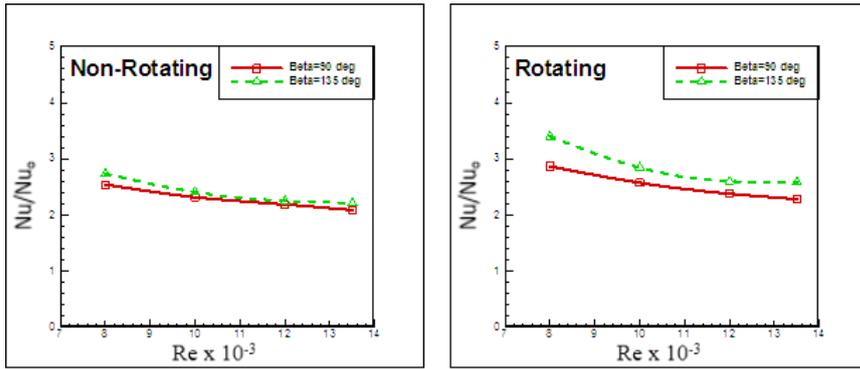


Fig. 5 Average Nusselt number ratio comparison.

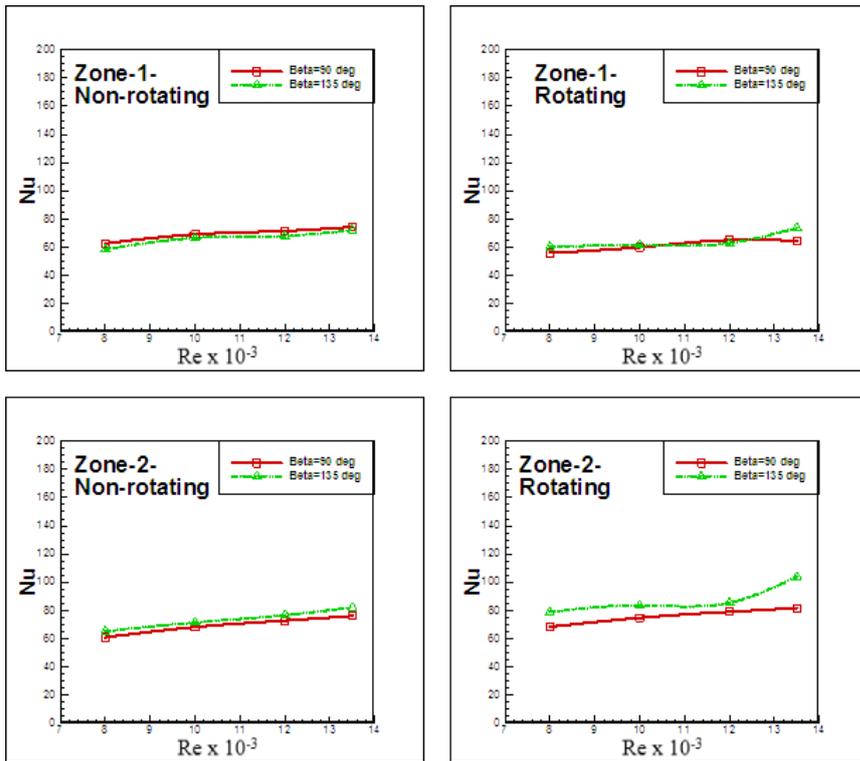


Fig. 6 Zones average Nusselt number comparison.

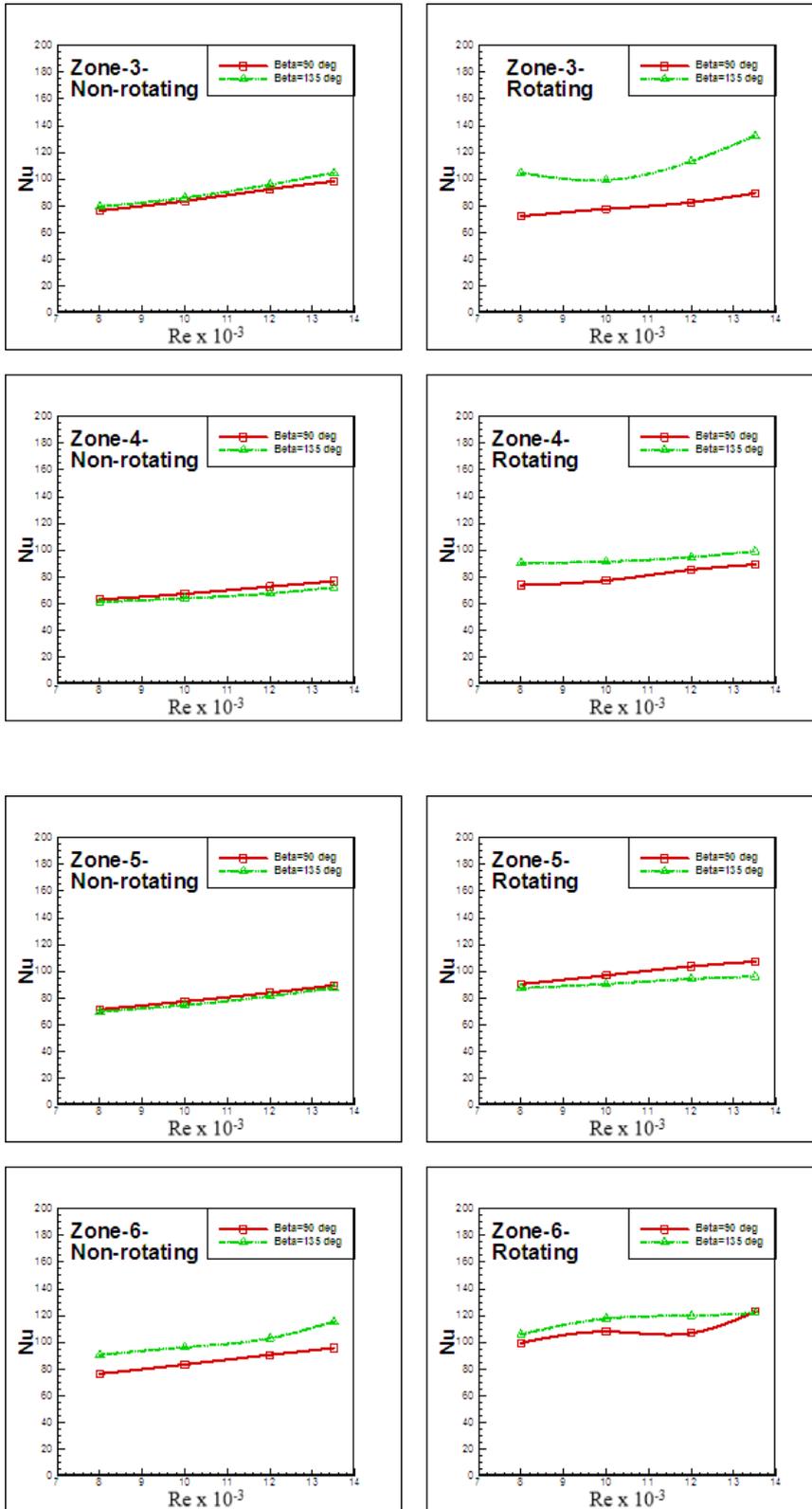


Fig. 6 Zones average Nusselt number comparison (continued).