Heat Transfer Enhancement for Rectangular Channels by Using Triangular-Shaped Ribs at High Reynold Numbers.

تحسين انتقال الحرارة لدكت مستطيل باستخدام مثيرات ذات مقطع مثلث عند ارقام رينولد عالية.

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ABSTRACT

In this paper, an experimental system was built to study the effects of triangular ribs inside a rectangular duct on the heat transfer and flow behavior. Boundary conditions were: inlet air temperature,(300^{o} K) and Reynolds numbers (Re=8082, 9698 and 11778). The surrounding constant temperature was (473 ^{o} K). The numerical simulations were done by solving the governing equations (Continuity, Reynolds Averaging Navier-stokes, and Energy equation) with (k-ε) model in three dimensions by using the SOLIDWORKS 3D CAD software. The effects of using triangular ribs fitted in rectangular passage channel on fluid flow and heat transfer characteristics were presented in this part. Ribs used with a pitch-rib height of 10, the rectangular channel of (30x60 mm) cross section, 1.5 mm duct thickness and 0.5 m long. The temperature, velocity distribution contours, cooling air temperature distribution at the duct centerline, the inner wall surface temperature of the duct, and thermal performance factor is found in this paper. The temperature distribution for the inner wall surface of the ribbed channel is lower than smooth one by (8.9 %) for case (1) and (Re=11778). The coolant air flow velocity seems to be accelerated and decelerated through the channel in the presence of ribs, so it was shown that the thermal performance factor along the duct is larger than 1, this is due to the fact that the ribs create turbulent conditions and increasing thermal surface area, and thus increasing heat transfer coefficient than the smooth channel. It concluded that, the triangular ribs with angle of 90^{o},case(1) is the best as compared to the other two cases in this study for cooling the rectangular duct. After modification and analysis have been done, Nusselt Number enhanced by (36%).

Keywords: Internal flow, heat transfer enhancement, rib tabulators cooling.

الخلاصة:

في هذا البحث تم بناء منظومة عملية لدراسة تأثير وضع مثيرات مثلثة المقطع داخل قناة مستطيلة على انتقال الحرارة وسلوك الجريان. الظروف الحدية كانت: درجة حرارة الهواء الداخل (300^{o}k) وارقام رينولد (Re=8082, 9698 and 11778). درجة حرارة محيط ثابتة (473^{o}k). تم التحليل العددي بواسطة حل المعادلات الحاكمة (الاستمرارية، معادلات نافير ستوك، و معادلة حفظ الطاقة) مع نموذج (k-ε) بالابعاد الثلاثة بأستخدام برنامج جاهز (SOLIDWORKS 3D CAD). تأثير استخدام المثيرات المثلثة المثبتة في ارضية قناة مستطيلة المقطع على جريان المائع و خواص انتقال الحرارة تم دراستها في هذا البحث. المثيرات التي تم استخدامها كانت بنسبة (بعد/ارتفاع=10) في قناة مستطيلة المقطع (30X60) سم، 1.5 ملم سمك و 0.5 م طول. درجة الحرارة، مخططات توزيع السرعة، مخططات توزيع درجة حرارة الهواء في مركز القناة، توزيع درجة الحرارة على جدار القناة الداخلي و معامل الاداء الحراري تم ايجادها في هذا البحث. وجد ان توزيع درجات الحرارة للجدار الداخلي للقناة باستخدام المثيرات المثلثة اقل منه للقناة بدون مثيرات وبنسبة (8.9%) للحالة رقم (1) و رقم رينولد (11778). سرعة الهواء تبدو في تزايد وتناقص داخل القناة بوجود المثيرات وعليه فأن معامل الاداء الحراري على طول القناة اكبر من (1) وهذا نتيجة لحقيقة ان المثيرات تولد حالة اضطراب وتزيد المساحة السطحية الحرارية وتزيد معامل انتقال الحرارة اكثر من القناة بدون مثيرات. تم الاستنتاج من هذا البحث ان المثيرات المثلثة مع مساحة مواجهة عمودية حالة رقم (1) هي الحالة الافضل بالمقارنة مع الحالتين الاخريين في هذه الدراسة لتبريد القناة المستطيلة المقطع. بعد التحديث والدراسة تم تحسين رقم نسلت بنسبة (36%).

الكلمات الافتتاحية: جريان داخلي، تحسين انتقال الحرارة، التبريد بمثيرات الاضطراب.

NOMENCLATURE

SYMBOL	DESCRIPTION	DIMENSION
A	Surface area	m^2
Cp	Heat capacity of air	J/kg.K
Dh	Hydraulic Diameter	m
e	Rib Height	m
f	Friction Factor	--
h	Heat transfer coefficient	$W/m^2.K$
k	Thermal Conductivity	W/m.k
Lc	Characteristic Length	m
$\dot{m}$	Mass flow rate	Kg/s
Nu	Nusselt Number = hD/k	--
Q	Heat Transfer Rate	W
Re	Reynolds Number = $\rho u D/\mu$	--
T_h	Temperature of the inner duct wall surface	^oK
T_c	Temperature of the air at the duct centerline	^oK
u	Flow velocity	m/s

1. INTRODUCTION

One of the most effective engineers prime movers is gas turbines, which is used for thrust and power obstetrics systems in converting the energy from its thermal form، which is happened in the combustion period, to mechanical form. Though the knowledge and technology in gas turbines have enhanced vastly during the years the requests on fewer emissions, greater efficiency, and more reliability are still required. The service life and the efficiency of a gas turbine is intensely depended on the components of the turbine, but these components are risky from high-pressure and temperature for the gas departed from the combustor (stoichiometric combustion) [1], and the progressive gas turbine machine can be operated with an excessive temperature condition of 1500 ℃ [2]. Unfortunately, this condition at rotor blade will surpass the temperature of melting the metal. So, there is a requirement for an innovative cooling method to overcome this problem. In research and the development efforts there are three main methods to challenging the thermal effects in gas turbines:

(1) New cooling technology.
(2) Development in thermal barrier coatings (TBCs).
(3) Improve high-temperature substrate materials.

This study focuses directly on enhancing the heat transfer in gas turbine blade by using rib arrays inside internal channels of heat exchanger systems.

Ribs are usually used to augment the heat transfer for the fluid (which is transport the energy) and the surfaces of heat transfer. It acts like a hitch for the flow and makes secondary flow like to forward and in reverse facing step so a local wall turbulence will happen on account of separated and reattached flow between the ribs, that will improve the heat transfer of the surface [3]. Many types of research show that there are specific geometrical parameters that can be affected on the heat transfer coefficient, from these parameters: blockage ratio, the passage aspect ratio, attack angle of the rib and the ratio between rib pitch to its height. Figure (1) shows different configurations of ribs used to enhance internal heat transfer.

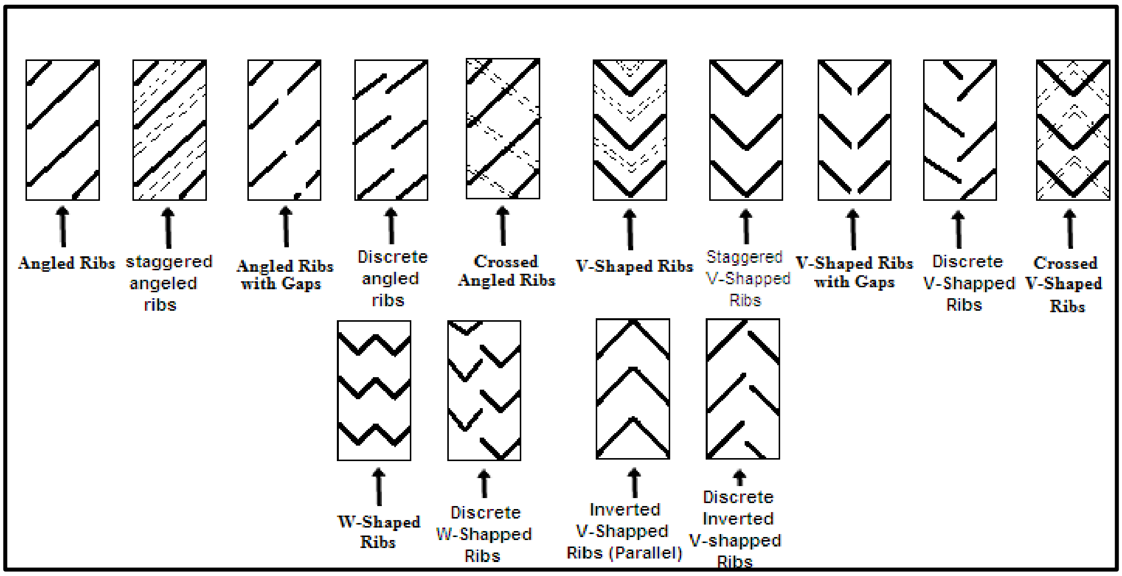

Figure (1). Different Rib Configurations. [4]

Park et al. [5] studied the effect of changes the aspect ratio and rib angles in the channels experimentally and he presented that the specific angles of the rib are suitable for altered aspect ratios, but especially the changing with 45°-60° ribs achieves well regarding heat transfer and pressure drop. Furthermore, angled ribs create greater heat transfer rates with minor pressure drop for the channels with small aspect ratios. Kukreja et al. [6] investigate experimentally a V-shaped channel with ribs of 45° and 60° V angle and conclude that the channel with 60° V-shaped ribs results in high heat transfer augmentation. The combined effects of rib angle and channel aspect ratio in rib roughened rectangular channels was studied by Han and Park [7], conclude that the square channel with angled ribs gives 30% better heat transfer better than the case of transverse rib. Alkhamis et al. [8] made an extended study by using 45-deg V-shaped ribs and they find that V-shaped ribs have greater heat transfer performance with a little pressure loss as compare with angled ribs. An investigation of the effect of rib shape on the heat transfer and pressure loss in the square channel was made by Chandra et al. [9]. They concluded that for a specific friction factor, the highest heat transfer enhancement happened with the square rib, while the performance of heat transfer is analogous to the tested rib types (slant-edged ribs, triangular, semi-circular ribs and circular). Ahn [10] tested a channel with different rib geometries and he advised that the triangular-shaped rib gave the optimum heat transfer performance. But Wang and Sunden [11] display different argument with conclusions that stated by Ahn [10]. Wang and Sunden made an experimental study by using a square channel with different-shaped ribs (equilateral-triangular, square, trapezoidal with increasing height, trapezoidal with decreasing height), they found that the highest heat transfer performance happened with the trapezoidal-shaped rib with decreasing height in the flow direction while producing the smallest reattachment length among all cases. Tuqa et al. [12] presented a numerical simulation to study the turbulent fluid flow and heat transfer for channel with different angles triangular ribs. With constant surface temperature and Reynolds number values of 20000 to 60000. They showed that, triangular ribs of angel 60° with Reynolds number of 60000 is the case of maximum value of average Nusselt number.

OBJECTIVE OF STUDY

The air is passed through the duct at different Reynolds number (8082, 9698 and 11778), and then the following parameter is to be studied.

1. To find out the Friction factor and Nusselt Number for the rectangular channel with and without ribs (triangular rib with three different cross flow area).
2. To find out the Nusselt Number ratio [Nusselt number with rib (Nu_r)/ Nusselt number without rib (Nu_o)] and Friction factor ratio [Friction factor with rib (f_r)/ Friction factor without rib (f_o)] for rectangular channel of all the ribs cases.
3. To find out the thermo-hydraulic performance factor for the rectangular channel with and without ribs cases.
4. Finally concluded the results and discussed the best combination of the model.

3. THEORETICAL PART

The following parameters will be discussed in this study:

a) Nusselt Number (Nu): it is a dimensionless term, its measure of the convective heat transfer which occurs at the surface and is defined as [13]:

$$Nu = h\,D_h/k \qquad \text{... (1)}$$

b) Friction Factor (f): the energy loss due to friction in a pipe or duct based on the fluid velocity and the friction resistance is known friction factor and it is a dimensionless term [12].

$$f = \frac{2}{(L/D_h)}\frac{\Delta P}{\rho U^2} \qquad \dots (2)$$

Thermo hydraulic Performance factor **(η):** This is defined by equation as follows and is Similar to enhancement of heat transfer at constant pumping power [12].

$$\eta = \left(\frac{Nu_r}{Nu_o}\right)/\left(\frac{f_r}{f_o}\right)^{\frac{1}{3}} \qquad \dots (3)$$

WhereNu_r, f_r, Nu_o and f_o are the Nusselt Numbers and friction factors for duct configuration with and without ribs respectively.

c) Reynolds number, according to the hydraulic diameter (D_h) and bulk mean velocity (U), as showed below:

$$Re=\rho u D/\mu \qquad \dots (4)$$

The numerical model was represented by a rectangular channel having 8 ribs on the bottom wall surface. Numerical analyses are conducted with SOLIDWORKS 3D CAD software and, realizable ($k - \varepsilon$) model is used. Steady state solutions were obtained. In all the equations, the second order discretization was used. Improved wall treatment is made to the Realizable k-ε Model. In all of the numerical investigation, the computational field includes a completely structured multiblock mesh structure, which refines near the rib vicinities and walls.

4. PHYSICAL MODELS

Considered the problem of air flowing through a rectangular duct, the bottom of the rectangular duct is provided with a plate having a number of ribs. The cross-section of rib is a triangle with three different cross flow area as shown in figure (2), has been provided.

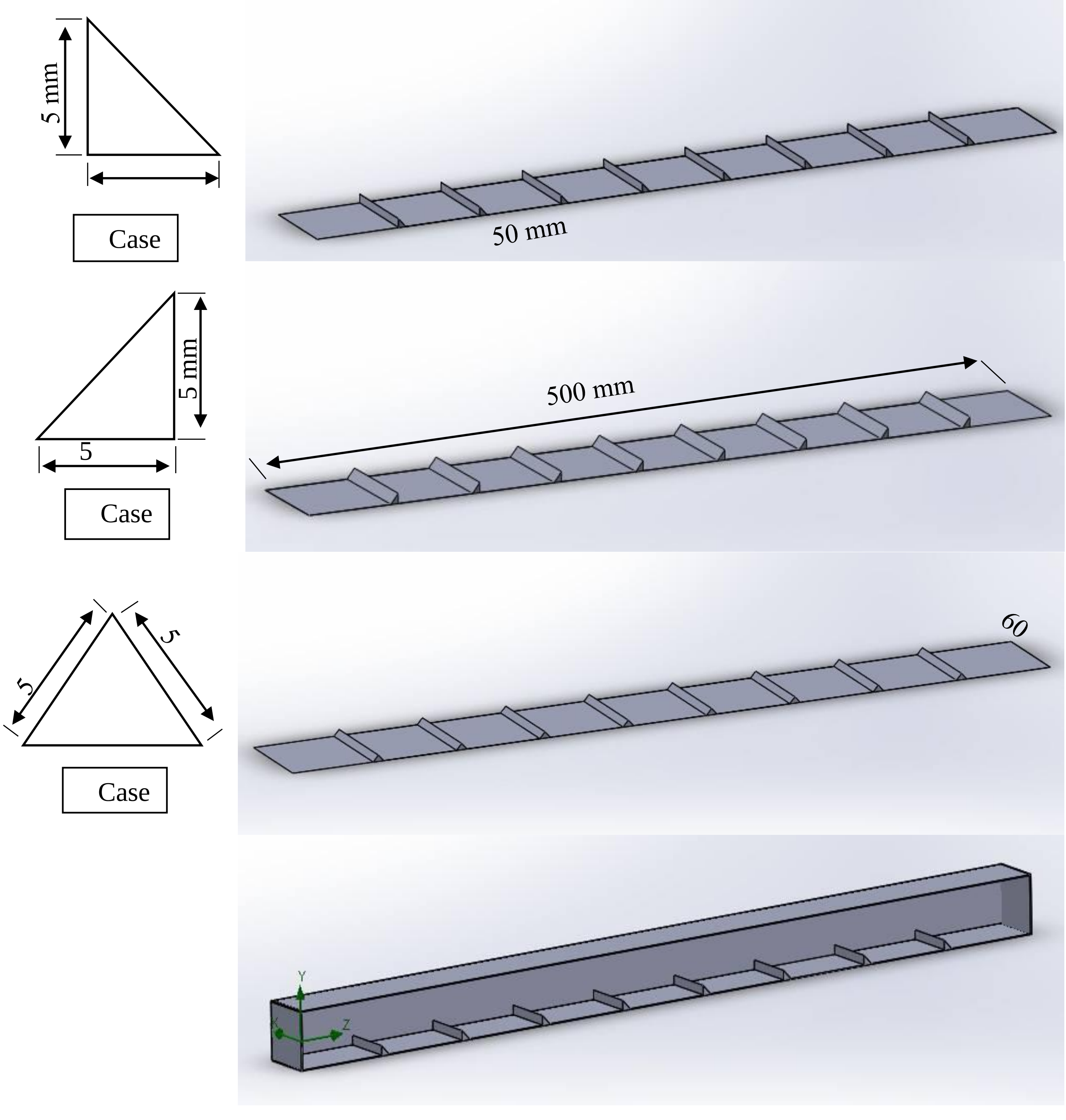

Figure (2): Physical geometry of ribs in bottom of the duct with front cross-section

5. EXPERIMENTAL PART

In the experimental part, the system shown in figure (3) is used. Figure (4) shows the schematic diagram of the system, which is consisted of an entry section, an experiment section, a centrifugal blower, perforated ribs, and measuring flow velocity, temperature, and pressure difference devices. Air was blowed by a fan with a variable speed and driven through the experiment section of the channel. The dimensions of the inner cross section of the channel were (60 x 30) mm and with (500) mm long. A heat input of 1000 W was given to the Ni-chrome heating wire put at equal spacing from the test section duct to ensure constant surrounding temperature inside the furnace. The furnace was insulated with alumina in order to avoid the losses of heat energy to the surrounding. The ribs were mounted on the bottom of the channel to improve the convective heat transfer. There are total four models created with three different ribs cross area cases and one is the smooth case by using SOLIDWORKS 3D CAD Modeling Software.

Figure (3): Photograph of the experimental rig

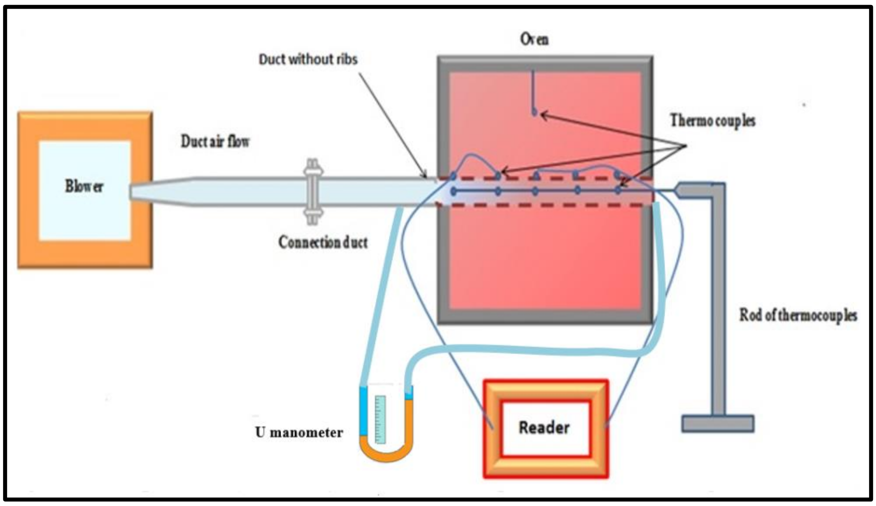

Figure (4): Schematic diagram of the experimental setup

6. EXPERIMENTAL PROCEDURE

Supply power to heaters to get the required hot air surrounding temperature and regulate the blower gate to obtain the required flow rate. Until reached the steady state conditions, record thermocouples reading which represent the inner wall surface temperatures for smooth channel (without ribs), and the air cooling temperature at the duct centerline. Repeat the above procedures for another flow rate of air. All the above steps are repeated for the channel with the three cases of ribs.

7. RESULTS AND DISCUSSIONS

In the experimental data, the Nusselt number and the friction factor for a smooth wall channel were first validated then; the heat transfers and friction factor for the rectangular duct with ribs were tested under a turbulent flow condition. To evaluate the enhancement in heat transfer in the ribbed roughened cooling channel, the distribution of the Nusselt number for the smooth (unribbed) channel is found experimentally as a reference. The tests are done with three different Reynolds numbers which are: 8082, 9698 and 11778, that calculated with respect to the channel hydraulic diameter and the mean inlet velocity.

Figure (5) presents the distribution of the inner duct wall temperature at constant surrounding wall temperature (473 ^{o}K) for three cases of ribs with respect to the smooth case and Reynolds number value of (11778). The wall temperature for a duct with ribs was less than that for the smooth duct. This was due to the fact that ribs make wakes which develop to vortices. This leads to increment the heat transfer in the duct wall to the coolant air, i.e. increasing the coolant air temperature. It is clear that case (1) is the best one compared to the other cases, in which it can show that the duct wall loss more heat and have the minimum temperature distribution. The frontal cross-sectional area of the ribs in case (1) gives the best normal area which causes the best separation of flow and leads to enhance the heat transfer between the coolant fluid and the wall.

Figure (6) show the variation of heat transfer in terms of Nusselt Number and Reynolds number for the channel of several ribs cross flow area cases. It is seen that Nusselt Number increase with the increase of Reynolds number. Reynolds number was calculated based on hydraulic diameter (D_h) of the rectangular channel. From the figure (6) it is seen that the Nusselt Number for case (1) is considerable high as compared to other cases. As shown in figure (6), the use of ribs leads to great enhancement in heat transfer in a comparable trend in compare with the smooth channel and the Nusselt number values, rise with the increase of the Reynolds number. The maximum difference of the averaged Nusselt number between the smooth and perforated ribs is found to occur at (Re=11778) with a value equal to (36%) for case (1).

From figure (7) later it is clearly seen that the results show an acceptable difference between the experimental and numerical Nusselt Numbers values. The experimental results give the higher values for Nusselt Number due to the systematic and random errors of the tests, like the insulation and the outdoor conditions.

Figure (8) shows that the Thermo hydraulic Performance factor depends upon the Nusselt ratio and friction ratio. It is seen that the Thermo hydraulic Performance factor for case (1) is highest as compared to the case (2) and (3).

Figure (9) represents the resulting contour of temperature distribution for the cooling air inside the duct for the case of smooth duct and duct with ribs for three different frontal areas with (Re =11778). The coolant air flow was separated and reattached, the heat transfer enhancement is noted at every inter rib space for all cases, which is due to the reattachment of flow after ribs. It was shown that case (1) shows the best enhancement where the normal cross area is best one to give high separation of flow which is lead to increase the vortices in this domain.

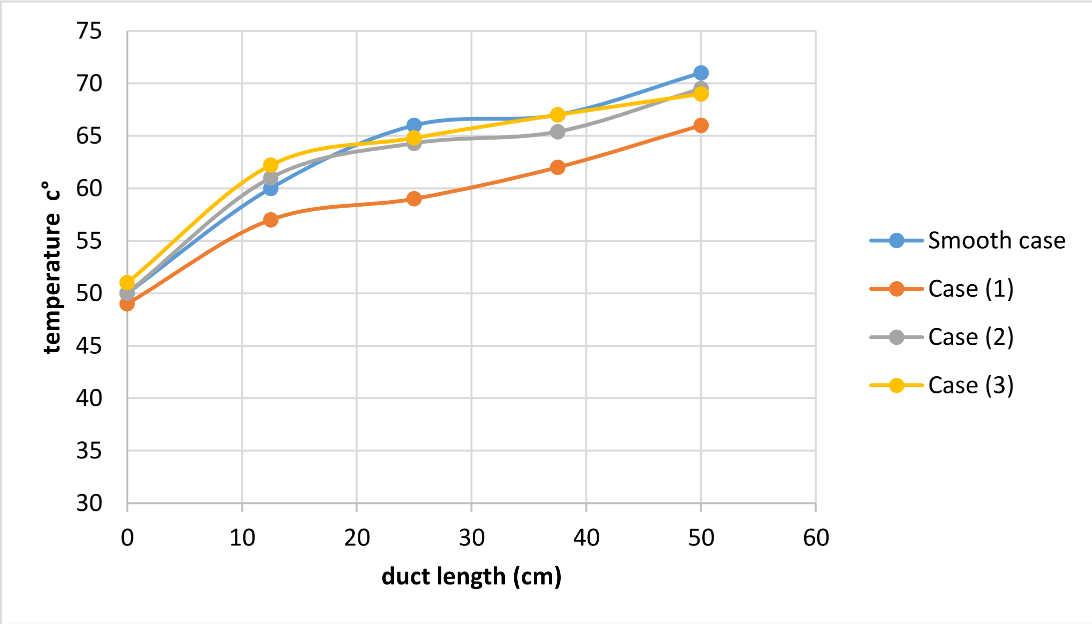

Figure (5) Temperature Distribution at the Duct Inner Wall for (Re= 11778) with the Ribs Three cases.

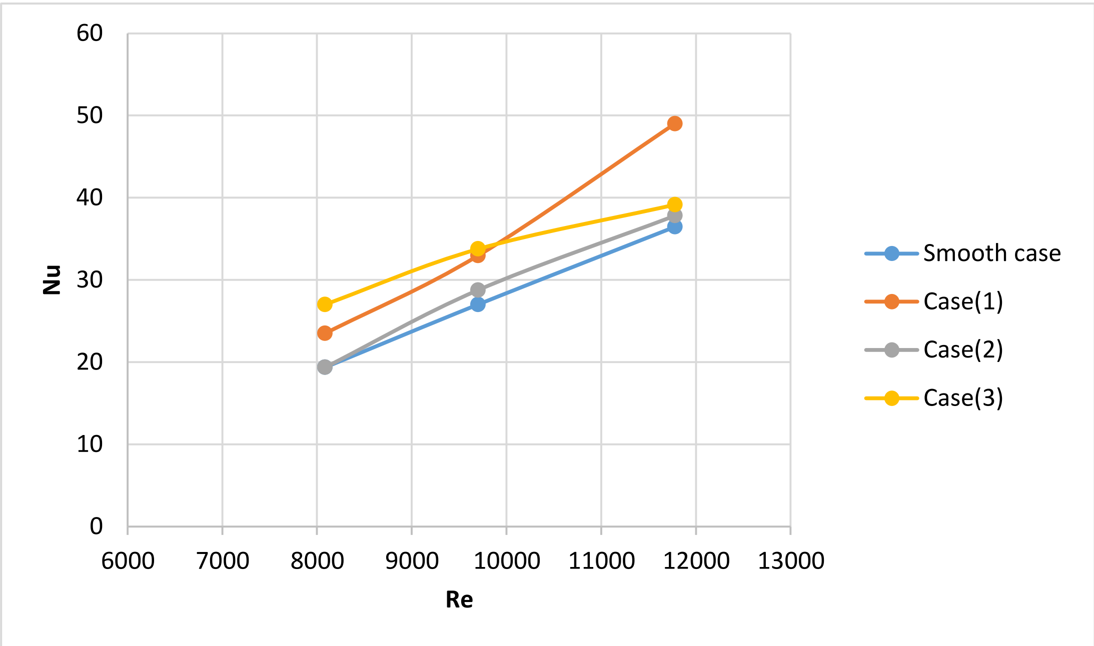

Figure (6). Variation of Nusselt Number and Reynolds Number for the Three Cases of Ribs with the Smooth Case.

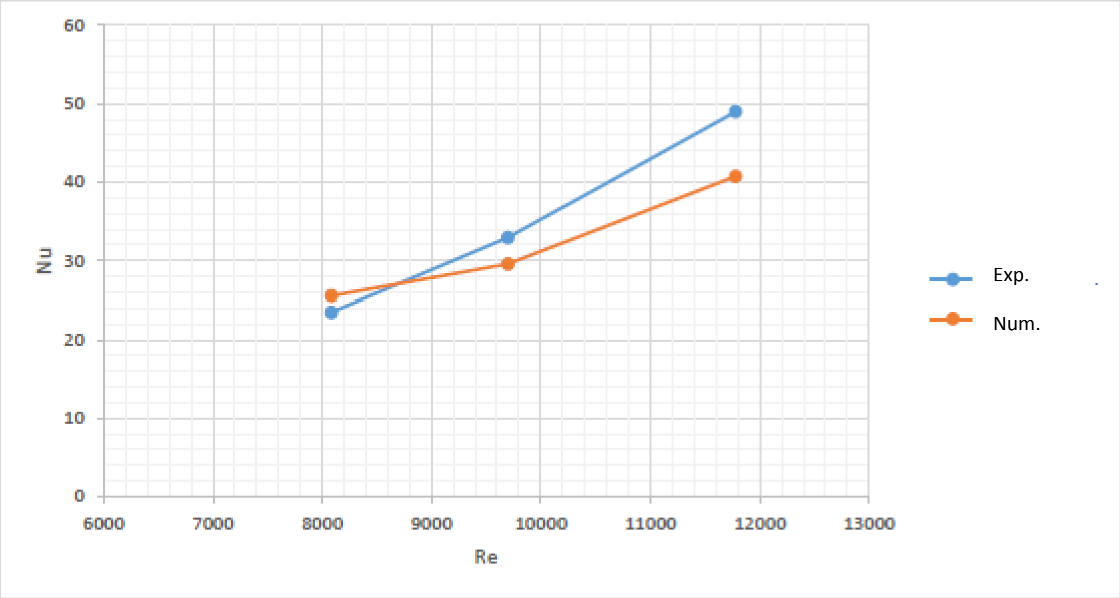

Figure (7). Comparison of Nusselt Number for Case (1) Experimentally and Numerically.

Figure (8) Thermal performance for all cases of ribs at (Re=11778)

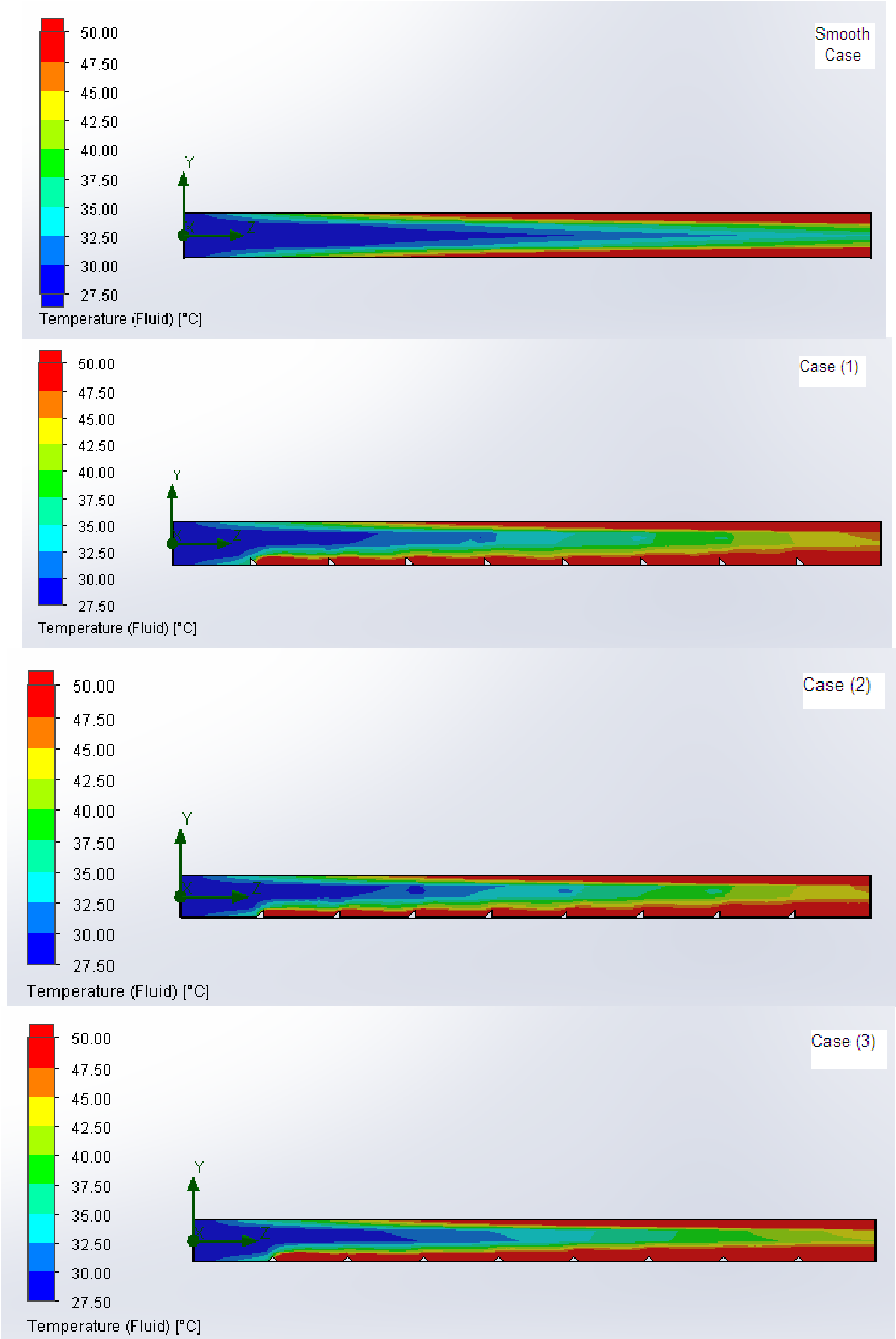

Figure (9) Temperature Distribution for the Coolant Fluid for all cases of Ribs at (Re=11778)

8. Conclusions

The following conclusions can be observed in the present work:

1. The temperature distribution on the inner surface for ribbed duct is lower than a smooth one.
2. Ribbed duct had improved the heat transfer rate, with compare to the smooth duct. The average heat transferred from surfaces with perforated ribs was greater than that on the smooth surface. In the boundary layer, a disturbance was created because of the ribs, which produced greater turbulence because of the separated and reattached flows.
3. The thermal performance factor along the duct is larger than 1, meaning that the ribs configuration performance always exceeds the smooth duct.
4. The greatest difference of the averaged Nusselt number between the smooth and perforated rib happens at Re = (11778) with a value equal to (36%).
5. The Friction Factor reduce with the increasing of the Reynolds number, for this parameter also the case (1) shows better results than case (2) and (3), because friction factor of case (1) is lowest in all range of Reynolds Number and this result leads to increase of thermo-hydraulic performance factor.

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