Heat Transfer Augmentation in Gas Turbine Blade Rectangular Passages Using Circular Ribs with Fins

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Abstract

In this paper, an experimental system was designed and built to simulate conditions in the gas turbine blade cooling and run the experimental part. Boundary conditions are: inlet coolant air temperature is 300K with Reynolds numbers (Re=7901) .The surrounding constant hot air temperatures was (673 K).The numerical simulations were done by using software FLUENT version (14.5), in this part, it was presented the effect of using circular ribs having middle fin fitted in rectangular passage channel on fluid flow and heat transfer characteristics. Ribs used with pitch-rib height of 10, 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 are presented in this paper. it can be seen that the duct with all ribs with middle fins was the better case which leads to increase the coolant air temperature by (10.22 %) and decrease the inner wall temperature by (6.15 %). 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.

Keywords: Blade cooling, Heat transfer Enhancement, Gas turbine, Rib turbulator Cooling.

الخلاصه

في هذا البحث تم تصميم وبناء نظام تجريبي لمحاكاه ظروف تبريد ريشه التوربين الغازي حيث كانت درجه حراره هواء التبريد الداخل (300 كلفن) ورقم رينولد (7901) ودرجه الحراره المحيطه (673كلفن) وتم اجراء المحاكاه العدديه باستخدام برنامج الفلونت (14.5) حيث تم دراسه تأثير الاضلاع الدائريه ذات الزعانف الوسطيه خلال مجرى مستطيل بابعاد (30 *60 ملم) وبسمك (1.5 ملم) وطول (0,5 م) على الجريان وانتقال الحراره .لاحظنا ان باستخدام الاضلاع الدائريه ذات الزعانف الوسطيه تم زياده حراره هواء التبريد الداخل بنسبه (10.22%) و نقصان بدرجه السطح الداخلي للقناه بنسبه (6.15%) وكان معامل الاداء الحراري اكبر من واحد وهذا يدل على ان الاضلاع الدائريه حسن بانتقال الحراره من خلال زياده المساحه السطحيه للانتقال وزياده الاضطراب الذي يؤدي الى زياده الخلط .

الكلمات المفتاحية :- تبريد الريشة ، تحسين انتقال الحرارة ، توربين الغازي ، التبريد بواسطة الاضلاع .

Nomenclature

Symbol	Description	Units
A	Surface area	m^2
C_p	Heat Capacity of air	J/kg.K
D_h	Hydraulic Diameter	m
e	Rib Height	m
f	Friction Factor	[-]
g	Acceleration due to gravity	m/s^2
H	Channel Height	m
h	Heat Transfer Coefficient	$W/m^2.K$
k	Thermal Conductivity	W/m.K
L_c	Characteristic Length	m
$\dot{m}$	Mass Flow Rate	kg/s
Nu	Nusselt Number	[-]
μ	Air Dynamic Viscosity	$N\ s/m^2$
P	Rib Spacing(Pitch)	m
Q	Heat Transfer Rate	W
pw	circumference	m
Re	Reynolds Number=$\rho uD/\mu$	[-]
T	Temperature	K

u	Flow Velocity	m/s

Introduction

Gas turbines are very efficient engineered first movers that transferring energy from thermal shape (Burning Phase) to mechanical shape- are extremely used for propulsion and power breeding systems. Though the technology in gas turbines has been amended extremely over the decades, so to high fuel prices , the requirement for higher efficiency, drop emissions and longer life have running to give notice the limit of current science and ability (**Yang, 1997**).

To make up the prompt evolution of in progress gas turbines, the operating condition (temperature) should be raised in order to upgrade the thermal efficiency and yield work of the advanced gas turbine engine. Therefore, the heat transferred into the turbine blade of is substantially raised as the turbine inlet temperature is constantly raised. Thus, it is fully significant to reduce the temperature of the turbine blades for a long resistance and safe work. The blade cooling must consist of cooling the main regions being that insecure to the hot gases (**Jonas , 2002; Han *et.al.*,2000**).

Shailesh *et.al.*,2013 had experimentally investigated the influence of the rib angle tendency on pressure drop and heat transfer in a square channel for a flow Reynolds number ranged from(5000 to 40,000),rib height to channel hydraulic diameter ratio (e/Dh) of 0.060 , rib spacing to rib height (p/e) of 10, and rib attack angle (90° and 60°).The 60° ribs with a gap yields about 3.8-fold enlargement in Nusselt number and about (7.4-fold) raise in the friction factor as compared with smooth passage channel. **(Arkan and Hasan ,2014)**presented an experimental and numerical investigation of heat transfer characteristics and thermal performance in 50 cm stainless steel tube long , inlet diameter of (30 mm) and outlet diameter of (60 mm) with constant surrounding hot air temperature of (1000, 1200 and 1400 K) using ANSYS Fluent 14.5. Results indicate that the use of internal ribs increase the heat transfer rate and found to possess the highest performance factors for turbulent flow. **(Liuo and Hwang ,1992)** investigated the effect of p/e and e/D_h ratios on the local and average heat transfer and friction factor for fully developed flow channel (aspect ratio = 4) with two opposite rib-roughened walls. The Reynolds number ranged from 5×10^3 to 5.4×10^3.The p/e ratio was 10, 15 or 20. The e/D_h was 0.063, 0.081 and 0.106. At a constant Reynolds number, the average friction factor and Nusselt number were increased with increasing e/D_h ratio, while these parameters decreased with rising p/e ratio at a constant e/D_h .**Zhang *et.al*.,1994** studied experimentally the effect of ribs spacing on the heat transfer coefficients and friction factors in channels have rectangular cross section area with two opposite ribbed-grooved walls. The rib-groove pitch to height ratios and Reynolds number were varied from 8 to 30 and 10^4 to 5×10^4,respectively. Results showed that the heat transfer and friction factor values for ribbed-grooved rectangular ducts decrease with increasing in groove pitch-to-rib height.

In this paper, the effect of fitting circular ribs with middle fin in a steel rectangular channel with internal flow of coolant air and uniform wall surface temperature will be inspected.

Governing Equations

The continuity equation represents the conservation of mass can be written as (**Anderson,1995**):

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \qquad (1)$$

ρ is the density of air, the fluid velocity at selected point in the fluid flow-field may be represented by velocity components *u*, *v*, and *w* at (*x*, *y*, *and z*), respectively.

It is impossible to model the turbulent eddies in the fluid flow by direct numerical simulation with the availability of computer resources. The Cartesian tensor equation for the RANS equations is described as [7, 8]:

$$\frac{\partial u}{\partial x_j}\left(\rho u_i u_j\right) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_i}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(\overline{\rho u_i' u'_j}\right) \quad \text{... (2)}$$

If the hot surface temperature of T_h on one side and T_c on the other side, the rate of heat transfer equation due to conduction is:

$$Q = UA\left(T_h - T_c\right) \quad \text{... (3)}$$

From the Nusselt number definition, the Nusselt number of heat transfer may be calculated from:

$$h = \frac{Nuk}{L_c} \quad \text{... (4)}$$

Where, L_c is the characteristic length (or hydraulic diameter, D_h) can be written as:

$$D_h = \frac{4A}{P_W} \quad \text{…(5)}$$

The Reynolds number of fluid flow is expressed as:

$$\mathrm{Re} = \frac{\rho u D_h}{\mu} \quad \text{...(6)}$$

Experimental Part

Figure (1) shows the main components of the experimental rig which designed and built in the Laboratories of Engineering College - Kerbala University, which is depicted in figure (2). This rig consists of: rectangular steel channel test section, blower ,U manometer ,control circuit, K-type thermocouples, heating system.

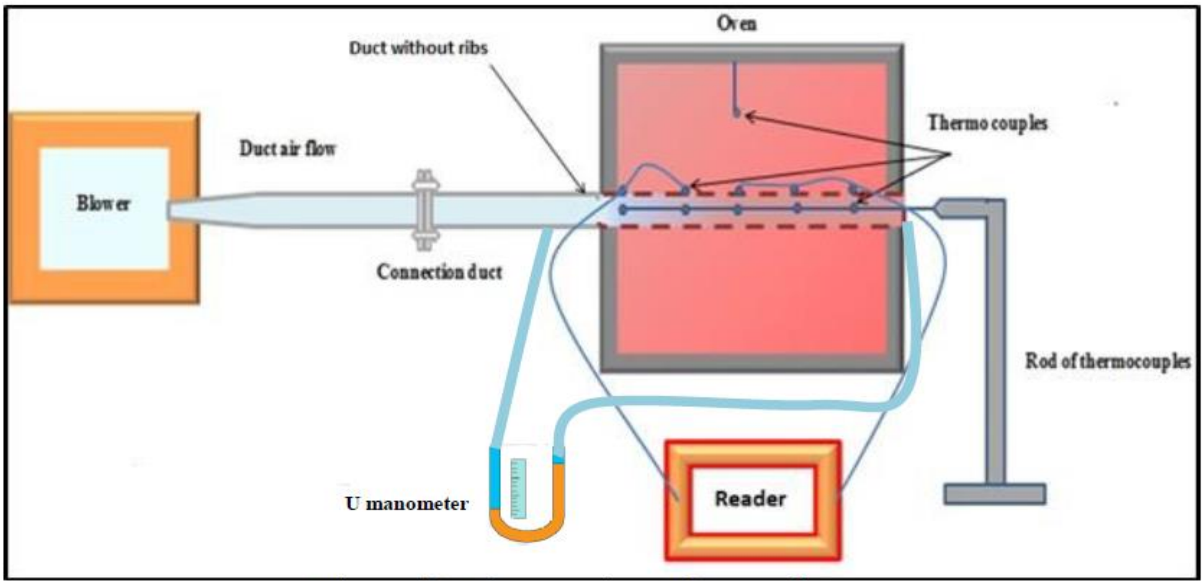


Figure (1) The experimental setup diagram

Figure (2) Photographic view of experimental rig

The air blower connected by elongated channel with the same cross-section dimensions, and then connected to the test section channel which is contained in a furnace box in a horizontal layout. The power supplied to the Ni-Cr heating wire was 1000 W oriented in a homogeneous manner around the test section channel. Furnace was well insulated with alumina as a good thermal insulation in order to avoid the heat losses to the surrounding. The thermocouples readings were observed each (45 minutes) until reaching the steady state condition. The air flow rate from blower is controlled by graduating switch.

A Circular stainless steel ribs of (27) mm outer diameter, (22)mm inner diameter and (5) mm thickness were manufactured ,there are two cases of ribs :

1- All ribs with middle fins (9 ribs) as shown in fig (4) case1

2- Ribs with fins and another without fins(5 ribs with middle fins + 4 ribs without fins) as shown in fig (4) case2

The ribs fitted in the (30*60) mm duct test section by the following procedures:

1. Drilling holes along the upper wall of the duct test section with 50 mm apparats.
2. Press all ribs by using a rod with the 50 mm spacing and then enter it inside the duct to ensure that all the ribs reached to the exact location inside the duct . Ribs were fixed in its position by using screws at the upper surface as shown in fig (3).

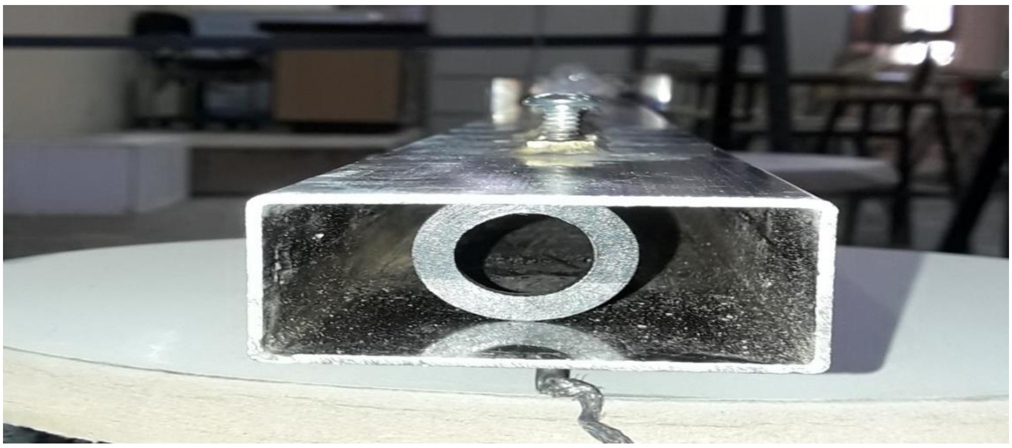


Figure (3) Ribs fitting in the duct test section

Experimental procedures

In the experimental work, the following steps were achieved:

1. Supply the electrical power to heaters and adjusting the controller to the required hot air surrounding temperature with measuring the current and the voltage, using a clamp meter.
2. Wait for a period of time to reach the steady state condition (approximately 45 minutes), then read and record all measurements required.
3. Regulate the blower gate to obtain the required flow rate (Re=7901.577), and wait to reach the constant air temperature around the duct test section (673 K).
4. Record thermocouples reading (T1, T2, T3, T4 and T5) which represent the inner wall surface temperature of smooth duct (without ribs), and the air cooling temperature at the duct centerline by thermocouple .

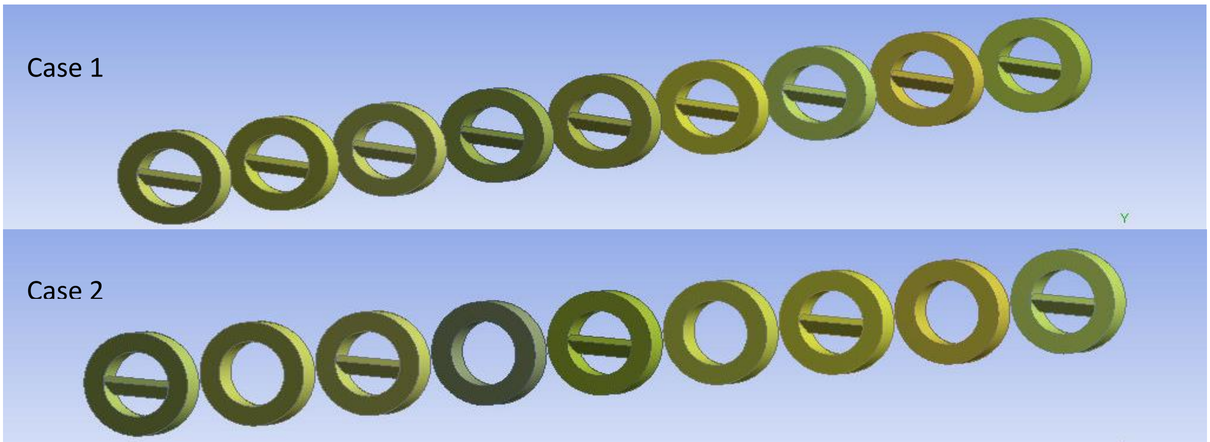


Figure (4) ribs geometry

CFD Methodology

In this investigation a 3-D numerical simulation of the conjugate heat transfer was conducted using the CFD code FLUENT 14.5. The CFD modeling involves numerical solutions of the conservation equations for mass, momentum and energy. These three equations are used to model the convective heat transfer process with the following assumptions, (a) steady 3-D fluid flow and heat transfer, (b) incompressible fluid and flow, and (c) physical properties of cooling fluid are temperature dependent.

Boundary conditions:

The boundary zone location is specified in the GAMBIT itself; the inlet, outlet and the wall condition location is specified.

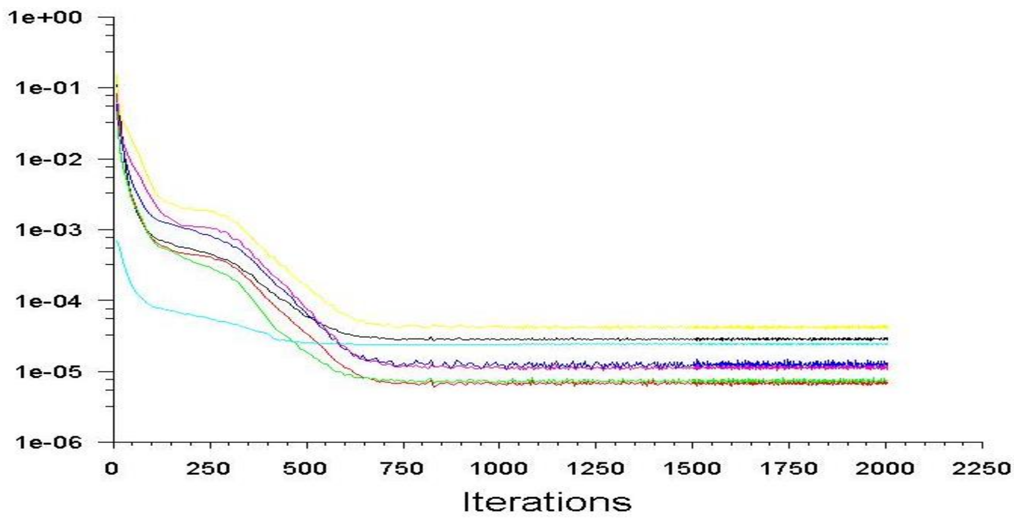
Fluid entry boundary condition:

The inlet air flow velocity is 3.2 m/s with constant temperature of 300 K.

Wall Boundary Conditions:

The duct wall is provided with wall boundary condition, a constant hot temperature is 673 k provided for plain and ribbed tube

In the present work, it requires 2000 iterations to reach the convergence. If the residuals are still not below the proper values, then additional iterations are requested. The output iterations are shown in Figure (5).



Figure(5) Convergence to solve discrete conservation equations

Results and Discussion

Figure (6) shows the cooling air temperatures at the channel centerline for a constant surrounding hot air temperature (673 K) , inlet air temperature (300 K) and coolant air flow of (Re=7901). The coolant air temperature at channel centerline seems to be higher than that for smooth channel (without ribs). Ribs make wakes which developed vortices and then leads to rise the heat transferred from the channel wall to cooling air, and thus the increase in the coolant air temperature can be obtained is (10.22 %) for case 1 and (8 %) for case2 .

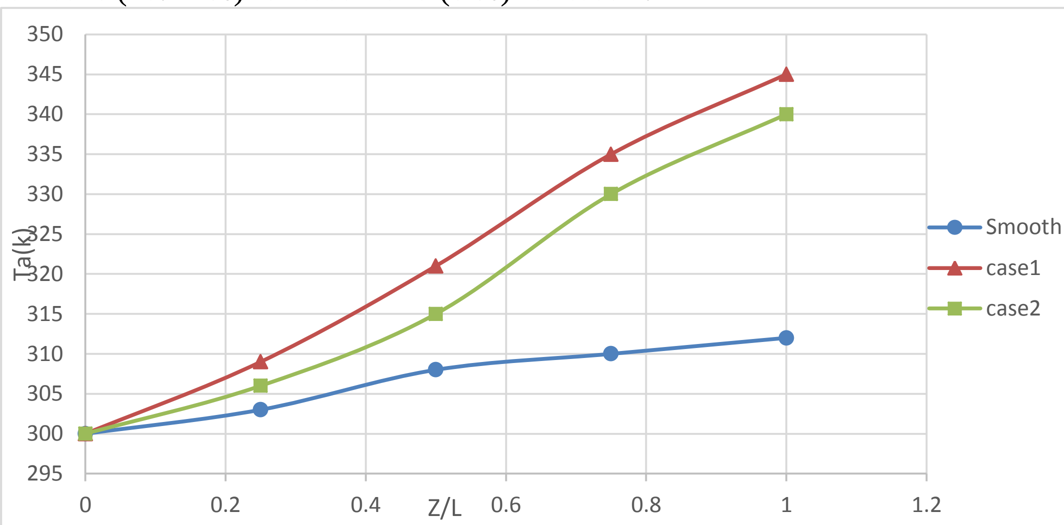


Figure (6) Cooling air temperature distribution along the channel centerline

Figure (7) presents the temperature distribution along the inner surface of the channel upper wall, it can be seen that the duct with finned ribs is the better case which leads to increase the coolant air temperature by (6.15 %) for case1 and (3.5 %) for case 2**.**

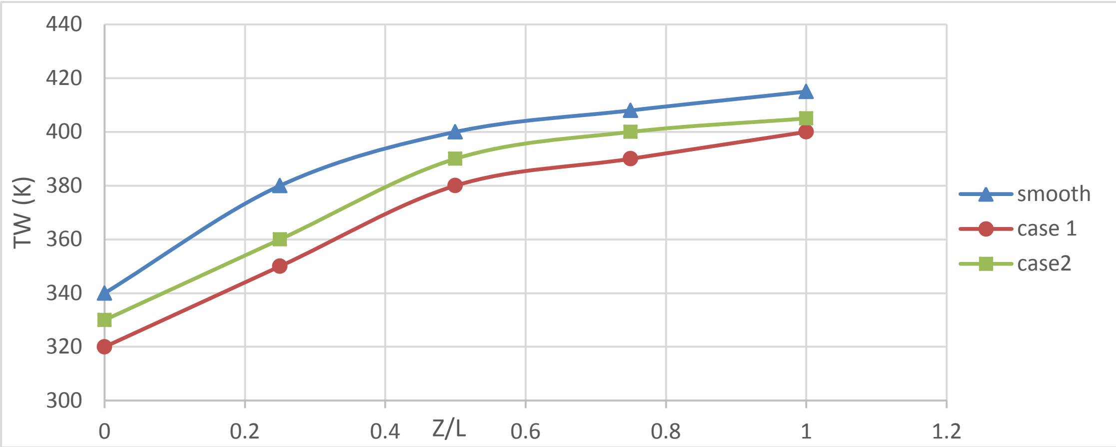


Figure (7) Temperature distribution along the channel inner wall

Figure (8) reveals the coolant air flow velocity along the channel centerline, it is clear that the coolant air flow velocity accelerated and decelerated through the channel, due to the ribs.

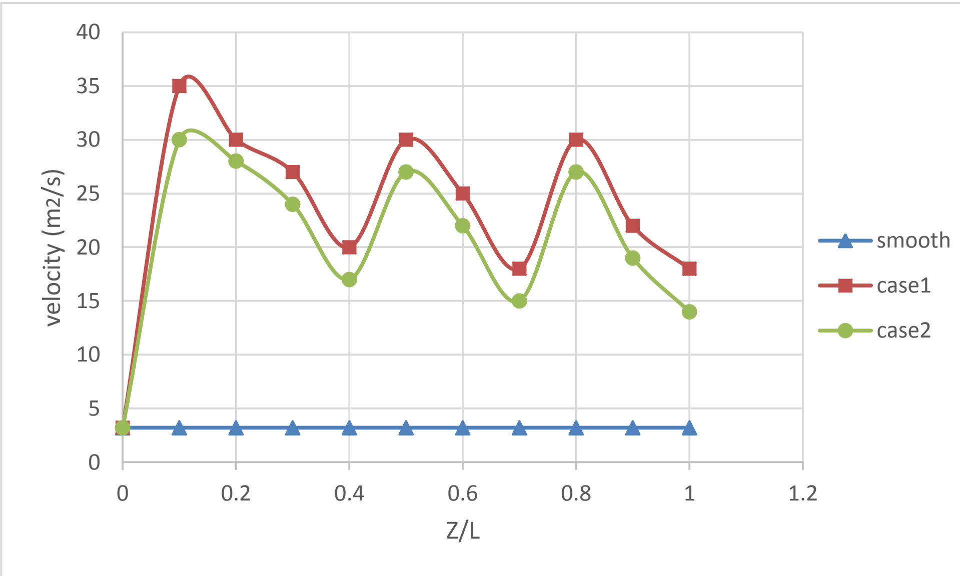


Figure (8) Cooling air velocity at channel centerline

Figure (9) thermal performance factor along the channel, it can be seen that the thermal performance is larger than 1, this means that the ribs configuration performance always exceeds the smooth channel. After the second rib, heat transfer decreases because ribs break down bend drive vortices that would moreover favor increment of heat transfer.

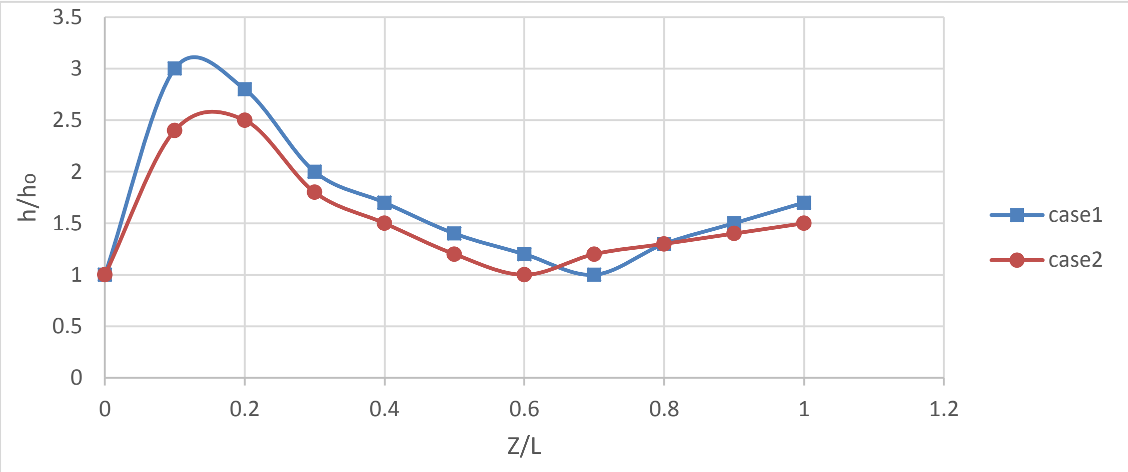


Figure (9) thermal performance factor along the channel

Figure (10) shows contour of temperature distribution for smooth rectangular channel for surrounding hot air temperature of (673 K) , inlet air temperature (300 K) and inlet air flow Reynolds number (Re=7901). It was shown that the cooling air temperature at duct centerline remained constant throughout the channel, while the coolant air temperature was unaffected in the channel centerline till (33 % for case1) and (50 % for case2) of the channel length and then sharply increased at the last (30 % for case 1) and (15 % for case 2) as shown in figure (11), (12). This is due to the effect of circulation generated by the ribs which clearly enhances the transfer of heat between the hot channel walls and the coolant air flow stream.

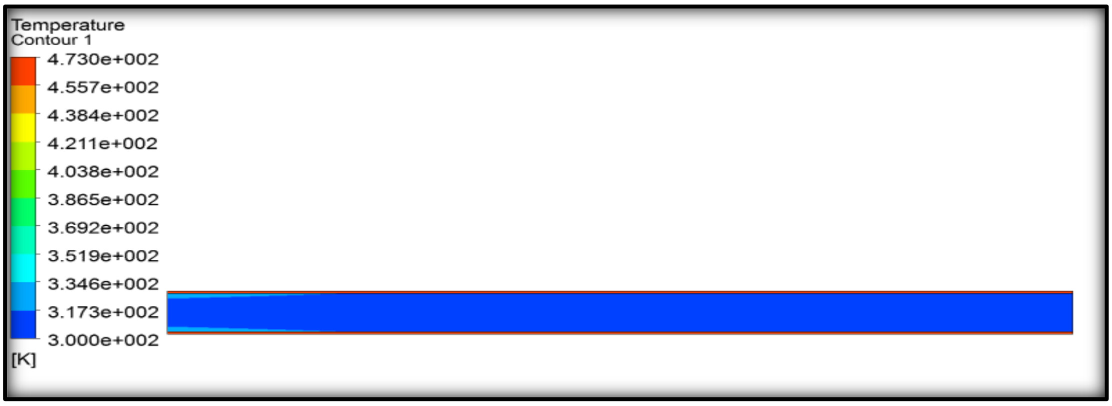


Figure (10) Temperature distribution contour for smooth duct

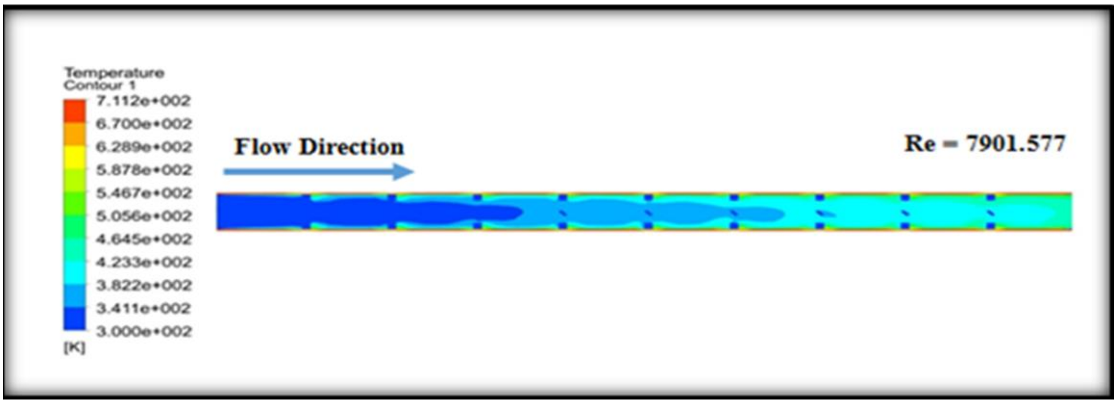


Figure (11) Temperature distribution contour for case1

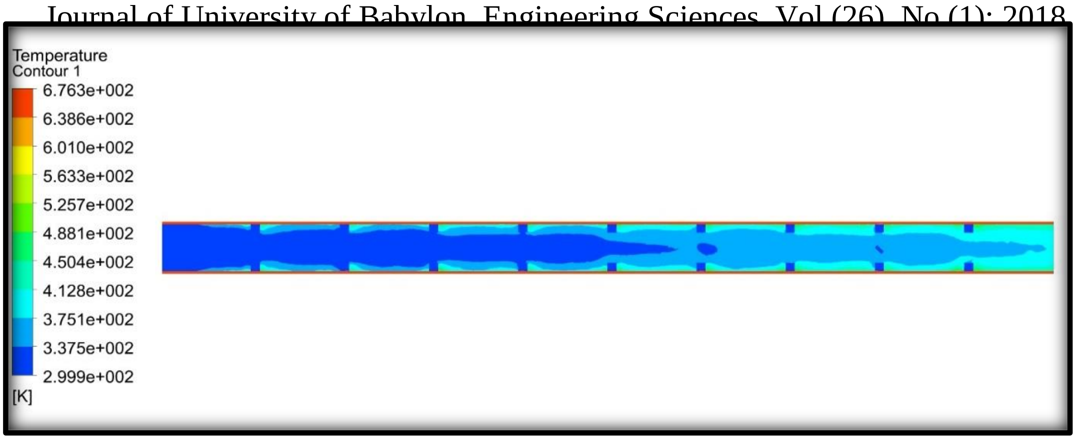


Figure (12) Temperature distribution contour for case 2

Figure (13) shows contours of temperature distribution for the rectangular channel with circular ribs having inclined fins at each rib location. It can be noted that after the first rib, air accelerates around the rib and boundary layer flow with separation is observed downstream of ribs which lead to make vortices and then enhances the heat transfer.

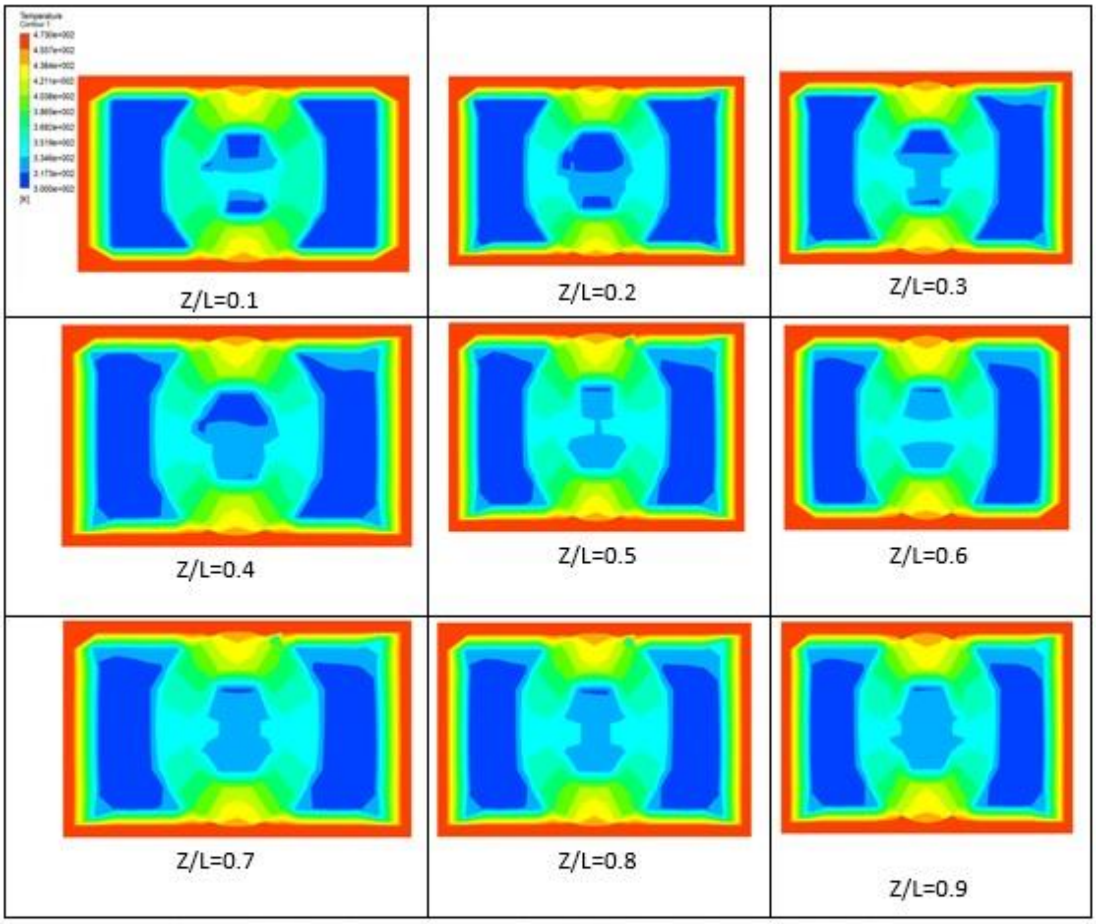


Figure (13) contour of temperature distribution at each rib for case 1

Figure (14) reveals that the cooling air velocity at channel centerline increased downstream constant throughout the channel, while figures (15), (16) presents the velocity distribution contour through a channel with ribs ,The coolant air flow was accelerated and decelerated through the duct, due to contraction and expansion for using these ribs.

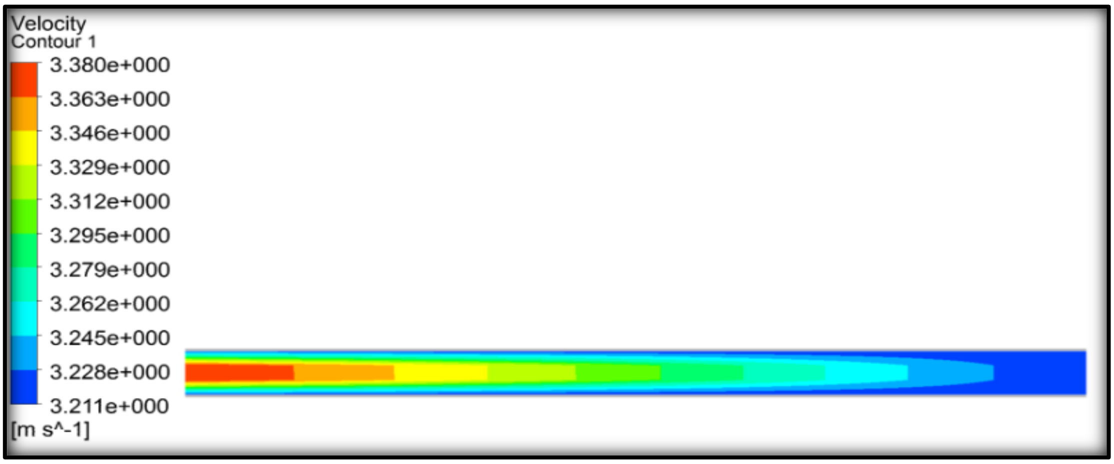


Figure (14) Contour of velocity distribution for smooth channel

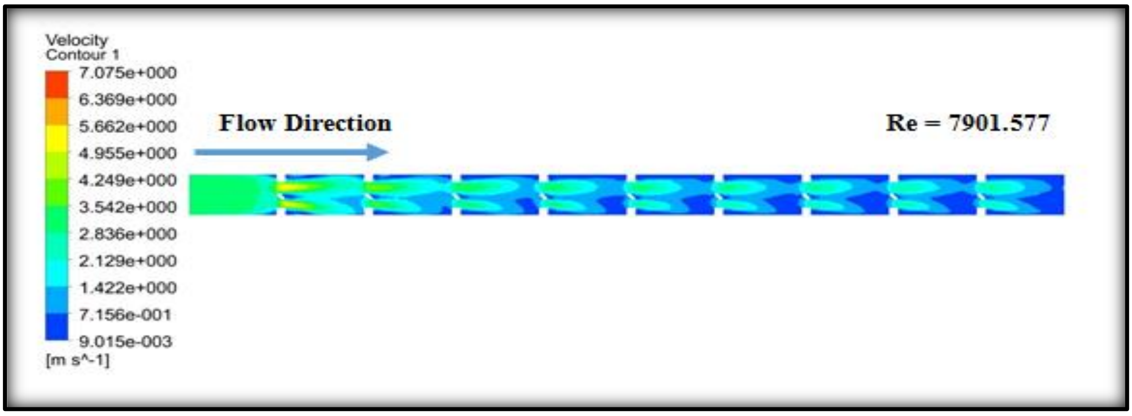


Figure (15) Contour of velocity distribution for case 1

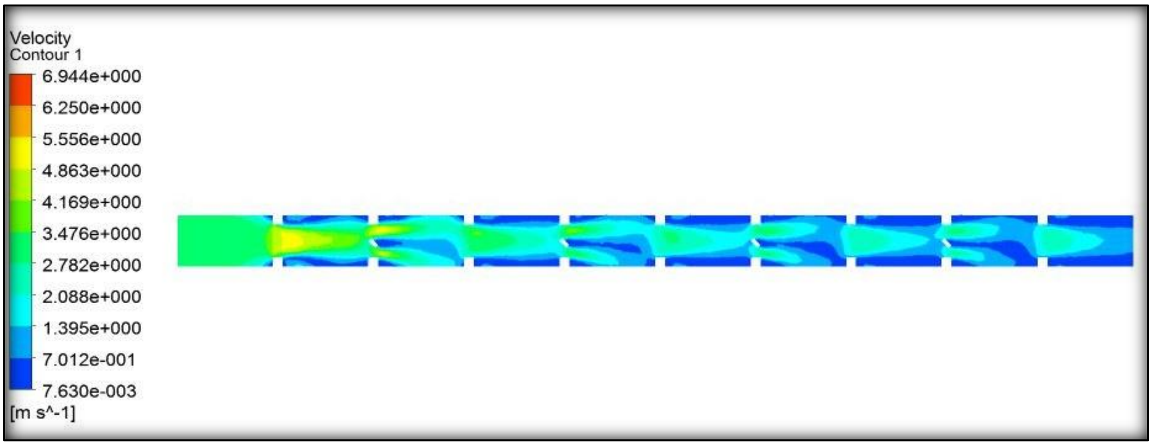


Figure (16) Contour of velocity distribution for case 2

Figure (17) presents the cooling air velocity at each rib location along the duct, it can be noted that as the flow approaches the first rib, it accelerates around it and boundary layer flow with separation at the downstream of the ribs. The effect of wake behind the rib decreases due to the increase in velocity through the circular ribs. The velocity at the right and left of the rib was seen to be increased.

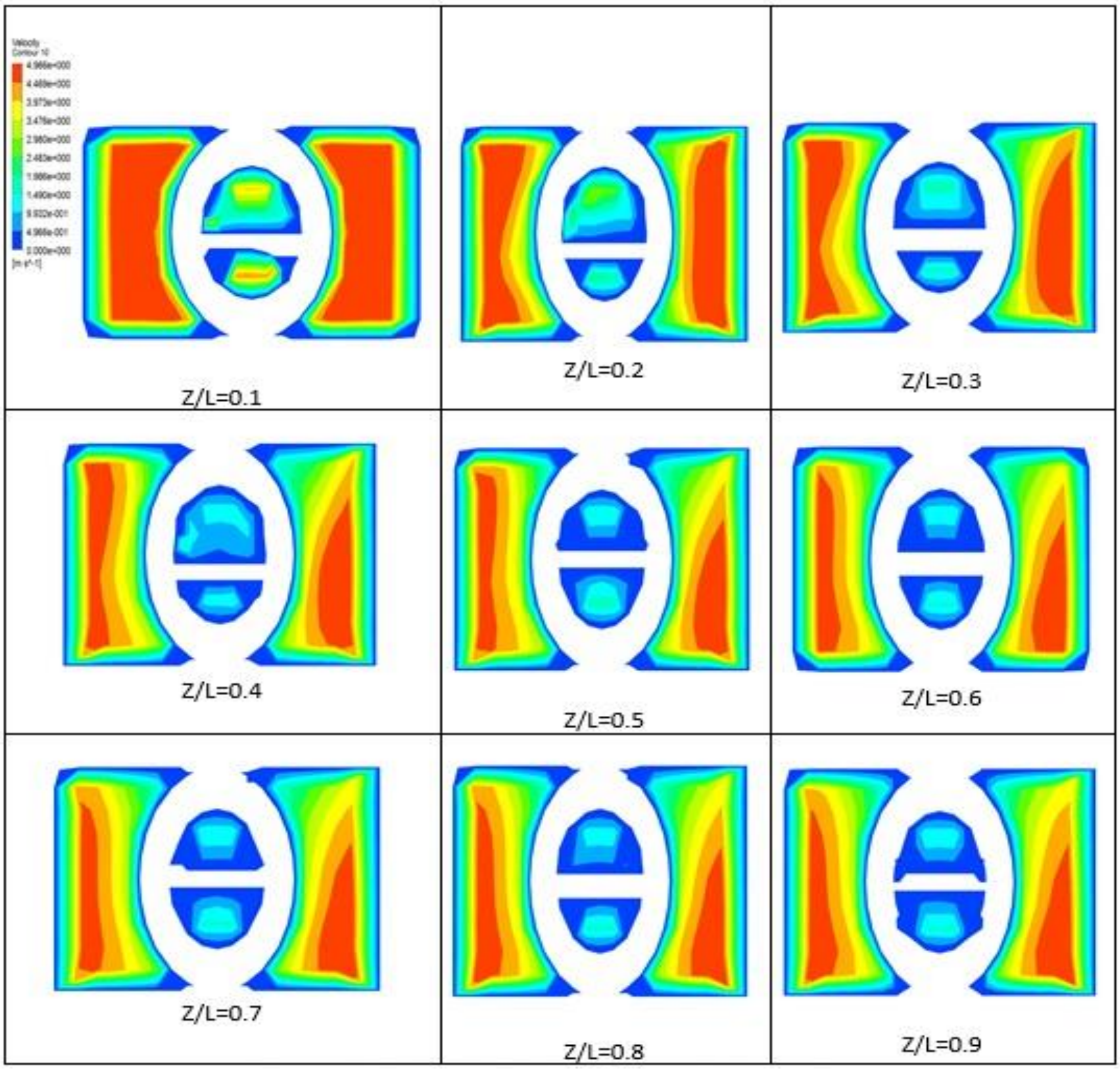


Figure (17) velocity distribution for channel with ribs location for case 1

Conclusions

From the present work, it can be obtain the following:

1. The coolant air temperature for duct with ribs was higher than that for smooth duct by (10.22 %) for case 1 and (8 %) for case 2.
2. The temperature distribution in the inner surface for ribbed duct is lower than smooth one by (6.15 %) for case1 and (3.5 %) for case 2.
3. The coolant air flow velocity was accelerated and decelerated through the duct, due to the contraction and expansion by the ribs.
4. Heat Transfer Coefficient increasing when using ribs in the duct because the increase in mixed flow (high turbulent) and increase the surface area of heat transfer , Heat Transfer Coefficient in the case (1) higher than case (2) .
5. The thermal performance factor along the duct is larger than 1, meaning that the ribs configuration performance always exceeds the smooth duct.

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