Review Of Heat Transfer Enhancement In Air-Cooled Turbine Blades

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Abstract: The gas turbine manufacturing is always looking for rise the thermal efficiency of gas turbine. One of the methods is by rising the turbine input temperature. Several major problems appear when rising the operating temperature such as: design melting temperature of the turbine gas is much higher than the melting temperature of the most materials, and the thermal stresses can be created by the fast spatial variations in temperature within the blade which can be in dangerous limits. To tackle the problems associated to thermal stresses, creep and oxidation which bound the lifetime of turbines, cooling of blades are necessary.

Keywords: Heat transfer, internal cooling, gas turbine, turbine blade.

1 INTRODUCTION
For modified gas turbines, the turbine inlet temperature of the gases is in the maximum range of (1700°C – 2000°C). These temperatures exceed the melting point of the turbine components. Therefore, it is essential to cool the turbine components so that they can withstand these extreme temperatures. With typical cooling techniques the temperature is decreased to almost 1000°C, so that they can withstand this extreme environment [1, 2].

2 GAS TURBINES CYCLE
Gas turbine systems operate on the thermodynamic cycle known as the Brayton cycle as shown in figure (1). In a Brayton Cycle the fresh air input to the compressor in order to compressed to a high pressure. The air mixed with fuel and combustion in the combustor. The burning products expand in the turbine then discharged to the environments. The turbine linked with generator to generate electricity [3].

A basic gas turbine engine contains four sections: intake, compression, combustion, and exhaust as shown in figure (2) the electrical starter rotated the compressor section to start the engine on the small engines or an air driven starter on big engines. The speed of compressor accelerates, air is carried to the inlet duct then compressed to a high pressure, and transported to the combustion chambers (burning section). Not whole of the compressed air is employed in the combustion, some of the compressed air bypasses the combustion chambers and flows within the engine to use in the internal cooling. The fuel controller injects the fuel through spray nozzles and ignited by igniter plugs. The fuel and air are mixture in the burning section, then burned in a continuous burning progression and gives a very high temperature typically about 2205 ºC, which heats the whole air mass to 871–1316 ºC. The mixture of gases and hot air expands and is moved to the turbine blades forcing in the turbine section to rotate the blades turbine [5].

3 INTERNAL COOLING
In general, gas turbine cooling is achieved by bleeding some relatively cool air from the compressor and using it inside the gas turbine blades to remove heat transferred into the blade from the hot mainstream. The cooling air flows through internal cooling passages inside the blade. These passages are specifically designed to maximize the heat transfer. In modern gas turbine blades, this is usually done by the use of contact area enhancers and turbulence promoters, such as ribs and pin-fins in the passages. Some of this cooling air is ejected onto the surface of the turbine blade to form an insulating film - with the goal of reducing contact of the blade with the hot mainstream gas [6]. There are a number of methods such as film cooling, rib

Fig. 1 Diagram of Brayton Cycle [4]

Fig. 2 Structure of a gas turbine engine [5]
turbulators, jet impingement, shaped internal cooling passages, dimple cooling, shown in Figure (3.a, b) to cool a modern gas turbine blade. The pin fin used to cool trailing edge, rib turbulators are employed to cool the internal passages and jet impingement is employed to cool the leading edge. This study focuses on the using rib turbulators or turbulence promoters to use in the internal cooling turbine blades [2].

Fig. 3 (a) Film cooling on a gas turbine blade (b) Interior cooling for a gas turbine blade, showing passages with pin fins and ribs [7]

4 EFFECT OF RIB TURBULATORS

The aim for presenting the ribs at regular spaces is to improve the heat transfer averages. Ribs are manmade protrusions that are sited in a controlled technique along the walls. The rib prompts a separation through flow and hence causes an increase in the frictional loss. The improvement of the heat transfer has a drawback in the rising pressure drop, which sometimes be able to several times larger than for the smooth passage. The heat transfer and pressure drop are strongly associated to the height of the rib. Though the ribs can be placed at different orientation, our study focuses on the ribs placed orthogonally (at 90 degrees) to the mainstream flow. The rib size and the space between the two following ribs, the pitch has great importance [2]. The performance of heat transfer in the ribbed passage depends on Reynolds number of the coolant air, rib shape, the passage aspect ratio. When the coolant passes over the ribs, the flow separates and reattaches as shown in Figure (4).

Fig. 4 Rib turbulators placed on opposite walls of the cooling passage showing flow separation and reattachment [8]

This separated and reattached boundary layer results in the increased heat transfer coefficient of the ribbed channel. The rib induces secondary flow that further improves the heat transfer from the wall to the coolant. The rib also induces turbulent mixing in the passages which increases the flow velocity [2]. Figure (5) shows the flow phenomenon of different rib spacing to rib height (P/e) and the separation then reattachment of it. For (P/e) rib spacing to rib height less than (7), the separation bubble is appear on top for each rib. While rib spacing to rib height is (7), the boundary layers between ribs and separation areas does not grow. For rib spacing to rib height is (10), that is thought to be best value as ultimate heat transfer is attained. Therefore at (P/e) equal to 10, peak break-up for near wall flow is reached that results in rise in the turbulence level and improve the exchange for fluid in the near wall area with main flow [1].

Fig. 5 Flow construction for various rib spacing to the rib height [9]

Liu and Hwang [10] investigated the effect of p/e and e/ Dh ratios on the local and average heat transfer and friction factor for fully developed flow in channel of (aspect ratio = 4) with two opposite rib-roughened walls. The Reynolds number ranged from 5x10^3 to 5.4x10^3. The p/e ratio were 10, 15 or 20 and e/Dh were 0.063, 0.081 and 0.106. they found that the local Nusselt number decreases as the pitch increases beyond 10. Aliaga et al. [11] made measurements of heat transfer and fluid flow visualization...
for p/e = 12 in a rectangular channel at a Reynolds number (1-1.5)×10^6. From Figure. (6.a), it can be imagined that the fully developed flow field was done after the third rib. The absence of a peak shape in the entrance region suggested a trapped vortex between the two ribs (Fig. 6.b). After the second rib, the periodic arrangement, featured by the ascending-descending profile, for the fully developed was achieved. This may indicate which the reattachment downstream first rib, if any happens at a higher distance than that for the continuously fully developed flow area.

Zhang et al. [12] studied experimentally the effect of ribs spacing on friction factors and heat transfer coefficients in rectangular channels 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. Zhang et al. results showed that the heat transfer and friction factor values for ribbed-grooved ducts were decreasing with an increasing in the value of the rib-groove pitch-to-rib height. Cavallero and Tanda [13] investigated the effect of a small p/e ratio (p/e = 4) on the local heat transfer distributions for a rectangular duct channel (AR= 5), where the thermal field did not reach fully developed. The inter-rib heat transfer distribution was characterized by a monotonic increase that was periodically repeated after 3-4 ribs, as shown in Figure. (7). This feature of the heat transfer distribution appeared by the presence of a trapped vortex between the ribs without a reattachment point.

Robert et al. [14] performed heat and fluid flow visualization in a trapezoidal high aspect ratio channel representing trailing edge of the turbine blade cooling channel. The transverse and sloping ribs were joined to two opposing long side walls and they were intended to function as secondary flow as well as turbulators to increase the heat transfer of the both rib-roughened and rear walls. The experiments and simulations were achieved with different rib inclinations, 90, 75, 60 and 45-deg. ribs, respectively, and two types of ribs, proportional and non-proportional ribs, respectively. The rib proportionality depended on the trapezoidal shape of the channel, so that the ratio of the rib height to the local channel width in the vertical direction was always equal to 1/3. The non-proportional ribs have a square cross section with height of rib e=1/3 w. The obtained results presented a complicated flow structure with a rib prompted secondary flow that conveyed air from channel core area towards rear wall. Sachin. et al. [15] presented the experimental research for improve heat transfer through a rectangular channel roughened using repeated V-shape ribs for the range of the Reynolds numbers from 5000 to 14000,The aspect ratio of rectangular channel (W/H) equal 8, angle of attack 60°, relative roughness height (e/Dh) equal 0.030, relative roughness pitch (p/e) equal 10 . The characteristics of heat transfer for roughened channel have been compared with smooth channel at similar flow condition. The maximum improvement in the Nusselt number and the friction factor is detected to be 2.57 and 2.85 times for smooth channel. Umesh et al. [16] presented an experimental work to discover the thermal and hydraulic performance in the fixed square passage using 45°inclined arc and V-shaped for the circle rib turbulators. The Reynolds numbers ranged from 45x10^3 to 75x10^3 and the aspect ratio of the passage (W/H=1) was considered. Rib shape geometry involved different blockage ratios (rib height to channel hydraulic diameter ratios ) of 0.083, 0.125 and 0.167, as well as pitch to height ratio (rib spacing ) is 10. The obtained results of the passage with various ribs shape confirmed which the rise in the rib width improved thermal performance for the passages. The optimal cooling configuration was obtained by the combined effect for the rib width, rib spacing and flow parameters. Shailesh et al. [17] had experimentally investigated the influence for rib angle tendency and effect of a hole provided in the integral ribs on pressure drop and heat transfer through a square channel. The experimental investigation has been performed with Reynolds number about (5000 to 40,000), relative roughness height (e/Dh) of 0.060 , relative roughness pitch (p/e) of 10, and rib attack angle of 90° and 60°. The 60° ribs with a gap yields about (7.4) times raise in the friction factor and about (3.8) times enhancements in Nusselt number compared with smooth passage and about (1.2) and (1.1) times that of 60° ribs without the gaps. Zahra [18] presented experimental investigations of local heat transfer through a rectangular passage with aspect ratio (W/H=4) used ribs can be continuous transverse as well as an obstacle. Liquid crystal thermography is employed to obtain the temperature fields in the heat transfer experiments. The flow Reynolds number, based on the channel hydraulic diameter, has the values 57000, 89000, and 127000, rib height to diameter ratio (e/Dh = 0.078), the rib-rib spacing to rib height (p/e) ratios =10, 20 and 30, in the first inter-rib. Sourabh Kumar [19] improved the performance by casting repeated continuous V- and broken V-shaped ribs on one side for the two pass square passages into the core of the gas turbine blade. Reynolds
numbers about (16,000, 56,000 and 85,000) which used within the turbulent flow. The heat transfer enhancement was 1.5 times higher in broken versus continuous ribs. The broken ribs performed more highly as the convection of the secondary flow is more due to the broken side ribs. Julker et al [20] presented an experimental research on turbulent flow characteristics in a rectangular channel for Reynolds number about of (15000 to 30000) fitted with the semicircular ribs for constant height (e = 3.5 mm). The duct height was 30 mm where the aspect ratio for a rectangular channel was (AR= 5). Four changed rib pitches for (28 3.5, 42, and 49 mm ) with constant rib height to channel height ratio (e/H = 0.116) and constant rib height to hydraulic diameter ratio (e/Dh = 0.07). Results showed that the maximum thermal enhancement is achieved by P/e = 10 among all other rib pitch to rib height ratio (P/e). kumar et .al [21] made an experimental work of heat transfer characteristics in the square channel. Square duct has an internal size of (4500 x 60 x 60 mm), which consists of an entrance section. The aim of this study was influence of square ducts on the convective heat transfer mechanism with Reynolds number ranged of (10,000-50000 ). The study showed significant improvements convection heat transfer can be achieved with the square duct. The Nusselt number increased at about 112% when compared with those from correlations of Dittus–Boelter. Mehmet et al. [22] performed experiments on forced convection of the heat transfer through a rectangular duct with the perforated ribs is used. The purpose of the work was to study the effect of the thermal performance of the ribbed channel. The Reynolds numbers of the duct flow range from 5375 to 36362 and the convective fluid was air , rib height to channel height ratio (e/H= 0.1). Results of the heat transfer for duct with perforated ribs are compared with results of a smooth duct that showed an increase 34.1% in heat transfer because of using ribs. Sivakumar et al. [23] presented experimental work of the heat transfer characteristics compared between smooth and three changed sized square ribbed divergent rectangular channels . The Reynolds numbers of the channel flow range from (20000 to 50000) with heights of the rib turbulators (e) were ( 3, 6 and 9 mm), the rib height to hydraulic diameter ratio of 0.035, 0.0697 and 0.1046 . The improved heat transfer rate of the ( 3 mm ) height rib divergent rectangular channel is more than (6, 9 mm) rib height rectangular divergent channel and the smooth channel. Chavan et al. [24] investigated experimentally the heat transfer improvement for divergent plain channel and divergent ribbed channel. The range for the Reynolds’s number from 5000-25000 (velocity 3.2 to 16 m/s). The ribs were placed in the inner surface of channel , rib height to hydraulic diameter ratio (e/ Dh = 0.045) and the relative roughness pitch (p/e) of 8 . The thermal performance of divergent rib duct is higher than the plain divergent duct. Thermal Performance for the divergent channel is increased by (34%)because of using the rib tabulator. Oronzo et al. [25] presented a numerical investigation of different-dimensions rectangular passage, ribboned walls was applied. Flow with different Reynolds number about ( 20000 and 60000),changed inclination angles (8 = 0°, 11°, 22° and 33°) of turbulators were used. Numerical simulations using the software FLUENT code. Simulations showed that the maximum Nusselt numbers were discovered by increasing the (5) angles, and the heat transfer rate was (1.34) times of the smooth channel at least Re = 20000. Khudhayer [26] presented the heat transfer and fluid flow through a passage with ribs turbulators. The problem was study with Reynolds numbers up to (32000). The effect of the number of ribs and the rib thickness on the flow, thermal field and step height, were studied. The calculated results were offered as velocity vectors,streamlines counters, and graphs of turbulent kinetic energy variation and Nusselt number. The effect for turbulence was modeled by using (k-e) model. The results showed that the size and strength of the recirculation regions behind the step are increased with increase for contraction ratio,The numerical analysis showed when increase Reynolds number cause increase in the heat transfer . Ahmed [27] carried out a numerical investigation to study heat transfer and turbulent flow characteristics in the ribbed square passages. The software Fluent 6.3 CFD has been applied. Reynolds numbers were from 10000 to 40000. Rib arrays for 45° V-shaped and 45° inclined and are attached in inline and staggered arrangements on upper and lower walls of the passage and compared with the 90° transverse ribs. The V-shaped and inclined rib pairs placed in the inline manner reveal heat transfer enhancement for about 17 - 50% higher than that for the 90° rib. Sagar et al. [28] presented a numerical simulation on square duct having hydraulic diameter (Dh ) of 0.05m. Air is working fluid with the flow rate in terms of Reynolds number ranging from 15,000 to 20,000. Details for rib height (e), pitch distance between turbulators (P) and turbulators angle are similar to experimental reference. The model is creating using ANSYS ICEM software. Numerical simulations were performed using the CFD software package ANSYS 14.5 FLUENT. Turbulence closure was achieved using k-e turbulence model, with enhance wall treatment for the simulation were used. The Nusselt number 2.0 times above at higher Reynolds number and friction factor 15 to 20 times above than those in the smooth duct without insert at pitch ratio 0.4. Sandeep et al. [29] carried out computational study to find heat transfer characteristics in rectangular channels with reverses pentagonal form at the same height for turbulent flow. The walls of the air side were ribbed with an array of the short ribs. The fluid in the channel was air and the heat transfer characteristics were find by measuring the overall heat transfer coefficients for the channel. The results showed the Nusselt number rises with rise of Reynolds number. A higher Reynolds number, 18,000 with the ratio of pitch to width for rib is 14.33 the maximum value for average Nusselt number was found to be 0.0226. Researchers [30-66] designed and built an experimental rig to simulate conditions in the gas turbine blade cooling. It was presented the effect of using different rib geometries fitted in different channel shapes and found the fluid flow and heat transfer characteristics. 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. Sohil et al. [67] presented a computational study to determine the average heat transfer coefficients and friction factors for turbulent flow through rectangular ducts with ribs. The
software ANSYS FLUENT 12.1 is applied to study the flow across the ribbed channel. The air was the working fluid flow in the channel, and the average heat transfer coefficients were find by measuring overall heat transfer coefficients for the channel. The results showed that the maximum enhancement of average Nusselt number is found to be 3.05 times that of smooth duct for relative roughness height of 0.07. Deepak et al. [68] investigated the three-dimensional CFD simulations to investigate heat transfer and fluid flow characteristics of artificially roughened rectangular channel using Ansys-CFX. Heat transfer characteristics of the rectangular channel are investigated for Reynolds numbers ranging from 8000 to 18000. Rectangular channel has an aspect ratio of 5, while the domain length for numerical analysis is kept 550 mm long the hydraulic diameter(Dh) of duct is 66 mm the relative roughness height (e/Dh) is 0.030, relative roughness pitch (p/e) is 10. The results showed the Nusselt number increases as the Reynolds and the Friction factor decreases as the Reynolds number increases. Navanath et al. [69] presented numerical study of heat transfer in the rectangular duct of cross section 2100x150x200 mm. using different internal ribs (V rib semicircular, Broken V rib 60 degree, Rectangular rib, V rib 60 degree, Rectangular rib with taper, Broken V rib (45 degree) ,Triangular rib , V-rib (45 degree), Triangular broken rib). The software ANSYS FLUENT is applied to study the flow across the ribbed channel. ribs spacing =50mm and pitch to height ratio (rib spacing ) is 8.3 with air as the working fluid at velocity equal to 1 m/s. The results showed the maximum heat transfer coefficient of 78 w/m²k for Rectangular taper rib.

5 CONCLUSION

Advanced gas turbines operating at extremely high temperatures, it is necessary to implement various cooling methods, so the turbine blades survive in the path of the hot gases. Simply passing coolant air through the airfoils does not provide adequate cooling; therefore, it is necessary to implement techniques that will further enhance the heat transfer from the airfoil walls. The internal heat transfer can be enhanced with jet impingement, pin-fin cooling and internal passages lined with turbulence promoters. The heat transfer distribution in cooling channels with turbulators has been studied for many years because a number of factors combine to affect the heat transfer.

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