IJEIT
The International Journal of Engineering and Information Technology
journal homepage:www.ijeit.misuratau.edu.ly
MISURATA UNIVERSITY

Achieving Best Turbine Blade Cooling Efficiency from a Cylindrical Cooling Hole with Reverse-Vortex Generator Using Numerical Simulation

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Abstract— **This investigation bargains mathematical investigation of the impact of kidney vortex generation (KVG) on the film cooling execution of a round cooling opening which has measurement (d = 20 mm) on a level plate. The collection between the stream cooling air and standard hot gases will result kidney vortex which will wipe out the film cooling adequacy. Two kind of reverse vortex generator (RVG) with various statures (H = 0.5 d and 0.25 d) are intended to eliminate the kidney vortex and examine best film cooling adequacy, where every one of them is mounted to the level plate upstream of the cooling opening by changing its parallel positions (A = 0.0 d, 0.25 d, 0.5 d and 0.75 d) concerning the opening community line and for each kind has diverse distance regard to opening focus line. The changing of blowing proportion (BP = 0.5-1.0) was considered in this investigation. The simulation model was used to simulate the formation of film cooling using a shear stress transfer (SST) model by Workbench 15 software. The outcomes have been introduced in terms of laterally averaged and maximum value of film cooling comparison diagrams, vorticy on x/d = 3.0 and film cooling distribution which clarified how the outcomes acquired. The Cases 04 and 07 gave the best situations for RVG where case 04 gave wide covered for laterally average film cooling efficiency (η = 0.35), while case 07 gave the highest expansion distribution and the maximum value for film cooling efficiency (η = 0.66). The best results are obtained when the blowing ratio (BP = 0.5) due to a lower velocity of cooling air which will made best interaction with the hot air (main stream)which will represent the hot gases in real case.**

Index Terms: **Inverse vortex pair (IVP), Blowing proportion (BP), Local temperture (θ), Film cooling efficiency (η)**

I. INTRODUCTION

Increasing demand to raise the efficiency of gas turbines request raising turbine inlet temperature. These days, the gas turbines work at the temperature range around 1800K-2000K, which is a lot higher than the softening temperature of the turbine parts materials. Such expanding of the turbine inlet temperature became conceivable due to utilization of cooling plan on the turbine parts. As define Film cooling is a jet flow of cold air to give a layer of cool air flow between the hot gases and the blade surface, decreasing heat transfer to the surface.

Kidney-rotating vortex pair (KRVP) produce due to raise up flow of cooling air. Then, at that point, the strategy for controlling KRVP has been concentrated since quite a while past. The shape of the cooling opening way out is known as one of the main factor which influences the film cooling execution and vortex structures [9]. As another idea for the controlling (KRVP), a few methods have been utilized in which surrounded of a cooling opening as utilizing ramp placed at the upstream of the film cooling openings [2],[3],[10].

A huge improvement can be achieved in the cooling execution of the film by using different types of exit openings shapes. Study of the effect of diffuser expansion angle, the angle of inclination of the cooling hole cooling tube length at fan inlet and circular holes [6]. Furthermore, the compound-angle holes, converged cooling holes, illustrate a better film cooling effectiveness than simple angle holes; however increase the heat transfer coefficient. Converged holes help to maintain the cooling jets closer to the surface and will decreases the mixing influence as well as improves the coverage of film cooling overall [1]. Other method was utilized to decrease the rise up of cooling air flow which is by utilizing a 10 deg span wise-diffused hole and found that the shaped hole produce better film cooling characteristics than commonly used circular holes. Subsequently, the coolant air flow had more infiltration into the mainstream when compared to the circular holes, and the film cooling execution was improved [7].

There are many researchers proposed other methods for example changing of blowing proportion in order to investigate better film cooling effectiveness along the blades surface. Blowing proportion (BP) strongly effect on the outcome. The efficiency of film cooling has increased along with the increase of blowing proportion. Nevertheless, when the blowing proportion exceeds more

Received 24 Agu , 2021; revised 17 Oct, 2021; accepted 25 Oct, 2021.

Available online 25 Oct, 2021.

than an optimum value, the efficiency will be reduced [6].

Some of explores concentrated about the influence of inclination angle to the film cooling. If the angle between cooling hole and the blade surface more steeper, there will be more coolant air mixes with the hot gas, which means there will be no contribution to the cooling of the surface [6]. Change the angle of inclination of the cylindrical hole 30, 60 ,and 90 degrees in a flat plate with a blowing proportion ranges from BP = 0.33 to BP = 2 and constant hole length in all the cases. It saw that 30 ° had the best film cooling adequacy with various blowing proportion [4][5].

This paper deals with a numerical study of the effect of the reverse vortex generator (RVG) on flat plate film cooling. This research is based on [8] in terms of the computational field, geometry of (RVG) and the limit conditions. The difference between current study and [8] study was just the modifying of (RVG) design. Where the primary design is based on [8] method whereas the secondary design is based on current method. In this search, film cooling efficiency, the stream field, and the temperature field were uncovered by utilizing CFD approach.

II. METHODOLOGY

A. Computational field

The computational field will be relating to the experiment geometry [8]. In this testing, the main stream air goes over the film cooling surface with a row of four circular cooling holes, which are 35^0 slanted against the plate. The opening length to diameter ratio is l/d ≅ 3.5. The cooling holes are feeding out with air from a plenum. The cooling holes have an internal diameter (d) of 20 mm. The opening pitch, p, was 60 mm (3d). The upstream distance is (10d) from the main edge of the cooling hole and downstream distance is (20d) behind the main edge of the opening. Height on the flat plate is (6d) and remainder of measurements displayed in figure1

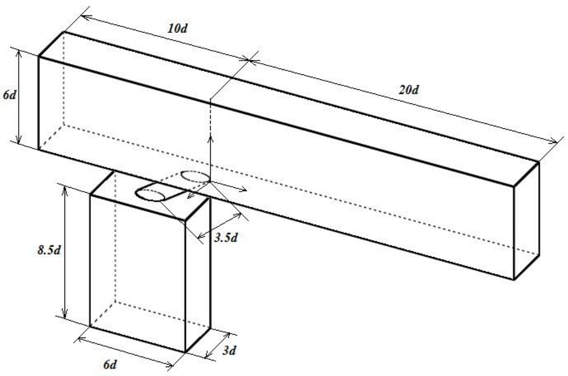


Figure1- Computational field without (RVG)

B. Geometry of RVG

The shaping of film cooling hole and reverse vortex generator (RVG) had several attempts were made to control the cooling air from the cooling hole by RVG proposed in this study. The utilized RVG had elliptic base and front shapes with filet at the root segment. In this examination, there are two groups of cases (RVG positions). The primary group, The stream wise distance from the center of a cooling hole to RVG, the length of the z axis orientations of RVG, the length of the x axis orientations of RVG, and the radius of a fillet were 1.5d, 1.0d, 0.5d, and 0.15d, respectively. As illustrated in figure 2. The height of RVG (H), and the off-set distance (literally position distance) of RVG from the center of a cooling hole (A) are listed in Table1.

Table 1- RVG Parameters for the primary group

Case	H	A
Basic	No RVG	No RVG
Case 1	0.5 *d*	0.0 *d*
Case 2	0.5 *d*	0.25 *d*
Case 3	0.5 *d*	0.5 *d*
Case 4	0.5 *d*	0.75 *d*

In the secondary group, the different only for the stream wise distance which was 1.5d to become 2d and the height of RVG (H) which was 0.5d to become 0.25d. The parameters for second group are listed in Table2.

Table 2- RVG Parameters for the secondary group

Case	H	A
Case 5	0.25 *d*	0.0 *d*
Case 6	0.25 *d*	0.25 *d*
Case 7	0.25 *d*	0.5 *d*
Case 8	0.25 *d*	0.75 *d*

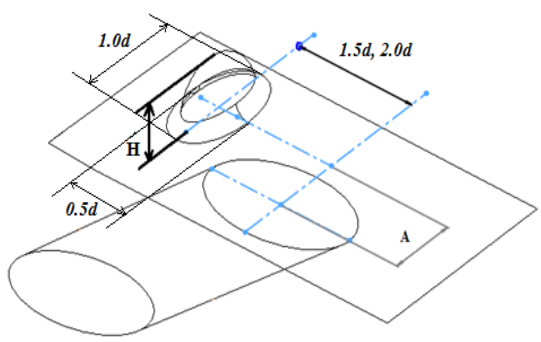


Figure 2-The geometry of reverse vortex generator (RVG)

C. Computational Meshing

The meshing has been carried out hybrid type which is definitely (Tetrahedrous mesh) in this research as illustrated in Figure 3. That means contains the organized and unorganized mesh. The body has constructed unstructured, however the organized mesh has inflation from film cooling surface (wall surface) due to it uses in order to resolve boundary layer at close to film cooling surface. The quantity of nodes and elements which utlized to simulate the air flow are 243390 and 981840 respectively.

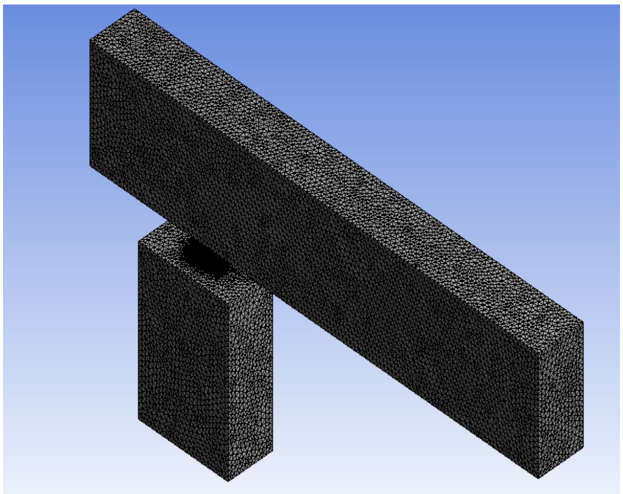


Figure 3- 3D view of the meshing model for all geometry cases using CFX model

D. *limiting conditions*

All tests depended on [8] in the wind tunnel at Reynolds number of 16500 depended on film cooling hole diameter (*d*).

Table 3 illustrates the parameters that given from experimental data.

Table 3- Limiting conditions parameters

Fluid model	**Turbulence model**	**SST**
Reynolds number of cooling air	*Re*	*16,500*
Hot air velocity	U_∞	*13m/s*
Hot air temperature	T_∞	*300 K*
Cool air temperature	T_c	*353 K*
Blowing proportion	*BP*	*0.5-1.0*
Hot air turbulence intensity	Ω_u	*1%*
Cool air jet turbulence intensity	Ω_c	*1%*
Density proportion	*DR*	*0.85*
Outlet pressure	*P*	*0 Pa*

Secondary cooling air velocity can be determined from blowing proportion (BP) equation as follow.

$$(BP = \rho_c U_c / \rho_\infty U_\infty) \quad (1)$$

ρ_c = Cool air density
ρ_∞ = Hot air density
U_∞ = Hot air velocity
U_c = Cool air velocity

III. RESULTS AND DISCUSSION

A. *Inverse vortex pair (IVP)*

Generally, the (IVP) is the main structure that can support the stream of secondary air extremely long in the stream field; it additionally lifts secondary air flow off the protected surface what allows hot air to enter under it.

Figure 4 illustrates the vortex scale distribution, The maximum value of vortex is 50 (s^{-1}) which is a positive value and represented as a blue color and the minimum value of vortex is -50 (s^{-1}) which is a negative value and represented as a purple color.

Figure 5 illustrates the vortices contours on the perpendicular plane at (x/d = 03). The pair of positive and negative vortices was the inverse vortex pair (IVP), and it promoted the secondary air to separate from the wall surface. The vortex pair of (B) was a twins vortex created when the hot air goes through the side and the upper surface of RVG, their rotation was inverse to IVP. In case 01 at BP = 0.5, IVP observed by z/d = 0.0 became large. It is due to IVP and kidney vortex pair (B) interacted each other in this region and the vorticy was emphasized. Concerning case 02, since the area wherein the vortex created from RVG moved from the middle, the IVP became asymmetrical. In case 03 and case 04, the vortex produced from RVG moved to the area as in figure 5 (a) (C) due to this vortex interacted with IVP, the coolant air became attached to the wall surface. From case 04 until case 08, there are scattering and separation of IVP which lets to coolant air to mixes very well with hot air and investigate high laterally coverage and maximum value of cooling efficiency along a wall surface. For high BP at 1.0, the high velocity for coolant air will increase IVP and make it large due to the jet of cooling air so high and does not attach the wall surface thus, the hot air does not influence of coolant air jet.

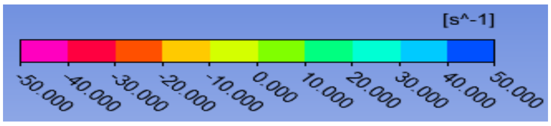
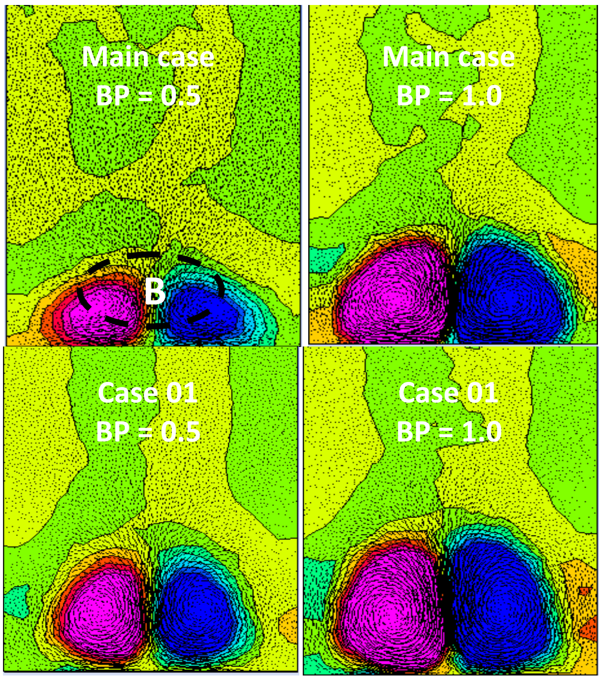
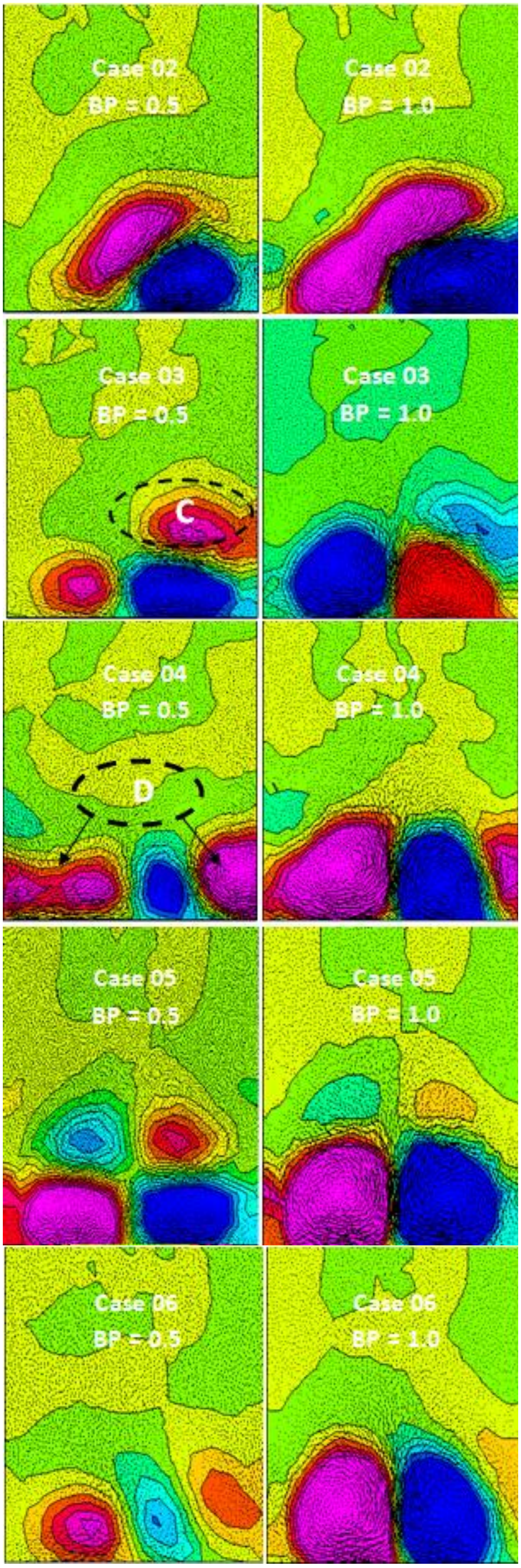


Figure 4- Vortex scale distribution.





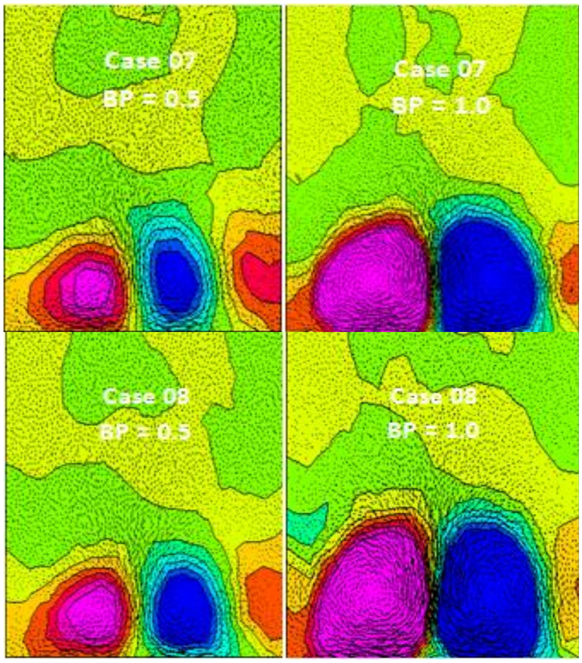


Figure 5- Vorticy in YZ plane at normal plane distance ratio x/d = 03 for all study cases at blowing proportion, BP = 0.5 and (b) BP =1.0

B. Local temperture (Non dimensional temperture)

Figure 6 illustrates non-dimensional temperature scale distribution, the maximum value of this ratio is 0.7 as the result was obtained by simulation and minimum value of this ratio is 0 that means, in this region, there is not interaction between the hot air and cooling air. Mathematically, it could be calculated by this equation (2):

$$\theta = \frac{T_f - T_\infty}{T_2 - T_\infty} \tag{2}$$

Where T_f the local fluid temperature is measured by film cooling temperature distribution, $T_\infty$is the mainstream hot air temperature at main inlet, and T_2 is the cool air temperature

Figure 7 illustrates non-dimensional temperature contours on the normal planes at x/d= 03 to wall surface for different BP obtained by CFD simulation. In main case, the core of nondimensional temperature in region (D) (As see in figure 7) was relatively high and close to the surface of a wall. In case 01, since film cooling efficiency distribustion did not expand too much on the wall surface direction, the nondimensional temperature did not expand on the wall surface as well due to the secondary air was attached the wall surface lightly and mixing of the mainstream (hot air) and the secondary air became strong. The nondimensional temperature was low in this case. In case 02, nondimensional temperature became asymmetrical by the vortex generated from RVG. Therefore, the nondimensional temperature was attached to the wall surface more than case 01 so the spatially averaged film cooling efficiency was higher than case 01. In case 03, the core of the nondimensional temperature attached to the wall surface comparing to case 01 and case 02. In case 04, although the nondimensional temperature attached to wall surface in the wall surface direction as case 03, the core position of nondimesional temperature is relatively high. Therefore, it is considered that the mixed effect of the mainstream (hot air) and secondary air was comparatively low at case 04. In case 05-case 08, the core of nondimensional temperature for all of them were attached to the wall surface, in case 05, case 07 and case 08 showed that there is similarity with main case for core non dimensional temperature so that is will result similar expand for film cooling efficiency distribution on the wall surface direction. However, in case 06 showed that there is similarity with case 04 for core nondimensional temperature. The best cases in BP = 1.0 was in case 03, case 04 and case 06 due to the core of non-dimensional temperature was attached to the wall surface. Since the vortex structure generated from a cooling hole became strong with increase in BP, it shows that the effect of interaction between mainstream (hot air) and the vortex generated from RVG decreased.

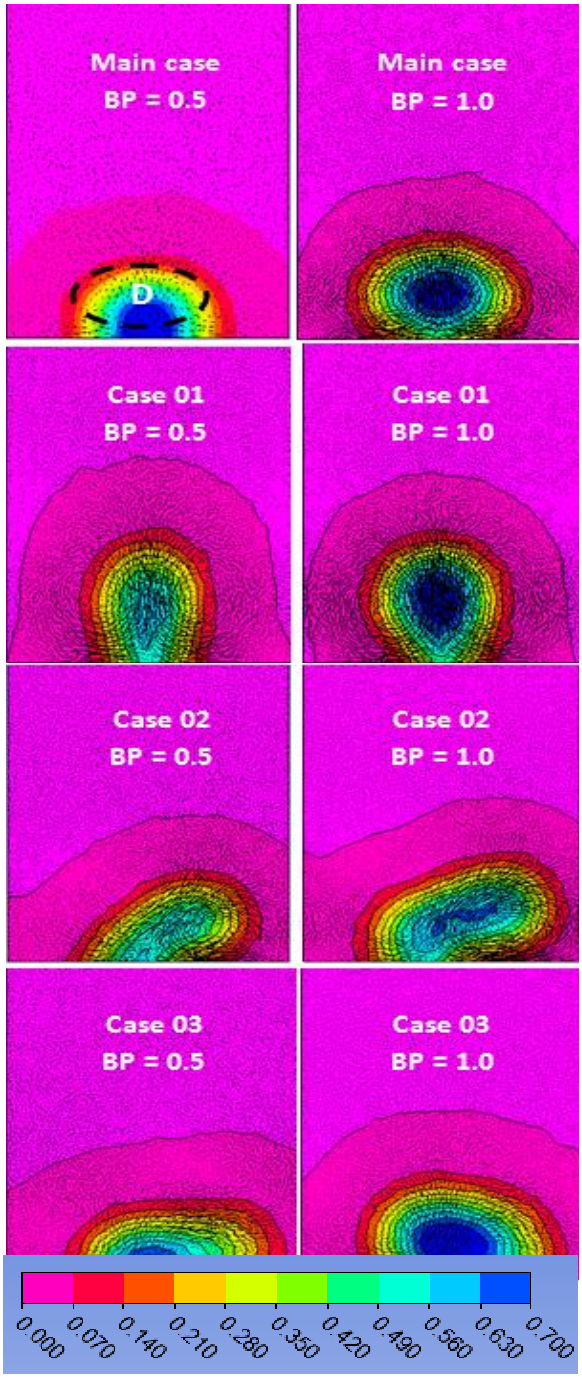


Figure 6 - non-dimensional temperature scale distribution

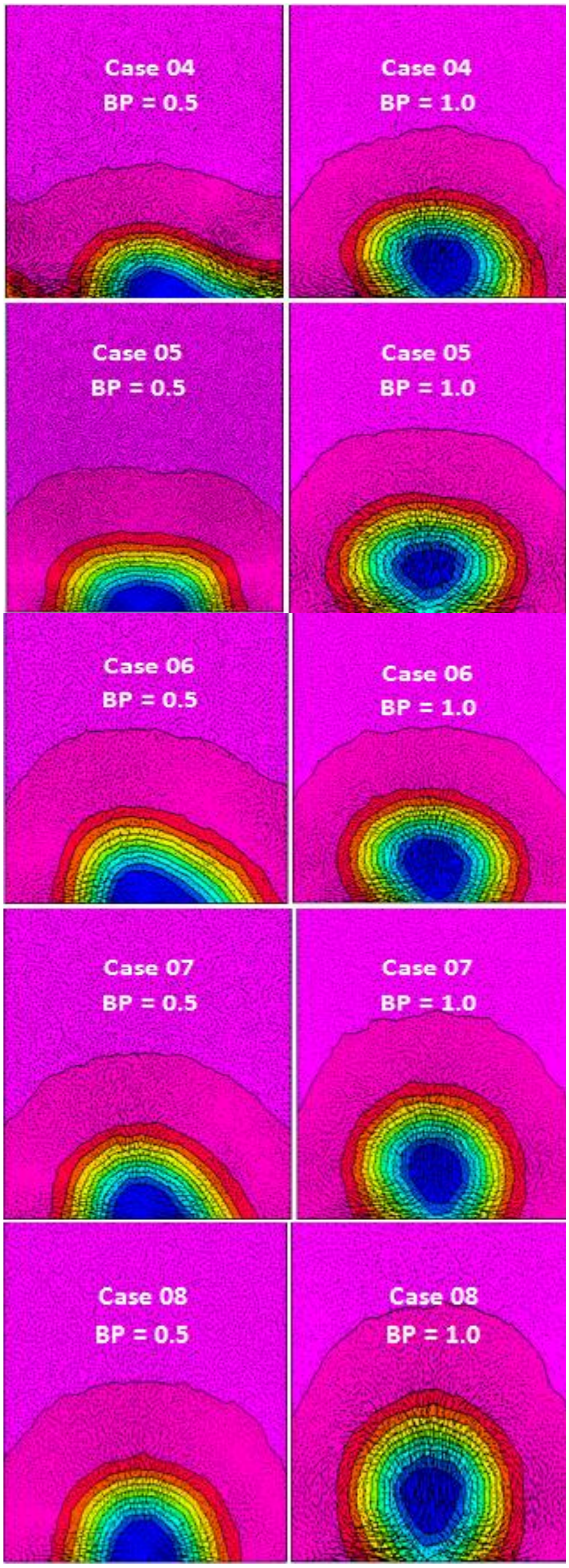


Figure 7- local temperature (non-dimensional temperature) on normal planes to the wall surface at x/d= 03 for all study case at BP = 0.5 and BP = 1.0

C. Film cooling efficiency

The film cooling efficiency is a mixture of the main stream (hot air) and the coolant air where it is responsible to transfer heat to wall surface (Blade surface), hence the film cooling efficiency is defined as in equation (3):

$$\eta = \frac{T_w - T_\infty}{T_c - T_\infty} \quad (3)$$

Where T_∞ is the main air temperature (hot air),
T_c is the coolant temperature and,
T_w is the wall temperature (Blade surface temperature).

Figure 8 illustrates the film cooling efficiency scale distribution (η).

Figure 9 (a) illustrates the distribution of film cooling efficiency on the wall surface at BP = 0.5 obtained by CFD simulation. In main case, the film cooling efficiency expanded symmetrical in mainstream direction on the middle line. In the case 01, the film cooling efficiency did not extend too much in mainstream direction and just the film cooling efficiency on the middle line due to the cooling air became high. In the case 02, the expansion and distribution of film cooling efficiency improved due to of the shifting (RVG) position and the explanation of that the mixing between hot air and cooling air promoted a little bit. On the other side, case 03 and case 04 had wide film cooling efficiency distribution in mainstream direction. The film cooling efficiency distribution was asymmetrical to the middle line due to the cooling air was bent by the vortex structure created from RVG. In the case 05 - case 08 had similar distribution of film cooling efficiency with that of main case except case 06 had similar distribution with case 04. The best cases that have long expansion in mainstream direction were case 05 and case 07. Nevertheless, the local film cooling efficiency in these cases were a little bit high and it was shown that mixing of the hot air and cooling air was promoted in these cases. Figure 9 (b) illustrates the distribution of film cooling efficiency on the wall surface at BP = 1.0. In these conditions, since the velocity in cooling air increased, the film cooling efficiency became low compared with the BP = 0.5 condition. Mostly, the expansion and distribution of film cooling efficiency was less than the cases at BP = 0.5. The best cases that have long expansion of film cooling efficiency were case 03, case 04 and case 06.

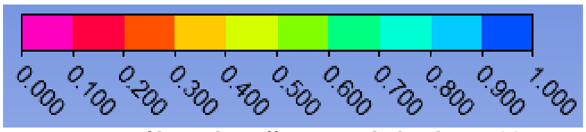


Figure 8- film cooling efficiency scale distribution (η)

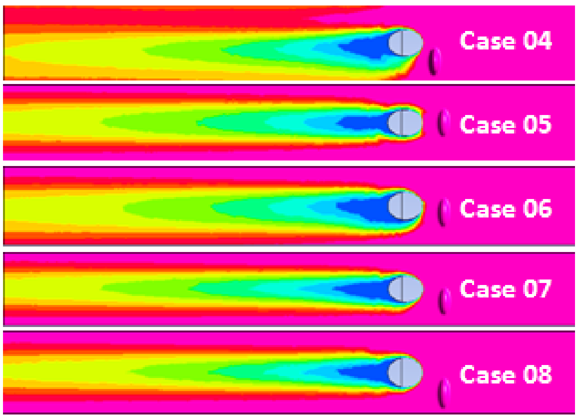


Figure 9 (a)- Distribution of film cooling efficiency on the wall surface at BP = 0.5

Figure 9 (b)- Distribution of film cooling efficiency on the wall surface at BP = 1.0

D. Comparison between all study cases

Figure 10 illustrates the comparison of laterally averaged film cooling efficiency between all cases of study. As found in the figure at BP = 0.5, the best cases of this comparison are case 04 and case 06, as well from figure the averaged film cooling efficiency for main case was higher than case 01 and case 02 that is due to the distribution of film cooling efficiency was symmetrical on the middle line and wide. From the case 03, the film cooling efficiency started increase a little bit to case 04 and then there is little bit decreased of film cooling efficiency for last four cases. Generally, last four cases were better than first four cases except case 04 that is due to the interaction between the hot air and cooling air was promoted and gave symmetrical distribution for film cooling efficiency on the middle line. At BP = 1.0, the case 03 was the best case.

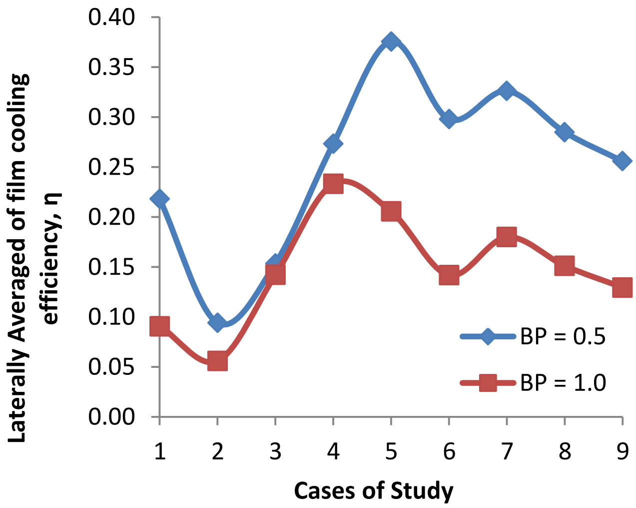


Figure 10- Comparison of laterally averaged film cooling efficiency between all cases of study at different BP

Figure 11 illustrates the comparison of maximum film cooling efficiency between all cases of study at different BP, as found in the figure main case and last four cases were the highest that due to the distribution of film cooling efficiency was symmetrical and expand on the middle line. The case 03 and case 04 were lower than last four cases that due to there are little bit shift of film cooling efficiency distribution on the middle line and coverage more region of area for wall surface.

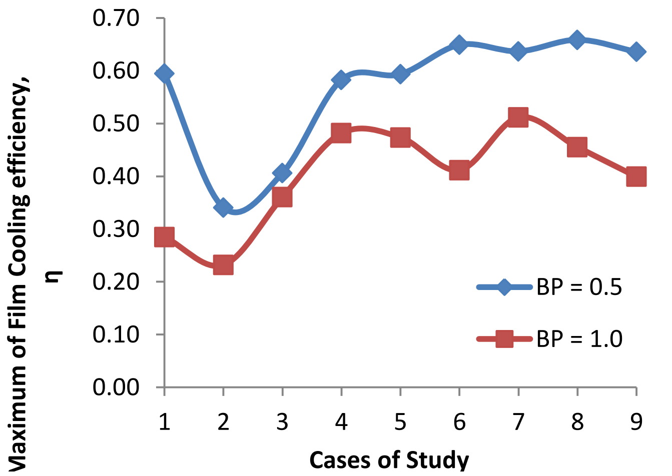


Figure 11- Comparison of Maximum film cooling efficiency between all cases of study at different BP

IV. CONCLUSION

From got results, the changing of (RVG) design improved the distribution of film cooling efficiency as well as increased the expansion of film cooling efficiency. Also, the analaysed results of cases show that in first group design and case 04 had a good coverage on the wall surface. While in second group design, all cases had symmetrical distribution better than the cases in first group design. The best case which gave best cooling efficiency was case 07 and value for film cooling efficiency ($\eta = 0.66$). In general, the best results obtained at blowing proportion (BP = 0.5) which gave best interaction between main stream (hot air) and cool air. While if the blowing proportion increased it will rise the jet cooling air so you have to change the design of cooling hole or angle of inclination to eliminate the rising jet of cooling air.

ACKNOWLEDGMENT

The author wish to thank everyone who give his time to teach me and give me knowledge. I would like to thank the College of Civil Aviation Misurata for their supported.

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