© Applied Science Innovations Pvt. Ltd., India Carbon – Sci. Tech. 8/2(2016)46-54

Carbon – Science and Technology

ISSN 0974 – 0546

http://www.applied-science-innovations.com

RESEARCH ARTICLE Received:10/03/2016, Accepted:15/04/2016

Computational study of turbine blade cooling with various blowing ratios

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Abstract: This paper presents computational analysis of centerline film cooling effectiveness using Navier-Stokes equation solver. Film cooling effectiveness has been varied along the downstream of cooling holes. The computational model has been validated with benchmark experimental literature. Computational study compares film cooling effectiveness over various blowing ratios (M) and various hole shapes. The k-ω shear stress transport model of FLUENT software has been used for the computational analysis. The hole geometry and blowing ratios have important effects on film cooling effectiveness. Computational results reveal that film cooling effectiveness increases with increase in blowing ratio whereas effectiveness decreases due to intermixing of coolant and mainstream flow and due to coolant jet lift off. The best results were obtained for fan-shaped hole with M=1.00. While for lower blowing ratio, coolant is unable to spread over a longer distance downstream of cooling holes.

Keywords: Blowing ratio, Film cooling effectiveness, Fan-shaped hole, Shear stress transport model.

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1 Introduction: Gas turbines operate under very high fluid temperatures that may damage turbine blades. In order to protect these blades, a comprehensive cooling technique known as film cooling has been implemented during blade design. Stability between excessive heating and cooling of the turbine blades has been maintained by designing this optimum cooling technique. Film cooling Effectiveness is a non-dimensional parameter used to characterize the film cooling performance.

Mathematically, film cooling effectiveness is defined as:

$$\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_{c,exit}}$$

where, T_{aw} is adiabatic wall temperature, T_∞ is free-stream temperature = 600 K & $T_{c,exit}$ is coolant inlet temperature = 300 K.

Researchers have performed many experimental and computational works. Few works which are relevant to my work are stated here. Vijay K. Garg and Raymond Gaugler [1] found that for a fixed coolant temperature, η passes through minima on suction side of shower head holes as coolant to mainstream mass flow ratio increases, while on pressure side of shower head holes, the η decreases with increase in coolant mass flow due to coolant jet lift off. Mahfoud Kadja and George Bergelest [2] used 2D numerical model and found that effective blade surface cooling is possible when M is increased but at the expense of reduced cycle efficiency. D. Lakehal et al. [3] found that resolving the viscosity affected near wall region with 1-equation turbulence model in a 2-layer approach increases the vortex strength and lateral spreading but reduces vertical spreading. In standard k-ε model with wall functions they found many complex features of flow like asymmetric behavior due to injection induced secondary flow vortex on left side and its absence on right side of the jet and initially strong and then slower decay of temperature in the core of the jet. Vijay K. Garg and David L. Rigby **[4]** found on blade side that coolant do not follow $1/7^{th}$ Power law profile and are highly skewed by mainstream flow for the holes. Vijay K. Garg **[6]** used Wilcox's k-ω model and found that coolant velocity and temperature distribution at hole exits do not follow $1/7^{th}$ Power Law and are highly skewed by mainstream flow. Distribution of k and ω at exit of holes are also skewed. Jr-Ming Miao and Chen-Yaun Wu **[8]** found that thermal flow field over film cooled test plate is dominated by strength of counter rotating vortex pairs (CRVP) due to interaction of coolant jet and mainstream. Structure of laterally diffused simple angle hole increases the lateral spread of cooling flow, thus improves the span wise-averaged film cooled η as B.R increases from 0.3 to 1.5. M.B. Jovanovic´ et. al. **[9]** observed that the imperfection placed 1diameter from the hole leading edge, deteriorates the η at moderate velocity ratios (around 0.5) whereas under same condition, same imperfection fixed at the hole exit improves the η. At velocity ratio 1.5, exit imperfection improves η. Turbulence intensity and imperfections placed deeper in the hole do not have a significant influence on film cooling η. Deepak Raj P.Y and Prof. Devaraj. K **[10]** performed computational work and determined that as blowing ratio increases, the local stream-wise effectiveness increases and causes coolant jet lift off. They also found that boundary layer developed in the wall jet region is removed due to suction created by film cooling holes and thus increasing heat transfer. Reattachment of coolant in the flow structure increases the effectiveness near the leading edge of the holes.

Thus, it is concluded from the literatures that film cooling effectiveness depend on various parameters such as blade rotation, temperature, location of hole, free stream turbulence, momentum flux ratio, wall heat conductivity, compressibility of coolant, density ratio, Reynolds number, etc. Therefore, the aim of this paper is to analyze centreline film cooling effectiveness using the SST k-ω turbulence model in FLUENT software. For simulation purpose, blade surfaces are considered as flat plates modelled in GAMBIT. Comparison have been made on various blowing ratios (M = 0.33, 0.50, 0.60, 0.83 and 1.00) for different hole geometries.

2. Computational Modelling:

2.1 Geometry: Figure (1) is the schematic side view of the computational model which has been made in GAMBIT for circular shaped hole. The coolant inlet areas of each shape that are circular, square and fan-shaped have been kept constant in order to have equal mass flow rate through each shape. The exit area is thrice the entry area for fan-shaped hole which has been determined from the work of Michael Gritsch et al. **[5]** and hence fan-shaped holes are categorized as expanded exits.

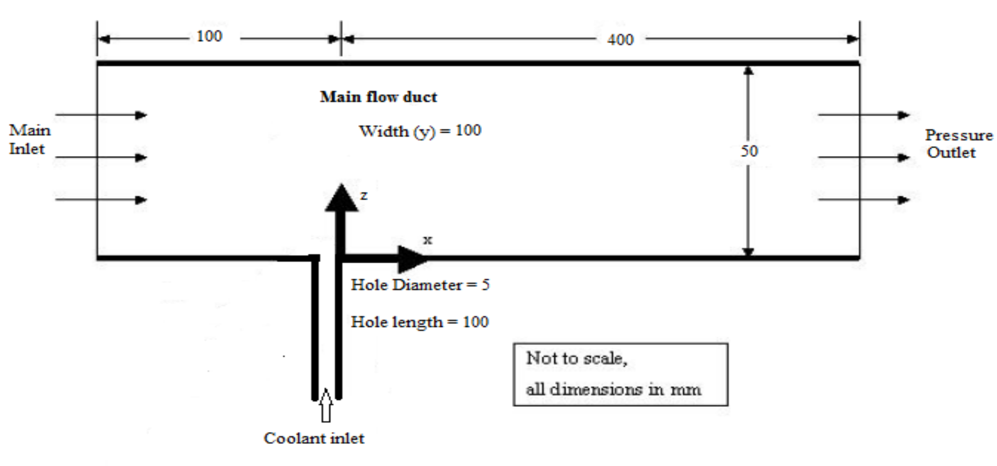

Figure (1): Computational Geometry.

2.2 Boundary Conditions: A velocity-inlet boundary condition has been specified with x-velocity equal to 25 m/s at main inlet, which has been maintained constant for all the blowing ratios. The mainstream and coolant temperatures are 600K and 300K respectively. The injection angle of coolant has been kept 90^0 with respect to the lower horizontal plate. Table (1) show the corresponding coolant velocity used according to different blowing ratios. Blowing ratio is the ratio of coolant to mainstream mass flux.

Table (1): Blowing Ratios and Coolant inlet velocities:

Sl. No.	Blowing Ratios (M)	Coolant Velocities (m/s)
01.	0.33	08.25
02.	0.50	12.50
03.	0.60	15.00
04.	0.83	20.75
05.	1.00	25.00

2.3 Grid Dependency: The meshes were created in Gambit version 2.1.2. Figure (2) depict mesh of a square hole case with 32,50,000 hexahedral cells made using copper scheme. Mesh for circular and fan-shaped holes consist of 28,38,225 and 25,65,000 hexahedral cells respectively. The grid dependency test have been carried out for M = 0.33. From Figure (3), it is concluded that the medium grid gave results which are almost similar to published experimental results. Therefore, grid size almost same as medium grid size have been used for further computational analysis.

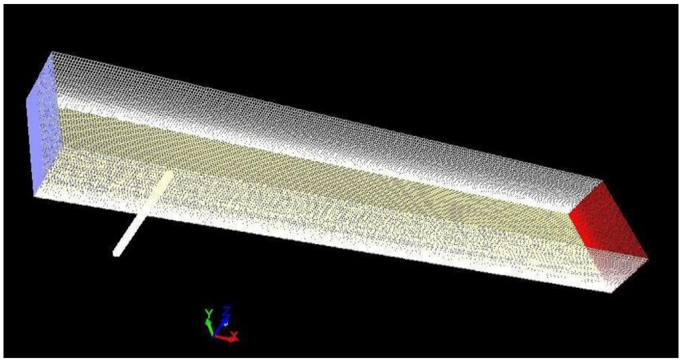

Figure (2): Meshed Geometry.

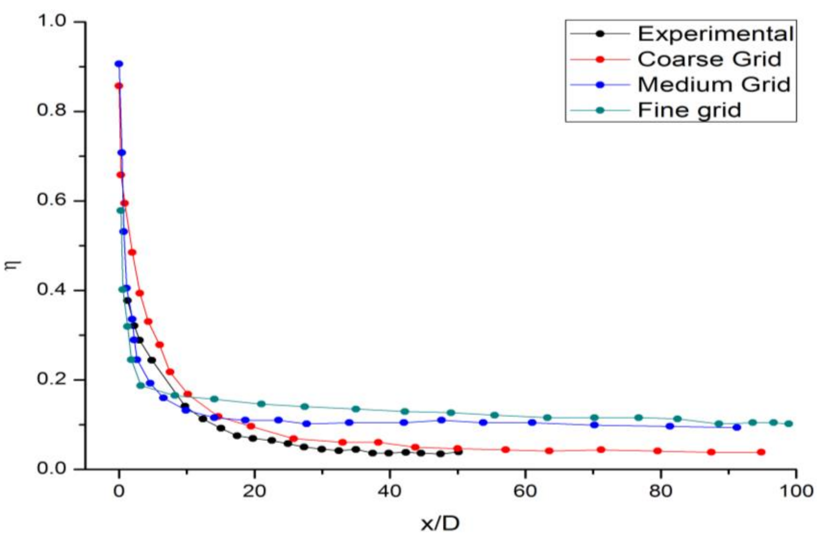

Figure (3): Grid Dependency Test.

2.4 Solver: The k-ω SST model is directly usable at the wall through the viscous sub-layer. This model can be used as a Low-Re turbulence model without any damping functions. Thus, for simulation purpose, the RANS shear stress transport model has been used in FLUENT 14. The SST formulation switches to k-ε behavior in the main free-stream flow structure and by this means it avoids the common k-ω problem which is model being too sensitive to the turbulence properties of main inlet free-stream. The SST k-ω model exhibit better performance in adverse pressure gradients as well as in separating flow structure. This model produces large turbulence levels in regions with large normal strain, like stagnation regions and regions with strong acceleration. The first transported variable is turbulent kinetic energy, k which determines the energy in the turbulence. The second transported variable is the specific dissipation, ω which determines the scale of the turbulence.

3. Results and discussion:

3.1 Validation: The computational model has been validated with benchmark experimental literature [7] for a circular hole case at M = 0.60 as shown in Fig. 4. The computational results are almost similar to the experimental results obtained by C.H.N. Yuen, and R.F. Martinez-Botas who studied film cooling effectiveness experimentally on a cylindrical hole with a stream wise angle of 30^0, 60^0 and 90^0.

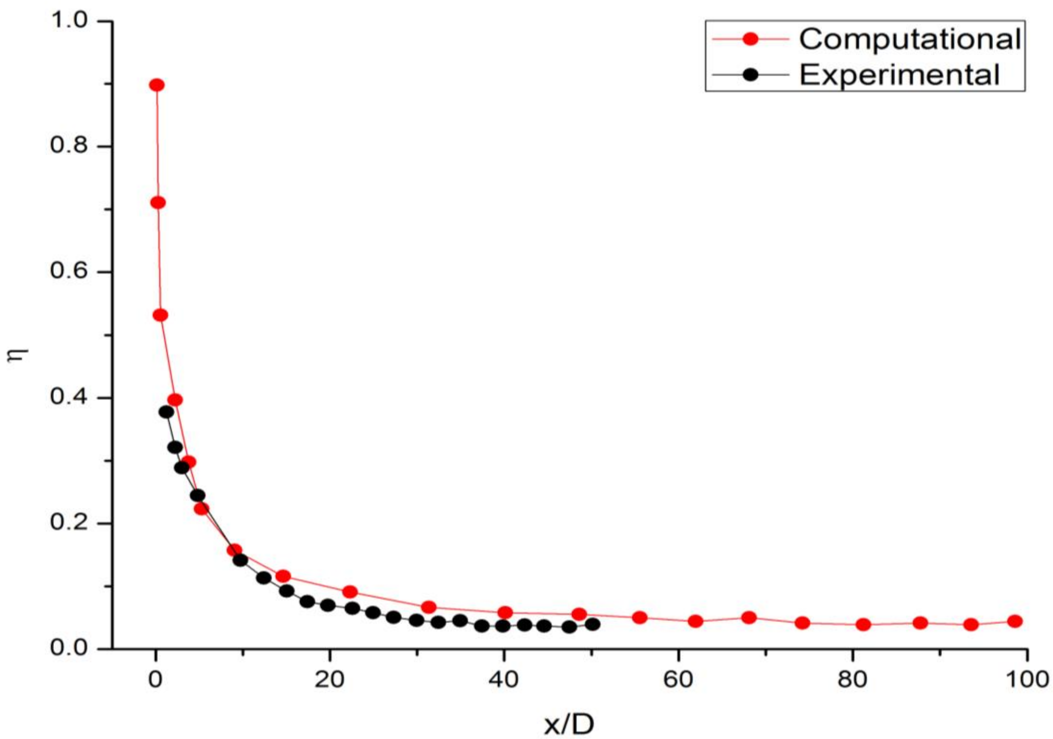

Figure (4): η for circular hole at M = 0.60

3.2 Hole shapes at different M: Figure (5), (6), (7), (8) and (9) show comparison of centerline effectiveness for single circular hole shape, square hole shape and fan-shaped hole at an axial distance of 100 mm from the trailing edge of the blade for M = 0.33, 0.50, 0.60, 0.83 and 1.00 respectively. As blowing ratio increases from 0.33 to 1.00, film cooling effectiveness also increases at coolant exit but it reduces further downstream of circular and square shaped holes due to rapid increase in intermixing of coolant and mainstream flow. Effectiveness at coolant exit for square hole case is less than circular hole due to presence of lateral separation of kidney vortices. Hence, at square hole exit coolant spread more laterally than longitudinally whereas in circular hole coolant could not spread laterally and cover narrow area downstream of the holes.

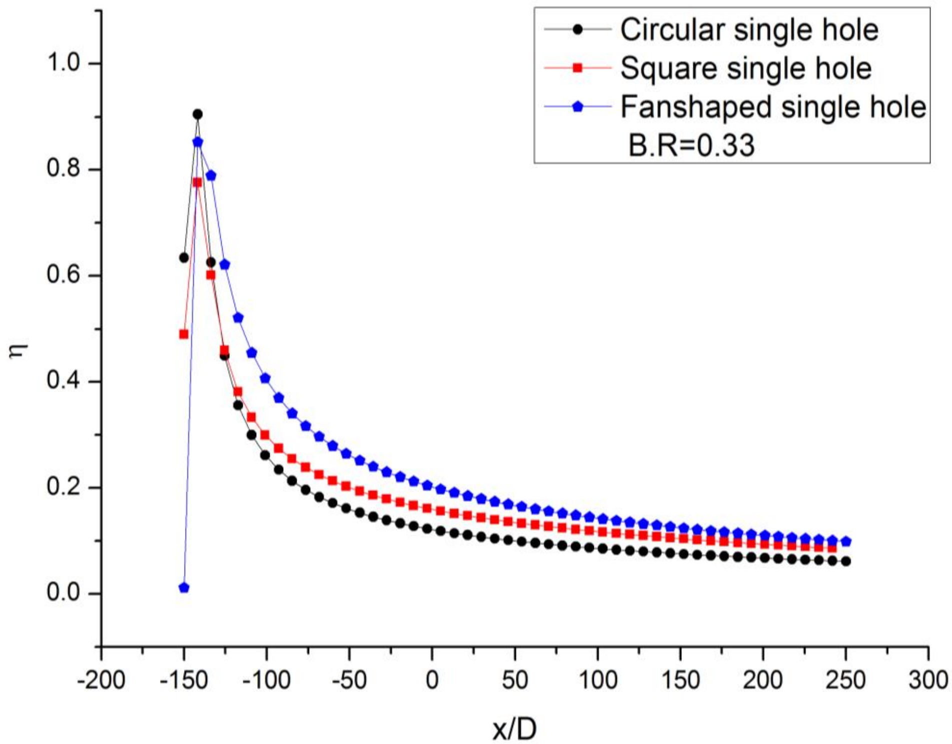

Figure (5): η for various shapes at M = 0.33

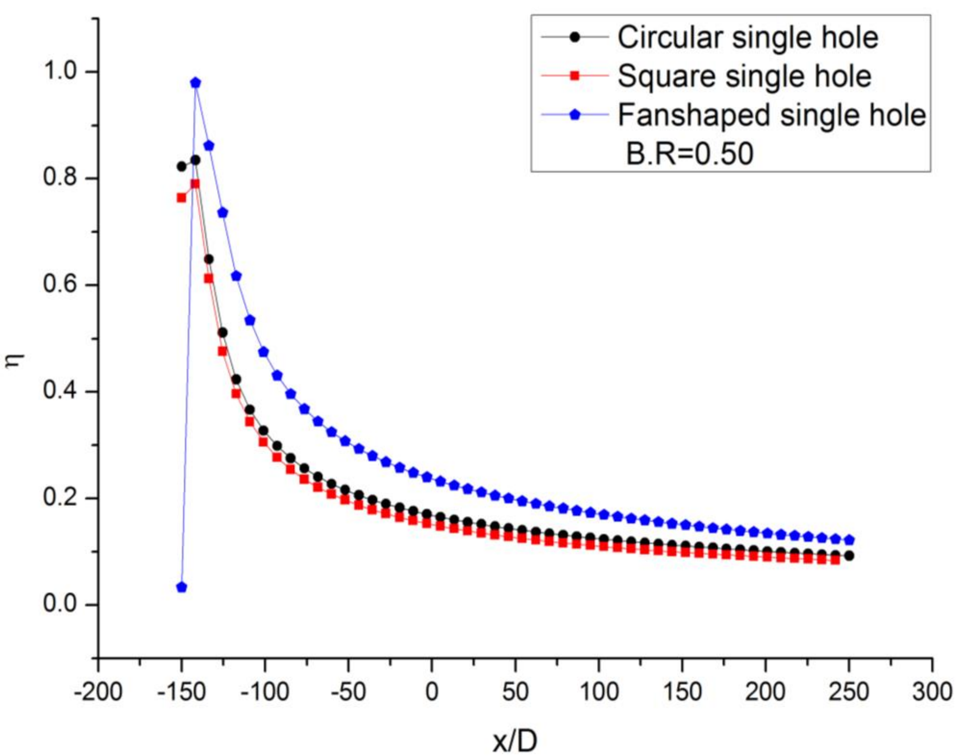

Figure (6): η for various shapes at M = 0.50

In fan-shaped hole, film cooling effectiveness is minimum at hole exit due to coolant jet lift off and thus main stream enters between coolant and wall leading to the fall of effectiveness. Soon coolant jet reattachment takes place and effectiveness almost approaches to unity. As blowing ratio increases from

0.33 to 1.00, centreline effectiveness also increases at coolant exit as well as further downstream of cooling hole.

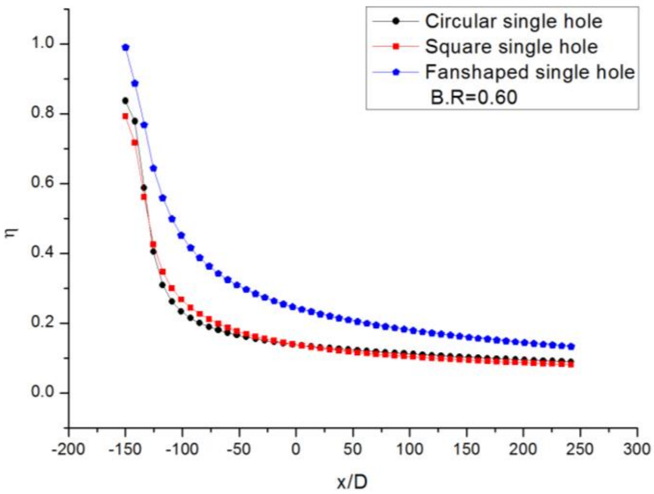

Figure (7): η for various shapes at M = 0.60

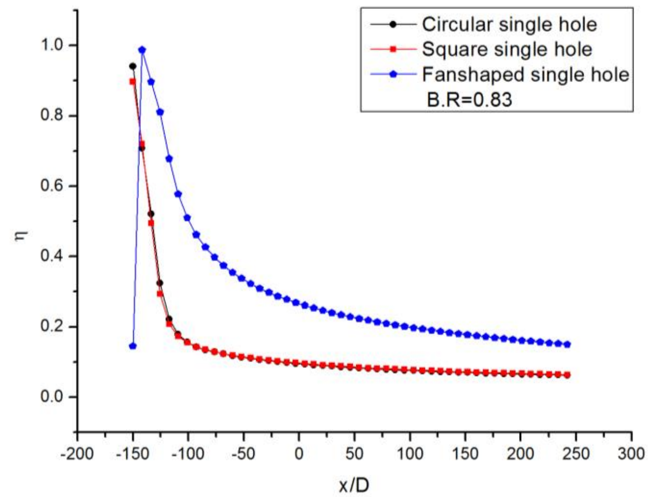

Figure (8): η for various shapes at M = 0.83

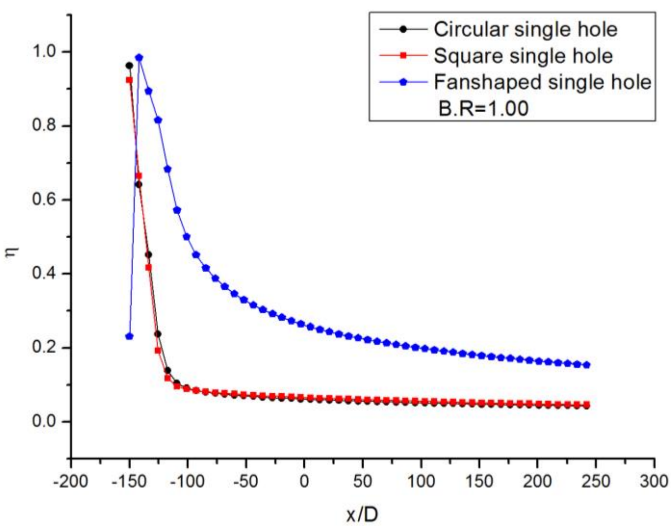

Figure (9): η for various shapes at M = 1.00

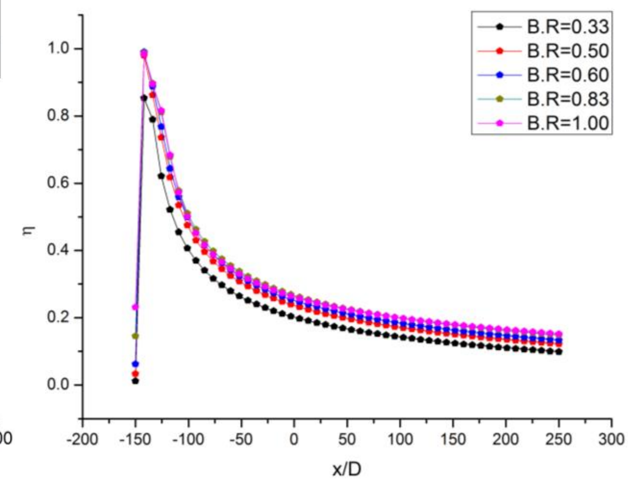

Figure (10): η for fan-shaped holes at different M

Out of all shapes, maximum effectiveness is achieved for fan-shaped hole for all blowing ratios. It is because of expended exits (exit area is more compared to other hole shapes) in fan-shaped holes. Therefore, it is possible to have higher effectiveness with constant coolant mass flow rate.

Figure (10) show comparison of centreline effectiveness for fan-shaped hole at M = 0.33, 0.50, 0.60, 0.83 and 1.00 respectively. Result shows that as blowing ratio increases effectiveness also increases for fan-shaped hole. Centreline effectiveness is maximum for M = 1.00. While centreline effectiveness decreases with increase in blowing ratios for circular and square hole shape due to increase in intermixing of coolant and mainstream. It also decreases because coolant spreads more laterally further downstream of cooling holes. But it is also visible from the results that the centreline effectiveness does not reach zero till the end of the plate and effectiveness is same for both circular and square hole at the end of the blade.

Figure (11) show comparison of centreline temperature curve for fan-shaped hole at M = 0.33, 0.50, 0.60, 0.83 and 1.00 respectively. Temperature curve is synonymous to the result of centreline effectiveness as effectiveness is inversely related to coolant temperature. Result reveals that cooling is effective till the end of the plate as temperature does not reach 600K and temperature corresponding to M = 1 is lower compared to other blowing ratios.

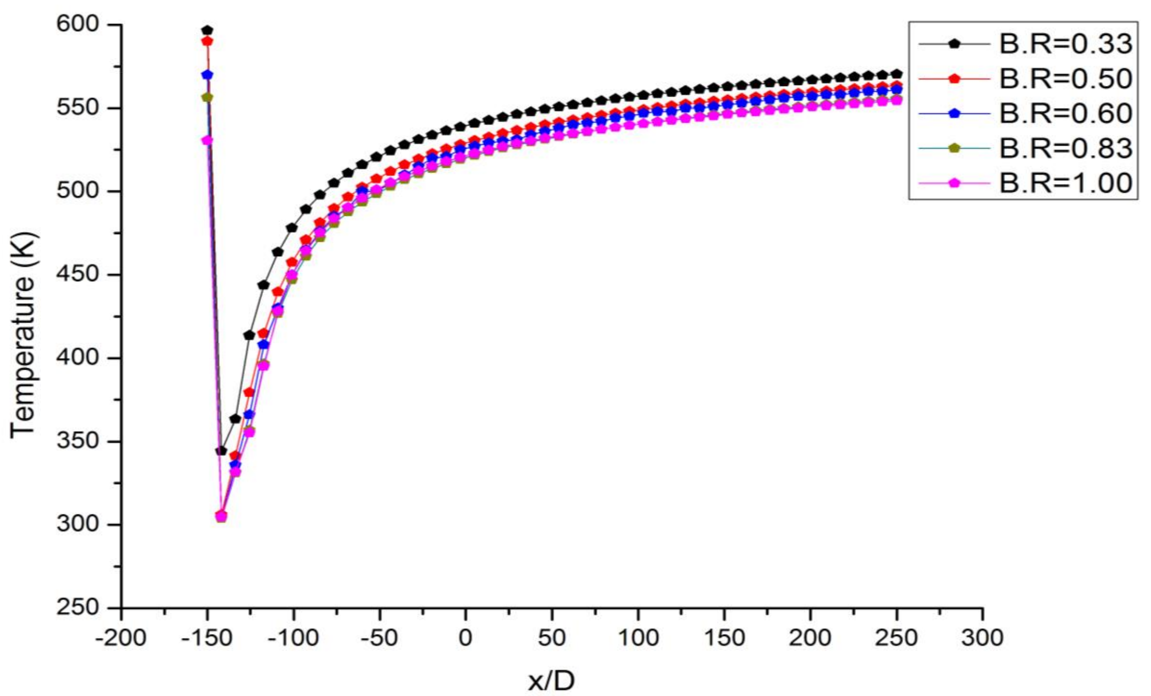

Figure (11): Temperature curve for fan-shaped holes at different M.

3.3 Temperature and velocity contours: Figure (12) represent temperature contour on lower plate for fan-shaped hole at M = 1.00. It shows that coolant also flow behind the hole and cool a small area. Near the hole surface, temperature almost reaches 300K.

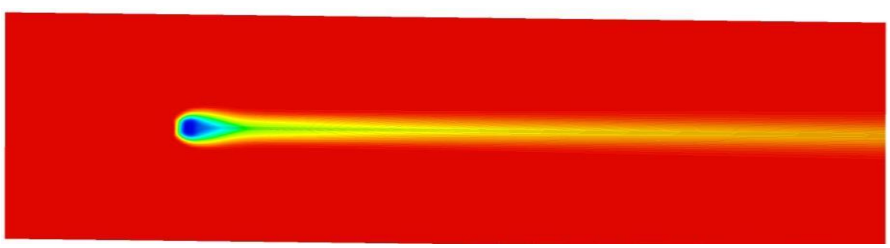

Figure (12): Temperature contour for fan-shaped hole at M = 1.00

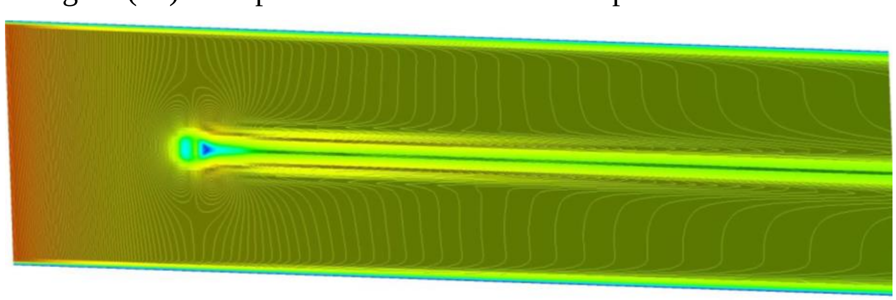

Figure (13): Velocity contour for fan-shaped hole at M = 1.00

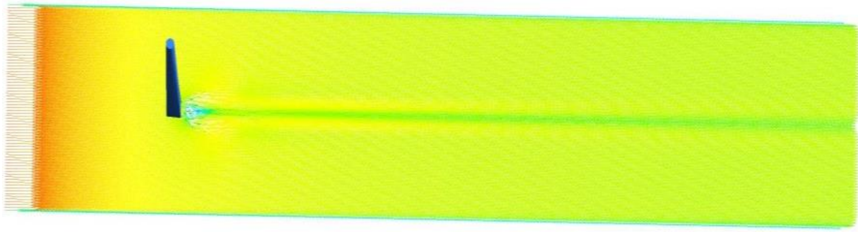

Figure (14): Velocity vector for fan-shaped hole at M = 1.00

Figure (13) and (14) represent velocity contour and velocity vector on lower plate for fan-shaped hole at M = 1.00. Near the hole, there is formation of recirculation zone which spreads laterally and does not affect the path of coolant flow downstream of the holes. As it is visible from the result that coolant flow path is smooth.

4 Conclusions: According to shapes, fan-shaped hole gave the best result. On comparing centerline effectiveness for single hole, it is observed that effectiveness is very less at the coolant exit for single square hole and fan-shaped hole compared to circular hole due to coolant jet lift off but after that effectiveness increase rapidly for fan-shaped hole compared to other two hole shapes because of coolant jet reattachment. Effectiveness decrease with increase in blowing ratios for circular and square hole shapes because large proportion of coolant mix with mainstream flow rapidly. Additionally, effectiveness increase with increase in blowing ratio as coolant spread more laterally as well as longitudinally. While for lower blowing ratio, coolant is unable to spread over a longer distance downstream of cooling holes. Thus, best results were obtained for fan-shaped hole with M = 1. Effectiveness of square is less in longitudinal direction compared to circular because of the presence of lateral separation of kidney vortices just downstream of the coolant hole exit. Coolant from square hole spread more laterally than longitudinally. From temperature curve result, it is visible that cooling is effective till the end of the flat plate that is 400mm downstream of the cooling holes as temperature does not reach 600K. Small amount of coolant flows behind the holes, thus cooling a smaller area. Recirculation zone near the hole spreads laterally and does not disturb the coolant flow path.

Acknowledgement: I take this auspicious opportunity to express my deep sense of gratitude to my guide Dr. A.K Roy, Associate Professor and my co-guide Mr. Prakhar Jindal, PhD. Scholar, Dept. of Mechanical Engineering, B.I.T. Mesra, Ranchi, for their pioneer guidance and innovative ideas and spending their precious time in scrutinizing my work at every stage, and this work would not see the light of completion without their continuous encouragement. I convey my deep sense of gratitude to Dr. Arbind Kumar, Head, Dept. of Mechanical Engineering, B.I.T, Mesra, Ranchi, for providing me with the requisite facilities and facilities extended by him while carrying out this work.

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