

Enhancing the effectiveness of film cooling

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Abstract: Advanced gas turbine stages are designed to operate at increasingly higher inlet temperatures to increase thermal efficiency and specific power output. To maintain durability and reasonable life, film cooling is needed in addition to internal cooling, especially for the first stage. Film cooling lowers material temperature by forced convection inside film cooling holes and by forming a layer of coolant about component surfaces to insulate them from the hot gases. Unfortunately, each cooling jet forms a pair of counter rotating vortices that entrains hot gas and causes the film cooling jet to lift off from the surface that it is intended to protect.

This paper gives an overview of efforts to enhance the effectiveness of film cooling. This paper also describes two new design concepts. One design concept seeks to minimize the entrainment of hot gases underneath of film cooling jets by using flow aligned blockers. The other design concept shifts the interaction between the approaching hot gas and the cooling jet to occur further above the surface by using an upstream ramp. For both design concepts, computational fluid dynamics results are presented to examine their usefulness in enhancing film cooling effectiveness.

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Nomenclature

a, b, c	spacing between blockers, height of blocker, and thickness of blocker
D	film cooling hole diameter
h	heat transfer coefficient
M	blowing ratio $= (\rho U_c) / (\rho U_\infty)$
p	static pressure
q_w	surface heat flux
T, T_c, T_∞	temperature, coolant inlet temp., hot gas temp.
T_w, T_{aw}	wall temperature, adiabatic wall temperature
U_c	average speed of coolant flow in hole
U_∞	speed of hot gas at freestream
u_τ	friction velocity: $(\tau_w / \rho)^{0.5}$
X, Y, Z	coordinate system defined in Fig. 2
y^+	normalized distance from the wall: $\rho u_\tau y / \mu$
y	normal distance from the wall
α, β	angle of ramp, distance to hole (Fig. 2)
η	adiabatic effectiveness/ normalized temperature: $(T_\infty - T) / (T_\infty - T_c)$
$\bar{\eta}$	laterally averaged adiabatic effectiveness (i. e., averaged along the spanwise direction)

μ	dynamic viscosity
ρ	density
τ_w	wall shear stress

Introduction

Gas turbines are widely used for both aircraft propulsion and electric power generation. One effective way to increase efficiency and specific power output is to raise the temperature of the gas entering the turbine component, which can be as high as the adiabatic flame temperature. Today, the desired inlet temperatures are about 2150 K for aircraft propulsion and 1850 K for electric power generation. Though these temperatures are still considerably lower than the adiabatic flame temperature, the best super alloys can handle at most 1325 K to 1450 K and thermal-barrier coatings can enhance that temperature at most by another 140 to 170 K. Thus, even the best materials available will suffer severe reductions in structural integrity through creep and thermal fatigue if they are not cooled.

Thus, cooling is needed for all components whose surfaces come in contact with the hot gases if a reasonable service life is to be achieved^[1,2].

Two widely used methods for cooling turbines are internal and film cooling^[1-3]. With internal cooling, lower temperature air, referred to as the coolant, is circulated inside each airfoil through a series of straight ducts and 180-degree bends. The coolant typically enters the internal passage inside each airfoil from the hub, and exits from its tip and/or trailing edge. Internal cooling lowers material temperature by forced convection in the passages. To enhance convection heat transfer without unduly increasing pressure loss, the passage walls are often roughened by ribs in the main body region of the airfoil and by pin fins in the trailing-edge region. Jet impingement cooling is often used to further enhance heat transfer in the leading-edge region. With film cooling, the coolant, introduced inside each airfoil for internal cooling, is forced to exit the airfoil through judiciously placed holes with the exiting coolant jets designed to adhere to the airfoil surface as much as possible with minimal disruption to the airfoil aerodynamics. Film cooling lowers material temperature by forced convection inside film-cooling holes and by forming a layer of coolant at the component external surfaces to insulate them from the hot gases.

For aircraft gas turbines, the coolant is extracted from the compressor. For power generation turbines, a designer has more flexibility in the source of the coolant because weight is a less important design constraint^[1]. Whether the coolant is supplied by the compressor or some other means, work input is needed. Thus, efficiency considerations demand that acceptable temperature distributions in the turbine material be achieved by using the minimal amount of coolant flow, while enabling a higher turbine inlet temperature.

There are two paths that can be taken to minimize the coolant flow. One is through innovation by devising more effective cooling strate-

gies. The other is through refinement of existing design paradigms. To be successful at either requires a good understanding of how design and operating parameters affect fluid flow and their impact on the cooling process. There are two reasons for this. The first is that current cooling strategies are already highly developed so that further improvements can be made only through an improved understanding of the underlying flow and heat transfer physics. The second is that as the minimum coolant flow is approached, the slightest increase in material temperature from insufficient cooling at any location within the turbine can reduce the turbine's service life by a factor of two or more.

The purpose of this paper is twofold. The first is to give a brief overview of film cooling and efforts to enhance the effectiveness of film cooling jets. The second is to describe two new film cooling design concepts. One design seeks to minimize the entrainment of hot gases underneath of film cooling jets by using flow-aligned blockers. The other design shifts the interaction between the approaching hot gas and the cooling jet to occur further above the surface by using an upstream ramp.

1 Overview on film cooling

For advanced gas turbines, the first-stage stator and rotor typically requires film cooling, which strives to form a blanket of cooler air next to the material surface to insulate the material from the hot gas^[3-6]. Many investigators have studied the effects of design and operating parameters on film cooling. These include film-cooling hole inclinations and length-to-diameter ratios, spacing between holes, geometry of holes including shaped holes, surface curvatures, free stream turbulence, embedded vortices from secondary flows, and unsteadiness from rotor-stator interactions. Excellent reviews on film cooling have been provided by Han, et al.^[7], Goldstein^[8], and Sundén and Faghri^[9]. In addition, a comprehensive bibliography of all work in film cooling have been provided by Kercher^[10,11].

Of the previous studies, Kelso and Lim^[12] and Haven, et al.^[13] showed the important role played by vortices in the evolution of film-cooling jets. One pair, referred to as the counter-rotating vortices (CRVs), was found to lift the jet off the surface that it is intended to protect and to entrain hot gases underneath it. The other pair, referred to as an anti-kidney pair, was shown to have a sense of rotation opposite to that of the CRVs, and so can counteract the undesirable tendencies of the CRVs. Thus, it is of interest to develop strategies to control the formation and strength of these vortices in a way that leads to more effective film cooling.

The importance of this problem to turbine cooling has led many investigators to study it. The goal has been to come up with design concepts to improve film-cooling effectiveness. One design paradigm is to modify the interaction between film-cooling jets. In this paradigm, there are two well-known designs. One is to use two staggered rows of film-cooling holes with each row being perpendicular to the freestream flow direction. With two rows, the film-cooling jets from the second row of holes entrain the cooler gas ejected from the first row of holes. The other is to use compound-angle holes, where film-cooling jets are injected at an angle that is not parallel to the freestream flow direction and can overlap each other. Both of these design concepts are widely used^[7,9].

Since the vorticity in the film-cooling jet originates from the flow in the film-cooling hole, the boundary layer upstream of the film-cooling hole, and the boundary layer/film-cooling jet interactions, other design paradigms have focused on modifying either the geometry of the film-cooling hole or the geometry just upstream or downstream of the film-cooling hole. On film-cooling-hole modification, the most straight forward design is to use a thin slot instead of one row of holes, so that that hot gas entrainment is confined to a much small region at the ends of the slot. However, the length of the slots cannot be too long in order to maintain structural integ-

ity. If shorter slots are used, then one could the two staggered rows of slots. As an alternative, shaped diffusion holes were developed, which uses the Coanda effect to cause the film-cooling jet to hug the curved surface and an expanding cross-sectional area near the hole exit to decrease cooling-jet speed and to increase lateral spreading. Haven, et al.^[13] and Hyams, et al.^[14] investigated the effects of shape holes on the vorticity dynamics of mainflow/film-cooling jet interactions. Instead of changing the geometry of film-cooling holes, Haven and Kurosaka^[15] investigated the effects of placing vanes inside holes that produce vortices in the same sense as the anti-kidney vortices. Zaman and Foss^[16] and Zaman^[17] investigated the effects of tabs placed at the film-cooling-hole exit. Shih, et al.^[18] proposed placing a strut or obstruction within each film-cooling hole that do not necessarily generate appreciable vorticity but can cause vortices inside film-cooling holes to be stretched and tilted in a way that would redistribute the magnitude of the vortices inside the CRVs and the anti-kidney pair. Bunker^[19] proposed creating a trench about each film-cooling hole exit to modify the boundary layer/cooling jet interactions. Na and Shih^[20] proposed modifying the geometry upstream of the film-cooling hole to improve film-cooling effectiveness. They suggested placing a ramp upstream of the film-cooling holes to modify the interaction between the approaching boundary layer flow and the film-cooling jet. Instead of modifying the vortex structure of film-cooling jets, Shih, et al.^[21] proposed flow-aligned blockers to ensure that the entrained gas is from the cooling jet itself instead of the freestream hot gas.

Still another design paradigm is to have no jets, but rather cooling flow oozing from many holes so that the entire surface is blanketed by a thin layer of coolant. This method, known as transpiration cooling, can be quite effective if the small film-cooling holes do not plug up by particles in the flow.

2 New design concepts for film cooling

In this section, the recently developed de-

sign concepts to improve film-cooling effectiveness the upstream ramp^[20] and flow-aligned blockers^[21] are described. The organization is as follows. First, the baseline problem without the ramp and blockers are described. Then, the ramp and the blocker design concepts are introduced. This is then followed by a description of the governing equations, the numerical method of solution, and the computed results.

2.1 Baseline problem

The ramp and blocker design concepts are presented in the framework of film cooling of a flat plate through one row of circular holes. This baseline problem, shown in Fig. 1, without the upstream ramp and without the flow-aligned blockers, is taken from the well-known benchmark study by Kohli and Bogard^[22] for which experimental data is available to validate computational studies. As shown in Fig. 1, the flat plate is cooled by film-cooling jets that emerge from a plenum through one row of circular holes. Each hole has a diameter D of 12.7 mm, a length of $3.5D$, and an inclination angle of 35° relative to a plane tangent to the flat plate. The hot gas and the coolant for film cooling are both taken to be air. The flow of the hot gas above the flat plate has a freestream temperature T_∞ of 298 K and a freestream velocity U_∞ of 20 m/s along the X -direction. The flow in the boundary-layer is assumed to be turbulent from the leading edge of the flat plate. The coolant has a temperature T_c of 188 K in the plenum. This gives a density ratio of 1.6. When the average velocity of the coolant in the film-cooling holes U_c is 6.25 m/s, the mass flux or blowing ratio M is 0.5. Other blowing ratios were also studied by varying the velocity at the inflow of the plenum that feeds the film cooling holes as described in the next subsection.

Two types of boundary conditions were applied on the flat plate for the heat transfer study. When the film-cooling adiabatic effectiveness is sought, the flat plate is made adiabatic. When surface heat transfer is sought, the flat plate is

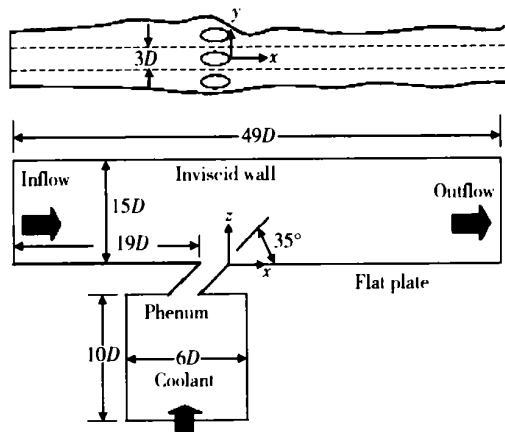


Fig. 1 Schematic of baseline problem: film cooling of a flat plate from a row of inclined circular holes (not drawn to scale).

maintained at a constant wall temperature T_w of 243 K. All other walls, including the walls of the film-cooling holes and the plenum, are made adiabatic. The back pressure at the outflow boundary above the flat plate is maintained at the standard atmospheric pressure.

For this problem, the computational domain is taken to be the region bounded by the solid and the dashed lines, where the dashed lines represent periodic planes (Figs. 1). Periodicity is assumed in the spanwise direction so that only the flow through one film-cooling hole needs to be examined. In addition, the “upper channel wall” (i.e., the wall without film-cooling holes) was moved closer to the wall with the film-cooling holes. This was done to reduce the size of the computational domain and hence computational cost. The errors incurred by this are minimized by making the “upper channel wall” sufficiently far away and by making it inviscid (i.e., the velocity there can slip despite the viscous nature of the flow) so that boundary layers will not form there.

2.2 Upstream ramp and flow-aligned blockers

The upstream-ramp design concept seeks to improve film-cooling effectiveness by modifying the interaction between the approaching boundary layer flow and the emerging film-cooling jet. The upstream ramp studied by Na and Shih^[20] is shown in Fig. 2. It has a length of $2D$, makes an

angle α with respect to the flat plate, and is located β upstream of the film-cooling hole. Na and Shih examined the following values of α and β : $8^\circ, 10^\circ$, and 14° for α and $0.5D$ and D for β .

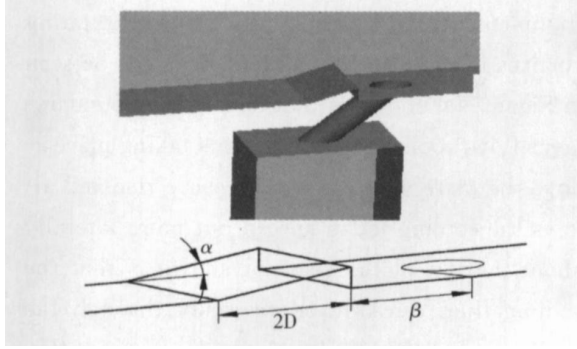


Fig. 2 Schematic of upstream ramp.

The flow-aligned blocker design concept seeks to improve film-cooling effectiveness by preventing or blocking hot gas from being entrained by the film-cooling jets. The flow-aligned blockers studied by Shih, et al.^[21] is shown in Fig. 3. Since the film cooling is for a flat plate in which the cooling jet emerges from a row of inclined holes, the flow-aligned blockers were taken to be pairs of parallel ribs or fence-like protrusions from the flat plate with rectangular cross sections. These “rectangular prism” blockers are located $1D$ downstream of the film-cooling hole. Each blocker has height b and thickness c , and separated by a distance a . Shih, et al.^[21] studied three values of a , b , and c : $0.8D$, D , and $1.2D$ for the spacing a ; $0.2D$, $0.4D$, and $0.6D$ for the height b ; and $D/20$, $D/10$, and $D/5$ for

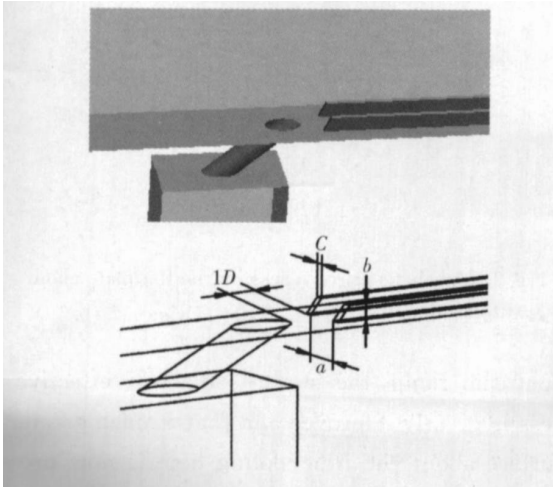


Fig. 3 Schematic of flow aligned blockers.

the thickness c .

2.3 Formulation and numerical method of solution

The baseline problem with and without the upstream ramp or the blockers was modeled by the ensemble-averaged conservation equations of mass (continuity), momentum (full compressible Navier-Stokes), and energy for a thermally and calorically perfect gas. The effects of turbulence were modeled by using the two-equation realizable $k-\epsilon$ model^[23]. In all cases, the integration of all equations is to the wall (i.e., wall functions were not used).

Na and Shih^[20] and Shih, et al.^[21] generated solutions to the aforementioned governing equations by using Version 6.1.18 of the Fluent-UNS code^[24]. Since only steady-state solutions were of interest, the SIMPLE algorithm was used. The fluxes at the cell faces representing advection were interpolated by using second-order upwind differences. The fluxes at the cell faces representing diffusion were interpolated by using second-order central differences. For all computations, iterations were continued until all residuals for all equations plateau to ensure convergence to steady-state has been reached. At convergence, the normalized residuals were always less than 10^{-6} for the continuity equation, less than 10^{-6} for the three components of the velocity, less than 10^{-8} for the energy equation and, and less than 10^{-5} for the turbulence quantities.

Na and Shih^[20] and Shih, et al.^[21] performed a grid-independent study and a validation study, which will not be repeated here. Interested readers are referred to those papers.

2.4 Results for the upstream ramp

For the upstream ramp shown in Fig. 2, Na & shih^[20] studied the nature of the flow and how that flow affects adiabatic effectiveness and surface heat transfer. In Fig. 4, it can be seen that when there is no ramp, there is a significant pressure rise on the flat plate, just upstream of the film-cooling hole. This high pressure region forms because the approaching boundary-layer

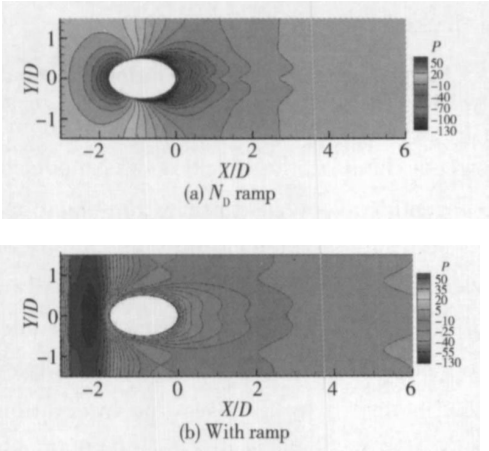


Fig. 4 Static gage pressure (Pa) on the flat plate about the film cooling hole ($\alpha=8.5^\circ$, $\beta=1D$, and $M=0.49$).

flow impinges on the film-cooling jet emerging from the hole. But, when there is an upstream ramp, the static pressure upstream of the film-cooling hole is quite low. This reduced pressure near the surface resulted because the ramp diverted the approaching boundary-layer flow upwards slightly so that it interacts with the film-cooling jet at a distance above the flat plate. The flow pattern created by the ramp is shown in Fig. 5. From this figure, it can be seen that there is a recirculating flow that extends from the backward-facing step of the ramp to the upstream end of the film-cooling hole. The recirculating flow entrains the cooler fluid from the

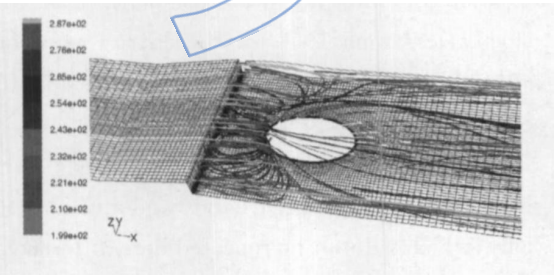


Fig. 5 Streamlines (colored by temperature) induced by the ramp with $\alpha=8.5^\circ$, $\beta=1D$, and $M=0.49$.

film-cooling jet and cools the wall bounding it. On this, it is important to note that this recirculating flow region will be filled with cooler fluid only if the reattachment of the separated region induced by the backward-facing step extends past the film-cooling hole. This is so that the hot gas will flow over the film-cooling jet. If the

hot gas reattached upstream of the hole, then hot gas will fill that region. Since the approaching boundary layer is made up of hot gas, deflecting it above the film-cooling hole also delays the entrainment of hot gases by the counter-rotating vortices in the film-cooling jets. This can be seen in Fig. 6. Not shown is that having the boundary-layer flow/cooling jet interactions taking place above the plate weakens the pressure rise and allows the cooling jet to spread out more laterally about the flat plate. Also not shown is that the cooling fluid can flow more easily through the cooling hole, which reduced the discharge coefficient associated with the hole. Figure 7 shows that with an

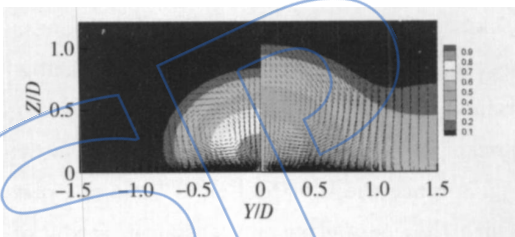


Fig. 6 Normalized temperature $(T_\infty - T)/(T_\infty - T_c)$ at $Y-Z$ plane located at $X/D=3$ ($M=0.49$, $\alpha=14^\circ$ that is located at $\beta=0.5D$). Left: No ramp. Right: With ramp.

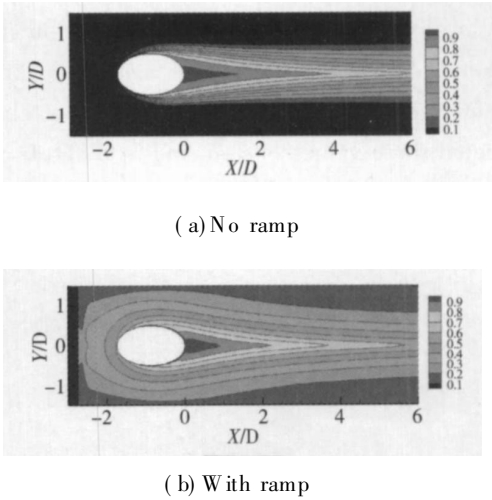


Fig. 7 Adiabatic effectiveness on the flat plate about the film cooling hole ($\alpha=8.5^\circ$, $\beta=1D$, and $M=0.49$).

upstream ramp, the surface adiabatic effectiveness is greatly improved and that a much greater surface about the film-cooling hole is now protected by the cooling jet, including the region upstream of the film-cooling hole. Na and Shih^[20]

examined the effects of the ramp angle and the distance from the backward-facing step of the ramp to the film-cooling hole, and made the following observations can be made. First, increasing the ramp angle (and hence the height of the backward-facing step), the higher is the laterally averaged adiabatic effectiveness, at least for the two angles studied. Second, placing the ramp $0.5D$ upstream gave higher laterally averaged adiabatic effectiveness than placing it D upstream. Bottom line is that placing a ramp upstream of a row of film-cooling holes increases laterally averaged adiabatic effectiveness significantly. The laterally averaged adiabatic effectiveness with ramp can be two or more times or higher than that without the ramp.

The average heat transfer rate per unit area for the entire flat plate with and without the ramp ($\alpha = 14.0$, $\beta = 0.5$, and $M = 0.49$) is -916.78 W/m^2 and -1094.70 W/m^2 , respectively. These numbers include the part of the plate upstream of film cooling, where surface heat transfer is highest. If that region is excluded (e.g., by deleting $19/49$ of -1094.70 W/m^2 which is a conservative estimate), then the average heat transfer per unit area with and without ramp is -492.30 W/m^2 and -670.22 W/m^2 , respectively. Thus, introducing a ramp can substantially increase adiabatic effectiveness and reduce surface heat transfer.

In closing, it should be noted that the ramp geometry studied is for an approaching boundary layer with a thickness of $0.14D$, a shape factor of 1.49 , and zero freestream turbulence.

2.5 Results for flow aligned blockers

For the rectangular-prism blockers shown in Fig. 3, Shih, et al.^[21] examined the effects of the three design parameters: $0.8D$, D , and $1.2D$ for the spacing between blockers a ; $0.2D$, $0.4D$, and $0.6D$ for the height of the blockers b ; and $D/20$, $D/10$, and $D/5$ for the thickness of the blockers c . They found the averaged adiabatic effectiveness to be the highest when $a = D$, $b = 0.6D$, and $c = D/5$. Though D may indeed be near optimum for the spacing between the blockers, optimum

values for the height and thickness of the blocker remain unclear since the effects of these two parameters remained monotonic in the range studied. It is anticipated that the optimal height is related to the blowing ratio, and the optimal thickness of each blocker is related to the spacing between film-cooling holes since there is a region between film-cooling holes that are unprotected by film cooling. Thus, a true optimal design even for the simple configuration considered here requires further study.

Figure 8 shows normalized temperature $(T_\infty - T)/(T_\infty - T_c)$ at two Y - Z planes, one located at $X/D = 3$ and one at $X/D = 7$ in which the blowing ratio is $M = 0.5$ with and without blockers. From this figure, it can be seen that the two blockers confine the cooling flow within it and prevents the entrainment of hot gases. By $X/D = 7$, the coolant is fairly well mixed along the spanwise Y direction so that the temperature variation is mostly along Z . Since the blockers are placed D downstream of the film-cooling hole exit, the cooling flow also wraps around the "outer" sides of the blockers. Thus, cooling extends beyond the blockers by as much as $0.2D$ beyond the blockers. This, of course, improved the film-cooling effectiveness outside of the blockers. Effects of blowing ratio can be seen in Fig. 9.

The average heat transfer rate per unit area with and without blockers is -781.34 W/m^2 and -1094.70 W/m^2 , respectively. These numbers include the part of the plate upstream of film cooling. If that region is excluded (e.g., by deleting $19/49$ of -1094.70 W/m^2), then the average heat transfer per unit area with and without ramp is -356.86 W/m^2 and -670.22 W/m^2 , respectively. Thus, introducing flow-aligned blockers can increase adiabatic effectiveness significantly as well as reduce surface heat transfer. But, the difficulty in implementing this method is that it is hard to predict the flow pattern. This design concept is best for problem for which the flow pattern is less complicated and do not vary appreciably during a duty cycle.

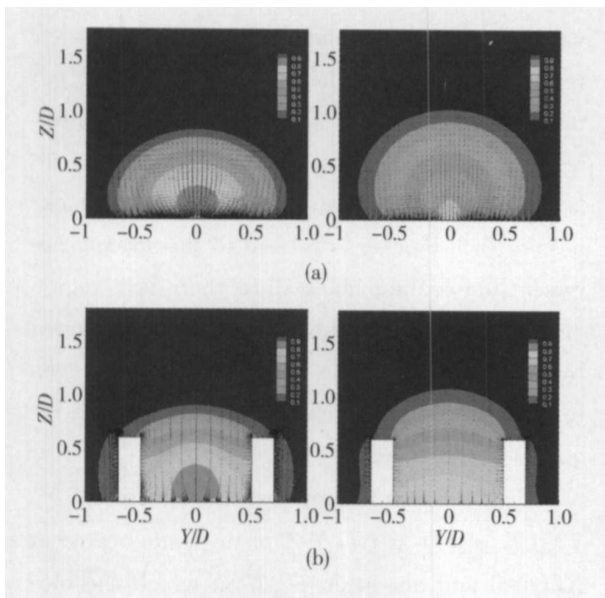


Fig. 8 Normalized temperature $(T_{\infty} - T)/(T_{\infty} - T_c)$ at $Y-Z$ planes located at $X/D = 3$ and 7 for $M = 0.5$.

(a) No blockers. (b) With blockers (Run 3).

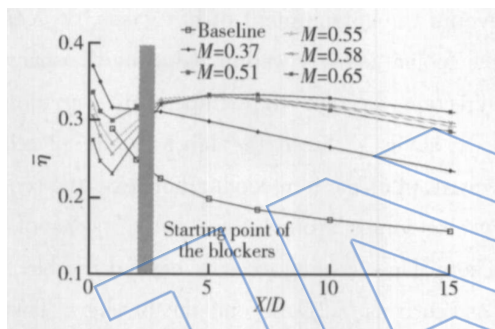


Fig. 9 Laterally averaged adiabatic effectiveness for several blowing ratios. Baseline has blowing ratio of 0.5.

3 Concluding remarks

Despite considerable efforts by many researchers and engineers, effective cooling remains a challenge, especially in regions referred to as “edges,” that is the end wall, the blade tip, the leading edge, and the trailing edge. In the end-wall region, film cooling is difficult because of the complicated secondary flows such as horseshoe and passage vortices that take place there. In the blade-tip region where the blade must not touch the shroud in the presence of thermal expansion, the leakage of hot gas from the pressure to the suction side causes major cooling problems. In the leading-edge region, the blade gets the hottest gas impinging on its surface where the boundary layer thickness is the

thinnest the pressure is the highest, and the curvature of the surface is the greatest. In the trailing-edge region, the thinness of the geometry severely limits design possibilities. Other issues in film cooling that require further study include surface roughness induced by erosion and deposition which changes surface heat transfer characteristics with service; TBC spallation which not only eliminates protection by the TBC but also modifies the flow and heat transfer in the region about the spallation; and thermal stresses induced by nonuniform cooling and thermal cycling. Solutions to the challenge of more effective cooling requires not just an improved understanding of the turbulent flows that can take place but their control through innovative designs.

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