ANALYSIS OF FLOW CHARACTERISTICS AND OVERALL COOLING EFFECTIVENESS OF ULTRA-HIGH IMPINGEMENT DISTANCE HEAT SHIELD

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ABSTRACT
Computational investigations is reported on a flat plate subjected to combined impingement and film cooling. Conjugate heat transfer and 3-D steady RANS approach with the k-ω SST turbulence model were used for analysis of the impingement and film cooling plates. The flow dynamic features of impinging jets and the coolant-mainstream interaction in presence of heat shield are described. The discharge coefficients are presented at blowing ratio M = 0.2 to 0.8 and Reynolds number Re 9300 to 75000 respectively. A comparison between the conventional heat shield and ultra-high impingement distance heat shield shows that the discharge coefficients values for the new heat shield are higher. The value of overall cooling effectiveness for the new heat shield is lower than the conventional structure at low blowing ratio, but at high blowing ratio, the values of overall cooling effectiveness for both the heat shields are almost equal. As the blowing ratio is increased from 0.2 to 0.8, the overall effectiveness values show an increasing trend for both the heat shields.

INTRODUCTION
An afterburner is a key component for the modern engines. The thrust augmentation provides by afterburning or reheating is critical for takeoff, maneuverability and acceleration. Therefore, there has been ongoing demand of higher temperature ratio to meet the goal of maximum thrust. As a consequence, the afterburner outlet temperature for recent high-performance engines exceeds 2200K (Stephen, 1960), which is far beyond the melting point of the afterburner cylinder.
The original flat-plate heat shield is a housing for enclosing the afterburner tube and receiving the coolant from the compressor for delivery along the tube, which is composed of several cylindrical cylinders. The cooling air can flow out from the joint between the cylinders, forming a film on the hot gas side wall and protect the surface from the hot exhaust gas (Stephen, 1960; Gérard et al, 1991). This structure has a low film cooling effectiveness so that requires a large quantity of cooling air to protect the wall.

(Howard et al, 1990) proposed a new structure of the liner that contains a plurality of double wall elements. Each element is generally annular and extends from a first axial part to a second axial part. The second axial part is placed inwardly towards the geometric center line of the afterburner duct, and both the two axial part have holes attached. This structure includes both impingement and film cooling techniques, which makes full use of cooling air. However, it is complicated and difficult to process and maintain.

Later, the transverse corrugated heat shield has been proposed. The wave axis of the latitudinal corrugated liner is parallel to the streamwise direction. This structure can effectively absorb the radial high frequency oscillation energy of 1000-2000HZ, which is mostly used in the turbojet afterburner. There remains a recirculation zone between its peaks and troughs, which can make full use of the cooling air and enhance heat transfer. (Zhang et al, 2000) analyzed the temperature distribution of the transverse corrugated liner under experimental and numerical simulation conditions.
Concluded that along the flow direction, the temperature of the wave wall gradually increases and has a certain range of temperature fluctuations. (Qu et al, 2016) analyzed the influence of blowing ratio, corrugation wavelength and corrugation amplitude on the cooling efficiency of the transvers corrugated heat shield. The results show that with the increase of the blowing ratio, the average adiabatic film cooling effectiveness increases, and there is an optimal blowing ratio M=0.2. Both the corrugation wavelength and the corrugation amplitude have a little influence on the average adiabatic film cooling effectiveness.

Today’s turbofan afterburner with low frequency energy of 100-500HZ has a large radial dimension. Therefore it is better to use the longitudinal ripple heat shield that can absorb the low frequency oscillation energy. The wave axis of longitudinal ripple heat shield is perpendicular to the streamwise direction. (Funazaki, 2001) came up with a longitudinal corrugated heat shield with certain range of hole diameter and hole pitch, which only drill holes in the downstream waves. They show that this structure can make the distribution of cold air more uniform and reduce the consumption of cold air. (Thomas, 1993) proposed a corrugated wall with uniformly hole and introduced a low cost manufacturing method for making the combustor liners. They show that this kind of structure can effectively reduce the radial temperature gradient and thermal stress, thus improving the fatigue stress. Also, more complex studies such as the influence of the cooling channel dimensionless height, corrugated wall dimensionless height and aerodynamic parameters on film cooling effectiveness have been conducted, the results show that the distribution of overall cooling effectiveness in the direction of streamwise is uneven, so there is a hidden danger of excessive thermal stress. (Nathan et al, 2016) used either crossflow, impingement, or a combination of both to supply the film cooling to investigate overall film cooling and impingement cooling effectiveness. The results show that the heat transfer coefficient generally increases with streamwise development, and increase with increasing blowing ratio. (Li et al, 2018) proposed a method to improve full-coverage effusion cooling effectiveness by varying the cooling arrangements and wall thickness. They found that adding impingement and pins to film cooling, and decreasing wall thickness increase the cooling efficiency significantly. (Sneha et al, 2018) provided spatially-resolved distributions of adiabatic cooling effectiveness for effusion surface and spatially-resolved distributions of surface Nusselt numbers impingement surface.

During the development of heat shield, its aerodynamic parameters, cooling effectiveness and coolant consumption have always been improved. However, its temperature non-uniformity restricts its development. The author proposed a impingement/film cooling apparatus with ultra-high impingement distance, the distance from the impingement surface to the effusion surface is 20 H. /d. The porous heat shield with an ultra-high impingement distance can effectively limit the oscillating combustion of the afterburner by generating the gas damping and turbulence. It also can effectively improve the uniformity of discharge coefficient and overall cooling effectiveness while maintaining a relatively high level of overall cooling effectiveness.

**ANALYSIS**

**Discharge Coefficient**

The discharge coefficient of the orifice is defined as the ratio of actual mass flow rate to that which should be pass through the cross-sectional area (Huning, 2008). It can be written in the form

\[ C_d = \frac{m_i}{m_{id}} \]  

The ideal mass flow rate can be calculated:

\[ m_i = p_i A \sqrt{\frac{2k}{k-1}} \frac{1}{RT_i} (\frac{p_2}{p_1})^{\frac{2}{k}} - (\frac{p_2}{p_1})^{\frac{k+1}{k}} \] 

Discharge coefficients CD of hole are impacted by several different factors: Reynolds Number, length to diameter ratio, cross flow at inlet and outlet, pressure ratio (Huning, 2008). The inlet or outlet approaching flow inclined to the orifice axis caused by crossflow is accompanied by flow loss, which decreases the discharge coefficient. Rohde et al (Rohde et al, 1969) experimentally investigated the discharge coefficient with cross-flow near the entrance of orifice. They considered the ratio of velocity head of the orifice jet to the velocity head of the main flow as a function of the discharge coefficient

\[ \theta = \frac{p^* - p_{jet}}{(p^* - p_{cro})} \] 

(Gritsch et al, 1998a; Gritsch et al, 2001) considered the influence of both entry and exit crossflow near the holes on the discharge coefficient. They plotted the discharge coefficient vs the ratio of jet to internal or external flux momentum. The formulas are:

\[ I_{jet/inCr} = \frac{(\rho u^2)_{jet}}{(\rho u^2)_{c}} \]

\[ I_{jet/extCr} = \frac{(\rho u^2)_{jet}}{(\rho u^2)_{s}} \]
ratio. The influence of crossflow on the discharge coefficient can be neglected when the momentum flux ratio exceed 2.

**NUMERICAL SIMULATION APPROACH**

**Combined impingement and film cooling Geometries**

The impingement and film holes are staggered arrangement as shown in Fig. 1. As shown in Fig. 2, to guarantee the complete development of the mainstream, the test plate is extended for 30 \(d_e\) on the upstream. To eliminate the impact of back flow, the test is extended for 40 \(d_e\) on the downstream. The length of \(H_c\) is 20 \(d_e\) for the ultra-high impingement distance model (model 1) and 5 \(d_e\) for the conventional impingement distance model (model 2). The length of \(L_c\) is 200 mm. The value of \(\Phi\) and \(b\) are fixed. The hole number of the jet impingement holes and film holes was shown in Fig. 3.

The jet hole pitch are the same in the X and Y directions for both impingement holes and film holes, which can be determined by:

\[
A = n_{st-j} n_{sp-j} X_j Y_j \Phi b = d_j^2 n_{st-j} n_{sp-j} \pi / 4
\]

\[
X_j = Y_j = d_j \frac{\pi}{4 \Phi}
\]

\[
A = n_{st-c} n_{sp-c} X_c Y_c b = d_c^2 n_{st-c} n_{sp-c} \pi / 4
\]

\[
X_c = Y_c = d_c \frac{\pi}{4 \Phi}
\]

Fig. 1 Schematic of the cooling pattern

**Boundary Condition and Numerical Approach**

The periodic boundary condition is assigned to the sidewalls in Y direction as shown in Fig. 4. The top and bottom of the effusion wall are set as couple wall. The impingement wall is adiabatic wall. The mass inlet boundary is applied at the inlet of mainstream and coolant. The mainstream Reynolds number from 9300 to 75000 is calculated based on the parameters of hot gas inlet. The mass flux of coolant depends on both of the blowing ratio \(M = \rho_f u_j / \rho_g u_e\) from 0.2 to 1 or the mainstream Reynolds number. The variables in blowing ratio depend on the aperture area of film holes and the parameters of coolant inlet. The fluid is ideal gas, and its specific heat and thermal conductivity depend on temperature. The fluid dynamic viscosity is determined by Sutherland’s formula. The commercial CFD software ANSYS Fluent 16.0 was employed in solving the steady RANS equations. The SST k-\(\omega\) turbulence model was applied, because the previous numerical computation research (Dees, et al, 2012; Mensch et al, 2014; Panda et al, 2012; Ramachandran et al, 2015; Zhou et al, 2016; Dyson et al, 2014; Zhao et al, 2016) indicated that better simulation results can be obtained by...
using the SST k-ω turbulence model. The criteria of convergence is that the residual value of energy is below 10^{-9}. As shown in Fig. 5, the value of $y^*$ of the first cell layer on the target wall was less then 1. To validate the numerical model, the numerical results were compared with experiment data, as shown in Fig. 6 (Qu, et al, 2017).

![Fig. 5 The value of $Y^*$ on the target wall](image)

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**RESULT AND DISCUSSION**

**Effect of Reynolds number on flow characteristic of impingement hole**

The case 1 has a larger level of discharge coefficient than case 2 as shown in Fig. 9. This is mainly due to the vena contracta of the impingement hole. The impingement holes have a small ratio of length to diameter $l_i/d_i$. To investigate the vena contracta, it was represented by the velocity contour. The number of contour levels was eight for each Reynolds number. The smallest cross-sectional area of high velocity region was applied to represent the vena contracta as shown in Fig. 8. Increasing the Reynolds number from 9300 to 42000 causes a large separation at the hole entrance and the flow cannot reattach to the hole sidewall, which reduces the vena-contracta area and actual mass flow rate. The line 2 and line 3 as shown in Fig. 8 are almost coincide, which means that the Reynolds number from 42000 to 75000 has less influence on vena contracta.

Fig. 9 also shows that the discharge coefficient of the downstream hole is larger than that of the upstream hole. This is due to the end of coolant plenum is closed and the coolant will flow back, which reduces the intensity of entrance crossflow.
Fig. 9  The discharge coefficient of impingement holes for Re = 9300, 42000, 75000, M = 0.3

Effect of Reynolds number on flow characteristic of effusion hole

Fig. 10  The discharge coefficient of film holes for Re = 9300, 42000, 75000, M = 0.3

The ratio of length to diameter of film hole is 0.75, which is a short hole too.

Fig. 10 shows that the discharge coefficient of the upstream effusion holes is negative (use the value of zero to represent) at Reynolds number Re = 9300. This is due to the pressure of the front-end of chamber in X direction is lower than the mainstream pressure. The impingement flow cannot eject through the effusion hole, and even the mainstream pour back into the chamber.

Fig. 10 also shows that the discharge coefficient of the downstream holes is higher than that of upstream holes. This is due to the outflow at upstream thickens the downstream boundary layer. The thick boundary layer reduces the flow area and accelerates the velocity of the mainstream as shown in Fig.8. According to the principle of mass conservation, when the density is assumed to be constant, the increase of velocity can reduce the pressure $p_g$, therefore increases the pressure ratio. Another reason is the film accumulation at the downstream provides the protection for downstream holes. The large low velocity region near the outlet of film holes reduces the influence of the mainstream crossflow, resulting in larger exit cross-sectional area compared with the upstream film hole as shown in Fig. 12.

Fig. 11  Velocity contour for center plane of film holes

Fig. 12  Comparison of upstream and downstream hole cross-sectional area (left: upstream; right: downstream)

Effect of blowing ratio on discharge coefficient of film hole

Fig. 13  The relation between discharge coefficient and blowing ratio for film holes
Fig. 14 The pressure ratio of film holes inlet and mainstream under different blowing ratio

Fig. 11 shows the effect of blowing ratio on discharge coefficient of film holes. With the increases of blowing ratio, the discharge coefficient increase. This is because of the enhanced suction of the film hole by the increased pressure ratio, which causes more penetration into the mainstream. For the case 1 in Fig. 13, there are some holes with a discharge coefficient of negative (use the value of zero to represent). This is due to the pressure ratio of these holes is less than 1 as shown in Fig. 14 and the insufficient pressure cannot eject the impingement flow in chamber through the film holes.

The impact of impingement distance on discharge coefficient of impingement holes

Fig. 15 Flow characteristic of impingement gap

Fig. 16 Vena contracta of film hole

Fig. 17 Discharge coefficient of high impingement distance model and low impingement distance model.

Fig. 15 shows a complex interaction between the impingement jet and effusion jet air feed. There remains a reverse flow in the impingement gap. The interaction of jet-impacting flow causes a upwash flow. The upwash flow will reattach to the impingement surface and form a crossflow. The effect of the crossflow on the exit cross-section area highly depends on the impingement distance. For the high-impingement-distance chamber, there remains a larger momentum loss for the upwash, and the larger the momentum loss of upwash, the smaller the momentum of crossflow. This means that for the higher impingement distance chamber, the exit cross-section area of impingement holes is larger as shown in Fig. 16. Therefore, the level of discharge coefficient for model 1 (high-impingement-distance gap) is larger by a factor of 0.02, compared to the model2 (low-impingement-distance gap) as shown in Fig. 17.

The effect of impingement gap on discharge coefficient of film holes

Fig. 18 shows that the difference between model 1 and model 2 is smaller at low blowing ratio M=0.2. With the increased of blowing ratio, the difference becomes larger. And for the most of film holes, the discharge coefficient of model 1 is larger than model 2 at blowing ratio M=0.8. This is due to the vortex near the effusion wall in chamber is enhanced by the increase of blowing ratio, which washes
away the impingement jet that should have blown into the film holes as shown in Fig. 19. And for model 2, the intensity of the vortex is larger due to the low impingement distance, which will cause less entrance than model 1.

From Fig. 18, we can also see that the distribution of discharge coefficient is more uneven for model 2. This is due to the impingement jet cannot be fully developed in small space for model 2, which resulting in uneven inlet condition for each hole.

\[ \eta = \frac{T_g - T_w}{T_g - T_c} \]  

(6)

The impact of impingement distance on overall cooling effectiveness

Overall cooling effectiveness is defined as follows:

**Fig. 18** Discharge coefficient of the third row of film holes of model 1 and model 2 at the blowing ratio of 0.2 and 0.8.

**Fig. 19** The vortex in impingement gap with short impingement distance

**Fig. 20** Contour of overall cooling effectiveness on interaction surface where the coolant interacts with the mainstream (for M = 0.6)

**Fig. 21** Laterally averaged overall cooling effectiveness

Fig. 20 show that as the impingement distance decreases from 20d to 5d, the overall cooling effectiveness is improved. The regions of high overall cooling effectiveness are concentrated near the impingement area. For the model 2, the spot area with high overall cooling effectiveness is more obvious and thus generates a higher gradient of temperature. The larger thermal stress caused by higher temperature gradient will reduce the thermodynamic efficiency and the service life of model 2.

The Fig. 21 shows that with the impingement distance decreases from 20d to 5d, the overall cooling effectiveness increases by approximately 0.1. The difference of overall cooling effectiveness \( \phi \) between model 1 and model 2 is smaller with the increase of blowing ratio. This is due to the stronger impingement cooling at larger blowing ratio reduces the influence of impingement distance. The Fig.18 also shows that the difference of overall cooling effectiveness \( \phi \) between model 1 and model 2 is smaller at large \( x/d_f \). This is due to the accumulated film at the downstream is the main factor affecting the cooling effectiveness.
CONCLUSION

Numerical study was applied to investigate the flow characteristic and overall cooling effectiveness of laminated configuration with high impingement distance. This paper also compared the uniformity of temperature and discharge coefficient of high and low impingement-distance heat shield.

Some conclusions are summary as follows:

Due to the short ratio of length to diameter \( l_c/d_c \) of impingement holes, with the increase of Reynolds number from 9300 to 42000, the discharge coefficient decreases. When the Reynolds number increases from 42000 to 75000, the influence of Reynolds number decreases, thus the increase of discharge coefficient mainly depends on the pressure ratio. Due to the influence of crossflow in coolant plenum, the discharge coefficient of the downstream impingement hole is larger than that of the upstream hole.

The increase of Reynolds number of films holes has less impact on the discharge coefficient at the hole number from 20 to 75 due to the higher difference of pressure ratio. The mainstream crossflow at the exit of film holes is weakened by the accumulation of film.

The discharge coefficient of impingement holes increases with the increase of blowing ratio. The discharge coefficient of film holes increases with the increase of blowing ratio. At blowing ratio \( M=0.8 \), the discharge coefficient value is on approximately the same level, due to the overall increase in pressure ratio.

The overall cooling effectiveness increases with the increase of blowing ratio. Because of the accumulation of film at downstream, the overall cooling effectiveness is larger at downstream.

The impingement-hole discharge coefficient of high impingement model is larger than that of low impingement model. And for the most of film holes, the discharge coefficient of high impingement model is larger than that of low impingement model.

The overall cooling effectiveness of conventional heat shield is larger than that of ultra-high impingement distance heat shield. But the overall cooling effectiveness of conventional heat shield is poor and the temperature gradient of model 2 is larger, which will reduce the thermodynamic efficiency and the service life.

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NOMENCLATURE

- \( \varphi \) overall cooling effectiveness
- \( l \) length of the hole
- \( d_c \) diameter of impingement hole, mm
- \( d_f \) diameter of effusion hole, mm
- \( H_c \) impingement distance, mm
- \( L_c \) the length of test plate, mm
- \( C_f \) discharge coefficient (actual mass flow rate/ideal mass flow rate)
- \( m \) actual mass flow rate
- \( m_a \) ideal mass flow rate
- \( \theta \) the ratio of velocity head of the orifice jet to the velocity head of the main flow
- \( p \) pressure
- \( p' \) total pressure
- \( I \) momentum flux ratio
- \( M \) blowing ratio, \( \rho'_1 U_c / \rho_u U_g \)
- \( Re \) reynolds number based on hydraulic diameter of hot gas inlet \( \rho_1 D / \mu_1 \)
- \( \delta_c \) Impingement wall thickness
- \( \delta_f \) film wall thickness
- \( \phi \) porosity of impingement wall
- \( X \) streamwise coordinate, m
- \( Y \) coordinate normal to the test surface, m
- \( Z \) spanwise direction, m
- \( b \) aperture area ratio of film hole to impingement hole
- \( n \) Holes number

Greek symbols

- \( \varphi \) Overall cooling effectiveness
- \( \rho \) density, kg/m³
- \( \mu \) molecular viscosity, Ns/m²

Subscripts

- \( intCr \) crossflow at the hole entrance
- \( extCr \) crossflow at the hole exit
- \( g \) hot gas
- \( c \) coolant
- \( f \) film
- \( l \) entrance
- \( e \) exit
- \( st \) streamwise direction
- \( sp \) spanwise direction
- \( id \) ideal
- \( jet \) jet gas
- \( cro \) crossflow
- \( st-c \) streamwise direction of impingement wall
- \( sp-c \) spanwise direction of impingement wall
- \( st-f \) streamwise direction of effusion wall
- \( sp-f \) spanwise direction of effusion wall

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