ABSTRACT

An experimental study has been conducted on the heat transfer performance in a multiple-jet impingement cooling system with pin fins and film cooling extraction. Transient liquid crystal thermography experiment were conducted to explore the heat transfer characteristics on these target plates. The ratio of jet-to-plate spacing was fixed to 1.5 and the Reynolds number based on jet diameter ranges from 15,000 to 30,000. The experimental results show that pin fin and effusion holes structure reduces crossflow strength in downstream region, improves and uniform heat transfer on the whole target plate obviously. When $Re_j=15,000$, there is a highest improvement in heat transfer on the endwall. Compared with flat plate, the averaged Nusselt number on the pin fin plate and the pin fin plate with effusion holes can be improved by up to 6.3% and 25.3%. Furthermore, three-dimensional CFD analysis was done to analyze the detailed flow structure and heat transfer characteristics in the impingement cooling systems with pin fins and film cooling extraction on the target plates.

INTRODUCTION

Advances in gas turbine technology for both aircraft propulsion and power generation are focused on increasing efficiency as well as maintaining or extending parts service life, and the key to improve efficiency is to increase the inlet gas temperature of turbine. The latest generation aircraft engine turbine inlet gas temperature exceeds 1700°C [1], which is much higher than melting temperature of material. Therefore, more efficient and advanced cooling technologies are urgently needed to ensure the safe and reliable operation for gas turbines. Impingement cooling is one of the most effective internal cooling methods. When the coolant flows directly impinge on target plate surface, the flow boundary layer is very thin and high heat transfer coefficient can be obtained.

In early stage, the research on impingement cooling characteristics has already begun. Factors like geometry of impingement hole, jet-to-plate spacing ratio, jet velocity and cross flow have been studied successively [2-6]. Impingement cooling on flat plate often has poor local heat transfer characteristics at non-stationary locations and thus affects the service life of materials seriously. Some scholars tried to combine impingement cooling with ribs, which not only increases the heat transfer area but also improves the turbulence in the cavity, so as to achieve the enhanced heat transfer. Chang et al. [7] experimentally found that placing rib between two jets can obtain better heat transfer performance. They also established corresponding formulas based on experimental data. Andrews et al. [9] reported that keeping the ribs spanwise direction consistent with the crossflow direction can achieve better heat transfer performance. Son et al. [10] have studied the impingement cooling on rough target plate surface using transient liquid crystal thermography method. The results show that employing circular and diamond ribs on flat plate can increase heat transfer performance by 22%-35%. Xing et al. [11] described heat transfer distribution on structured surface with micro-rib by transient liquid crystal thermography experiment.

A more efficient cooling technology is the combination of impingement cooling and film cooling. The arrangements of effusion holes may change flow field of the impingement cooling system, which is less considered in former studies. On the other hand, double wall cooling is an efficient cooling technology for next generation aircraft engine and gas turbine [12-13]. Thus, in this paper, impingement heat
transfer performance in a double-wall cooling system with pin fins and effusion holes was studied. Heat transfer, pressure loss and flow structure characteristics were investigated by experiments and numerical computations.

**EXPERIMENT SETUP**

A schematic of experimental device is given in Fig.1. The airflow is drawn into the wind tunnel by a variable frequency blower and its mass flow rate is measured by a vortex flowmeter. Then, the air temperature is heated up to around 45°C, after getting through a high-power mesh heater. The wind tunnel is made of 15mm thickness Plexiglas, which provides good thermal insulation for the test section. After passing through a channel, the air flows then into the inlet plenum and enters the test section. The test section is equipped with thermocouples, pressure holes and target plates with thermograph liquid crystal, shown in Fig. 2. The measured data is transmitted to a LABVIEW data acquisition system and recorded. At last, the fluid flows into outlet plenum and setting chamber to exit. A 3CCD camera is placed at the back of outlet plenum, and colour changes on the target plate surface can be clearly recorded during the experiment through the outlet plenum wall made of high transparent Plexiglas. Lights are installed on each side of the test section to make the colour bright, and the test section is covered with black opaque screen to avoid interference from ambient light.

**Figure 1 Schematic of experimental setup**

Three target plates are used in the experiments, which are the flat plate, pin fin plate and pin fin plate with effusion holes. The target plates are made of 20mm thickness Plexiglas and all the pin fin structure are designed to be full-high, which connects the impingement plate and target plate without gaps. The geometric structures of pin fin plate and pin fin plate with effusion holes are shown in Fig. 3. Besides, the ratio of jet to plate spacing \(H / D_j\) is equal to 1.5, \(D_p = D_e = 10\)mm, \(P_X / D_p = 5\) and \(P_I / D_p = 5\).

**Figure 2 Schematic of test section**

**Figure 3 Schematic of target plates**

The method of transient liquid crystal thermography (TLC) is used to determine the heat transfer in the jet impingement system, which is referred to Ireland [14]. The liquid crystal was sprayed uniformly on the surface of target plate endwall. Since the measurement time is short, the heat transfer in the Plexiglas target plate can be considered as a transient heat conduction process of one dimensional half infinite plate, which can be described by equation (1):

\[
k \frac{\partial^2 T}{\partial z^2} = \rho c \frac{\partial T}{\partial t}.
\]  

(1)

Boundary conditions are:

\[
t = 0, T = T_0
\]

\[
z = 0, \quad \frac{k \partial T}{\partial z} = h(T_B - T_w)
\]

\[
z \to \infty, T = T_0
\]

Because the heat transfer on the surface of pin fin and effusion holes does not satisfy the one-dimensional heat conduction hypothesis, the heat transfer characteristics on
endwall is only measured in this experiment. By solving equation (1), the dimensionless temperature at the wall is obtained:

$$\frac{T_w - T_0}{T_B - T_0} = 1 - \exp(h^2 \frac{l}{k \rho c}) \times \text{erfc}(h \sqrt{\frac{l}{k \rho c}}) . \quad (2)$$

Eq. (2) is only valid for an ideal temperature step rise within the flow. However, in reality the thermocouples record a time-dependent variation of the mainstream temperature, which can be simulated by a series of small temperature step rise. By using Duhamel’s superposition theorem, the solution for the heat transfer coefficient at every location is then represented as:

$$T_w - T_0 = \sum_{i=1}^{N} \left(1 - \exp(h^2 \frac{(l - l_i)}{k \rho c}) \times \text{erfc}(h \sqrt{\frac{(l - l_i)}{k \rho c}})(T_{B,i} - T_{B,i-1}) \right) \quad (3)$$

Solving Eq. (3) by an iteration method, the heat transfer coefficient $h$ can be determined.

The jet Reynolds number is defined as:

$$Re_j = \frac{V_j D_j}{\nu} . \quad (4)$$

The local Nusselt number is defined as:

$$Nu = \frac{h D_j}{\lambda} . \quad (5)$$

The experimental uncertainties have been determined using Kline and McClintock’s method [15]. According to the measurement instruments, the maximum measurement uncertainties of the flow rate are $\pm 2.0\%$, the length as well as the pressure drop are $\pm 1.0\%$ and $\pm 2.5\%$ respectively; The measurement uncertainty in temperature are $\pm 0.3^\circ C$, time detection is $\pm 0.1 \text{s}$, and the air properties are $\pm 1.0\%$, and the thermal conductivity of the Plexiglas is $\pm 0.01 \text{W/} \text{m} \cdot \text{K}$. The uncertainty due to the lateral conduction in the test plate can be below $2.0\%$. Therefore according to the standard uncertainty analysis, the measurement uncertainties of $Re_j$ and $Nu$ are respectively about $\pm 3.0\%$ and $\pm 8.5\%$.

**NUMERICAL SETUP**

In order to obtain a thorough understanding of the flow structure and heat transfer details in the impingement cooling system, three-dimensional steady CFD simulation was carried out. ANSYS CFX 14.5 was used for calculation, and k-omega SST turbulence model was chosen. Since this paper is focused on the internal heat transfer in a double-wall cooling structure, the simulation domain does not include the fluid within effusion holes, which is set to be pressure outlet, and the pin fin surface is adiabatic. To be more similar with experiment, the inlet plenum is taken into account for CFD model. The geometric model and boundary conditions are shown in figure 4.

![Figure 4 Schematic of computational model and boundary conditions](image)

The grid first layer spacing of end wall surface is $0.01 \text{mm}$ for structural mesh, thus averaged $Y$ plus is less than $2$ to meet the requirements of k-omega SST model. Grid independence check for pin fin plate has been done, and the difference of physical values between three meshes (cell sizes from 7.3 mil. to 12.0 mil.) is less than $1\%$. Figure 5 shows GCI analysis [17] of the three meshes at $Re_j = 15,000$, which indicates grid number has little influence on the physical values.

![Figure 5 GCI analysis for spanwisely averaged Nusselt numbers on pin fin plate at $Re_j = 15,000$](image)

**RESULTS AND DISCUSSION**

**Experimental results**

Figure 6 shows the local Nusselt number distribution on the endwall, which is obtained by liquid crystal thermograph experiment at $Re_j = 30,000$. The Nusselt numbers reach high values at the jet stagnation points and decreases rapidly from centre to around. From upstream to downstream, with increasing cross flow strength, the Nusselt numbers increase firstly and then reach maximum value at the fourth stagnation region. Then, with the cross flow getting further stronger in downstream region, jets become offset and the Nusselt numbers decease. The existence of pin fins and effusion holes structure makes the jet stagnation region move towards upstream, indicating that downstream crossflow effects is reduced. On the other hand, in the diagonal area between jets there are usually low heat flux regions. The structure of pin fin and effusion hole makes the airflow mix in these regions, thus improves and uniformes heat transfer.
Figure 6 Experimental data of local Nusselt number distributions on target plates at $Re_j=30,000$

Figure 7 Comparisons of experimental data of averaged Nusselt numbers on endwall

Figure 7 gives a comparison of experimental data of heat transfer on the flat plate with literature data from Ei-Gabry [18], and the averaged Nusselt numbers on pin fin plate and pin fin plate with effusion holes are also provided. It can be seen that in this paper averaged Nusselt number on flat plate agree reasonably well with the literature data, and the pin fin plate with effusion holes has maximum averaged heat transfer values while the flat plate has minimum averaged heat transfer values at the same jet Reynolds number. Besides, the heat transfer correlations have been given for the pin fin plates and pin fin plate with effusion holes, which were based on experimental data.

Pin fin plate:

$$\bar{Nu} = 0.081Re^{0.73}Pr^{1/3}, \quad 15,000 \leq Re \leq 30,000. \quad (6)$$

Pin fin plate with effusion holes:

$$\bar{Nu} = 0.102Re^{0.69}Pr^{1/3}, \quad 15,000 \leq Re \leq 30,000. \quad (7)$$

Figure 7 also provides the fitted curve for the correlations above, which agree well with experimental data, with deviation no more than 5.8%.

Figure 8 shows the Nusselt number ratio and pressure loss ratio compared with flat plate. From Figure 8 (a), it can be seen that pin fin and effusion holes structure improves the heat transfer on the endwall, and the heat transfer enhancement decreases as the jet Reynolds number increases. When $Re_j=15,000$, compared with flat plate there is a highest heat transfer enhancement and the averaged Nusselt number of pin fin plate and pin fin plate with effusion holes improves by 6.3% and 25.3% respectively. While when $Re_j=30,000$, the improvements are 0.7% and 13.2% respectively. From Figure 8 (b), it can be seen that pressure loss of pin fin plate is higher than flat plate by 18.0% while pressure loss of pin fin plate with effusion holes is less than flat plate by 7.3% at $Re_j=15,000$. Besides, the pressure loss ratio is almost independent of jet Reynolds number.

Numerical results

In order to strengthen the understanding of flow features in impingement cooling systems, CFD analysis was done. Figure 9 shows the local Nusselt number distribution on the endwall of target plates obtained by CFD. Figure 10 provides a comparison of spanwisely averaged Nusselt numbers on the endwall of flat plate obtained from experiments and CFD. It can be seen that the experimental and numerical results agree reasonably well with each other, with deviation no more than 8.1%.
Figure 9 Local Nusselt number distribution from CFD results on target plates at \( Re_j = 30,000 \)

Figure 10 Spanwisely averaged Nusselt number on flat plate obtained from experiment and CFD at \( Re_j = 30,000 \)

Figure 11 Three-dimensional pathlines in the jet impingement system at \( Re_j = 30,000 \)

Figure 11 shows three-dimensional pathlines between the third and fourth jet in the impingement cooling system. From Figure 11 (a) it can be seen that on the surface of flat plate wall flow firstly washes against the endwall surface, then collides with the upstream wall flow produced by the downstream jet, which makes it separates from the wall and forms a vortex. From Figure 11(b) it can be known that on the surface of pin fin plate the initial formation stage of wall flow is similar to that of flat plate, however, a horseshoe vortex is generated near the end of pin fin and the heat transfer in this area is strengthened. Then the horseshoe vortex develops downstream and forms another vortex at the edge of downstream stagnation region. Figure 11(c) makes it clear that on the surface of pin fin plate with effusion holes the flow field shows a different pattern. The collision of the upstream and downstream jet occurs almost in the middle of two stagnation regions, and the downstream jet can also impinge onto the lower parts of upstream pin fins and enhances the heat transfer in this region.

Figure 12 shows distributions of absolute velocity ratio and streamlines in the longitudinal central plane at \( Re_j = 30,000 \). The effect of cross flow is not obvious for jet impingement onto the pin fin plate with effusion holes, because the existence of effusion holes reduces the cross flow in downstream region obviously. For pin fin plate with effusion holes, the vortex is closer to the pin fin surface and impingement plate compared with the flat plate and pin fin plate, indicating that the crossflow effects are reduced.

Due to the influence of cross flow, the mass flow of different impingement holes is different. As can be seen from Figure 13, the flow rate reaches the maximum value at the fourth impingement hole and is relatively lower at the third and fifth holes.
The contributions to the work by Dr. Chaoyi Wan is appreciated.

**NOMENCLATURE**

- \( D_j \) Impingement jet diameter [mm]
- \( D_p \) Pin fin diameter [mm]
- \( D_e \) Effusion hole diameter [mm]
- \( H \) Jet-to-plate spacing [mm]
- \( P_x \) Pin fin’s streamwise spacing [mm]
- \( P_y \) Pin fin’s spanwise spacing [mm]
- \( c \) Heat capacity [W/(kg·K)]
- \( \bar{h} \) Heat transfer coefficient [W/(m²·K)]
- \( \lambda \) Fluid conductivity [W/(m·K)]
- \( \rho \) Density [kg/m³]
- \( \nu \) Dynamic viscosity coefficient [m²/s]
- \( z \) Coordinate along the plate depth [m]
- \( \text{Re}_j \) Reynolds number, based on jet diameter
- \( Nu \) Local Nusselt number, based on jet diameter
- \( \overline{Nu} \) Spanwisely averaged Nusselt number
- \( \overline{Nu} \) Area-averaged Nusselt number
- \( t \) Time [s]
- \( T_w \) Wall temperature [K]
- \( T_B \) Jet airflow temperature [K]
- \( T_0 \) Initial wall temperature [K]
- \( \Delta P \) Pressure loss [pa]
- \( V_j \) Jet inlet velocity [m·s⁻¹]

**REFERENCES**


