CHAPTER 5

Impingement/Effusion Cooling Methods in Gas Turbine

Hyung Hee Cho & Beom Seok Kim
Department of Mechanical Engineering, Yonsei University, Korea.

Abstract

The thermal efficiency of gas turbine engines is strongly dependent on the turbine inlet temperature. The inlet temperature is limited by potential structural failure of hot engine components, which is mainly attributable to high thermal stresses and material strength reductions due to high wall temperatures. To extend the life of the hot components in gas turbine systems, various cooling methods have been developed over the past several decades. In this chapter, we discuss the fundamentals of an impingement/effusion cooling method that has recently been developed for use under high heat flux conditions. The cooling mechanisms and the flow and heat transfer characteristics are explained as determinants of the cooling performance and component reliability. Since impingement/effusion cooling is a combined cooling technique, it can be controlled by a number of factors, including the hole pattern, angle of the effusion holes, target surface configuration, and gap spacing. Accordingly, we classify the principal variables affecting cooling performance, and explain how heat transfer and fluidic characteristics can be controlled by each variable based on thermo-physical aspects. In addition, various applied techniques involving surface modifications, such as curved surfaces and combined rib and pin-fin structures, are discussed in this chapter. We review the effects of the impingement/effusion cooling systems, and present a detailed account on experimental approaches as well as numerical analyses regarding heat and mass transfer. The development process for an impingement/effusion cooling system is described, together with suggested directions for further advances in this field.

Keywords: Gas turbine cooling, impingement/effusion cooling, impingement cooling, effusion cooling, heat transfer.
1 Introduction

Impingement/effusion cooling is an advanced cooling scheme that has recently been applied to gas turbine engines. In this technique, the inner surfaces of the hot components, such as combustor liners and blade walls, are cooled by the impingement of cooling air. The outer surfaces, which are in contact with hot gases, are protected by effusion film cooling. This scheme involves two flow streams: jet impingement on a target plate and effusion flow through holes in the target plate. In addition, a concomitant crossflow can be employed in the internal passages of an actual cooling system for a combustor or a turbine blade. The diagrams of Fig. 1 illustrate impingement/effusion cooling of a turbine blade and a combustor. Since cooling performance is considered in the structural design of these components, it is important to know the consequent heat transfer and flow characteristics in a complicated system. For improved cooling performance of hot components via impingement/effusion cooling, we must therefore have insight into both the overall and local heat transfer characteristics in terms of various performance factors.

Specific impingement/effusion cooling system configurations are characterized by geometrical features, such as the jet plate-to-effusion plate distance, the injection hole diameter, the jet-to-jet spacing, and the hole pattern. Figure 2 shows a diagram (with nomenclature) of the geometrical configuration of a typical impingement/effusion cooling system. In this system, we define two principal variables

Figure 1: Impingement/effusion cooling of a turbine blade (left) and a combustor (right).

Figure 2: Diagram of impingement/effusion cooling in a staggered array.
that influence the flow and heat transfer characteristics: the ratio $H/d$ of the gap spacing to the hole diameter (which represents the distance between the jet injection plate and the effusion plate) and the jet-to-jet spacing (which represents the conformal distance between adjacent impingement jets). Aside from these, other variables can also affect the flow and heat transfer characteristics.

Array jet impingement is a common method of cooling hot solid surfaces. The heat transfer under an array of impinging jets is generally superior to that achieved by typical convective heat transfer methods. The array jet impingement cooling has an advantage that it is easy to adjust the location of interest and dissipate a large amount of heat effectively. However, a crossflow is formed by the spent air from an array of impinging jets in a confined space, and the amount of crossflow increases as the flow moves downstream. According to numerous studies on array jet impingement, this crossflow degrades heat transfer performance. Impingement/effusion cooling systems have been studied to reduce the crossflow effect and improve the heat transfer. Of course, diverting air for cooling purposes has the effect of reducing the engine efficiency. Hence, it is essential to develop schemes that can guarantee maximum thermal protection with a minimal flow of coolant. Researchers have studied the effects of numerous variables on impingement/effusion cooling. Nazari and Andrews [1], Al Dabagh et al. [2], and Andrews and Nazari [3] examined the effect of the number of holes on film cooling performance in impingement/effusion cooling. Cho and Goldstein [4] investigated the effect of the hole arrangement on local heat/mass transfer characteristics inside the effusion plate. They found that a high transfer rate is induced by strong secondary vortices and flow acceleration, and that the overall transfer rate is approximately 45%–55% higher than with impingement cooling alone. Cho and Rhee [5], Cho et al. [6], and Ekkad et al. [7] investigated the heat/mass transfer and flow characteristics of an impingement/effusion cooling system with respect to various experimental conditions, such as gap distance, Reynolds number (Re), hole arrangement, and size. To enhance the cooling performance in an impingement/effusion cooling system, Funazaki et al. [8, 9] and Yamawaki et al. [10] investigated the effects of circular pin-fins on impingement/effusion cooling. Taslim et al. [11] examined the behavior of impingement/effusion cooling on roughened walls with bumps. Metzger and Bunker [12] and Hong et al. [13, 14] investigated the effects of surface curvature on impingement/effusion cooling. Thus, heat transfer in impingement/effusion cooling methods can be enhanced in terms of various parameters. Moreover, the heat transfer characteristics of impingement/effusion cooling are closely related to those of impingement jet cooling.

It is necessary to understand heat transfer in a cooling system with crossflow, since crossflow inevitably occurs in the internal passages of actual combustor or turbine blade cooling operations. Rhee et al. [15] studied the effect of crossflow on the heat transfer characteristics of an array of impinging jets, and compared the results with the cases of effusion holes on the injection plate. Ekkad et al. [7] examined the effect of the crossflow orientation. They showed that the crossflow orientation significantly affects the heat transfer distribution, even when the crossflow rate is low. Rhee et al. [16] conducted an experimental investigation of the
effects of crossflow on the heat transfer. They reported that the overall heat/mass transfer rate on the effusion (target) plate decreases as the crossflow rate increases, which is similar to the trend for array jet impingement. To enhance the heat transfer coefficients and improve the uniformity of the heat transfer, Rhee et al. [17] and Hong et al. [18, 19] combined ribs and pins on the target surface. Their results indicate that surface modifications via specialized turbulators are effective for enhancing heat transfer by reducing the negative effect of crossflow.

2 Heat Transfer of Impingement/Effusion Cooling

2.1 Fundamentals of impingement/effusion cooling

2.1.1 Basic concepts of impingement/effusion cooling

Among the various cooling schemes, transpiration cooling is known to be the most efficient method, since coolant flow through a porous material can guarantee uniform heat transfer characteristics on the surface. However, porous material has mechanically weak structures and cause clogging problems. To resolve/reduce these problems, a cooling method using two perforated plates (an injection plate and an effusion plate) has been developed for impingement/effusion cooling. A detailed diagram of this impingement/effusion cooling method is shown in Fig. 3. The inner surfaces of the effusion plate, such as combustor liners or blade walls, are cooled by the impingement of cooling air. The outer surfaces of the effusion plate, which are exposed to hot gas, are protected by an effusion film cooling air. Thus, the technique involves two flow schemes: jet impingement on a plate and effusion flow through the holes of an effusion plate.

2.1.2 Heat transfer characteristics of array jet impingement

In array jet impingement, the fluidic characteristics can be described as shown in Fig. 4. At a stagnation point of the jet stream, the heat transfer can be characterized in terms of the fundamental fluid dynamics of a single jet. Indeed, the heat transfer and fluid flow characteristics in the stagnation region are similar to those of a single jet application. As the impinged jet flow spreads out in all directions from the stagnation point on the surface, a wall jet is created and then a wall jet boundary

![Figure 3: Conceptual diagram of the impingement/effusion cooling method.](https://www.witpress.com/issn_1755-8336/on-line)
layer is developed. Depending on the thickness of this boundary layer, the wall jet flow changes from laminar to turbulent, and the heat transfer will be increased in the transition region. These phenomena appear with both a single impinging jet and array impingement jets. However, the array impingement jets have different heat transfer characteristics due to the flow interactions between adjacent wall jets. As Fig. 4 indicates, a pair of secondary vortices could be induced by the creation of primary vortices due to wall jet interactions. At first, the wall jets spread out from the stagnation points of the impinging jets. Wall jets from adjacent impinging jets collide, generating primary vortices. The primary vortices flow upwind, and heat transfer is enhanced in the region between the wall jets owing to the secondary vortices. These fluidic phenomena are dependent on the ratio of the gap spacing to the hole diameter. We will discuss the heat transfer characteristics in detail in Section 3.2.

The heat transfer and flow characteristics of array impingement jets are significantly affected by crossflow. Figure 5 shows a diagram of crossflow generation. This crossflow is generated by the spent air flow supplied from upstream impingement jets, and makes stagnation points steering to crossflow direction. Collisions between the crossflow and the impinging jet flow produce horseshoe-like vortex, and thus generate a nonuniform heat transfer distribution. This can reduce the overall heat transfer coefficients and uniformity. Therefore, effusion holes and/or additional obstacles are employed to reduce the crossflow and increase the local turbulence of the flow, respectively. It has been verified that this modified cooling

![Figure 4: Fluidic structures from array impingement jets [5].](image1)

![Figure 5: Crossflow from the spent air in array impingement jets.](image2)
method produces a uniform heat transfer distribution for array impingement jets with a crossflow [15, 17, 18]. We will introduce the effect of effusion holes in the next section.

2.2 Comparison of impingement jet and impingement/effusion cooling

To compare impingement jet and impingement/effusion cooling, we describe some exemplary approaches for evaluating the heat transfer performance via an array jets. Rhee et al. [15] tried to investigate the effects of spent air flow on heat/mass transfer, with and without effusion holes, in array impingement jets. They presented a quantitative assessment of the local cooling performance using contour plots of the Sherwood number (Sh) for array jet impingement and impingement/effusion cooling at Re_d = 10,000, as shown in Fig. 6. For array impingement jets without effusion holes, the crossflow generated by spent air has a significant influence on the heat/mass transfer characteristics of the target plate for small gap distances (H/d ≤ 2.0, where H and d denote the gap between the plates and the hole diameter, respectively). For small gap distances, the heat/mass transfer coefficients at the stagnation points increase as the flow moves downstream, since the turbulent intensity of the jet core is increased by the interaction between the impinging jets and the crossflow stream. The heat/mass transfer coefficients in the middle region

Figure 6: Contour plots of Sherwood numbers for array impingement jets (left) and impingement/effusion (right) cooling at Re_d = 10,000 [15].
are reduced by the re-entrainment of spent air and the development of a thermal boundary layer. However, for large gap distances \( H/d \geq 4.0 \), the crossflow has little effect on the heat/mass transfer of the target plate due to the large cross-sectional area of the flow path.

For the array impingement jets with effusion holes on the same plate, uniform and periodical heat/mass coefficients are obtained on a target plate. In particular, the overall transfer performance of \( Sh \) increases according to the decrease of \( H/d \), since the spent air can be discharged through the effusion holes in the injection plate, and flow re-entrainment can be significantly reduced. The effects of crossflow, which are strongly dependent on \( H/d \), are evaluated quantitatively in Fig. 7. The effect of the effusion holes is clearly indicated for small gap distances \( H/d < 2.0 \), while the crossflow effect is diminished for large gap distances. Thus, impingement/effusion cooling provides higher average heat transfer and a more uniform heat transfer distribution than array jet impingement for small gap distances. From those results, we can demonstrate that heat transfer characteristics are strongly affected by the geometrical variable \( H/d \) and the presence of effusion holes.

3 Major Variables for Impingement/Effusion Cooling

3.1 Effect of hole pattern and arrangement

Impingement/effusion cooling combines impingement jet cooling and effusion cooling, and uses two perforated plates: an injection plate and an effusion plate. In the design process, the plates must be properly arranged to obtain the desired local and overall cooling performance. The effusion holes play two main roles: they increase the local heat transfer coefficients and improve the uniformity of the heat transfer on the overall surfaces by removing the effects of crossflow. Therefore, the hole arrangement is an important factor in the construction of an impingement/effusion
cooling system. In this section, we explain the effect of the hole pattern on the local/overall heat transfer performances.

Many researchers have focused on the evaluation of heat transfer characteristics in terms of hole arrangement. Hollworth and Dagan [20] and Hollworth et al. [21] compared the heat transfer characteristics of staggered and inline arrangements of holes. Cho and Rhee [5], Cho et al. [6], and Rhee et al. [22] evaluated the heat transfer characteristics and tried to improve local performance with various hole patterns, including staggered, shifted, inline, square, and hexagonal arrangements, as illustrated in Fig. 8. Based on these previous studies, we can infer that the thermo-fluidic behavior of effusion streams through effusion holes is dependent on the types of effusion hole arrangement, as is the resulting local/overall cooling performance.

As a representative result, Fig. 9 shows contour plots of Sh for staggered, shifted, and inline hole arrangements under the constraints Re$_d$ = 13,500 and H/d = 1.0 [6]. With the staggered arrangement, great enhancement of heat/mass transfer is achieved by jet impingement, flow acceleration into the effusion holes, and wall jet interaction in the middle regions. In particular, the peak values are even higher in the middle region than at the stagnation points, mainly because the small hole spacing reinforces the wall jet effects. As a result, the overall heat transfer rate is remarkably improved. With the shifted arrangement, low heat transfer regions are formed around the effusion holes, and the additional peaks in the middle region are lower than those of the staggered arrangement due to the geometry. Also, when the gap distance is small, an asymmetric distribution is formed due to the instability of the vortex movement; however, there are no asymmetric patterns
with the larger gap distance or hole spacing. With the inline arrangement, the Sh levels are much lower than those of the other hole arrangements, since most of the impinging flow is discharged directly.

The arrangement of jet or effusion holes is an important design factor for local heat transfer performance. Figure 10 shows the flow and vortex characteristics of impingement/effusion cooling with various hole arrangements. The staggered array is a candidate for improved heat transfer characteristics, particularly between injection holes and effusion holes. Peak values of the convective heat transfer coefficients are observed near stagnation regions, and square cells with high heat transfer coefficients are formed around the impingement jet holes due to interactions between adjacent wall jets. These flows are exhausted to the effusion holes, and accelerate when approaching the effusion holes so that heat transfer is slightly increased at these points.

The hole number ratio is defined as the ratio of the number of impinging jet holes to the number of effusion holes per unit area. The hole pattern is subject to the hole number ratio. Unlike staggered, shifted, and inline arrangements, square and hexagonal arrangements can have different hole number ratios. The hole number ratios of the staggered, square, and hexagonal arrays shown in Fig. 8 are 1:1, 3:1, and 2.5:1, respectively. The local heat transfer characteristics are shown in Fig. 11 [22]. The local Sh has higher values with the staggered arrangement than with the square and hexagonal arrangements, due to the high impinging jet flow velocity, which is related to the low hole number ratio. However, spatial difference of heat transfer coefficients in the middle and center region of the impinging jets, and they result in unfavorable uniformity of the heat transfer. With the square arrangement, although the overall Sh levels are lower than those of the staggered arrangement, a uniform heat transfer distribution is obtained because the number of injection holes is increased. Equilateral triangular cells with high heat transfer rates are formed with the hexagonal arrangement, and a fairly uniform heat transfer distribution is also obtained in this case.
Figure 10: Flow and vortex patterns in impingement/effusion cooling for the various hole arrangements [6].

Figure 11: Contour plots of Sherwood numbers for (a) staggered, (b) square, and (c) hexagonal arrays with $H/d = 1.0$ and $Re_d = 10,000$ [22].
Figure 12 shows the average Sh for the different hole arrangements at various Reynolds numbers. The staggered arrangement has more efficient heat transfer characteristics than the other hole arrangements. However, the staggered arrangement is hampered by lower uniformity than the square and hexagonal arrangements. In light of these disparities of the overall and local heat transfer characteristics, we must therefore carefully consider the requirements and then select a suitable hole pattern and hole number ratio when designing an impingement/effusion cooling system.

3.2 Effect of plate spacing

The ratio $H/d$ of plate spacing to hole diameter represents the gap distance between the injection plate and the effusion plate. This factor directly affects the distribution of heat transfer coefficients in the impinging jet region, since the distance between the jet and the effusion plate is related to the flow momentum and the development of downstream flows after impingement. It has already been demonstrated in many studies that the overall heat transfer also increases with decreasing $H/d$ [23, 24].

Figure 13 shows contour plots of the Sh distribution for various $H/d$ [5]. The ratio of plate spacing to hole diameter is controlled between 0.33 and 10.0 as a principal variable. The local Sh patterns have no significant disparities and show good symmetry at $x/d = 0.0$ in all cases. For small gap distances, the interaction between adjacent impinging jets is very weak. The flow in the gap spacing develops from laminar to turbulent like a flow developing in a duct. The local heat/mass

![Graph](image-url)
transfer coefficient then reaches a secondary peak in the region apart from the stagnation point by 1.5\(d\).

However, for large \(H/d\) (\(\geq 2\)), the turbulent intensity of the jet core increases. Therefore, the heat transfer in the stagnation region increases with the gap spacing. The heat/mass transfer at the stagnation point reaches the highest value at \(H/d = 6.0\). However, when \(H/d\) is large, the secondary peak disappears and the heat/mass transfer decreases monotonically since the wall jet is already transitioning to the turbulent regime outside the stagnation region.

\(H/d\) is also a factor affecting local performance, as Fig. 14 indicates. Figure 14 (a) shows the local heat transfer distributions for \(z/d = -3.0\) (along the stagnation line). In Fig. 14 (a), there are two peak values at \(x/d = \pm 3.0\) and \(\approx \pm 1.5\) when the gap distance is small, and these peak values are maximized at \(H/d = 0.33\). As the gap distance increases, the heat transfer at \(x/d = \pm 3.0\) increases, and the heat transfer at \(x/d = \pm 1.5\) decreases. The Sh values exhibit uniform distributions in the middle region (Fig. 14 (b)), and these values are almost the same for moderate gap distances due to the interactions and flow acceleration. However, the heat/mass transfer rates are about 25\% lower for \(H/d = 0.33\) compared with the other cases because of the weak interactions between the wall jets. Table 1 lists the average Sh for different gap distances at \(Re_d = 10,000\). The average Sh values are 6–10 times higher for impingement/effusion cooling than for a single layer with effusion alone. The average Sh has a maximum value at \(H/d = 0.33\), and decreases with increasing gap distance. Although the average Sh attains the highest value for \(H/d = 0.33\), the slope of the local Sh distribution is very steep. Therefore, from the point of view of thermal stress, impingement/effusion cooling with an extremely small gap distance is not recommended.

### 3.3 Effect of Reynolds number

The Reynolds number is one of the principal variables in heat transfer. In an actual gas turbine, the mass flow rate of the coolant is constrained by the operating conditions for each component. The mass flow rate of coolant is typical factor affecting the overall/local heat transfer performance in impingement/effusion cooling schemes. Increasing Re simply stands for the increase of the mass flow...
rate of the coolant. Thus, the heat transfer coefficient can be increased by reinforcing the fundamental flow momentum, which is proportional to Re. It has been shown that the local heat transfer characteristics in impingement/effusion cooling can also be improved by controlling Re.

The local heat/mass transfer distribution is characterized by Re with $H/d = 1.0$, as shown in Fig. 15 [5]. The Sh distributions exhibit a similar pattern for all Reynolds numbers. However, the quantitative levels and secondary peak values of Sh are clearly dependent on the Reynolds number, increasing linearly with it. Figure 16 shows the local Sh distributions along $x/d = 1.5$ for a staggered hole arrangement at $H/d = 2$ for various Reynolds numbers [6]. Generally, the heat/mass transfer coefficient at a stagnation point is proportional to $Re^{0.52}$ for a simple nozzle.

Figure 14: Local Sherwood number distributions for various gap distances [5].

Table 1: Average Sherwood numbers for different gap distances with a staggered array holes and $Re_d = 10,000$ [5].

<table>
<thead>
<tr>
<th>$H/d$</th>
<th>0.33</th>
<th>1.0</th>
<th>2.0</th>
<th>6.0</th>
<th>10.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\bar{Sh}$</td>
<td>100.8</td>
<td>96.0</td>
<td>92.8</td>
<td>82.9</td>
<td>68.2</td>
</tr>
</tbody>
</table>
Figure 15: Local plots of Sherwood number for different Reynolds number with $H/d = 1$ in a staggered array [5].

Figure 16: Distribution of Sherwood number along $x/d = 1.5$ on the effusion surface with a staggered hole arrangement and $H/d = 2.0$ [6].

geometry and a gap distance of $2d$. The results normalized by $Re^{0.52}$ are also presented. The left-hand graph shows the raw Sh distributions, and the right-hand graph shows the normalized Sh distributions. As the left-hand graph clearly indicates, the Sh levels increase continuously with Re, and the patterns are similar for all $Re_d$. When we compare the normalized values, some interesting features are
observed related to Re, particularly in the laminar-to-transient region. Although Re is different for each case, the normalized values around the stagnation point are virtually identical. However, there are some discrepancies between the normalized values in the middle region (laminar-to-transient). In other words, the values are about 20% lower in the low Reynolds number region (Re_d ≤ 5070) than in the high Reynolds numbers region (Re_d ≥ 8,090). In addition, the minimum peak values are at x/d = 0.25 for the lower Reynolds numbers, and at x/d = 0.35 for the higher Reynolds numbers. This means that Re affects the heat/mass transfer primarily in the middle region, and its effect is different from that observed in the stagnation region. This is possibly due to differences in the strength of adjacent wall jets and the interactions between them.

Overall average Sh values are shown in Fig. 17 for three hole arrangements with H/d = 2.0. Although the effusion hole arrangements are different, the catalyzed/accelerated fluidic motions obtained by increasing the Reynolds number lead to improved heat transfer in each case. In particular, it is possible to formulate correlations of the following type for the overall heat transfer characteristics in the confined regions:

\[ \overline{Sh} = C \text{Re}_d^a \]  \hspace{1cm} (1)

where C and a are constants subject to the structural design variables. For example, for Re_d = 5,000–12,000 in a staggered jet pattern with H/d = 1.0, the overall average Sh on the target plate can be modeled by the equation

\[ \overline{Sh} = 0.562 \text{Re}_d^{0.552} \]  \hspace{1cm} (2)

Figure 17: Area-averaged Sherwood number for impingement/effusion cooling with various hole arrangements and H/d = 2.0 [6].
For Reynolds numbers ranging between 3,000 and 13,500 and $H/d = 2.0$, the overall average $Sh$ value on the effusion plate can be expressed as follows for different hole arrangements. For a staggered hole arrangement,

$$\overline{Sh} = 0.253 \, Re_0^{0.67} \tag{3}$$

For a shifted hole arrangement,

$$\overline{Sh} = 0.213 \, Re_0^{0.68} \tag{4}$$

For an inline hole arrangement,

$$\overline{Sh} = 0.254 \, Re_0^{0.61} \tag{5}$$

The overall average values are proportional to $Re^{0.67}$ or $Re^{0.68}$ for the staggered and shifted arrangements, whereas the heat/mass transfer coefficients in the stagnation region are relative to $Re^{0.52}$.

### 3.4 Effect of surface curvature

For actual gas turbine components exposed to high operating temperatures, there is a demand for increased local/overall heat transfer efficiency. This demand could be satisfied by applying impingement/effusion cooling to various blade regions. The impingement/effusion cooling method is particularly efficient to reduce the highly concentrated thermal loads along the leading edges of the blades.

The powerful merits of this cooling technique have led some researchers to focus on the development/improvement of impingement/effusion cooling systems. As a series of these processes, studies of heat transfer characteristics on curved surfaces have been conducted based on simplified surface conditions for the leading edges of actual turbine blades. Hong et al. [25] evaluated local heat/mass transfer performance on a curved effusion plate under impingement/effusion cooling conditions. They verified the effects of curved surfaces on fluidic and heat transfer characteristics by comparing the corresponding results obtained with a flat surface. Figure 18 shows the two different ducts used to investigate the effect of surface geometry, one with a flat surface and the other with a concave surface [25]. Note that the effusion holes on the concave surface are slightly shifted to the outside with respect to the spanwise direction ($z/d$) compared with the flat surface.

Figure 19 shows the resulting $Sh$ contour plots on the flat and concave surfaces. The overall heat/mass transfer distributions on the flat and concave surfaces are similar. However, there are some disparities between the $Sh$ distributions due to the differences in surface geometry. In the stationary case in particular, the $Sh$ variations are more uniform on the concave surface than on the flat surface. In the outer region ($z/d > 3.0, z/d < -3.0$), relatively high $Sh$ values are produced by the curvature effects.

Figure 20 compares the local $Sh$ values of the flat and concave surfaces at $z/d = 0.0$ to those at $x/d = 3.0$. These plots clearly illustrate the detailed effects of the surface geometry. The concave surface exhibits a peak value similar to that of the flat surface along the stagnation line (Fig. 20 (a)). However, the influence of the
surface geometry is much more apparent along the spanwise direction ($z/d$), as shown in Fig. 20 (b). In light of interactions with other performance factors, the Sh differences between the concave and flat surfaces increase with $z/d$, and the concave surface is more efficient with higher heat/mass transfer values than the flat surface. This can be explained in terms of the curvature effect, which results in the generation of a thin boundary layer and increased flow mixing due to the difference in the flow path lengths.
Table 2 lists the quantitative results for the average Sh and uniformity values on the flat and concave surfaces. The average Sh is 10% higher on the concave surface than on the flat surface. To estimate the uniformity of the heat/mass transfer coefficients on flat and concave surfaces, the uniformity can be expressed by

\[ 1 - 0.01\sigma \]  

where \( \sigma \) is the standard derivation of the whole measured Sh data for each case. Based on the experimental results from reference [25], we conclude that a concave surface can provide enhanced uniformity compared with a flat surface. This is
because the local/overall Sh is improved along the spanwise direction by the curvature effect. Thus the curvature effect leads to better and more uniform heat/mass transfer on an effusion plate.

3.5 Effect of crossflow

In the actual cooling systems of combustor walls and turbine blades, crossflow exits through the internal passages. The main source of crossflow is spent air from the impingement jets or initial crossflow in the internal passages. Crossflow in the passages adversely affects impingement/effusion cooling since it deflects the impinging jets and sweeps away the wall jet flow by decreasing the momentum of the impinging jets. Therefore, to improve the cooling performance of these hot components, one must determine the basic heat transfer characteristics of impingement/effusion cooling with respect to the crossflow.

Most previous studies have dealt with the effects of crossflow in impingement cooling. Behbahani and Goldstein [26] and Metger and Korstad [27] investigated the effect of crossflow on the heat transfer/flow characteristics of array jet impingement. The heat transfer is less efficient in the upstream region due to initial crossflow. The stagnation regions of the individual jets, which are determined by the local heat transfer coefficients, move further downstream as the amount of crossflow increases with the accumulation of spent air from upstream jets. Based on these fluidic phenomena, it has been demonstrated that the presence of crossflow decreases the average heat transfer performance in comparison to that of a system without crossflow.

Periodic effusion holes can be used to reduce the unfavorable effects of crossflow. Crossflow can be discharged through effusion holes in an effusion plate, which helps improve the uniformity of the overall heat transfer on the plate. Accordingly, some researchers have investigated fluidics and heat transfer characteristics in impingement/effusion cooling systems with crossflow.

The blowing ratio is an important variable pertaining to the initial crossflow in impingement/effusion cooling. It is the ratio of the flow rate of the crossflow to that of the impinging jets, and is defined as

$$M = \frac{Q_c}{Q_i}$$

where $Q_c$ and $Q_i$ denote the flow rates of the crossflow and the injected jets or effused flow, respectively. Rhee et al. [16] verified the effect of the initial crossflow

<table>
<thead>
<tr>
<th>Surface</th>
<th>Stationary $Sh$</th>
<th>Uniformity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flat surface</td>
<td>33.1</td>
<td>0.90</td>
</tr>
<tr>
<td>Concave surface</td>
<td>35.6</td>
<td>0.93</td>
</tr>
</tbody>
</table>
in terms of various blowing ratios in an impingement/effusion cooling system. To investigate the effect of crossflow, the blowing ratio was varied from 0.5 to 2. Figure 21 shows a diagram of the injection and effusion hole arrangements. A staggered arrangement of holes between the injection and effusion plates was adopted, and the crossflow was injected through the passage, which is formed between the injection and effusion plates. Figure 22 shows the Sh distributions for different blowing ratios under the conditions \( Re_d = 10,000 \) and \( H/d = 2.0 \). The local Sh without crossflow has a coincidently repetitive pattern along the lateral direction (Fig. 22 (a)). The Shs are high around the stagnation points, and decrease with the development of the wall jet boundary layer after impingement.

For a low blowing ratio of \( M = 0.5 \) (Fig. 22 (b)), the Sh distribution is similar to that in the case without crossflow, since the effect of the crossflow is weak. However, the heat/mass transfer patterns for the cases with relatively high blowing ratios are totally different from those of the case without initial crossflow. The injected jets are deflected, and the wall jets are swept away by the crossflow. Therefore, the Sh contours have nonuniform and asymmetric distributions. Nevertheless, the Sh contours exhibit a periodic distribution except along the first row of injection holes, since a certain amount of the crossflow and spent air can be discharged through the effusion holes, and the balance between the flow entrance and exhaust is preserved. Additional peaks between the effusion holes, which are induced by the wall jets, appear in the cases with \( M = 0.0 \) and 0.5. On the other hand, heart-shaped low heat transfer regions are formed between the effusion holes without any additional peaks when \( M = 1.0 \) (Fig. 22 (c)), since the wall jets swept away by the crossflow are sucked into the effusion holes. However, the regions affected by the impinging jets have much lower heat transfer efficiency at the relatively high blowing ratios of \( M = 1.5 \) and 2.0. The heat/mass transfer rates on the effusion target plate generally decrease as the blowing rate increases due to the reinforced crossflow momentum and diminished effect of jet impingement.

Figure 23 shows the spanwise average Sh on the effusion plate for the various cooling methods. The impingement/effusion cooling technique achieves the highest heat transfer performance in all tested cases, especially as regard to the average Sh, because the effects of crossflow and flow re-entrainment of spent air are reduced by the installation of effusion holes. In array jet impingement, the blowing ratio of the crossflow increases with increasing \( x/d \), and the thermal boundary layer is thickened. Thus, the re-entrainment of spent air into the impinging jets also increases. As a result, the average Sh decreases and the peak values move
downstream as the fluid flows downstream. However, in impingement/effusion cooling, some of the crossflow and/or spent air is discharged through the effusion holes. Hence, the effects of crossflow and flow re-entrainment are diminished throughout the entire domain, except for the inlet region. Consequently, the average Sh distributions exhibit periodic patterns, and the peak Sh levels are almost the same over the entire domain with or without initial crossflow.

Figure 22: Contour plots of Sherwood number for various blowing ratios with $Re_d = 10,000$ and $H/d = 2.0$ [16].
The overall average Sh values on the target plate are shown in Fig. 24 for various cooling methods and blowing ratios. Impingement/effusion cooling is 20%–30% more efficient for heat transfer than array jet impingement, and 3.5–9 times more efficient than convective cooling (crossflow only). For impingement/effusion cooling with an initial crossflow, the overall average Sh on the target surface can be modeled by the following equation, particularly in the confined regime of $M = 0.5–2.0$ with $Re_d = 10,000$ and $H/D = 2.0$:

$$\overline{Sh} = 70.86M^{-0.304}$$  (8)

According to the correlated results, we can infer that the heat transfer rate decreases monotonically with increasing blowing ratio. Therefore, it is necessary to reduce the crossflow rate to improve the overall cooling performance.

3.6 Effect of surface modification

As we reviewed the unfavorable effects of crossflow on local/overall heat transfer, we must consider crossflow behavior in the design of cooling systems. However, it is not easy to regulate the exact amount of crossflow in a cooling system. How
much one can reduce the low heat transfer regions is the key to enhance the cooling performance of a cooling system with crossflow under confined operating conditions. The installation of rib turbulators or pin-fin structures on the target or effusion surface shows promise for improving the cooling performance with small geometrical alterations.

3.6.1 Rib turbulators

Rib turbulators block partial flow streams and break the boundary layer near the wall. These characteristics help to enhance the heat transfer on a wall. The beneficial effects of rib turbulators on heat transfer have led some researchers to incorporate rib turbulators in impingement/effusion systems to reduce the crossflow effect. Haiping et al. [28] performed experiments on jet impingement cooling using a rib-roughened surface. Andrews et al. [29] investigated how impingement cooling is affected by the orientation of transverse ribs with respect to the crossflow. They reported that rib turbulators significantly reduce the effects of crossflow, and that heat transfer can be enhanced by the presence of ribs, especially ribs oriented normal to the crossflow.

Rhee et al. [17] studied the effect of rib arrangements on flow and heat/mass transfer characteristics in an impingement/effusion cooling system with initial crossflow. As shown in Fig. 25, five different rib turbulator configurations were installed on effusion plates. The numerals (90 and 45) and characters (D, U, V, and A)
denote the attack angle of the ribs and the rib arrangement, respectively. D or U indicate that the ribs are installed downstream or upstream, respectively, from the effusion hole, and V or A indicate that the ribs are concave or convex, respectively, in the downstream direction at the centerline of the effusion holes. Figure 26 shows Sh contour plots for impingement/effusion cooling with the various rib configurations at $M = 1.0$, including additional contour plots for rib attack angles of 90° and 45°. According to Fig. 26 (b–d), the stagnation regions have almost the same locations with 90° ribs and without ribs. However, local enhancements by the rib turbulators make the overall heat/mass transfer distributions more uniform. In addition, the low heat transfer regions are locally weakened and removed by the ribs, although the effects of jet impingement are dominant over the entire domain. This is because the ribs act as a block against near-wall crossflow, creating a large circulating flow, so that the wall jets spread more widely in the upstream and lateral directions, and cover those regions effectively. Figure 26 (e) and (f) show Sh contour plots for rib configurations with 45° angles. The overall heat/mass transfer features on the effusion surface are slightly different from those obtained with the 90° rib configurations, since the effects of jet impingement and flow effusion are dominant in this cooling system. The positions and levels of the stagnation points are almost the same as those of the 90° cases, and relatively high heat/mass transfer is achieved around the effusion holes, since the flow is accelerated and disturbed in that region. However, additional local Sh peaks and low heat transfer regions are formed due to the secondary flow induced by the angled ribs.

Table 3 lists the area-averaged Sh values for impingement/effusion cooling with the various rib configurations at $M = 1.0$. The rib turbulators enhance heat transfer by 4%–15% in comparison to the case without ribs. The 90DU case has the highest heat/mass transfer coefficients among the various rib configurations, since the shorter rib spacing efficiently protects the jet cores against the crossflow. However, the average values are similar to those from the cases with rib except for 90DU. It means that the attack angle of the ribs is not a particularly significant parameter for average heat/mass transfer characteristics in an impingement/effusion cooling system with initial crossflow. Therefore, rib turbulator position could be an important parameter for controlling the initial crossflow in an impingement/effusion cooling system.
3.6.2 Pin-fins

Circular pin-fins are considered a possible means of reducing the adverse effect of crossflow, and improving both cooling performance and durability. They also offer the advantages of being conductive heat spreaders and promoters of the structural strength of the system. However, there are not enough studies of pin-fins.
in impingement/effusion cooling. Andrews et al. [30] and Annerfeldt et al. [31] carried out experimental investigations of pin-fins, but these were confined to jet impingement cooling. Funazaki et al. [8, 9] and Yamawaki et al. [10] examined the effects of circular pin-fins on impingement/effusion cooling.

Hong et al. [18] studied the effects of pin-fins on heat/mass transfer characteristics in impingement/effusion cooling with an initial crossflow. Five different pin-fin arrangements are shown in Fig. 27, referred to as CP0, CP1, CP2, CP1U, and CP1D, respectively. Figure 28 shows contour plots of Sh for impingement/effusion cooling with pin-fins at $M = 1.0$. When circular pin-fins are installed between the injection and effusion plates, they tend to generate vortices and wakes, which are efficient to reduce the local low heat/mass transfer regions. Moreover, the pin-fins prevent wall jets from being swept away, increasing the local heat transfer in the injection region. However, pin-fins produce lower average Sh values than rib turbulators due to partial blockage effects.

Table 3: Overall average Sherwood number for the various rib configurations at $M = 1.0$ [17].

<table>
<thead>
<tr>
<th>Rib configuration</th>
<th>Array impinging jets</th>
<th>Impingement/effusion cooling</th>
<th>Enhancement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without ribs</td>
<td>60.8</td>
<td>71.0</td>
<td>~15%</td>
</tr>
<tr>
<td>90D</td>
<td>–</td>
<td>77.4</td>
<td>–</td>
</tr>
<tr>
<td>90U</td>
<td>–</td>
<td>78.3</td>
<td>–</td>
</tr>
<tr>
<td>90DU</td>
<td>72.7</td>
<td>81.4</td>
<td>~11%</td>
</tr>
<tr>
<td>45V</td>
<td>63.5</td>
<td>78.5</td>
<td>~20%</td>
</tr>
<tr>
<td>45A</td>
<td>–</td>
<td>76.5</td>
<td>–</td>
</tr>
</tbody>
</table>

Figure 27: Diagrams of the various circular pin-fin arrangements [18].
Figure 28: Contour plots of Sherwood number for impingement/effusion cooling with circular pin-fins at $M = 1.0$ [18].
The alteration of pin-fin positions along the centerline of the injection holes directly affects the flow path of the impinging jets and wall jets, resulting in the change of local heat/mass transfer distribution. When pin-fins are installed in front of the impinging jets, the blockage effect on the crossflow enhances the heat/mass transfer. However, when pin-fins are installed just behind the impinging jets, they block the wall jet and decrease the heat/mass transfer.

By installing circular pin-fins, the overall heat/mass transfer can be improved by 5%–11% in comparison to the case without pin-fins at \( M = 1.0 \) (as shown in Fig. 29). Increasing the number of pin-fins leads to augmentation of the overall Sh value. However, despite the presence of pin-fins, the CP0 and CP1D cases exhibit little enhancement of the overall Sh value. Even though pin-fins are effective for enhancing local/overall heat transfer, they also lead to increased pressure drop. As seen from Fig. 30, the pressure drop is clearly dependent on the types of installed pin-fin structures. The pressure drop through the channel becomes greater as the number of installed pin-fins and the blowing ratio increases. Although the pressure loss through the channel is less than 10% of what is experienced by perforated plates without pin-fins, the pressure loss for the CP2 case is approximately four times higher than that of the case without pin-fins. Compared with rib turbulators, pin-fins incur a pressure loss that increases steeply with respect to the blowing ratio.

Based on these previous studies, we can summarize that the installation of additional structures effectively reduces the influence of crossflow, and leads to enhanced heat transfer. However, it is also accompanied by a pressure loss through the channel due to the increased blockage area. Therefore, you should simultaneously consider the respective pros and cons of heat transfer and pressure loss when incorporating turbulating structures into a design.

Figure 29: Overall average Sherwood number for various pin-fin arrangements at \( M = 1.0 \) [17, 18].
References


