Vortex Structure Effects on Impingement, Effusion, and Cross Flow Cooling of a Double Wall Configuration

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Abstract. A variety of different types of vortices and vortex structures have important influences on thermal protection, heat transfer augmentation, and cooling performance of impingement cooling, effusion cooling, and cross flow cooling. Of particular interest are horseshoe vortices, which form around the upstream portions of effusion coolant concentrations just after they exit individual holes, hairpin vortices, which develop nearby and adjacent to effusion coolant trajectories, and Kelvin-Helmholtz vortices which form within the shear layers that form around each impingement cooling jet. The influences of these different vortex structures are described as they affect and alter the thermal performance of effusion cooling, impingement cooling, and cross flow cooling, as applied to a double wall configuration.

1. Introduction

To provide insight into the effects of different vortex structures, as they affect and alter the thermal performance of effusion cooling, impingement cooling, and cross flow cooling, as applied to a double wall configuration, new cooling data are provided using a unique experimental facility which is designed and assembled at the Propulsion Research Center of the University of Alabama in Huntsville. The facility is designed to provide full coverage film cooling effectiveness data and impingement cooling effectiveness data, wherein the film cooling flow is supplied using either cross flow, impingement flow, or a combination of both together. The facility uses three independent flow channels to provide double wall cooling arrangements which model the same configurations from operating gas turbine engines. Presented are new data which illustrate the performance of a sparse effusion hole array which is supplied by two different configurations: (i) a cross flow arrangement, and (ii) an array of impingement jets.

For the effusion cooled surface, presented are spatially-resolved distributions of surface adiabatic film cooling effectiveness, and surface heat transfer coefficients (measured using transient techniques and infrared thermography). For the impingement or cross flow cooled surface, presented are spatially-resolved distributions of surface Nusselt numbers (measured using steady-state liquid crystal thermography). Of interest are techniques which enhance the cooling capabilities of effusion cooling, especially at cool side locations between entrances to individual effusion cooling holes. To produce
this cool side augmentation, impingement jet arrays at different jet Reynolds numbers, from 7930 to 18000, are employed. Experimental data are given for a ratio of jet-to-target plate distance to effusion hole diameter of 14, impingement plate thickness of 3.0 effusion hole diameters, and spanwise and streamwise impingement hole spacing such that coolant jet hole centerlines are located midway between individual effusion hole entrances. For the effusion cooling, streamwise hole spacing and spanwise hole spacing (normalized by effusion hole diameter) are 15 and 4, respectively. Effusion hole angle is 25 degrees, and effusion plate thickness is 3.0 effusion hole diameters. Considered are overall effusion blowing ratios from 3.3 to 7.4, with subsonic, incompressible flow.

Experimental data are presented for a freestream velocity variation, which is adjacent to the effusion cooled boundary layer, which is approximately constant with streamwise distance. With this configuration, the ratio of the inlet area to the exit area of the main stream flow duct is 1. Of particular interest are effects of impingement jet Reynolds number, effusion blowing ratio, streamwise development, and mainstream Reynolds number. Also included are comparisons of impingement jet array cooling results with results associated with cross flow supply cooling. As such, the present experimental results are employed to expand the range of applicability of current correlations, and for development of new design correlations for experimental conditions not previously considered. Also of interest are the effects of a variety of different types of vortices and vortex structures on thermal protection, heat transfer augmentation, and cooling performance of impingement cooling, effusion cooling, and cross flow cooling.

2. Experimental Apparatus and Procedures
Details on the experimental apparatus and procedures which are employed are provided by Rogers et al. [4] and Ren et al. [5,6]. Those discussions provide a complete description of relevant measurement details for the presently presented results. Figure 1 shows a side, cross-sectional view of the test section, including optical instrumentation arrangements, from Rogers et al. [4]. Here, the film cooling plate is double-wall cooled, with effusion cooling on the hot, mainstream side, and either cross flow or impingement cooling on the cold side.

![Figure 1. Side, cross-sectional view of the test section, including optical instrumentation arrangements, from Rogers et al. [4].](image)

3. Experimental Results With Cross Flow Cooling – Effusion Cooled / Hot Surface of Effusion Plate
The data within this section are obtained with cross flow cooling. Presented first are measured results for the effusion cooled / hot surface of the effusion plate.

Figure 2 presents the surface distribution of adiabatic film cooling effectiveness for a main flow velocity of 5 m/s, a Reynolds number of about 97,267, and an average blowing ratio of 3.4. The main flow temperature is 306K. These data are obtained for an extent of the test surface which spans more than four rows of effusion holes. The data in Figure 2 show that adiabatic effectiveness values are lower between and away from the holes and higher downstream of the film cooling holes. Figure 2 also shows overall increases in adiabatic effectiveness as x/d increases from 0 to 70, due to increased accumulations of film near to the downstream parts of the test surface.
Figure 3 gives a surface distribution of heat transfer coefficient in dimensional form for a main flow velocity of 5 m/s, a Reynolds number of about 93,642, and an average blowing ratio of 5.5. Here, heat transfer coefficients range from approximately 20 W/m²K to 140 W/m²K along the test surface. Higher heat transfer coefficient values are generally observed just upstream and around the holes. Lower heat transfer coefficient values are observed away from the holes. These variations are tied to local magnitudes of three-dimensional turbulent transport, which is associated with jet mixing and shear near jet edges.

Within Figures 4 and 5, the effects of varying blowing ratio on adiabatic film cooling effectiveness distributions and heat transfer coefficient distributions are presented and described for a sparse effusion cooling hole array supplied by a cross flow arrangement. The data in Figure 4 show that line-averaged adiabatic cooling effectiveness generally increases as blowing ratio increases, when compared at a particular x/dₑ location, indicating that surface protection is improved as film concentration increases near the surface. Also evident are larger variations with blowing ratio BR, and overall increases of adiabatic effectiveness, as the flow advects downstream, and x/dₑ increases. The average blowing ratios for the present data are 3.4, 4.4, 5.5, 6.4, and 7.3.

The data in Figure 5 show that line-averaged heat transfer coefficient values also generally increase as the blowing ratio increases from 3.4 to 7.3, when compared at a particular x/dₑ location. Lower values of heat transfer coefficient indicate less mixing between the film cooling gas and the hot main flow gas, which leads to better thermal protection of the surface from the hot main flow gas.

4. Experimental Results With Cross Flow Cooling – Cross Flow Cooled / Cold Surface of Effusion Plate

The data within this section are obtained with cross flow cooling. Presented here are measured results for the cross flow cooled / cold surface of the effusion plate.

Figure 6 gives the local, spatially resolved Nusselt number distribution with main flow velocity of 5 m/s, main flow Reynolds number of approximately 95,000, and a blowing ratio of 3.4. Figure 7 then presents the streamwise variation of line-averaged Nusselt numbers. Here, line-averaged Nusselt numbers increase with the blowing ratio from 3.4 to 6.4, and then remain approximately constant as the blowing ratio increases from 6.4 to 7.3, when compared at a particular x/de location. Note the increases of local and line-averaged Nusselt numbers in the vicinity of effusion hole entrances. Such behavior is due to increases of local flow advection speeds as cross flow fluid approaches and then enters into effusion hole entrances. Here, line-averaged Nusselt numbers are determined as the average of local, spatially resolved Nusselt numbers at a constant streamwise (x/dₑ) location, for a range of spanwise locations from y/dₑ = 0 to y/dₑ = 8.2.
Figure 3. Local, spatially-resolved surface heat transfer coefficient distribution with main flow velocity of 5 m/s, with a Reynolds number of around 93,642, a main flow temperature of 307 K, and a blowing ratio of 5.5, for hot / effusion cooled side of effusion plate, with coolant supplied by cross flow [5].

5. Vortex Structure Associated With a Single Film Cooling Jet in Cross Flow
Figures 8 and 9 show schematic diagrams of the vortices, which are associated with a single film cooling jet in cross flow. Included are jet wake vortices, jet shear-layer vortices, horseshoe vortices, and counter-rotating vortices (or kidney vortices). Note that an upwash region is located between the two vortices within the counter-rotating vortex pair. This upwash region often results in advection of film coolant concentrations away from the test surface, a phenomenon referred to as film coolant lift-off. One consequence of lift-off is decreased film coolant concentrations, and decreased thermal protection near surfaces, at locations downstream of film cooling holes. According to Walters and Leylek [7], this counter-rotating secondary flow vortex structure downstream of the jet exit is the most significant mechanism affecting film cooling performance. The dominant vorticity within this structure emanates from the film hole. Its origin is a result of streamwise oriented vorticity contained within the boundary layers which develop within individual film holes.

Figure 4. Streamwise variation of line-averaged adiabatic film cooling effectiveness with main flow velocity of 5 m/s, with a Reynolds number of around 95000, main flow temperature of 307 K, and the blowing ratios of 3.4, 4.4, 5.5, 6.4, and 7.3, for hot / effusion cooled side of effusion plate, with coolant supplied by cross flow [5].
Figure 5. Streamwise variation of line-averaged heat transfer coefficient with main flow velocity of 5 m/s, with a Reynolds number of around 95000, main flow temperature of 307 K, and the blowing ratios of 3.4, 4.4, 5.5, 6.4, and 7.3, for hot / effusion cooled side of effusion plate, with coolant supplied by cross flow [5].

Figure 6. Local, spatially resolved Nusselt number distribution with main flow velocity of 5 m/s, main flow Reynolds number of approximately 95,000, crossflow temperature of 294 K, and blowing ratio of 3.4, for coolant / cross flow side of effusion plate, with coolant supplied by cross flow [5].

Figure 7. Streamwise variation of line-averaged, spatially resolved Nusselt number for different blowing ratios with main flow velocity of 5.1 m/s, main flow Reynolds number of approximately 95,000, and crossflow temperature of 294 K, for coolant / cross flow side of effusion plate, with coolant supplied by cross flow [5].
6. Experimental Results With Impingement Jet Array Cooling – Impingement Cooled / Cold Surface of Effusion Plate

The data within this section are obtained with impingement jet array cooling. Presented first are measured results for the impingement cooled / cold surface of the effusion plate.

Figure 10 provides a comparison of local, spatially-resolved surface Nusselt number variations for the coolant / cross flow side of the effusion plate, for different blowing ratios, for \( \text{Re}_{\text{MS}} = 142,000-155,000 \). Here, data are given for \( \text{BR} \) values of 3.3, 4.3, 5.5, 6.3, and 7.4. Note that \( x/de = 0 \) is defined at the upstream edge of the area of present spatially-resolved measurements. Also included within Figure 10a are locations of the impingement holes as well as the effusion hole entrances.

According to the results within Figure 10, regardless of the value of blowing ratio \( \text{BR} \), the highest local Nusselt numbers are present at smaller \( x/de \) locations, relative to the locations of the impingement hole centerlines. Such \( \text{Nu} \) augmentation regions are associated with impingement jet stagnation locations, which are positioned at different \( x/de \) locations, relative to the impingement holes. This is because of turning and re-direction of the impingement jets as they cross the impingement passage. The re-direction is a result of the static pressure variations within the passage, with lowest values near the entrances of the effusion holes. With this physical situation, coolant within impingement jets exits the impingement holes, turns to smaller \( x/de \) locations, and then impacts upon the coolant side of the effusion plate. Afterwards, the coolant is then re-directed along the plate surface. Afterwards, it enters into individual effusion holes, which are also located at smaller \( x/de \) locations, relative to adjacent impingement hole locations.
The resulting coolant advection trajectory is likely aided by secondary flows within the impingement cooling passage. According to Al Dabagh et al. [1], internal flows within the impingement jet passage are complex, and are dominated by the counter-rotating vortices between individual impingement jets, with flows directed opposite to the jets, at locations between individual jets. Cho and Rhee [2] describe similar jet interactions, with secondary vortices which form between these strong primary vortices. According to these investigators, the strength and complexity of such interactions change dramatically, as the locations of impingement and effusion holes are altered, relative to each other.

Figure 11 shows line-averaged surface Nusselt numbers (determined from the results given in Figure 10), as they vary with streamwise development for different blowing ratios for Re_{MS}=142,000-155,000. Here, line-averaged magnitudes are determined over y/de values from 0.0 to 8.0. Within this figure, solid rectangles denote effusion hole entrance streamwise locations, and dashed rectangles denote impingement hole streamwise locations. Here, line-averaged Nusselt numbers often increase as the blowing ratio increases from 3.3 to 7.4, when compared at a particular x/de location. As for the results which are presented within Figure 10, Figure 11 shows that highest local Nusselt numbers are present at smaller x/de locations, relative to the locations of the impingement hole centerlines. The lowest local Nusselt numbers are then present at and near effusion hole entrance locations. Responsible is the turning of the impingement jets through the impingement passage, the collection of coolant along the surface (at x/de locations just larger relative to effusion hole entrances), followed by the entrance of the coolant into individual adjacent effusion holes.

Figure 12 then compares spatially-averaged surface Nusselt numbers, as they vary with streamwise development. These data are also presented for the coolant / cross flow side of the effusion plate, for different blowing ratios and Re_{MS}=142,000-155,000. Each spatial-average is determined over one complete period of impingement array hole pattern, which is equivalent to a streamwise extent of 15 effusion hole diameters. Here, spatially-averaged Nusselt numbers associated with x/de=42.5 increase continually as the blowing ratio increases from 3.3 to 7.4. Values associated with x/de=12.5 and x/de=27.5 increase as the blowing ratio increases from 3.3 to 5.5. However, afterwards, spatially-averaged Nusselt numbers are nearly invariant as the blowing ratio increases further from 5.5 to 7.4.

Figure 10. Comparison of local, spatially-resolved surface Nusselt number variations for different blowing ratios for Re_{MS}=142,000-155,000 for coolant side of effusion plate, with coolant supplied by an impingement jet array [6]. (a) BR=3.3. (b) BR=4.3. (c) BR=5.5. (d) BR=6.3. (e) BR=7.4.
Figure 11. Comparison of line-averaged surface Nusselt numbers with streamwise development for different blowing ratios for Re_Ms=142,000-155,000 for coolant side of effusion plate, with coolant supplied by an impingement jet array [6]. Solid rectangles denote effusion hole entrance streamwise locations. Dashed rectangles denote impingement hole streamwise locations.

Figure 12. Comparison of spatially-averaged surface Nusselt numbers with streamwise development for different blowing ratios for Re_Ms=142,000-155,000 for coolant side of effusion plate, with coolant supplied by an impingement jet array [6].

7. Experimental Results With Impingement Jet Array Cooling – Effusion Cooled / Hot Surface of Effusion Plate

The data within this section are obtained with impingement jet array cooling. Presented are measured results for the effusion cooled / hot surface of the effusion plate.

Figure 13 gives surface, local heat transfer coefficient variations in dimensional form for BR=7.4 and Re_Ms=142,000. Here, heat transfer coefficients range from approximately 20 W/m²K to 140 W/m²K along the test surface. Higher heat transfer coefficient values are generally observed just upstream, around, and along a trajectory downstream of each hole. Lower heat transfer coefficient values are generally present away from the holes. These variations are tied to jet mixing, vortex development around jet coolant concentrations, increased shear near jet edges, and the resulting augmentations of local three-dimensional turbulent transport.

Figure 14 presents distributions of surface, local adiabatic film cooling effectiveness variation for BR=6.3 and Re_Ms=146,000. These data show that adiabatic effectiveness values are increased along the entire test plate, including around and downstream of the film cooling holes. The overall high values of adiabatic effectiveness are due to abundant effusion coolant, resulting from efficient insertion of coolant at the entrance of each effusion hole from an adjacent impingement jet.

Figure 15 shows line-averaged, dimensional heat transfer coefficient values, as they vary with
streamwise development, for different blowing ratios for $\text{Re}_{\text{MS}}=142,000-155,000$. Here, line-averaged values are determined over $y/d_e$ from 13 to 25. These data are provided for the effusion cooled / hot surface of the effusion plate for a sparse effusion cooling hole array supplied by an impingement jet array. Here, line-averaged heat transfer coefficient values increase as the blowing ratio increases from 3.3 to 7.4, when compared at a particular $x/d_e$ location. Lower values of heat transfer coefficient indicate less mixing between the film cooling gas and the hot main flow gas, which leads to better thermal protection of the surface from the hot main flow gas.

Figure 16 shows the variation of line-averaged adiabatic film cooling effectiveness values with streamwise development for different blowing ratios for $\text{Re}_{\text{MS}}=142,000-155,000$. Within this figure, values given in Figure 16a represent coolant stagnation temperatures determined using cross flow stagnation conditions, and values given in Figure 16b represent coolant stagnation temperatures determined using impingement plenum stagnation conditions. For both sets of results, line-averaged adiabatic cooling effectiveness values generally increase as blowing ratio increases from 3.3 to 5.5, when compared at a particular $x/d_e$ location, indicating that surface protection is improved as film concentration increases near the surface. However, as the blowing ratio increases from 5.5 to 6.3, and then to 7.4, less variation with blowing ratio is evident. This is believed to be due to abundant coolant along and near to the test surface, which is provided by efficient insertion of coolant into effusion hole entrances by individual impingement jets.

8. Vortex Structure Associated With a Single Impingement Jet

Figures 17 and 18 show numerically-predicted, and experimentally measured development of Kelvin-Helmholtz vortices, which are associated with a single jet. The results in Figure 17 are provided for a turbulent jet, whereas the results in Figure 18 are provided for a laminar jet.

Figure 17 shows vortex formation occurs along the edge of a free axisymmetric jet. This is caused by Kelvin-Helmholtz instability from the shear force between the high velocity air jet and the surrounding stagnant air. Small vortices are created close to the nozzle and grow in size as they are carried along the flow until they are big enough to fully interrupt the jet. The vortices then collapse creating a fully turbulent flow. This visualization is produced by injecting smoke into the nozzle air flow as the fluid is forced through a 100 mm diameter nozzle. A laser sheet provides a two-dimensional view of the flow. The associated turbulent Reynolds number is approximately 10,000.

Figure 18 presents numerical prediction results for laminar jet flow with Kelvin-Helmholtz instability and vortices. Provided are contour variations of specific entropy through the center of the computational domain, at times of 0.0, 3.0, 6.0 and 10.0 seconds, with Kelvin-Helmholtz dynamic time scale of 3.4 seconds.

Figure 13. Surface, local heat transfer coefficient variation for $\text{BR}=7.4$ and $\text{Re}_{\text{MS}}=142,000$ for hot / effusion cooled side of effusion plate, with coolant supplied by an impingement jet array [6].
According to Lee et al. [8], local and spatially-averaged Nusselt numbers beneath impingement jets with different jet-to-target plate distances are generally the result of the competing influences of two different phenomena: (i) the coherence of individual jets and the strength of adjacent shear layers, and (ii) the development and advection of vortices generated by Kelvin-Helmholtz instabilities. The coherence of individual jets and the intensity of the adjacent shear layers, are strongest just after each jet emerges from an impingement plate hole. With this situation, maximum local jet velocities are distributed over the central part of each jet, with the largest gradients of velocity present in the thinnest shear layers which surround each jet. As each jet advects downstream and surrounding fluid is entrained to become part of each jet, the local velocity profile becomes more rounded with adjacent shear layer velocity gradients which are less intense as jet fluid diffuses and advects in lateral directions. If such jets impinge upon a surface, the highest local and spatially-averaged surface Nusselt number augmentations are expected to be present just after jets emerge from originating holes, when they are most coherent with the most intense adjacent shear layers [8].

Another mechanism for local heat transfer augmentation is formation, development, and surface impact of Kelvin-Helmholtz generated vortices. In contrast to the diminishing jet coherence with streamwise development, these vortices continue to develop as they advect in the streamwise direction. Their ability to augment thermal transport is a result of the mixing they induce between the jet and surrounding non-jet fluid. The resulting gradients of density (if the flows are compressible) and stagnation temperature within the vortices are key elements in their capability to augment local surface heat transfer coefficients and Nusselt numbers [8].
The relative influences of the jet/shear layer coherence and Kelvin-Helmholtz generated vortices are reflected in the $Z/D$ values associated with the highest Nusselt numbers. For higher jet Reynolds numbers $Re_j$, jet/shear layer coherence seems to be more important, since $Z/D=1.5$ gives optimal surface Nusselt numbers for an impingement array. For lower Reynolds numbers, the Kelvin-Helmholtz vortices have greater influences since $Z/D$ from 3.0 to 5.0 give optimal impingement Nusselt numbers [8].

9. Comparisons of Experimental Results With Impingement Jet Array Cooling and Cross Flow Cooling – Effusion Cooled / Hot Surface of Effusion Plate

Figure 19 shows comparisons of line-averaged heat transfer coefficient values with streamwise development for impingement flow coolant supply and cross flow coolant supply [4]. Figure 20 then shows comparisons of line-averaged adiabatic film cooling effectiveness with streamwise development for impingement flow coolant supply and cross flow coolant supply. Here, data in Figures 19a and 20a are provided for BR=4.3, and data in Figures 19b and 20b are provided for BR=6.3. These data comparisons are provided for main flow Reynolds numbers of 142000 to 155000.

When compared at particular values of streamwise location $x/de$ and blowing ratio $BR$, results in Figures 19a, 19b, 20a, and 20b generally show lower heat transfer coefficients, and significantly higher adiabatic effectiveness for the impingement supply arrangement, relative to the cross flow supply arrangement. Thus, for the same Reynolds number, blowing ratio, and streamwise location, significantly increased thermal protection is provided when the effusion coolant is provided by an array of impingement cooling jets. Within the present investigation, this is due to increased abundance of effusion coolant along the test surface, which is provided by efficient insertion of coolant into effusion hole entrances by individual impingement jets.
10. Summary and Conclusions

A variety of different types of vortices and vortex structures form near to and around individual effusion coolant concentrations, including jet wake vortices, jet shear-layer vortices, horseshoe vortices, and counter-rotating vortices (or kidney vortices). Of these different types of vortices, the counter-rotating secondary flow vortex structure downstream of the jet exit is the most important in regard to influences on local and overall film cooling performance. This is a result of the secondary advection and augmented turbulence transport influences of individual vortices within the pair, as well as the upwash region, which is located between the two vortices within the counter-rotating vortex pair.

Local and spatially-averaged Nusselt number magnitudes beneath impingement jets are generally the result of the competing influences of two different phenomena: (i) the coherence of individual jets and the strength of adjacent shear layers, and (ii) the development and advection of vortices generated by Kelvin-Helmholtz instabilities. The coherence of individual jets and the intensity of the adjacent shear layers, are strongest just after each jet emerges from an impingement plate hole. With this situation, maximum local jet velocities are distributed over the central part of each jet, with the largest gradients of velocity present in the thinnest shear layers which surround each jet. In contrast to the diminishing jet coherence with streamwise development, Kelvin-Helmholtz generated vortices continue to develop as they advect in the streamwise jet direction. Their ability to augment thermal transport is a result of the mixing they induce between the jet and surrounding non-jet fluid. The resulting gradients of density (if the flows are compressible) and stagnation temperature within the vortices are thus key elements in their capability to augment local surface heat transfer coefficients and Nusselt numbers, as impingement jets impact upon a target surface.

Within the present paper, the influences of these different vortex structures are described as they affect and alter the thermal performance of effusion cooling, impingement cooling, and cross flow cooling, as they are employed within a double wall cooling configuration. Presented are data for a sparse effusion cooling hole array. Non-dimensional streamwise and spanwise film cooling hole spacing, relative to effusion hole diameter, are 15 and 4, respectively. The facility is designed to provide full coverage film cooling data and impingement cooling effectiveness data, wherein the film cooling flow is supplied using either cross flow, impingement flow, or a combination of both together. Here, data are given with a cross flow supply only (with no impingement cooling), as well as for an impingement jet array supply only (with no cross flow). For both arrangements, experimental results are provided for 5 different crossflow Reynolds numbers, which are associated with 5 different values of blowing ratio BR. Presented are distributions of surface adiabatic cooling effectiveness and surface heat transfer coefficients for the hot side of the effusion test plate, and spatially-resolved distributions of surface Nusselt numbers for the coolant side of the effusion test plate.

With a cross flow only coolant supply, for the hot mainstream / effusion cooled side of the test plate, line-averaged adiabatic film cooling effectiveness generally increases, and line-averaged heat transfer coefficient also generally increases, as the blowing ratio becomes larger, when compared at a particular streamwise location and mainstream Reynolds number. On the coolant / cross flow side of the effusion test plate, increases of local and line-averaged Nusselt numbers occur in the vicinity of effusion hole entrances, due to increases of local flow advection speeds as cross flow fluid approaches and then enters into effusion hole entrances.

With an impingement jet array coolant supply, for the hot mainstream / effusion cooled side of the test plate, when compared at particular values of streamwise location and blowing ratio BR, lower heat transfer coefficients, and significantly higher adiabatic effectiveness are often present for the impingement supply arrangement, relative to the cross flow supply arrangement. Thus, for the same impingement jet Reynolds number, mainstream Reynolds number, blowing ratio, and streamwise location, significantly increased thermal protection is provided when the effusion coolant is provided by an array of impingement cooling jets (compared to a cross flow channel supply arrangement). On the coolant / cross flow side of the effusion test plate, regardless of the value of blowing ratio BR, the highest local Nusselt numbers are present at smaller x/de locations, relative to the locations of the
impingement hole centerlines. Such Nusselt number augmentation regions are associated with impingement jet stagnation locations, and re-positioning of individual impingement jets, relative to impingement hole exits, from turning and re-direction of coolant concentrations as they advect across the impingement passage.

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Figure 17. Visualization of turbulent jet induced Kelvin-Helmholtz vortices, including initial formation and subsequent vortex distortion and break-up.

Figure 18. Numerical prediction of laminar jet flow with Kelvin-Helmholtz instability and vortices. Contour variations of specific entropy through the centre of the computational domain at times of 0.0, 3.0, 6.0 and 10.0 seconds, with Kelvin-Helmholtz dynamic time scale of 3.4 seconds.

12. Nomenclature

| Symbol | Definition |
|--------|------------|
| BR     | film cooling blowing ratio, $\frac{\delta_{ef} V_{ef}}{\rho_{m3} \gamma_{m3}}$ |
| DH     | hydraulic diameter |
| $d_{ef}$ | film cooling effusion hole diameter |
| $h$    | heat transfer coefficient |
| $\bar{h}$ | line-averaged heat transfer coefficient |
| $Nu$   | local Nusselt number |
| $\overline{Nu}$ | line-averaged Nusselt number |
| $\overline{Nu}$ | spatially-averaged Nusselt number |
| Re     | Reynolds number |
| $v$    | time-averaged velocity |
| $x$    | streamwise coordinate |
Figure 19. Comparisons of line-averaged heat transfer coefficient values with streamwise development for hot / effusion cooled side of effusion plate for impingement flow coolant supply and cross flow coolant supply [6]. (a) BR=4.3. (b) BR=6.3.

Figure 20. Comparisons of line-averaged adiabatic film cooling effectiveness with streamwise development for hot / effusion cooled side of effusion plate for impingement flow coolant supply and cross flow coolant supply [6]. (a) BR=4.3. (b) BR=6.3.
\( y \)  
spanwise coordinate

**Greek symbols**

\( \eta \)  
adiabatic film cooling effectiveness

\( \bar{\eta} \)  
line-averaged adiabatic film cooling effectiveness

\( \rho \)  
density

**Subscripts**

\( DH \)  
main flow value, determined using hydraulic diameter of main flow passage

\( ef \)  
effusion jet value

\( ms \)  
main flow value

\( MS \)  
main flow value, determined using hydraulic diameter of main flow passage

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