Heat transfer from multiple row arrays of low aspect ratio pin fins

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1. Introduction

Pin fin arrays are generally used to promote heat transfer in applications ranging from cooling of electronic equipment to cooling gas turbine airfoils. Placed in a channel flow, pin fins serve to enhance heat transfer through increased surface area and increased turbulence. Despite the many applications of pin fins, there is relatively sparse data available, particularly for short pin height-to-diameter ratios.

One particular application of short pin fins is cooling the trailing edge of a turbine airfoil, which is a particularly difficult engineering challenge because of its thin cross-section. It is a challenge because the external heat transfer coefficients associated with the mainstream core flow passing along the external airfoil surface are very high and the thin trailing edge section does not allow for many cooling designs to meet structural integrity requirements. The cooling method most commonly used for the trailing edge section of the blade are cylindrical pin fins with relatively low aspect ratios (pin height-to-diameter ratios). These pin fins, which are most commonly used in staggered or in-line array configurations, are often placed in the internal cooling channels of the airfoils.

The addition of pin fins increases the wetted heat transfer area and turbulence levels of the channel flow thereby augmenting the convective heat transfer. The most common cooling designs for the trailing edge employ pin fins with aspect ratios between 0.5 and 4.0 [3]. Because of the large range of pin aspect ratios that lead to differing flow effects, such as the pin fin wake interaction with their associated base endwalls, interpolation between aspect ratios does not provide accurate results [4]. The heat transfer characteristics of low aspect ratio pin fin arrays has not been as widely studied as the heat transfer in high aspect ratio pin fin arrays commonly used in heat exchanger applications.

This paper presents experimental results that show how the addition of pin fin arrays augments convective heat transfer. In these studies, the heat transfer from the surface of the pins as well as the heat transfer from the duct walls, also known as endwalls, is measured for a wide aspect ratio channel. Heat transfer and pressure drop are measured for multiple row arrays of pin fins to determine the independent effects of spanwise and streamwise array spacings.

2. Review of relevant literature

The development of internal cooling methods using pin fin arrays can be largely attributed to experiments conducted in the past 50 years by numerous researchers. The work in the literature has focused on measuring heat transfer of pin surfaces as well as on the channel wall surfaces known as endwalls. Fig. 1 illustrates the pin spacing definitions used throughout this paper. Results in the literature show that pin and endwall heat transfer vary greatly depending on Reynolds number and pin spacing. The following section describes the pin fin array heat transfer characteristics determined by various researchers in the literature.

2.1. Nusselt number development

Metzger et al. [5], Chyu [6], Yeh and Chyu [7], and Ames et al. [8] all observed an initial increase in Nusselt number through the
entrance rows of multiple row pin fin arrays preceding a slight decrease through the downstream rows. Metzger and Haley [9] explained the development trend observed by Metzger et al. [5] by measuring turbulence intensities through an array. Similar to heat transfer, turbulence increased through the entrance rows in the array to reach a peak value and gradually decreased through the downstream rows of the array.

Chyu [6] measured spanwise average Nusselt number through seven rows of in-line and staggered pin fins. He concluded that high heat transfer in an array occurs as a result of wake impingement from upstream pins and flow acceleration between two spanwise neighboring pins. Chyu [6] determined for a given array, the maximum occurred in the second row for an in-line array and the third row for a staggered array, which is consistent with the findings of Metzger et al. [5]. Hwang and Lui [10] also observed that their maximum Nusselt number occurred at the second row, as with the findings of Chyu [6].

Yeh and Chyu [7] studied the heat transfer through ten rows of staggered pin fins at aspect ratios of 1.0 and 2.8 and they observed the same development trend through both arrays. The heat transfer increased through the first three rows and decreased through the remainder of the array. The difference between the maximum heat transfer and the tenth row was approximately 16%. Metzger et al. [5], who also measured heat transfer through a ten row array, found a similar result with a decrease of approximately 12% between the row having the maximum heat transfer and the tenth row.

Ames et al. [8] measured heat transfer around the circumference of a pin fin in each row of an eight row staggered array. The measured heat transfer distribution around the cylinder in row one was typical for a single circular cylinder in cross flow with high heat transfer on the surface near stagnation decreasing to the point of separation. Wake shedding downstream of stagnation increased heat transfer near the trailing edge of the pin. A gradual increase in heat transfer was observed through row two while a sharp increase was measured between rows two and three. This development trend was very similar to the trend observed by Metzger et al. [5] and Yeh and Chyu [7] who also studied Nusselt number development through staggered arrays. Results were further investigated by Ames et al. [11] finding that peak heat transfer occurred in row three because of the violent wake shedding from row two, and they found the intensity of wake shedding correlated well with the heat transfer rates in the wakes of pins.

Lyall et al. [12] measured pin and endwall Nusselt numbers for a single row of pin fins at various spacings and found good agreement with correlations for the first row in a multiple row array developed by Metzger et al. [5] and Chyu [6]. Hwang and Lui [10] developed a correlation to predict Nusselt numbers in the first row of an in-line array located in a wedged duct. They compared their correlation with the correlations developed by Metzger et al. [5] and Chyu [6] and found good agreement.

### 2.2. Effects of pin spacing

Multiple studies have been conducted to determine the effects of pin spacing on heat transfer in pin fin arrays with aspect ratios between 0.5 and 4.0. Arrays with height-to-diameter aspect ratios near one are commonly used in airfoil applications because pin fins increase the convective surface area while providing the structural integrity necessary in the trailing edge region of the airfoil.

Metzger et al. [13] studied the effects of streamwise spacing by independently varying streamwise spacing relative to spanwise spacing. For streamwise spacings ranging from 1.05 to 5.0, they found that array-averaged Nusselt number decreased with increased streamwise spacing. They concluded that closely spaced arrays yielded higher heat transfer than widely spaced arrays.
Simoneau and VanFossen [14] obtained average Nusselt numbers on a single heated pin in cross flow. They found that the Nusselt number on a single pin was between 7% and 15% lower than the Nusselt number on the same pin when placed in an array with a spanwise and streamwise spacing of 2.67 and an aspect ratio of three. The increased turbulence levels created by the surrounding array increased the convective heat transfer from the surface of the pin.

Lyall et al. [12] studied pin and endwall heat transfer in single rows of pin fins with a height-to-diameter ratio of one at spanwise spacings of two, four, and eight. Although no correlations were developed, Lyall et al. [12] found that the pin heat transfer for the geometries with spanwise spacings of four and eight had a stronger Reynolds number dependency than the array with the spanwise spacing of two which is consistent with the findings of Metzger et al. [5] who observed increased Reynolds number dependence with increased pin spacing. The combined endwall and pin heat transfer, also known as array-averaged heat transfer, had comparable Reynolds number dependency for all three geometries tested by Lyall et al. [12].

Lyall et al. [12] found that pin heat transfer increased with increased spanwise spacing; however, endwall heat transfer decreased with increased spanwise spacing. They attributed decreased endwall heat transfer to the decreased interaction between pins as the spacing was increased. Consistent with Simoneau and VanFossen [14], the array-averaged Nusselt number also decreased with increased spanwise spacing because the endwall surface area was much greater than the pin surface area at an aspect ratio of one.

The variation in pin spacing dependence data shown in the literature suggests that pin spacing has a large effect on endwall heat transfer and a small effect on pin heat transfer. For the current study, experiments were conducted through a range of Reynolds numbers at four different pin spacings to explore the effects of spanwise and streamwise spacing on array heat transfer and pressure drop.

<table>
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<th>S2/d</th>
<th>H/d</th>
<th>Umax/Ubulk</th>
<th>d/Dh</th>
<th>Ap/Aavit</th>
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Fig. 2. Schematic of the closed loop test facility used for pin fin array testing.
2.3. Relative pin and endwall heat transfer

Many researchers have focused on the relationship of pin heat transfer relative to endwall heat transfer. The past studies are in agreement that heat transfer is higher on the pin than the endwall; however, the relative difference between pin and endwall heat transfer is in question.

VanFossen [15] found for staggered arrays with $S/d = S2/d = 3.46$ and $0.5 < H/d < 2.0$, pins had a 35% higher heat transfer coefficient than the endwalls. Metzger et al. [16] found for staggered arrays with $S1/d = S2/d = 2.5$, $H/d = 1$ and $S1/d = 2.5$, $S2/d = 1.5$, $H/d = 1$, pin heat transfer was 80–100% higher than endwall heat transfer. Chyu et al. [17] found that pin heat transfer was at most 20% higher than the endwall heat transfer for staggered arrays with $S1/d = S2/d = 2.5$ and $H/d = 1$. Chyu et al. [18] found for staggered pin fin arrays with $S1/d = S2/d = 2.5$ and $H/d$ ranging from 2 to 4, pin heat transfer was 30–70% higher than endwall heat transfer and was more sensitive to Reynolds number than $H/d$.

Yeh and Chyu [7] studied the relationship between pin and endwall heat transfer at pin height to diameter aspect ratios of one and 2.8. In the array having an aspect ratio of one, the pin heat transfer was as much as 10% higher than the endwall heat transfer while the array having an aspect ratio of 2.8 had 20% higher heat transfer on the pin than on the endwall much like the findings by Chyu et al. [17].

Lyall et al. [12] studied relative pin and endwall heat transfer for single rows of pin fins at an $H/d = 1$. They found that the pin-to-endwall Nusselt number ratio depended highly on Reynolds number and spanwise spacing. The highest ratio of pin-to-endwall Nusselt number occurred at the lowest Reynolds number and the widest spanwise spacing. For a spanwise spacing of eight pin diameters, the pin-to-endwall Nusselt number ratio ranged from 2.3 at the highest Reynolds number to 3.4 at the lowest Reynolds number. For a spanwise spacing of two pin diameters, the pin-to-endwall Nusselt number ratio ranged from 1.5 at the highest Reynolds number to 2.0 at the lowest Reynolds number.

Results from the literature show that pin spacing affects pin and endwall heat transfer as well as the development of heat transfer through an array; however, the independent effects of streamwise and spanwise spacing have never been explored in detail. In the current study, pin and endwall heat transfer are measured in a spatially resolved manner to determine the independent effects of streamwise and spanwise spacing on heat transfer development and relative pin and endwall heat transfer.

3. Experimental facility

To measure the convective heat transfer in pin fin arrays, different array geometries were placed in a wide aspect ratio test section. Experiments were conducted to determine endwall heat transfer across arrays having seven rows of copper pin fins with varied spanwise ($S1/d$) and streamwise ($S2/d$) spacings. Experiments were also conducted using non-conductive instrumented pins to measure the heat transfer on an individual pin. Recall, definitions for these parameters are shown in Fig. 1 while Table 1 shows the geometric specifications for each array that was studied.

It is important to note that the $S2/d = 1.73$ and $S2/d = 3.46$ spacings were used because those row spacings are necessary to have an equilateral triangle arrangement of pins with $S1/d = 2$ and $S1/d = 4$ respectively.

Fig. 2 shows a schematic of the closed loop facility, similar to the one used by Lyall et al. [12], in which flow travels in the clockwise direction. As flow entered the plenum, it first encountered a splash plate, which prevented the inlet air from propagating through to the test section without mixing. The air then passed through a finned tube heat exchanger that was used to ensure a constant air temperature to the test section. As the flow exited the plenum, it was directed through a rounded test section inlet to ensure uniform flow entering the duct. Strips of 60 grit sandpaper were added at the duct entrance to uniformly trip the boundary layer across the entire span of the channel.

The test section of the facility, as indicated in Fig. 2, was constructed as a wide aspect ratio duct, as shown in Fig. 3, with a width of 61 cm and a height of 0.95 cm giving a width to height ratio of 64:1. The test section contained an unheated entrance length equal to 40 hydraulic diameters measured from the exit of the plenum to a heated endwall. Flat heaters were attached to the top and bottom endwalls creating a constant heat flux boundary on which arrays of copper pin fins were placed. Pressure taps were placed in the ceiling of the test section to obtain static pressure measurements throughout the channel. Friction factors were also calculated to characterize pressure drop across each array.

Downstream of the test section, the flow entered an extension that transitioned the flow from a rectangular duct to a round pipe. A venturi flowmeter was placed 15 pipe diameters downstream of the extension and was followed by a pipe length of eight diameters that was needed to ensure accurate flow measurements. The maximum velocity, $U_{max}$, and bulk velocity, $U_{bulk}$, through the test section could be calculated using the known flow area through the
channel with and without pin fins, respectively. The flowrate for each experiment was adjusted using the variable speed blower, shown in Fig. 2, to set the air velocity through the test section to achieve the desired Reynolds number. Pressure relief valves, also shown in Fig. 2, were installed on both the suction and the discharge sides of the blower. These relief valves were used to adjust the channel operating pressure to equilibrate the pressure inside the test section with that on the outside of the test section. Pressure equilibration was necessary because the top heater served as the endwall surface with no other support than the pin fins. Without adjusting the operating pressure of the channel, a pressure differential between the test section and external surroundings caused the endwall to either bow outward or contract inward. Channel pressure was set to achieve contact between the pin fins and the bottom heater and thermally conductive paste was applied to the pin fin surfaces mating with the bottom endwall to ensure adequate thermal contact for each experiment. It is important to note that the flowrate was quantified directly downstream of the test section such that it was an accurate measure of the flow through the test section.

As previously stated, heaters placed along the top and bottom walls of the duct were used to create a constant heat flux boundary condition on the endwalls of the pin fin array. Pins were attached to the heaters with a high thermal conductivity epoxy. The heaters were powered by individual power supplies that were placed in series with a precision resistor. Accurate current measurements could be made by measuring the voltage across the precision.
resistor. Power supplies were adjusted independently to provide the same power to the top and bottom endwall heaters. The net convection heat flux to the flow was calculated using,

\[ q_{\text{net}}^0 = \frac{P - q_{\text{loss}}^0}{2L \cdot W} + \frac{0.5N\pi \cdot d^2}{\lambda} \]

where \( P \) is the total power to the heaters, \( q_{\text{loss}}^0 \) is the calculated heat flux loss due to conduction, \( L \) is the length of each heater, \( W \) is the width of each heater, \( N \) is the number of pins, and \( d \) is the pin diameter. The term in the denominator is the wetted area which incorporates the pin surface area into the overall heat flux area.

The loss due to conduction was estimated using thermocouple measurements made on the outside of the test section walls to measure loss temperature (see Fig. 3). In general, the heat losses were less than 5% of the total supplied by the heater at the lowest Reynolds number and decreased with increasing Reynolds number. Knowing the net heat flux to the flow, the heat transfer coefficients were found using the calculated bulk air temperature as the fluid reference temperature. It is important to note that bulk air temperature increased with spanwise distance through the array because of added heat from the endwall heaters. The heat transfer coefficient was based on the measured local wall temperature and local bulk temperature.

The temperatures on the external side of the endwall heater were measured with an infrared (IR) camera through a 17 cm by 33 cm rectangular opening. A Zinc Selenide window, shown in Fig. 3, was placed in the rectangular opening between the exposed heater surface and the surroundings to minimize heat loss and to allow for IR transmittance. Between the external side of the heater and the Zinc Selenide window was a thin air gap which minimized conductive losses from the heater. During testing, a 5 cm thick piece of insulation was placed on the external side of the window. After steady-state was achieved the insulation was removed to allow camera access to the heater surface. Measurements were acquired over a time period of less than two minutes with minimal heat loss to the environment. No noticeable difference in temperature was indicated when the foam was removed.

Each heater was composed of inconel strips arranged in a serpentine fashion and encapsulated in Kapton. A thin layer of copper was vacuum deposited to the external side of each endwall heater to ensure an even heat flux distribution along the endwall. Although each heater was composed of many layers of material, the total thickness was only 254 \( \mu \)m. Within the IR window area, the copper heater surface was painted black to increase the...
emissivity for viewing with the IR camera. Three thermocouples were attached to the back side of the heater with thermally conductive epoxy and were used to calibrate the IR images.

In addition to the endwall studies, experiments were conducted to acquire pin heat transfer data using two instrumented pin fins. Pins were constructed from balsa wood and the surface of each was covered with inconel heating foil. A schematic of a balsa pin is shown in Fig. 4. Balsa wood was used because of its insulating properties to minimize conduction losses through the pin. Three thermocouples were embedded around the circumference of each balsa pin below the inconel heater. Each thermocouple was placed at a different circumferential location to determine a heat transfer coefficient at the leading edge, the trailing edge, and 90° from the leading edge on the side of the pin. Comparisons with the literature showed that these three measurement locations gave a good average of pin heat transfer coefficients. During pin heat transfer tests, the array was placed on a non-conductive surface to prevent conduction losses to the endwall. Both pins were placed in the row of interest and the results from both pins were averaged to decrease the uncertainty of the results for each row. Experiments were performed through the full range of Reynolds numbers for each row individually.

4. Uncertainty analyses

An uncertainty analysis was performed using the uncertainty propagation method described by Moffat [19]. The uncertainty was calculated at low and high Reynolds numbers for the highest and lowest values of pin and endwall Nusselt numbers. The calculated Reynolds number uncertainty was 4.0% and 2.5% at Reynolds numbers of 5000 and 30,000, respectively. The pin Nusselt number (\(Nud_{dp}\)) uncertainty was 5.5%. The uncertainty for the duct Nusselt number (\(Nu\)) was less than 13% for all cases tested. The highest contributor to the uncertainty of Nusselt number was the calibration of the IR images. The combined pin and endwall, also referred to as array-averaged, Nusselt number (\(\overline{Nuo}\)) uncertainty was less than 8% for all cases tested. Repeatability tests were conducted for various geometries with the results falling within calculated uncertainty values.

5. Effects of Reynolds number on endwall and pin heat transfer

Convective heat transfer in pin fin arrays depends not only on pin spacing, but Reynolds number as well. This section discusses the effects of Reynolds number on endwall and pin heat transfer. Two definitions of Reynolds number are used throughout this section. The duct Reynolds number (\(Re\)) is based on the bulk velocity through the channel and uses the channel hydraulic diameter as the characteristic length. The array Reynolds number (\(Re_{d}\)) is based on the maximum velocity through the array and uses the pin diameter as the characteristic length. Two definitions of Nusselt numbers are also used. To keep with convention, the array Nusselt number and array Reynolds number are used when comparing to results from the literature. The duct Nusselt number (\(Nu\)) uses the channel hydraulic diameter as the characteristic length while the array Nusselt number (\(Nud_{d}\)) uses the pin diameter as the characteristic length. The array Nusselt number is commonly used throughout the paper to draw comparisons between different array geometries based on endwall Nusselt number (\(Nud_{d,e}\)) and pin Nusselt number (\(Nud_{dp}\)).

Figs. 5 and 6 show contour plots that illustrate the endwall Nusselt number augmentation (\(Nud_{d,e}/Nu_{o}\)) through a range of Reynolds numbers for the \(S1/d = 2, S2/d = 1.73\) and \(S1/d = 4, S2/d = 3.46\) arrays, respectively. The Nusselt number augmentation is the ratio of the measured endwall Nusselt number (\(Nu\)) to the correlation for a smooth duct Nusselt number (\(Nu_{o}\)) at the same duct Reynolds number [2]. Each contour plot represents the same physical area independent of pin spacing differences, which is why fewer pins are shown in Fig. 6 relative to Fig. 5. These contour plots show that augmentation decreases with increasing Reynolds number for both pin spacings. This decrease is observed because at low Reynolds numbers the pin fins greatly increase the turbulence relative to an empty duct while at high Reynolds numbers the turbulence is only slightly increased by the pin fins. The empty duct Nusselt number (\(Nu_{o}\)) has a Reynolds number exponent of 0.8 which is higher than any of the Reynolds exponents previously reported in the literature [5–7,10,17]; therefore the trend of decreasing augmentation with increasing Reynolds number is consistent with relevant literature [2].

Fig. 7 shows the spanwise average endwall augmentation with respect to row number for the \(S1/d = 2, S2/d = 1.73\) and \(S1/d = 4, S2/d = 3.46\) geometries. The Nusselt number for each row was calculated by taking a spanwise average of the Nusselt numbers from half of a row spacing upstream to half of a row spacing downstream of the row of interest. It is important to note that the data under each pin was not included in this average and are represented by the blue circles in the contour plots. Fig. 7 confirms the effect seen in Figs. 5 and 6 by showing that the augmentation
is significantly lower at high Reynolds numbers than at low Reynolds numbers at every location in the array. The row-by-row augmentation development through the array is independent of Reynolds number showing the same trend in every case. Further analysis of the Nusselt number development on the endwall is discussed in later sections of this paper.

The circumferential variation of heat transfer around the instrumented pin fin was measured and is compared with results from the literature in Fig. 8. The general trend of pin Nusselt number around the circumference of the pin fin shows a maximum Nusselt number at the leading edge of the pin fin where the flow stagnation occurs. The Nusselt number then decreases to reach a minimum value at the point of laminar boundary layer separation. Separation occurs at the point on the surface of the pin where the adverse pressure gradient is high enough to cause the velocity gradient on the surface of the pin to become zero. For a single pin in cross flow, separation generally occurs closer to the pin leading edge at low Reynolds numbers than at high Reynolds numbers, but the location of separation can depend on other factors such as the spacing of the pins placed downstream. Downstream rows of pins can affect the adverse pressure gradient which directly affects the location of separation.

The minimum Nusselt number at the separation point is sometimes followed by a sharp increase in Nusselt number caused by the transition to turbulent flow. The Metzger et al. [13] and Ames et al. [8] results in Fig. 8 both show a sharp increase in Nusselt number following separation. Downstream of the sharp increase, there is another decrease in Nusselt number which is caused by the further development of the turbulent boundary layer. The decrease can be seen in the Metzger et al. [13] data in Fig. 8. The increase in Nusselt number sometimes seen at the trailing edge of the pin fin is caused by mixing in the wake region.

Even with the limited number of circumferential measurements, the Nusselt numbers on the pin in this study show trends consistent with the trend described above. For the high Reynolds number case in Fig. 8, separation followed by wake mixing causes increased heat transfer towards the trailing edge. In low Reynolds number cases, wake mixing near the trailing edge is not intense and low heat transfer is observed both in the current study and by Ames et al. [8].

Fig. 9 shows the average pin Nusselt numbers (\(\bar{N}_\text{Nu}_{pin}\)) for the \(S/d = 2\), \(S/d = 1.73\) and \(S/d = 4\), \(S/d = 3.46\) geometries compared to results from the literature. Results for pin heat transfer are given in terms of \(N_u\) and \(Re_p\) to be consistent with the literature. The pin Nusselt numbers for the two spacings are almost identical falling within experimental uncertainty of one another. These results differ, however, from the single row findings of Lyall et al. [12]. Lyall et al. [12] found that the pin Nusselt number was lower at \(S/l = 2\) than at \(S/l = 4\). They concluded that the data did not scale with the pin Reynolds number because the pin results did not scale with velocity alone. The multiple row geometries tested here do scale with pin Reynolds number which indicates that, in multiple row arrays, the maximum velocity is the main driving mechanism for convective heat transfer on the pins.

The Zukauskas [20] inner row correlation predicts Nusselt numbers for large aspect ratio pin fin arrays with \(S/l\) ranging from 1.3 to 2.6 and \(S/d\) ranging from 0.6 to 3.9 and \(H/d > 8\). The Zukauskas [20] correlation agrees well with the results of the two geometries presented here but has a slightly higher Reynolds number dependence. The Chyu et al. [17] and the Yeh and Chyu [7] correlations have similar Reynolds number dependence as the geometries tested here but differ in magnitude.

Using the endwall and pin heat transfer data, Fig. 10 shows the array-average Nusselt number augmentation for the \(S/l = 2\), \(S/d = 1.73\) and the \(S/l = 4\), \(S/d = 3.46\) geometries. The array-averaged Nusselt numbers are calculated using the area fractions shown in Table 1 to calculate an area weighted average of the pin and endwall Nusselt numbers for a given geometry. The endwall contribution to the array-average Nusselt number was calculated by averaging the data from half of a row spacing upstream of the first row to half of a row spacing downstream of the last row in each array. The augmentation is greatest at the lowest Reynolds number for both spacings and gradually decreases as Reynolds number is increased. As stated previously, adding pin fins increases the turbulence by a larger amount at low Reynolds numbers than at high Reynolds numbers. Augmentation for both pin spacings shows the same dependency on Reynolds number.

6. Effects of pin spacing on endwall heat transfer

The endwall heat transfer measurements have a dominant effect on the array-averaged heat transfer in arrays with low pin aspect ratios because the endwall area is significantly larger than the...
pin area as shown in Table 1. Because endwall heat transfer is dominant, much can be learned about the array-averaged heat transfer by exploring the effects seen on the endwall. Specifically, this section focuses on the effects of spanwise and streamwise pin spacing on endwall heat transfer. The results are presented in terms of array Nusselt number (Nud) and array Reynolds number (Re) to be consistent with the literature.

Fig. 11 shows the average endwall Nusselt numbers (\(\overline{Nud}_{d,e}\)) for the four geometries tested in this study. The general trend in Fig. 11 shows that heat transfer increases as pin spacing decreases. The data show that endwall heat transfer is more dependent on streamwise spacing than spanwise spacing for these geometries tested. The conclusion can be drawn that the pin wake interaction between streamwise rows of pins induces more convective benefit than interactions between adjacent pins in a given row.

The Reynolds number dependence on pin spacing effects is illustrated in Fig. 11. The four cases agree well at low Reynolds numbers but diverge at high Reynolds numbers. The S2/d = 1.73 cases have a higher magnitude and Reynolds number dependency than the S2/d = 3.46 cases at high Reynolds numbers. The S2/d = 1.73 arrays are more tightly spaced and promote more wake interaction between streamwise pin rows than the arrays with the wider streamwise spacing. This streamwise dependence is consistent with the findings of Metzger et al. [13] who showed a similar heat transfer dependence on Reynolds number for pin fin arrays with varying streamwise spacing.

Fig. 12 shows the Nusselt number development trends through the S1/d = 2, S2/d = 1.73 and S1/d = 2, S2/d = 3.46 arrays, which are very different. The Nusselt number in the S2/d = 3.46 array reaches a peak value at row two, and then remains constant through the array. In contrast, S2/d = 1.73 requires a longer development length with array Nusselt numbers continuing to increase through row four. The longer development length for S2/d = 1.73 results in a higher fully developed Nusselt number. Recall that Chyu [6] determined for a given array that the maximum Nusselt number occurs at the row having the first direct wake impingement from upstream rows. The S1/d = 2, S2/d = 3.46 array is widely spaced in the streamwise direction relative to the spanwise direction causing the direct wake shedding from row one to generate a maximum local Nusselt number in row two. Likewise, the first direct wake shedding from upstream rows in the S2/d = 1.73 array is not experienced until rows three and four which is where the Nusselt number reaches its fully developed value. These development trends can be observed qualitatively in Figs. 5 and 6.

The largest difference between fully developed values of the S2/d = 1.73 and S2/d = 3.46 arrays occurs at the highest two Reynolds numbers which is where the two trends for the respective geometries diverge in Fig. 11. The difference in development trends between the two streamwise spacings shown in Fig. 12 is more obvious at high Reynolds numbers because the tight streamwise spacing increases the wake interaction and hence the fully developed Nusselt number value.

It is important to note that in Fig. 12 the heat transfer for the S2/d = 1.73 array increases through the first three rows, reaches a maximum at row four and gradually decreases through the remainder of the array. Metzger et al. [5] and Yeh and Chyu [7] observed the same decrease through the array downstream of the location where the maximum heat transfer was measured. The trends observed in Fig. 12 imply that heat transfer development is dependent on the ratio of streamwise to spanwise spacing (S2/S1). Of the geometries reported in this paper, the only geometry that experienced fully developed heat transfer before rows three and four was the case having S2/S1 greater than 1. In the case with S2/S1 > 1, the rows are spaced too far apart for wakes from consecutive rows to interact. In the cases with S2/S1 < 1, mixing is enhanced when the wakes from row one interact with row two and when wakes from row two interact with row three. In other words, when the rows are spaced close together the wakes build up until they are fully developed whereas when S2/S1 > 1 the wakes do not build up because the rows are too far apart.

Fig. 13 shows that the pin-to-endwall Nusselt number ratio depends on Reynolds number and pin spacing. The highest ratio occurs at the lowest Reynolds number and the widest pin spacing. As the Reynolds number is increased, the ratio decreases. This result is similar to the findings of Lyall et al. [12]. The dependence of pin-to-endwall Nusselt number ratio on pin spacing can be attributed to the effects of pin-wake interactions on endwall heat transfer. As the array spacing increases, wake interaction decreases. As the wake interaction decreases, the average heat transfer on the endwall decreases. The decrease in endwall heat transfer increases the pin-to-endwall Nusselt number ratio. Metzger et al. [16] also found a pin-to-endwall Nusselt number dependence on pin spacing. They determined that the pin-to-endwall heat transfer ratio was \(\overline{Nud}_{d,p}/\overline{Nud}_{d,e}\) = 1.8 at S1/d = 2.5, S2/d = 2.5 and \(\overline{Nud}_{d,p}/\overline{Nud}_{d,e}\) = 2.0 at S1/d = 2.5, S2/d = 1.5. VanFossen [15] reported a ratio of 1.35 and Chyu et al. [17] measured a maximum pin-to-endwall Nusselt number ratio of 1.2. The results reported in Fig. 13 show
ratios ranging from 1.4 at the highest Reynolds number and tight-
est array spacing to 1.85 at the lowest Reynolds number and wid-
est array spacing.

Fig. 14 shows endwall augmentation contour plots for each of
the four geometries tested at $Re = 13,000$. The contours show
that augmentation decreases as the array spacing increases in the span-
wise as well as the streamwise direction. The development trends
shown in Figs. 7 and 12 can be seen in the contour plots in Fig. 14
as well. As previously discussed, when the streamwise spacing is
much larger than the spanwise spacing as it is in Fig. 14b, the Nus-
selt number reaches a fully developed value further upstream in
the array than in the arrays having larger spanwise spacing than
streamwise spacing. The differences in wake interaction between
rows of different arrays are shown in Fig. 14. Wake interaction in-
creases through row 4 in Fig. 14a, c, and d while wake interaction is
maximized at row two in Fig. 14b.

The array-averaged Nusselt numbers for all four geometries
studied are shown in Fig. 15. The array-averaged Nusselt numbers
for the $S_1/d = 2$, $S_2/d = 1.73$ and $S_1/d = 4$, $S_2/d = 3.46$ geometries
were calculated using the area-average of the pin and endwall Nus-
selt numbers. Pin results were not obtained for the $S_1/d = 2$, $S_2/
= 3.46$ and $S_1/d = 4$, $S_2/d = 1.73$ geometries because the pin Nus-
selt number showed little dependence on pin spacing. A pin Nus-
selt number correlation developed from the results in Fig. 9 was
used to calculate the pin contribution to the array-average Nusselt
numbers for these geometries. Fig. 15 shows that the two arrays
with $S_2/d = 1.73$ have the same convective heat transfer, which is
higher than the two arrays with $S_2/d = 3.46$. The array with the
widest spanwise and streamwise spacings has the lowest heat
transfer across the entire range of Reynolds numbers. The agree-
ment between the two arrays with $S_1/d = 1.73$ shows that when
the streamwise spacing is small, a decrease in spanwise spacing
does not significantly increase the heat transfer. These results sug-
gest that the streamwise spacing plays a larger role than spanwise
spacing in affecting the convective heat transfer through pin fin
arrays.

7. Pressure loss measurements

Pressure loss measurements were made for all geometries
tested in this study. Pressure loss results were quantified using ar-
ray friction factors which are plotted for all geometries in the cur-
rent study in Fig. 16. The results in Fig. 16 indicate that pressure
loss caused by spanwise spacing effects is much more significant
than the pressure loss caused by streamwise spacing effects. The pressure loss associated with spanwise spacing is a direct result
of flow blockage while the pressure loss associated with stream-
wise spacing is the result of wake interactions between pin rows.
Clearly, the results presented here illustrate that pressure loss is
more sensitive to flow blockage than wake interaction effects.

The objective of pin fin arrays is to maximize heat transfer aug-
mentation while minimizing the pressure drop through the array.
Webb and Eckert [21] developed a parameter to quantify the effi-
ciency of roughness elements in heat exchanger design taking into
account both pressure drop (pumping power) and heat transfer.
The thermal performance is defined by,

$$\eta = \frac{\bar{Nu}/Nu_0}{(f_{array}/f_0)^{1/3}}$$

Fig. 17 shows the thermal performance plotted with respect to Rey-
olds number for the four geometries tested in the current study.
The trends in Fig. 17 were generated using correlations for Nusselt
number and friction factor augmentation determined experimen-
tally for each geometry in the current study. It is interesting to note
that the two arrays having equal pin and endwall area fractions (as
shown in Table 1) have almost identical thermal performance.
Because the thermal performance has a greater dependence on heat transfer than pressure drop, the array with the tightest spanwise and streamwise spacing has the highest thermal performance while the array with the widest spanwise and streamwise spacing has the lowest thermal performance.

8. Conclusions

Experiments were conducted to determine the independent effects of spanwise and streamwise spacing on heat transfer and pressure loss through multiple row arrays of pin fins. The general trends showed that the heat transfer in an array of pin fins increased with decreased spanwise and streamwise pin spacings with a stronger dependence on streamwise spacing than spanwise spacing. Heat transfer development through the array was also found to be dependent on the ratio of streamwise to spanwise spacing. If the streamwise spacing was larger than the spanwise spacing, the heat transfer reached a peak farther downstream in the array, because the pin wake interaction was enhanced by closely spaced rows. The pin Nusselt numbers obtained for different array geometries were in good agreement with one another indicating little dependence on pin spacing for those geometries studied.

The independent effects of spanwise and streamwise spacing on pressure loss were also investigated. In contrast to the heat transfer results, which were shown to be dependent on streamwise spacing, the friction factor was found to be more dependent on spanwise spacing than streamwise spacing because of the dominant effect of flow blockage on pressure loss. Therefore, friction factor could be minimized and heat transfer could be maximized by simply decreasing the streamwise spacing of a given pin fin array. Based on Nusselt number augmentation and friction factor augmentation, however, the thermal performance was highest in the array with the tightest spanwise and streamwise spacing.

The results obtained for the range of geometries tested in this study improves the understanding of heat transfer and pressure drop in low aspect ratio pin fin arrays. These results provide essential information for use in the design of internal cooling channels for turbine airfoils. The spatially resolved results presented here showed that not only does pin spacing have a large effect on array-averaged heat transfer but it has a large effect on the development of heat transfer through an array as well. The results from the current study showed that heat transfer can be maximized while minimizing pressure drop by increasing the spanwise spacing and decreasing the streamwise spacing of pin fins.

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