ABSTRACT
Cylindrical pins, often called pin fins, are used to create turbulence and promote convective heat transfer within many devices, ranging from computer heat sinks to the trailing edge of jet engine turbine blades. Previous experiments have measured the time-averaged heat transfer over a single pin as well as the flow fields around the pin. However, in this study, focus is placed on the instantaneous heat flux around the centerline of a low aspect-ratio pin within an array. Time-mean and unsteady convective heat flux are measured around the circumference of an isothermal heated test pin via a microsensor located at the surface. The pin is positioned at various locations within a staggered array in a large-scale wind tunnel. Reynolds numbers from 3,000 to 50,000, based on pin diameter and maximum velocity between pins, are tested with a streamwise spacing of 1.73 diameters between rows, a spanwise spacing of 2 diameters, and a pin height of 1 diameter. The time-averaged and standard deviation of convective heat flux around the pin is higher over most of the pin surface for pins in downstream row positions of an array relative to the first row pin, except in the wake which has similar levels for all rows. For a given pin position in the array, as the Reynolds number increases, the point of minimum heat transfer moves circumferentially upstream on the pin fin, corresponding to earlier transition of the pin boundary layer. Also, for a given Reynolds number, the minimum heat transfer point on the pin circumference moves upstream for pins further into the array, due to the high turbulence levels within the array which cause early transition. For a single pin row with no downstream pins, heat transfer fluctuations are very high on the backside of the pin due to the significant unsteadiness in the pin wake, but heat transfer fluctuations are suppressed for a pin with downstream rows due to the confining effects of the close spacing. The results from this study can be used to design pin-fin arrays that take advantage of unsteadiness and increase overall convective heat transfer for various industry components.

INTRODUCTION
Arrays of cylinders with fluid flowing through them are used in many types of heat exchange devices to enhance convective heat transfer by increasing the turbulence of the working fluid. In many cases, the cylinders also increase the available surface area for convective heat transfer. For an individual cylinder at moderate to high Reynolds numbers, flow around the cylinder is complex and often separates from the cylinder surface in an unsteady manner (von Karman vortex shedding phenomenon). This unsteadiness impacts the local and overall cylinder heat transfer. When cylinders are arranged in an array, wakes from upstream cylinders can further influence the instantaneous and mean flow behavior on downstream cylinders and thus affect the heat transfer on downstream cylinders.

The ratio of spanwise height of a cylinder to its diameter (H/D) is also important. For some applications, such as the trailing edge of turbine blades or in compact electronics cooling, the endwalls that bound the flow are close to each other, resulting in low aspect ratio cylinders, referred to here as pin fins. For aspect ratios on the order of one (examined in this study), the overall heat transfer in the array is thought to be influenced nearly equally by both the pin fin and the endwall flow, unlike long cylinder arrays where the heat transfer is dominated by the cylinder.

There are many studies which have examined the time-averaged heat transfer in pin fin arrays, but few that have presented unsteady heat transfer measurements. However, the unsteadiness is a primary feature of the array, and understanding its nature may lead to improved array designs that take advantage of the unsteadiness for overall improved heat transfer performance. To that end, this study presents measurements of the unsteady heat transfer around the circumference of a pin at various locations within a staggered array and relates the results to previously published measurements of the pin wake flowfield.
**NOMENCLATURE**

D = pin fin diameter, 63.5 mm

D_H = channel hydraulic diameter

f = frequency

F_r = Frossling number, \( F_r = \frac{N_u D}{\sqrt{Re_D}} \)

G = voltage gain applied to signal by amplifier

h = heat transfer coefficient, \( h = \frac{q'}{T_s-T_m} \)

H = pin fin height and channel height, \( H=D \)

HFM = heat flux microsensor

I_{RTD} = current through RTD

k = thermal conductivity of air

LES = Large Eddy Simulation

\( Nu_D \) = Nusselt number based on D, \( Nu_D = \frac{hD}{k} \)

PSD = power spectral density

q" = heat flux

R = resistance of RTD

RANS = Reynolds-averaged Navier Stokes

\( Re_D \) = Reynolds number based on D and \( U_{max} \), \( Re_D = \frac{D\cdot U_{max}}{\nu} \)

R_i = initial (room temperature) resistance of RTD

RTD = Resistance temperature detector

St = Strouhal number, \( St = \frac{fD}{U_{max}} \)

t = time

TKE = in-plane turbulent kinetic energy

T_m = channel mean temperature

T_s = pin surface temperature

u = local x-velocity

\( U_m \) = channel mean velocity upstream of array

\( U_{max} \) = maximum velocity between pins, \( U_{max}=2\cdot U_m \)

v = local y-velocity

V = voltage

w = local z-velocity

X = streamwise direction

Y = spanwise (cross-stream) direction

**Greek**

\( \nu \) = kinematic viscosity

**Superscripts**

' = fluctuating quantity

\( \overline{\cdot} \) = time-average of quantity

\( \overline{\cdot}' \) = standard deviation of fluctuating quantity

**RELEVANT PAST STUDIES**

Local convective heat transfer around a cylinder in crossflow has been studied by many researchers (e.g. [1-4]) and there is not space here to enumerate all of them. However, the general consensus is that the local time-average heat transfer is high at the stagnation point and decreases around the circumference of the cylinder due to boundary layer development. At approximately 85° downstream (location can vary with Reynolds number, turbulence level, roughness, and other parameters), the local heat transfer reaches a minimum. Heat transfer behavior beyond the minimum point is a function of the Reynolds number, where for low \( Re_D (<4,000) \) there is little change, but for high \( Re_D \), there can be a significant increase due to the unsteadiness in the separated shear layer around the backside of the cylinder, which increases convective transport [5].

Due to the importance of the unsteady flow around a cylinder, a few researchers have examined the local instantaneous heat transfer. A series of studies by Scholten and Murray [6, 7] investigated freestream conditions on cylinder heat transfer. For low freestream turbulence, Scholten and Murray [6] found that heat transfer fluctuations at the cylinder leading edge were weak but coupled to the cylinder vortex shedding frequency, while backside fluctuations were strong but with more random behavior. A companion study [7] found that high turbulence levels from a bar grid reduced the dependency of the time-resolved heat transfer on vortex shedding events. Nakamura and Igarashi [8] measured unsteady heat flux around a cylinder using the same type of sensor as in this study. For Reynolds numbers above 6,000, they noticed a local plateau in the time average and fluctuating heat transfer at around 150° from the stagnation, corresponding to the separation of the recirculating flow at the cylinder backside.

Cylinder heat transfer enhancement due to upstream wakes is particularly important in an array of cylinders, which are found in many types of heat exchangers, and yet little work exists on the unsteady convective heat flux. Scholten and Murray [9] found heat transfer fluctuations were well correlated to vortex shedding events in the first row of a three-row cylinder array, but there was less direct correlation for downstream rows due to increased random turbulence in the array.

There are, however, many studies that investigate the time-average behavior of the heat transfer in a cylinder array. Low aspect ratio cylinder arrays (also known as pin fin arrays) are unique relative to long cylinders due to the potentially significant impacts of the close endwalls. Brigham and van Fossen [10] measured lower array-average heat transfer for aspect ratios less than four, and Ostanek and Thole [11] found a shorter wake closure length for low aspect ratio pins, relative to an infinitely long cylinder. Many studies indicate how the row-averaged convective heat transfer increases significantly for the first few rows in an array, and then levels off further into the array [12-15]. The initial increase in the first few rows is due to the elevated levels of turbulence from upstream pin wakes, and in subsequent rows, the turbulent flow in the array becomes fully developed. Local static pressures and time-average heat transfer around a short aspect ratio pin were presented by Ames, et al. [16]. They found high heat transfer downstream of the third row due to the turbulent wake of row 1, even when corrected for an effective approach velocity determined by upstream pin wakes and blockage. Hotwire measurements by Ames and Dvorak [17] showed significant increases in pin wake fluctuations with Reynolds number and subsequent large backside heat transfer levels.

More recent work has examined the spatio-temporal flowfield in low aspect ratio pin fin arrays. Delibra, et al. [18] applied a hybrid RANS-LES model to compute the flow and heat...
transfer in a low aspect ratio array and commented on the importance of both large-scale vortex features as well as small-scale turbulence on heat transport. Ostanek and Thole [19] investigated various pin streamwise spacings in the same short aspect ratio pin fin array as in this study. They found that periodic vortex shedding was strongly suppressed, and its Reynolds number dependence was increased, by closely spaced arrays. Proper orthogonal decomposition analysis by Ostanek and Thole [19] indicated that periodic unsteadiness was not responsible for an increase in array heat transfer that was observed for tight array spacings by Lawson, et al. [14]. It is still unclear exactly how the large-scale and small-scale unsteadiness in a low aspect ratio pin fin array interact to influence the overall heat transfer.

This study examines the time-average and unsteady convective heat transfer around the circumference of a low aspect ratio cylinder at various locations in a staggered array. In this way, this work extends the studies of Nakamura and Igarashi [8] and Ames, et al. [16], and provides important information about the local heat transfer behavior that can be correlated with the time-resolved flowfield results of Ostanek and Thole [11]. A more complete understanding of the turbulent convective heat transfer in a staggered array may aid in the development of improved non-uniformly staggered arrays ([12, 20]) or arrays tailored to the specific performance required in an application.

**DESCRIPTION OF EXPERIMENT**

The wind tunnel used in this experiment is a recirculating layout driven by a blower, as shown in Figure 1. The blower provides flow to a supply plenum that is the same width as the rectangular flow channel that makes up the test section, but the plenum has a height that is 14 times the test section height (height is the dimension out of the page in Figure 1). A quarter-round convergence is mounted upstream of the rectangular flow channel to smoothly direct flow from the large plenum into the channel. The rectangular channel has a height of 63.5 mm (dimension out of the page in Figure 1), a width of 1.13 m (width to height aspect ratio of 17.8), and a length of 5.87 m. A staggered pin fin array begins approximately 20D downstream of the start of the rectangular channel. Previous measurements have verified a fully developed flow condition entering the array [11]. The array contains seven rows of staggered pin fins with diameter D=63.5 mm in the X-direction at a streamwise spacing of X/D = 1.73 and a spanwise spacing of Y/D = 2 (spacings remained constant throughout this study). All pins in the array span the entire 63.5 mm height of the channel and have thin foam tape at their ends, so that no flow goes over the pin ends, only around the circumference. Also, the pin rows extend across the entire width of the channel as indicated in Figure 1. In Figure 1, the rows are the vertical columns of pins, with row 1 at the left and row 7 at the right. The flow from the rectangular channel dumps to an exhaust plenum with a heat exchanger to control flow temperature, and then passes through a venturi to measure overall mass flow through the array so that channel mean velocity (U_m) and Reynolds number can be determined. The Reynolds number reported in this study is based on the pin diameter and the maximum velocity U_max between pins, where U_max=2*U_m for the Y/D=2 spacing. U_max is determined by using conservation of mass to estimate the local velocity at the minimum Y-direction spacing between pins in a row, as indicated by Figure 1. In this study, Reynolds numbers were varied from 3,000 to 50,000.

The test pin fin with an integrated heat flux microsensor (HFM), represented in Figure 1 as an orange pin in row 3, is positioned in various rows within the array. The test pin is made of aluminum in two separate halves, so that the HFM can be installed at the pin midplane; see exploded assembly in Figure 2. A central hole aligned with the axis of the pin houses a heater cartridge that heats the aluminum pin uniformly in order to provide a constant temperature to the entire surface of the cylinder. The pin Biot number, based on the spatially averaged heat transfer coefficient around the pin, ranges from 0.006 to 0.035 over the range of heat transfer coefficients measured. The cylinder also contains locating pins to ensure that its two halves are aligned, and screws to hold the pin together. The pin can be rotated around its axis while installed in the wind tunnel, so that the HFM can be positioned at any desired circumferential location. Just like the other pins in the array, the test pin height is the same as the test section height so that there is no gap on either end of the pin. Thin foam tape is attached to the ends of the test pin to thermally insulate the pin and prevent leakage flow, but also allows for circumferential rotation of the pin to position the sensor. No other pins in the array are heated in this...
study, so that the approach channel temperature to the pin is constant.

The HFM (Vatell Corporation) is a thermopile-based heat flux gauge that measures differential temperatures across a very thin dielectric substrate. An integrated resistance temperature detector (RTD) is located around the gauge, so that both local heat flux and surface temperature are measured. The HFM has an overall circular sensing area of 6 mm in diameter, including the RTD around the heat flux gauge; thus it spatially integrates the heat flux over approximately 10° of the pin cylinder circumference (see Figure 2). The gauge has a thin high-emissivity coating (used during the radiant sensor calibration) and has a 0-95% rise time of approximately 900 µsec [21]. Voltage output from the gauge is sent to a custom-designed low noise amplifier provided by Vatell Corporation, and then to a data acquisition system (DAQ). The DAQ samples the amplified voltage signal at a rate of 1100 Hz for 10 seconds (11,000 samples) and records it to disk for later processing. For this study, the sensor and amplifier were brand-new and were simultaneously calibrated by the manufacturer, in a thorough process involving a standardized reference gauge [21].

The test procedure consists of initially zeroing pressure transducer and HFM voltage outputs prior to turning on the blower. Then, the desired channel Reynolds number is achieved by adjusting the blower variable frequency drive. Electric current is supplied to the heater cartridge and the wind tunnel runs for an hour to ensure a steady pin temperature has been reached. Figure 3 indicates the consistency of the pin surface temperature at each circumferential location for a wide range of Reynolds numbers and row positions. Once a steady temperature condition is confirmed, voltage data is recorded from the HFM at 10° intervals along the circumference by rotating the pin about its axis. The circumferential location of the HFM around the pin fin is determined by an angular coordinate system, with 0° denoting the upstream stagnation point and 180° at the back of the pin fin. A few measurements at locations between 180° to 360° were obtained to check the symmetry of the heat flux measurements around the circumference, which were nominally 15% different between corresponding sides of the pin. As discussed later, part of the disagreement in symmetry is likely due to natural convection that occurs in this test section.

A custom post-processing script is used to apply the sensor calibration to the instantaneous recorded voltages from the HFM to obtain instantaneous total heat flux. In this script, the measured RTD voltage is used to determine the RTD resistance in Eq. (1), and then the calibration for the RTD, provided by the manufacturer during calibration, is applied to the resistance in Eq. (2) to determine the sensor temperature:

\[
R(t) = \frac{V_{\text{RTD}}(t)}{I_{\text{RTD}} + g_{\text{RTD}}} + R_1 \quad \text{Eq. (1)}
\]

\[
T_s(t) = c \times R(t) + d \quad \text{Eq. (2)}
\]

where \(c\) and \(d\) are the temperature sensitivity [ohms/°C] and temperature offset [°C] calibration constants, respectively. The instantaneous heat flux is determined by converting the measured HFM voltage using Eq. (3):

\[
q^*(t) = \frac{V_{\text{HFM}}(t)}{g_{\text{HFM}}} \quad \text{Eq. (3)}
\]

where \(g\) is the temperature correction coefficient for the sensor [µV/(°C*W/cm²)] and \(h\) is the sensitivity of the sensor [µV/W/cm²], both of which are provided during the manufacturer calibration.

Total heat flux measured by the sensor is corrected for radiation losses by assuming exchange with large isothermal surroundings at the mean air temperature in the channel. Note that the pin is painted flat black to match a similar coating on the HFM surface that was applied to increase its absorptivity during the radiation calibration performed by the manufacturer. Because of this, radiation losses in this experiment are approximately 16% of the total heat flux. Conduction losses are assumed to be negligible, since the sensor is located at the pin midspan and the endwalls of the tunnel are constructed of low thermal conductivity Lexan and glass. After obtaining the convective heat flux, the script determines the instantaneous Nusselt number using the instantaneous convective heat flux, pin surface temperature, and channel freestream temperature, and calculates the mean and standard deviation of the processed signal.

Measurement Validation

To validate the sensor measurements, the time-mean Nusselt number was obtained for the heated pin located in a single-row array (no downstream pins), as well as in the first row of a multi-row array with six rows of pins downstream (total of seven rows). The results are compared to previously published results (Ames, et al. [16], Ostanek and Thole [22], Kirsch, et al. [23]) in Figure 4. This figure shows the mean Frossling number, \(Fr\) (Nusselt number over the square root of Reynolds number, which is a common scaling for cylinder heat transfer) versus pin circumferential position. Note that the Ames, et al. study [16]
was for a different pin spacing (X/D=Y/D=2.5) and for pins with H/D=2. The disagreement between Ames, et al. [16] and Ostanek and Thole [22] from 100° to 180° is due to the different streamwise pin spacing; Ostanek and Thole found that closer streamwise spacings (small X/D) suppress backside pin heat transfer.

In Figure 4, the pin heat transfer in this study is slightly higher near the stagnation, and lower around 100 degrees and 130 degrees, relative to prior studies. There are likely two possible contributing factors for the disagreement. Due to the orientation of the tunnel, the test pin axis is horizontal (the cylinder axis is parallel to the lab floor), so for high pin temperatures, natural convection can occur in a direction transverse to the channel flow direction. For quiescent air, the front stagnation for natural convection would be at 90° in our pin coordinate system, and the convection plume would ascend from the 270° position. We estimate that natural convection could be 28% of the forced convection at $Re_D=3,000$, and 12% of the forced convection at $Re_D=20,000$. Unfortunately, it is difficult to correct for this since the local value of the natural convection coefficient would need to be determined, and it is unlikely that a measurement of natural convection coefficients at quiescent conditions would be appropriate to correct the measured local convection in the mixed-mode conditions in the experiment, since the thermal boundary layer on the pin would be influenced by both the natural and the forced convection.

Another possible consideration is the nature of the HFM surface. Although the heat flux sensor is installed so its edges are flush with the pin fin surface, the sensor face itself is flat (not curved). Over the 10° arc that the sensor covers, this means that the maximum deviation of the flat sensor face from the ideal cylinder curvature is 0.12 mm (0.004”). The manufacturer does not produce curved sensors, and adding a coating to create a curved surface would negatively affect the sensor response.

The temporal response of the sensor was checked by examining the FFT of the instantaneous heat flux measurements at 90° for a pin in a single row array. At this circumferential position, the FFT contains a strong amplitude peak at a frequency of 12 Hz, for $Re_D=20,000$ (see Figure 5). The estimated Strouhal number for the heat flux fluctuations, based on pin diameter and maximum velocity, is St=0.16, which compares very well to the non-dimensional vortex shedding frequency of St=0.17 in the wake of a single row low aspect ratio pin reported by Ostanek and Thole [11]. Note also that the expected important frequencies in the pin fin array were well within the maximum sampling frequency of the HFM.

### Uncertainty Analysis

Overall uncertainty in the experimental measurements were found using a propagation of errors method at a 95% confidence level [24]. Ostanek and Thole [11] reported an uncertainty of 3.5% for $Re_D$ for this wind tunnel. Bias uncertainty in heat flux, as determined by the manufacturer, is 3.0%. To determine precision uncertainty, 10 samples were taken at circumferential locations of 90° and 180° for row 3 at $Re_D=20,000$. Precision uncertainty of time-average heat flux is approximately 1% for both locations, and precision uncertainty of RMS of heat flux is 0.8% and 1.1% at 90° and 180°, respectively. Overall uncertainty for the difference between pin surface and freestream temperature is approximately 1.4%, and thus the overall uncertainty in time-average Nusselt number is 3.71%. However, due to the indeterminate contribution of natural convection described earlier, a more conservative estimate of overall uncertainty is 13%. The estimated circumferential position accuracy of the center of the sensor is ± 1.5°; recall that it spatially integrates over a 9.5° arc.

### RESULTS

The mean and unsteady heat transfer around the pin is described as a function of Reynolds number and row position for the low aspect ratio staggered array. The results are explained in the context of flowfield measurements obtained previously by Ostanek and Thole [11, 25] for the same low aspect ratio pin.
geometry. Finally, the effect of the closely-spaced multi-row array versus a single row array is analyzed.

Time-Averaged Heat Transfer in an Array

The time-averaged Nusselt number was calculated for tests at Reynolds numbers of 3,000, 10,000, 20,000, and 50,000 while the HFM pin was positioned in rows 1, 3, and 5 in the array. Figure 6 plots the time-averaged Nusselt number against the circumferential position around the pin for rows 1 and 5 at all Reynolds numbers. In this and subsequent figures, a circumferential position of 0° corresponds to the forward stagnation point. Figure 6 shows that as the Reynolds number increases, the magnitude of heat transfer increases across all rows. This effect occurs because increasing Reynolds number results in decreased boundary layer thickness, which in turn increases the convective heat transfer around the pin. Also from this figure, it is apparent that the heat transfer in a downstream row is higher than the first row, due to the presence of freestream turbulence from upstream pin wakes. For a given Reynolds number and row position, the time-average heat transfer around the pin is high at the stagnation and decreases along the circumference, reaching a minimum at approximately 90°-110° depending on Reynolds number. The heat transfer then increases along the backside of the pin due to transition, as well as turbulence in the pin wake.

The Frossling number, \( Fr = \frac{Nu_D}{\sqrt{Re_D}} \), is another way to present the cylinder heat transfer in a way that generally collapses the effect of Reynolds number, and is commonly used in cylinder heat transfer nomenclature. Figure 7 shows the time-mean Frossling number for row 1 at all of the Reynolds numbers in this study. Due to the natural convection described earlier that may be significant at low \( Re_D \), the \( Fr \) number at the stagnation does not collapse to a common value for all of the Reynolds numbers. However, it is apparent that \( Fr \) follows the same general pattern of decrease from 0° to 90° for all Reynolds numbers as the laminar boundary layer on the pin surface grows. The location of the minimum heat transfer point is a function of \( Re_D \). For a high \( Re_D \) of 50,000, the minimum \( Fr \) occurs earlier along the pin surface at 90°, while the minimum \( Fr \) for \( Re_D \) of 3,000 does not occur until 110°. This is due to boundary layer transition to turbulence occurring further upstream as the Reynolds number increases [26]. Beyond the minimum \( Fr \) point for \( Re_D \) of 10,000 and higher, there is a sharp increase due to the turbulent boundary layer which then drops briefly as that boundary layer grows. Finally, beyond 140°, heat transfer on the backside of the pin starts increasing. As will be described later, this increase in backside heat transfer is due to separation of the boundary layer from the pin surface and increased turbulence in the wake of the pin. The increase in time-mean heat transfer at the backside of the pin is larger for high \( Re_D \) versus low \( Re_D \), because of the larger range of turbulent scales that would be present.

Compared to Row 1, time-mean \( Fr \) at the stagnation point increases in rows 3 and 5 as seen in Figures 8 and 9, respectively. This increase is due to higher freestream turbulence in the flow caused by upstream rows. The freestream turbulence also impacts the circumferential location of the minimum \( Fr \), where for rows 3 and 5 the minimum point occurs between 80° and 90° regardless of \( Re_D \). In this case, the freestream turbulence causes boundary layer transition to occur earlier along the pin, and at a common location regardless of \( Re_D \). Beyond the first minimum heat transfer point for \( Re_D > 20,000 \) in Figure 8, the mean \( Fr \) peaks as the pin boundary layer transitions to turbulence and then decreases as the turbulent boundary layer grows along the pin. Because of the turbulent pin boundary layer, separation is delayed until after the second \( Fr \) minimum around 130°. As was seen in row 1, beyond a circumferential location of 140°, the turbulent wake of the pin results in increasing heat transfer at the backside of the pin.

A comparison across Figures 7-9 shows that from rows 1 to 5 for \( Re_D > 10,000 \), the backside time-mean heat transfer on a pin is nearly the same, despite the significant increase in heat transfer on the forward stagnation for rows downstream of row 1. This might be expected due to the vigorous turbulence in the pin wake.

![Figure 6. Time-average Nusselt number around the pin for a range of row positions and Reynolds numbers.](image1)

![Figure 7. Time-average heat transfer around a first row pin at various Reynolds numbers.](image2)
for high Reynolds numbers. However, at a low $Re_D$ of 3,000, the first row backside heat transfer (Figure 7) is relatively low compared to the downstream rows (Figures 8-9). To explain this, measurements of the turbulent kinetic energy (TKE) by Ostanek and Thole [11] are presented in Figure 10. This figure shows contours of the sum of the squares of the in-plane velocity fluctuations ($u$ and $v$), normalized by the square of the channel maximum velocity, for $Re_D=3,000$ and 20,000. Because the $w$-velocity fluctuation component was not measured, this is not a true turbulent kinetic energy, but it does indicate the in-plane turbulent fluctuation intensity. Note also that these fluctuations include both the random and deterministic (vortex shedding) flowfield fluctuations; that is, vortex shedding events are not filtered out of the data. For downstream rows, Figure 10 shows higher normalized turbulence in the flowfield around the pins for $Re_D=3,000$ versus $Re_D=20,000$. This result is due to the smaller range of turbulent scales for the low Reynolds number. For low $Re_D$, deterministic fluctuations (vortex shedding) are a larger contribution to overall turbulence, and also are not as effective at dissipating the upstream pin wake turbulence, resulting in higher normalized turbulence in downstream rows. This is consistent with hotwire measurements by Ames, et al. [16] and Ames and Dvorak [17]. Note that the highest turbulence levels are located in the shear layers between the flow around the pin and the separated flow in the wake. Turbulence levels closer to the pin surface are likely attenuated by the pin, although measurements were not obtained in that region.

Examination of Figure 10 for the first row wakes indicate that the in-plane fluctuations at the backside of the first row pin are relatively low at $Re_D=3,000$, compared to higher levels at a $Re_D$ of 20,000. The higher level of fluctuations would be expected to increase local heat transfer in the separated flow on the backside of the pin, which is shown in Figure 7. However, in the downstream rows shown in Figure 10, local turbulence levels on the backside of the pin in the wake, close to the pin surface, are nominally the same. This similarity is consistent with the similar $Fr$ magnitudes seen at 180° in row 5 (Figure 9).

**Unsteady Heat Transfer in an Array**

In addition to the time-average heat transfer, the HFM enabled acquisition of the unsteady heat transfer at each circumferential location, which is unique in the literature for low aspect ratio pins. Figure 11 shows an example of the time trace from the HFM, presented as instantaneous Nusselt number, for three circumferential positions around a pin in the first row of the array, at $Re_D=20,000$. It can be seen that the signals are turbulent, and as was just discussed, the time-mean value of heat transfer is also high at both the front stagnation (0°) and backside of the pin (180°), but the heat transfer fluctuations are much lower for the stagnation of the first row pin relative to its backside. The high $Nu_D$ at the front of the pin is due to the developing thermal boundary layer, while the unsteady cylinder wake contributes to the high $Nu_D$ at the backside. At 90° for the pin, both the mean value and unsteadiness in $Nu_D$ are low.
The unsteady Nusselt number was processed to extract the fluctuations and their standard deviation, and is shown in Figure 12 as the standard deviation of the Frossling number. Note that \( \text{Re}_D \) is not determined in an instantaneous sense, so only the standard deviation of \( \text{Nu}_D \) is determined from the HFM, and then is normalized by the square root of the \( \text{Re}_D \) for that case.

In Figure 12, heat transfer fluctuations are relatively low at the first row pin stagnation due to a lack of disturbances in the fully developed approach flow, and remain relatively constant up to a circumferential position of about 70° along the pin. At the minimum time-mean heat transfer point (see Figure 8), the fluctuations are also at a minimum. The fluctuations increase for locations beyond 100° along the pin circumference due to the vortex shedding and turbulent wake of the pins. The increase in fluctuations along the pin is also somewhat dependent on Reynolds number, with low \( \text{Re}_D \) showing only a small increase in fluctuating heat transfer, but above \( \text{Re}_D = 10,000 \), there is a more significant increase in backside fluctuation levels. However, there is little difference in the pin backside fluctuation between Reynolds numbers for \( \text{Re}_D > 10,000 \), suggesting that the vortex shedding mechanism is insensitive to Reynolds number above that value (up to the maximum tested). This corresponds to the lack of variation in backside time-average heat transfer between various \( \text{Re}_D \) cases as seen in Figures 7-9.

Figures 13 and 14 show the standard deviation of the fluctuating Frossling number for rows 3 and 5, respectively. In contrast to row 1, stagnation point fluctuating heat transfer is much higher, with levels that are nominally five to six times larger than in row 1. Fluctuation levels decrease along the pin circumference up to the minimum fluctuation point, and then increase again along the backside of the pin. The minimum fluctuation level moves forward along the pin toward stagnation as the Reynolds number increases, indicating an earlier start to boundary layer transition.

In rows 3 and 5 (Figures 13-14), the standard deviation of \( \text{Fr} \) appears to have a significant Reynolds number dependence. At low \( \text{Re}_D \), there is a dramatic variation in pin heat transfer fluctuations, with low fluctuations at approximately the same location as the time-mean minimum \( \text{Nu}_D \). As \( \text{Re}_D \) increases to 50,000, the normalized heat transfer fluctuations become nearly constant around the pin, such that the fluctuating \( \text{Fr} \) at 0° is lower than for a \( \text{Re}_D \) of 3,000, but is higher at 90°. This same type of \( \text{Re}_D \) dependence can also be seen in the in-plane turbulent kinetic energy measurements shown in Figure 10. For low Reynolds numbers in rows 3 and 5, the in-plane TKE at the forward stagnation is higher than at the backside of the pin. However, at \( \text{Re}_D = 20,000 \), the in-plane TKE at the forward stagnation is only slightly larger than the TKE at the back side of the pin. Based on the flowfield in Figure 10, the variation of the fluctuating heat transfer around the pin circumference would be expected to be more spatially uniform, as is observed in our measurements.
Ostanek and Thole [11, 19] found significant suppression of vortex shedding for arrays with streamwise row spacing below X/D=2.16 diameters due to constriction of the cylinder wakes in the tightly packed array. To understand how this would impact the mean and unsteady heat transfer around the cylinder for the closely spaced array in this study (X/D=1.73), a single row case was also studied, where there were no downstream pins behind the first row. This allows the maximum amount of unsteady motion for the cylinder wakes.

Figure 15 shows the flowfield measurements of Ostanek and Thole [11], where both the time-average velocity magnitude and the in-plane turbulent kinetic energy are presented for the wake of the cylinder. For the first row of the array, the wake of the cylinder is constrained, and the resulting in-plane TKE is mostly located in the separated shear layer that curls around the side of the cylinder. There is also a moderate level of in-plane TKE in the wake where the shear layers are breaking down and perhaps some mild vortex shedding is occurring. In contrast, the wake for the single row pin is much longer and is not constrained, and does not have as distinct of a shear layer. The in-plane TKE is generally higher throughout the wake due to vortex shedding as well as to stochastic turbulence (breakdown of shear layers).

The comparison between local time-mean Fr for a first row pin in an array, versus a single row pin, is shown in Figure 16 for ReD=20,000. Also shown on this figure is the unsteadiness in heat transfer, presented as the standard deviation of fluctuating Fr. The two mean Fr cases agree well regarding the level of heat transfer on the forward stagnation and up to 110° on the pin surface. However, between 120°-150°, the single row pin exhibits higher mean and unsteady heat transfer. Unfortunately, no flow measurements were obtained in this area, but it is conjectured that the single row pin generates strong vortex shedding that contributes to the high mean and fluctuating Fr. Further along the circumference of the pin, the time-mean Fr is similar between the two cases, suggesting that the stochastic turbulence in the wake dominates the unsteady heat transfer.

Figure 17 shows a frequency analysis of the unsteady heat transfer signal at three locations around the circumference of the pin, presented as a power spectral density (PSD) magnitude. At 0°, the magnitude of the low frequency events is low and there is only a short fall-off range where the amplitudes decrease with frequency. This behavior is the same for both a first-row pin in the array as well as for a single row pin. However, at 90°, there is a distinct difference between the amplitudes, with the single row pin experiencing larger low-frequency events. In addition, there is a strong peak at approximately 12 Hz, which is the expected shedding frequency from this low aspect ratio cylinder as described earlier. This is not seen in the first row result where the pin has close downstream rows, but Ostanek and Thole [19] also noted that close array spacing suppresses vortex shedding. Finally, at 180°, the PSD amplitude at low frequencies is high for...
both cases (although still higher for the single row pin) and there is a larger frequency fall-off region, implying a large range of turbulent scales in the wake of the pin.

Due to the suppression of vortex shedding that occurred for the close array spacing studied in this work, it was difficult to discern a strong peak in the PSD amplitudes for the multi-row array (see first row result at 90° in Figure 17), and thus we were unable to analyze changes in the behavior of the vortex shedding with row position or Reynolds number. However, this lack of strong magnitude at this array spacing is consistent with the conclusions of Ostanek and Thole [11] from examination of their time-resolved flowfield measurements. Future work should examine more widely spaced arrays which are known to have a stronger vortex shedding amplitude, as well as synchronized flowfield and unsteady heat transfer measurements.

CONCLUSIONS

A heat flux microsensor (HFM) with high response rate was designed into an isothermal heated cylinder (pin) that was inserted at various row locations in a low aspect ratio staggered array. The array geometry was fixed at a streamwise spacing X/D=1.73 and a spanwise spacing Y/D=2. The HFM could be rotated around the pin circumference to obtain spatial variation of mean and instantaneous heat transfer at the pin midspan. A range of Reynolds numbers from 3,000 to 50,000 was studied for the pin in rows 1, 3, and 5 of the seven-row array, as well as in a single row array with no downstream pins.

Measurements indicate a minimum in the time-average and fluctuating heat transfer at approximately 90°-110° from the pin stagnation, directly upstream of where boundary layer transition begins. This point of minimum heat transfer moves upstream with increasing Reynolds number in the first row, but is relatively insensitive to Reynolds number in downstream rows due to high freestream turbulence from upstream pin wakes. This turbulence also causes high time-mean stagnation heat transfer on downstream rows, but does not have a significant impact on the heat transfer around the backside of the pin due to the dominance of its own wake on its local heat transfer.

For low Reynolds numbers, normalized heat transfer fluctuations (standard deviation of the temporal heat transfer signal) around the circumference of downstream-row pins varies in a manner similar to the time-mean result, with high values at 0° and 180° and low values around 90°. However, for high Re_D, heat transfer fluctuation magnitudes are more uniform around the pin. These different behaviors suggest that the unsteady heat transfer in high Re_D cases is significantly affected by the large range of turbulent scales in the flow, and not just the unsteadiness of vortex shedding from the pin.

For a single row pin, the wake is much less confined than in the closely-spaced multi-row array tested in this study, and more significant vortex shedding occurs, as determined by increased heat transfer fluctuations and a definitive spike in the power spectral density distribution of the unsteady heat transfer signal.

Overall, this work provides additional insight into the effect of the unsteady flowfield on pin heat transfer, and could be used to understand how to optimize array spacing to take advantage of wake shedding and upstream turbulence for maximum heat transfer enhancement per row. Future work could also consider the simultaneous measurement of instantaneous velocity fields and pin heat transfer to develop more physics-based modeling of high freestream turbulence effects on heat transfer, in a manner similar to Hubble, et al. [27].

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