THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

98-GT-175

HEAT TRANSFER CONTRIBUTIONS OF PINS AND ENDWALL IN PIN-FIN ARRAYS: EFFECTS OF THERMAL BOUNDARY CONDITION MODELING

M.K. Chyu, Y.C. Hsing and T. I.-P. Shih
Department of Mechanical Engineering
Carnegie Mellon University
Pittsburgh, PA 15213

V. Natarajan
The BOC Group Technical Center
Murray Hill, NJ 07947

BREAK

ABSTRACT

Short pin-fin arrays are often used for cooling turbine airfoils, particularly near the trailing edge. An accurate heat transfer estimation from a pin-fin array should account for the total heat transfer over the entire wetted surface which includes the pin surfaces and uncovered endwalls. One design question frequently raised is the actual magnitudes of heat transfer coefficients on both pins and endwalls. Results from earlier studies have led to different and often contradicting conclusions. This variation, in part, is caused by imperfect or unrealistic thermal boundary conditions prescribed in the individual test models. Either pins or endwalls, but generally not both, were heated in those previous studies. Using a mass transfer analogy based on the naphthalene sublimation technique, the present experiment is capable of revealing the individual heat transfer contributions from pins and endwalls with the entire wetted surface thermally active. The particular pin-fin geometry investigated, S/D = X/D = 2.5 and H/D = 1.0, is considered to be one of the optimal array arrangement for turbine airfoil cooling. Both inline and staggered arrays with the identical geometric parameters are studied for $5{,}000 \leq Re \leq 25{,}000$. The present results reveal that the general trends of the row-resolved heat transfer coefficients on either pins or endwalls are somewhat insensitive to the nature of thermal boundary conditions prescribed on the test surface. However, the actual magnitudes of heat transfer coefficients can be substantially different, due to variations in the flow bulk temperature. The present study also concludes that the pins have consistently 10 to 20% higher heat transfer coefficient than the endwalls. However, such a difference in heat transfer coefficient imposes very insignificant influence on the overall array-averaged heat transfer, since the wetted area of the uncovered endwalls is nearly four times greater than that of the pins.

Nomenclature

D	pin diameter
H	pin height
h	heat transfer coefficient, equation (2)
h_m	naphthalene mass transfer coefficient, equation (1)
k	thermal conductivity of air
m	mass transfer flux of naphthalene from pin-fin surface
n	power index
Nu	Nusselt number, hD/k
Pr	Prandtl number, ν/α
q	heat flux from pin-fin surface
Re	Reynolds number, $U_m D/\nu$
S	pin spacing in spanwise direction
Sc	naphthalene-air Schmidt number, ν/K
Sh	pin-resolved naphthalene mass transfer Sherwood number, $h_m D/K$
T	temperature
U_m	mean flow in the minimum flow area
X	pin spacing in streamwise direction

Greek Symbol

α	thermal diffusivity
K	naphthalene-air diffusion coefficient
ν	kinematic viscosity of air
$\rho_{v,w}$	vapor mass concentration or density of naphthalene on pin-fin surface
$\rho_{n,b}$	vapor mass concentration or density of naphthalene in flow bulk
ρ_s	density of solid naphthalene

Subscript

A	array-averaged
b	bulk
C	column-averaged
N	normalized by mean value

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Stockholm, Sweden — June 2–June 5, 1998
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1998

R row-averaged
w wall, fin surface

INTRODUCTION

Heat transfer with flow over an array of circular pin-fins has been the subject of extensive research in the past because of its importance in a wide variety of engineering applications. Pin-fins with small height-to-diameter ratios, namely on the order of unity, are often used as heat transfer augmentation devices for cooling of turbine airfoils. They are especially effective for internal cooling passage near the blade trailing section where the pin-fins (the so-called pedestals) also serve a structural purpose in bridging the narrow span between the pressure and suction surfaces. Under this context, heat transfer and pressure loss for short, circular pin-fin arrays have been studied extensively since the early 1980's. Substantial contribution was made by groups at NASA-Lewis (VanFossen, 1982; Simoneau and VanFossen, 1984; Brigham and VanFossen, 1984) and Arizona State University (Metzger et al. 1982a, 1982b, 1982c, 1984, 1986). Their findings collectively suggest that the transport phenomena associated with short pin-fins are considerably different from that of the conventional "cylinders in crossflow" problem (Zukauskas, 1972) where the cylinders are much longer, say H/D > 8, and without endwalls present. Important results concerning staggered pin-fin arrays from these studies were compiled in a review article by Armstrong and Winstanley (1987). Using a heat/mass transfer analogous system, Chyu (1990) later reported that the presence of fillets at the cylinder-endwall junction reduces the heat transfer from a pin-fin array. Without fillets, his mass transfer results agree favorably with those using direct heat transfer measurements. Recently, Chyu et al. (1996) found that square or diamond shaped pin elements may be more effective for heat transfer augmentation than circular pins. However, actual manufacturing of sharp-edged pins may be difficult.

One issue concerning the total heat transfer from a pin-fin array as well as the actual split of heat transfer between the pins and uncovered endwalls is of great importance for the thermal design of turbine airfoils. In reality, both pins and endwalls participate in the process of cooling the component. However, to ease the experimental setup and heat transfer measurements, virtually all the previous studies have in one way or another involved certain approximations of thermal boundary conditions in the laboratory models. Either the pins or the endwalls, but generally not both, are heated or thermally active. The well known correlation obtained by Metzger and his associates (1982a, 1982b, 1982c) are based on heat transfer measurements on the uncovered portion of the endwall, while the pins are made of non-conductive basswood. On the other hand, Simoneau and VanFossen (1984) used a single, heated pin and tested it in different positions within the array to determine the row-resolved heat transfer coefficient. The mass transfer work by Chyu et al. (1990, 1996) and Natarajan and Chyu (1994) had the surface of all the pin elements coated with a thin layer of naphthalene and the pins were situated on an aluminum tooling plate without naphthalene coating. This setup is analogous to a heat transfer situation in which each pin is isothermally maintained, i.e. the pin has a fin efficiency of unity, and the endwall is adiabatic. Using the same mass transfer technique, Chyu and Goldstein (1991) examined the detailed distribution of mass (heat) transfer coefficient on a naphthalene coated endwall while the pins were kept mass transfer inactive.

Only a few studies in the pin-fin literature are focused explicitly on the difference of heat transfer between the pins and endwall. VanFossen (1981) reported that, for S/D = X/D = 3.46 and $0.5 \leq H/D \leq 2$, the heat transfer coefficient on the pin surface is approximately 35% higher than that of the endwall. His conclusion is drawn by comparing the overall heat transfer from pins made of highly conductive material with that of nonconductive wooden pins. Using the same approach, Metzger et al. (1984) reported that, for $2.5 \leq S/D \leq 3.5$, $1.5 \leq X/D \leq 3.5$, and H/D = 1.0, the pin surface heat transfer coefficient is approximately double the endwall value. In a separate study Metzger et al. (1982b) suggested that the heat transfer coefficient measured on the endwall is somewhat insensitive to the fact whether the pin is conductive or non-conductive. Using the mass transfer analogy, Chyu (1990) reported comparable heat transfer coefficient on the surface of the pins and endwall for both inline and staggered arrays, S/D = X/D = 2.5, H/D = 1. While all pins were made mass transfer active in his study, only one-row of the endwall was active during the experiment, so the boundary condition is not a perfect representation of the real situation. Al Dabagh and Andrews (1992) later employed a transient heating technique to evaluate the heat transfer contributions from the pins and the endwall for three staggered arrays, S/D = 2.0, X/D = 1.5 and H/D = 0.7, 1.0 and 2.2. Contradicting to all the previous findings, their results indicated that the endwall heat transfer coefficient is 15 to 35% higher than the pins.

The present study is an attempt to resolve the aforementioned controversy. It is speculated that the inconsistent results among the earlier studies may be attributable to the limitation as well as the uncertainty involved in the experimental systems. The convective transport with the flow passing through a pin-fin array is so complex that it can result in highly non-uniform local heat transfer coefficients distributed over the entire wetted surface. Conventional thermal methods based on thermocouple measurements of several selected points over a heat transfer surface may not be representative of the overall system behavior. In addition, heat loss is always a concern and is virtually unavoidable. Finally, individual studies have used different ways of prescribing thermal boundary conditions in their test rigs, which may lead to different results.

Given the priority to quantifying the heat transfer difference between the pins and endwalls, the mass transfer analogous approach based on naphthalene sublimation emerges as one of the most effective choices for the present use. Since the surface of all pins and endwalls can be separately coated with naphthalene, the concern of overall system "mass loss" as well as "internal mass leakage" between pins and endwall is virtually non-existent. In an unprecedented attempt, the entire test section, which includes all the pin wetted surface and the uncovered endwalls attached to both ends of the pins, is coated with naphthalene, hence is mass transfer active. By analogy, the mass transfer system as such is equivalent to a heat transfer system whose wetted surface is entirely isothermal. Mass transfer of individual pins and row-resolved endwall segments can be determined by weighing the amount of naphthalene sublimed during the test period. The weighing approach has been well documented and proven to be a highly accurate method for determining regional average data by numerous studies in the past.

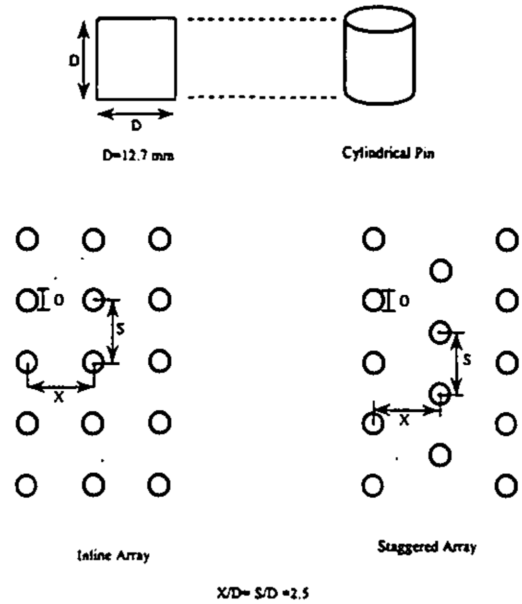


Figure 1 Pin-Fin Geometry

EXPERIMENTAL APPARATUS AND PROCEDURES

The geometry of pin-fin arrays as well as the housed channel is the same as that of an earlier study performed by one of the present authors (Chyu, 1990). The pin-fin elements are perfectly cylindrical in shape and made of aluminum rods approximately 12.5 mm in diameter. To facilitate a rational comparison, the two arrays studied here, an inline array and a staggered array, have identical geometric parameters: $H/D = 1$, $S/D = X/D = 2.5$. Figure 1 shows the geometry of the two arrays. The staggered array with these array parameters is deemed to be the most optimal configuration for turbine blade cooling. There are seven rows for both arrays, but the total number of pins is different. For the inline array, each row has five pins giving a total of thirty five pins. On the other hand, the number of pins per row in the staggered array alternates with five in the odd number row and four in the even number row, a total of thirty-two pins.

The test channel which houses the pin-fin arrays is made of aluminum tooling plate and has a rectangular cross-section, 159 mm wide and 12.7 mm high. Downstream of the test section is a 230 mm long discharge duct, which is approximately ten times the hydraulic diameter of the test section. Inlet flow to the array is introduced through a rectangular opening (25.4 mm wide and 12.5 mm high) on one side of the test channel. The center of the first row of pins is about a two pin-spacing from the furthest edge of the test section. Attached to the opening is an approximate 455 mm long aluminum duct which serves as the entrance section. Airflow is supplied by a 50 hp compressor and is monitored by a pressure regulator, a control valve, and an orifice before reaching the test section.

Each experiment starts with preparation of naphthalene coating on the pin-fin surface. This coating is achieved by dipping the cylinder, with both ends taped, into a pool of nearly boiling, molten naphthalene. During the dip, the cylinder is held by a tweezers at both ends and immersed in the liquid naphthalene for about one second. The naphthalene solidifies mm thick layer on the cylinder surface. Such a coating process generally results in a quality surface, so no additional polishing or machining procedure is necessary. Similar coating process also applies to preparing the endwall samples. The endwall on either top or bottom surface of the test channel is comprised of seven separated segments. Each segment, which represents the endwall of a given row, extends one-half pitch upstream as well as downstream to the pin axis of that row and spans across the entire width of the array. After the coating process, the pins and endwall segments are stored in a tightly sealed plastic box for at least 15 hours to ensure that they attain thermal equilibrium with the surrounding air.

Before a test run, all pins and endwall segments are separately weighed using an electronic balance with an accuracy of 10^{-2} mg in a 166 g range. They are then screw-mounted together, forming the test section of the desired array geometry. Having assembled the test section, the compressed air is induced and flows through the channel for about 30 minutes. During the test run, the system temperature is determined by the average readings of four thermocouples embedded in the inner surface of the two endwalls. The system temperature is used to evaluate the naphthalene vapor concentration, $\rho_{v,w}$, on the fin surface. Since the value of $\rho_{v,w}$ is very temperature sensitive, an isothermal system is highly desirable. A test is considered to be a failure and discarded if any two of the four thermocouple readings differ more than 0.2 C. After the test run, all pins and endwall segments are unscrewed and removed from the channel and each is weighed again. The weight difference for each individual components yields the amount of naphthalene sublimed during the test run. The average naphthalene mass sublimed is approximately 20 mg. The repeatability for all the test runs is within 8%.

HEAT / MASS TRANSFER ANALOGY AND DATA REDUCTION

The convective heat transfer coefficient, h, of each participating element is given by

$$h = q / (T_w - T_b) \quad (1)$$

where T_w and T_b are the element wall temperature and the bulk mean temperature in the channel, respectively. By analogy (Eckert, 1976), the mass transfer coefficient, h_m, of each cubic fin is

$$h_m = m / (\rho_{v,w} - \rho_{n,b}) \quad (2)$$

where $\dot{m}$ is the mass transfer rate per unit area which can be calculated from the weight change of the coated pin or endwall segment before and after the experiment. The naphthalene concentration at the wall, $\rho_{v,w}$, is obtained by evaluating the time-averaged naphthalene vapor pressure using the pressure-temperature correlation of Ambrose et al. (1974) in conjunction with the ideal gas law.

The value of naphthalene concentration in the bulk flow at a given row is determined by the total mass transfer upstream. The increase in bulk concentration naphthalene vapor within the domain of a specific row j can be expressed as

$$\Delta\rho_{n,b} = M_j / Q \quad (3)$$

where M_j is the mass transfer per unit time from all participating surfaces of the entire row j, and Q is the volumetric air flow rate through the channel. Since the air flow at the channel inlet is naphthalene free, it leads to

$$\rho_{n,b} = \sum_{1}^{j-1} M_j / Q \quad (4)$$

As the mass transfer system is essentially isothermal, the naphthalene vapor pressure and vapor concentration at the wall are constant. As mentioned earlier, this is equivalent to a wall boundary condition of constant temperature in heat transfer, and the pins thus can be regarded to have an ideal 100% efficiency.

The dimensionless local mass transfer coefficient, Sherwood number, Sh, is defined by

$$Sh = h_m D / K \quad (5)$$

where K is the naphthalene-air diffusion coefficient which is determined by taking the Schmidt number equal to 2.5; i.e.

$$Sc = \nu / K \quad (6)$$

Since the naphthalene concentration in the boundary layer is extremely small, the kinematic viscosity, ν, uses the value of air under the operating conditions. By analogy, Sherwood number can be transformed to its heat transfer counterpart, Nusselt number (Nu), using the relation

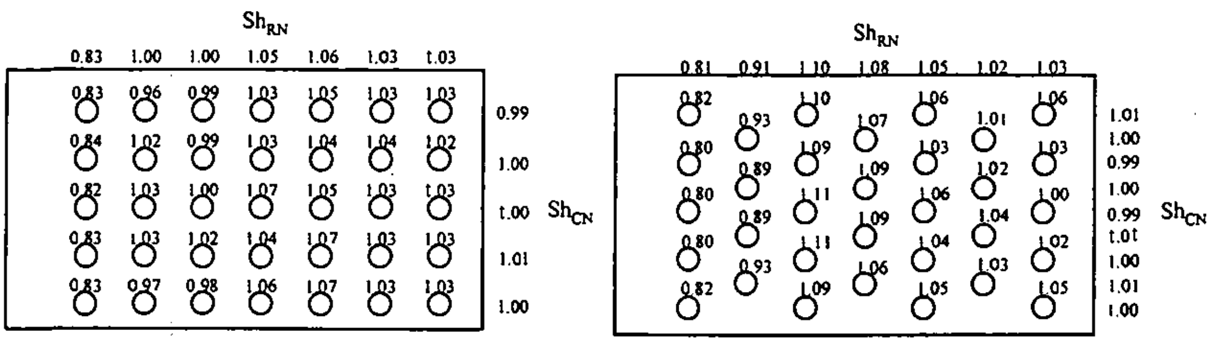
$$Nu / Sh = (Pr / Sc)^n \quad (7)$$

where Pr is the Prandtl number and the power index n is approximately equal to 0.4, according to Sparrow and Ramsey (1978). Using air (Pr =0.7) as the coolant, a direct conversion relationship can be obtained,

$$Nu = 0.6\, Sh \quad (8)$$

RESULTS AND DISCUSSION

To illustrate the quality of the present experiment, Figure 2 shows the relative magnitude of mass transfer among all individual pins. Here both pins and endwalls are mass transfer active. The numerical value marked atop each pin is the ratio of its individual pin-resolved Sherwood number, Sh, to the Sherwood number averaged over all pins in the array. The value marked outside the figure border are the row-average (Sh_{RN}, top border) and column-average (Sh_{CN}, right border) of the normalized Sherwood number. By analogy, these normalized values are equal to the ratios of corresponding Nusselt number in heat transfer. One significant feature revealed in Fig. 2 is the excellent data symmetry for both arrays, as the value of Sh_{CN}



(a) Inline Array

(b) Staggered Array

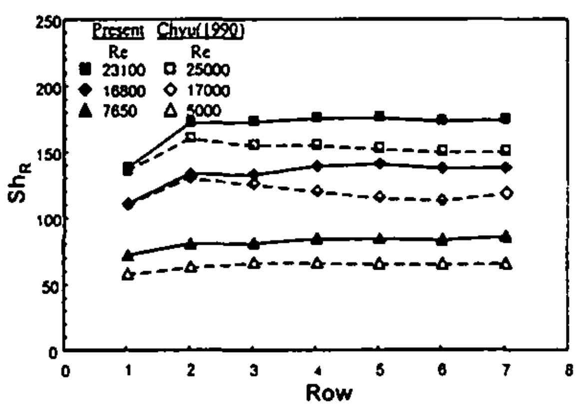
Figure 2 Relative Heat Transfer Coefficient on Pin Elements, Re = 16,800 varies less than ±1% across the array span. Although Fig. 1 displays the results for Reynolds number about 16,800 only, the same observation prevails over the entire range of the Reynolds number tested.

As a contrast to the invariant nature of Sh_C, the row-averaged mass transfer, Sh_R, varies relatively significantly along the streamwise direction, as shown in Fig. 3. In addition, the magnitudes of Sh_R depend also on the Reynolds number, the array geometry, as well as the participating boundary layer. According to Metzger et. al. (1982a) and Chyu (1990), the maximum Sh_R exists at the second row and the third row for the inline and staggered array, respectively, provided that the Reynolds number is sufficiently high, say Re ≥ 10,000. The maxima become less obvious when Re decreases. Despite such a similar findings, their conclusions, in fact, are based on quite different tests and both with partially active boundary conditions. Metzger et. al. (1982a) measured the row-averaged heat transfer over the endwall while the pins are made of basswood and thus are heat transfer inactive. Chyu (1990), on the other hand, measured the mass transfer from the pins and kept the endwall inactive. Since the array geometry of the latter is identical to that of the present study, the effects of mass transfer (thermal) boundary condition on the pin mass (heat) transfer can be realized by comparing the corresponding results between the two studies, as exhibited in Fig. 3. While the general trend of Sh_R versus row number and Re is by and large uninfluenced by the difference in boundary condition, the value of Sh_R is consistently higher, by approximately 10 - 20%, for the present study in which both pins and endwalls are mass transfer active. Such a difference is a result of simple interpolation, since the corresponding cases shown in Fig. 3 do not have identical Reynolds numbers.

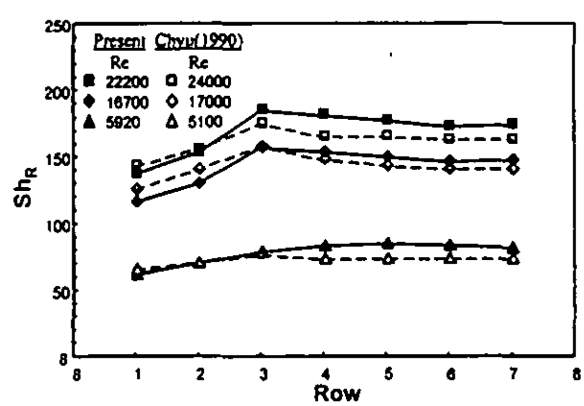
Based on the same data shown in Fig. 3, Figure 4 gives a different perspective of the results by presenting the enhancement factor relative to a smooth channel without pin array. The ordinate of Fig. 4 has dual meaning: Sh_R/Sh_o and Nu_R/Nu_o, where Sh_o and Nu_o are the Sherwood and Nusselt number, respectively, for the corresponding smooth channel case without pin array. The value of Sh_o is determined from the Dittus-Boettler equation for heat transfer (Kays and Crawford, 1980)

$$Nu_{Do}/Pr^{0.4} = 0.023\ Re_D^{0.8} \tag{9}$$

Where Re_D is the channel Reynolds number and Nu_{Do} is the fully developed Nusselt number based on the channel hydraulic diameter, D_h. To be compatible with the characteristic length used for Sh, Sh_o can be expressed by

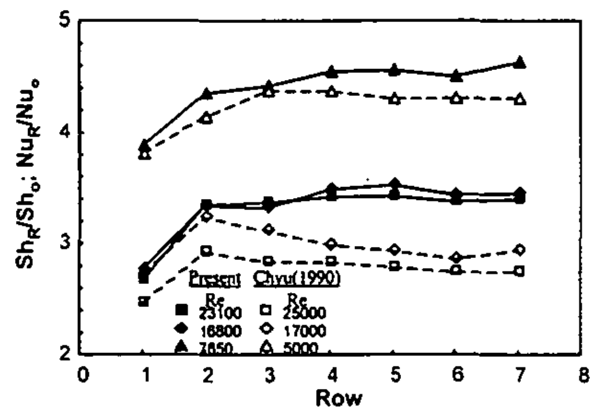


(a) Inline Array

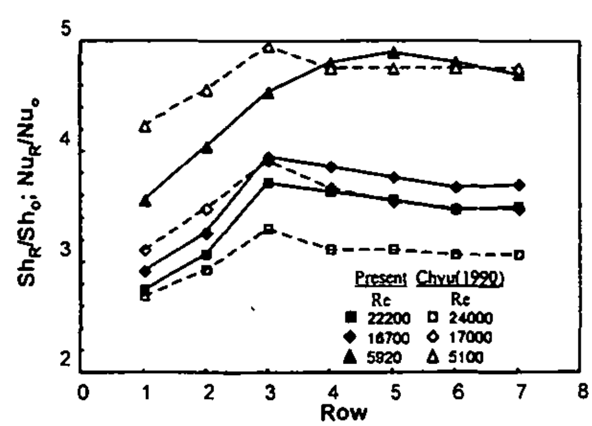


(b) Staggered Array

Figure 3 Heat Transfer Coefficient on Pin Surface



(a) Inline Array



(b) Staggered Array

Figure 4 Normalized Heat/Mass Transfer Coefficients

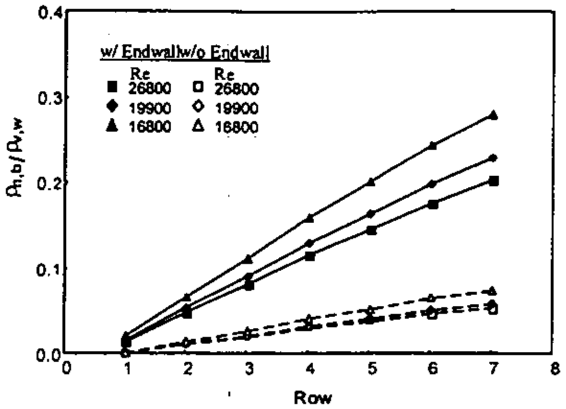


Figure 5 Variation of Bulk Concentration in Staggered Array

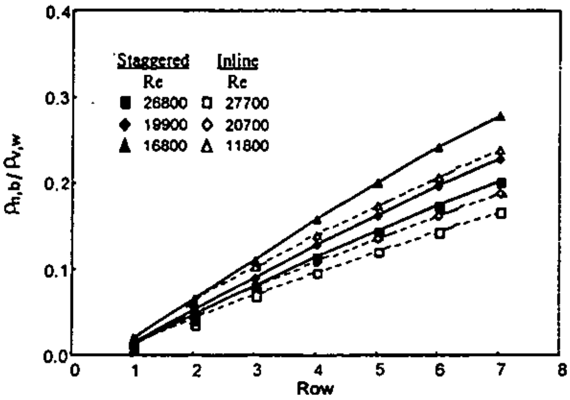


Figure 6 Variation of Bulk Concentration

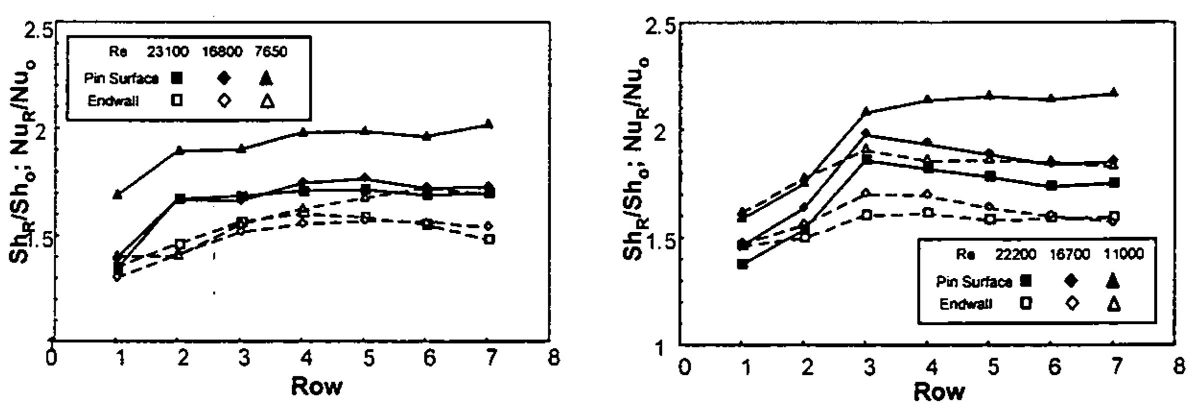
$$Sh_o/Sc^{0.4} = Nu_o/Pr^{0.4} = (D/D_h)\,(Nu_{Do}/Pr^{0.4})$$

$$= 0.0124\ Re_D^{0.8} \qquad (10)$$

Mass transfer results obtained from the present test channel without pins agree favorably with this correlation with an average error about 8% over the entire range of Reynolds number. In addition to the effects of boundary condition, the results shown in Fig. 4 also implies that the enhancement factor decreases with Reynolds number. While more detailed discussion and quantitative correlations are to be given later, the enhancement factor for the pin surface alone is roughly about 4 for Re ~ 5,000 and 3 for Re ~ 20,000.

One significant effect of thermal boundary conditions on the convective transport is to alter the nature of temperature distribution in the flow bulk. For given channel geometry and Reynolds number, the change of bulk temperature in a channel is expected to be greater with a fully active channel wall than without. By analogy, such a notion is equally applicable to mass transfer, as evidenced by Eqs. (3) and (4). Figure 5 reveals a comparison of the streamwise variation of naphthalene vapor concentration in the flow bulk with or without active uncovered endwall for the staggered array. The abscissa of the figure marked as "row" includes the pins in the row and the uncovered endwall ranging one-half longitudinal pitch upstream and downstream to the pin axis. The ordinate represents the magnitude of bulk mass concentration of naphthalene, ρ_{nb}, normalized by its counterpart near the wall, ρ_{vw}. Since ρ_{vw} is constant in the present naphthalene sublimation system, the ratio ρ_{nb} / ρ_{vw} is equivalent to the value of $T_b - T_i / T_w - T_i$ in heat transfer.

Most evident in Figure 5 is the substantial contribution of the mass transfer from the endwall toward the increase in bulk concentration. The difference in ρ_{nb} with endwall inclusion is nearly 4 to 5 times higher than without. Because the area of uncovered endwall over the entire pin-fin array accounts for nearly 80% of the total wetted area in the channel, complete negligence of the endwall heat transfer not only is unrealistic but may also be inaccurate in presenting a sensible heat transfer coefficient based on bulk temperature. Although the level of surface mass transfer rate under a highly turbulent flow is dominated primarily by the local hydrodynamics, rather than the details of thermal boundary condition, the corresponding local mass transfer coefficient, however, can vary significantly with the boundary condition via the change in bulk concentration in the test channel. Since $\rho_{vw} - \rho_{nb}$ is smaller toward downstream for the case of fully active boundary than that of only the pins active, the resulting heat transfer coefficient is higher for the



(a)Inline Array

(b) Staggered Array

Figure 7 Heat Transfer Coefficient on Pin Surface and Endwall

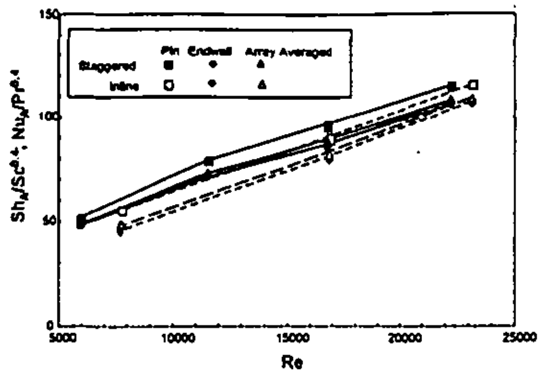


Figure 8 Array-Averaged Heat Transfer Coefficient

former, provided that m is either a constant or decreases with a lesser extent than that of $\rho_{vw} - \rho_{nb}$ toward downstream. This notion substantiates the finding displayed in Fig. 4, where the row-average mass transfer coefficient is consistently higher for the case with the entire test surface active. Regardless of the nature of boundary condition, the change of bulk mass concentration decreases with the Reynolds number. In addition, the staggered array tends to have a stronger increase in bulk concentration than the inline array for a given Reynolds number, as illustrated in Fig. 6. Such a difference between the two arrays appears to be widened toward downstream.

As a major finding from the present study, Figure 7 shows the comparison of heat transfer coefficient on the pin surface and the uncovered endwall region. Similar to Fig. 4, the data shown in Fig. 7 are presented as enhancement factors relative to the corresponding values of fully developed smooth channel, Sh_R/Sh_o or Nu_R/Nu_o. Note that Sh_R or Nu_R here represents the row-averaged Sherwood number or Nusselt number for either the pins or the endwall. The difference in the row-average transfer coefficient between the pins and endwalls varies to a certain extent with the Reynolds number, the array configuration, and the row position. Except for the first one or two rows, the heat transfer coefficient on the pin surface for both arrays is consistently higher than that of the endwall, by 10 to 20%. The difference widens with a decrease in Reynolds number. Although the overall trend that a pin has a higher heat transfer coefficient than the endwall agrees favorably with most of the earlier findings, the margin of difference observed here is relatively low. VanFossen (1981) reported a 35% differential, while Metzger et al. (1984) cautiously claimed that the difference is nearly 100%. On the other hand, the present finding contradicts the observation by Al Dabagh and Andrews (1992) who reported that the heat transfer coefficient on the pin surface is lower instead, by approximately 15 to 35%.

Further examination of Figs. 7(a) and 7(b) reveals that one of the most significant differences between the two arrays lies in the entrance region. With an exception for the case of the inline array with the lowest Reynolds number (Re = 7,650), the pins and uncovered endwall have very comparable magnitude of heat transfer coefficient in the first row for the inline array and the first two rows for the staggered array. This difference, on the other hand, may be fundamentally insignificant, since the second row of the staggered array is effectively part of the first row, as no pins situate directly in front of the row.

$$Sh_A/Sc^{0.4} = Nu_A/Pr^{0.4} = a\,Re^b$$

Array Configuration		a	b
Inline	Pin Surface Fin Surface Averaged Pin & Fin	0.155 0.052 0.068	0.658 0.759 0.733
Staggered	Pin Surface Fin Surface Averaged Pin & Fin	0.337 0.315 0.320	0.585 0.582 0.583

Table 1 Nu-Re Correlations

The most useful information for the turbine designers probably is the area-average heat transfer coefficient over the entire array. Figure 8 shows such area-average data, in the form of $Sh_A/Sc^{0.4}$ or $Nu_A/Pr^{0.4}$, varying with the Reynolds number for both arrays. As expected, the staggered array overall has a higher average heat or mass transfer coefficient than the inline array. The difference is about 10 to 20% for Re ≥ 20,000 and becomes less significant when the Reynolds number gets higher. Also included in the figure are the corresponding area-average data respectively for pins and endwalls. Largely because the endwall accounts for nearly 80% of total wetted area for either array, the array-average results are virtually equal to the corresponding endwall averages, rather than the pin averages. In other words, the 10 to 20% higher heat transfer coefficient on the pin surface has a marginal effect on the overall array heat transfer. However, this notion may change for different arrays in which pin surface accounts for a large portion of the array wetted area. Such a situation happens when an array has dense pin arrangements, i.e. smaller pitches, or taller pins.

Table 1 gives the power correlations between the average Nusselt number and Reynolds number. The inline array shows a stronger Reynolds number dependence than the staggered array, b = 0.733 versus 0.583. From the geometric standpoint, this trend is not surprising. The inline array is expected to preserve more channel flow characteristics as the pins in adjacent rows form two virtual walls for a channel. On the other hand, flow passing through the staggered array should experience stronger separation and stagnation effects. In the limiting case, the turbulent Nu-Re correlation should reveal a power index nearly 0.8 for smooth channel flow and 0.5 for separated flow over stagnant obstacles.

Compared to the well-known correlation reported by Metzger et al. (1982a) for the staggered array, i.e.

$$Nu/Pr^{0.4} = 0.080\,Re^{0.728}$$

for H/D = 1, S/D = X/D = 2.5, 1,000 ≤ Re ≤ 100,000,

the present correlation shows a relatively weaker dependence on Re, 0.583 Vs 0.728. A combination of different test range in Reynolds number and thermal boundary condition may be responsible for such a deviation. Metzger's correlation is based on heat transfer data measured over the endwall only, while the pins are virtually adiabatic. Despite this seemingly apparent disparity in Reynolds number dependence, the actual magnitudes of Nu derived from the two correlations, in fact, agree surprisingly well (< 10%) within the present test range, 5000 < Re < 25,000.

CONCLUDING REMARK

The information revealed from the present mass transfer measurements is expected to provide important guidelines for resolving a long-standing issue concerning the heat transfer contribution from different participating segments in a pin-fin array. The present study inherits two distinct features that may make its results more reasonable and reliable than those in the past. The first feature has to do with the implementation of an equivalent isothermal boundary condition over the entire test channel. This feature provides a baseline to quantify the effects of partially active or imperfect boundary conditions on the data accuracy of heat transfer coefficient. The second feature is the attainment of a true row-averaged data, which includes all the non-uniformity existing in the domain of a measured segment. Further affirming the general observation of VanFossen (1981) and Metzger et al. (1984), the present data collectively indicate that the heat transfer coefficient on the pin surface is higher than that on the uncovered endwall, by approximately 10 to 20%. However, the two earlier studies reported differences higher than the present finding. The primary cause of such disagreement is deemed to be a combination of imperfect boundary conditions and measurement techniques that may be insufficient for resolving the highly complex heat transfer characteristics inherited in pin-fin arrays. In more practical perspective, the 10 to 20% difference in heat transfer coefficient may have little impact on the overall array-averaged heat transfer. This result is mainly because the endwall accounts for nearly 80% of the wetted area for the present array configuration, $H/D = 1$, $S/D = X/D = 2.5$. Hence an experimental approach focused solely on the endwall measurement is expected to be more representative than focused on the pins.

REFERENCES

Al Dabagh, A.M. and Andrews, G.E., 1992, "Pin-Fin Heat Transfer: Contribution of the Wall and the Pin to the Overall Heat Transfeer," ASME Paper 92-GT-242.

Ambrose, D. Lawenson, I.J. and Sprake, C.H.S., 1975, "The Vapor Pressure of Naphthalene," *J. Chem Thermo.*, pp. 1173-1176.

Armstrong J. and Winstanly, D. 1987, "A Review of Staggered Array Pin Fin Heat Transfer for Turbine Cooling Applications," ASME Paper 87-GT-201

Brigham, B.A. and VanFossen, G.J., 1984, "Length-to-Diameter Ratio and Row Number Effects in Short Pin Fin Heat Transfer," *J. Engrg. for Gas Turbine and Power*, Vol. 106, pp. 241-246.

Chyu, M.K., 1990, "Heat Transfer and Pressure Drop for Short Pin-Fin Arrays with Pin-Endwall," *J. Heat Transfer*, Vol. 112, pp. 926-932.

Chyu, M.K. and Goldstein, R.J., 1991, "Influence of Cylindrical Elements on Local Mass Transfer from a Flat Surface," *Int. J. Heat and Mass Transfer*, Vol. 34, pp. 2175-2186.

Chyu, M.K., Hsing, Y.C. and Natarajan, V., 1996, "Convective Heat Transfer of Cubic Fin Array In a Narrow Channel," ASME Paper 96-GT-201.

Eckert, E.R.G., 1976, "Analogies to Heat Transfer Processes," *Measurements in Heat Transfer*, ed. by Eckert, E.R.G. and Goldstein, R.J., Hemisphere Publishing Corp. New York.

Metzger, D.E., Berry, R.A. and Benson, J.P., 1982a, "Developing Heat Transfer in Rectangular Ducts with Staggered Arrays of Short Pin Fins," *J. Heat Transfer*, Vol. 104, pp. 700-706.

Metzger, D.E. and Haley, S.W., 1982b, "Heat Transfer Experiments and Flow Visualization of Arrays of Short Pin Fins," ASME Paper 82-GT-138

Metzger, D.E., Fan, Z.X., Shepard, W.B., 1982c, "Pressure Loss and Heat Transfer Through Multiple Rows of Short Pin Fins," *Heat Transfer 1982*, Vol. 3, Hemisphere Publishing Corp., pp. 137-142.

Metzger, D.E., Fan, C.S. and Haley, S.W., 1984, "Effects if Pin Shape and Array Orientation on Heat Transfer and Pressure Loss in Pin Fin Arrays," *J. Engrg. for Gas Turbine and Power*, Vol. 106, pp. 252-257.

Metzger, D.E. and Sheppard, W.B., 1986, "Row Resolved Heat Transfer Variations in Pin Fin Arrays Including Effects of Non-Uniform Arrays and Flow Convergence," ASME Paper 86-GT-132.

V. Natarajan and M.K. Chyu, "Convective Heat Transfer from Finite Cylinders Mounted on a Plane Wall," Paper 4-EC-15, Vol. 3, pp. 71-77, 10th International Heat Transfer Conference, Brighton, UK. August 14-18, 1994.

Simoneau, R.J. and VanFossen, G.J., 1984, "Effect of Location in an Array on Heat Transfer to a Short Cylinder in Crossflow," *J. Heat Transfer*, Vol. 106, pp. 42-48.

Sparrow, E.M. and Ramsey, J.M., 1978, "Heat Transfer and Pressure Drop for a Staggered Wall-Anached Array of Cylinders with Tip Clearance," *Int. J. Heat Mass Transfer*, Vol. 21, pp. 44-50.

VanFossen G.J., 1982, "Heat Transfer Coefficient for Staggered Arrays of Short Pin Fins," *J. Engrg. for Power*, Vol. 104, pp. 268-274.

Zukauskas, A.A., 1972, "Heat Transfer from Tubes in Cross Flow," *Advances in Heat Transfer*, Vol. 8, pp. 116-133.