General Disclaimer

One or more of the Following Statements may affect this Document

- This document has been reproduced from the best copy furnished by the organizational source. It is being released in the interest of making available as much information as possible.

- This document may contain data, which exceeds the sheet parameters. It was furnished in this condition by the organizational source and is the best copy available.

- This document may contain tone-on-tone or color graphs, charts and/or pictures, which have been reproduced in black and white.

- This document is paginated as submitted by the original source.

- Portions of this document are not fully legible due to the historical nature of some of the material. However, it is the best reproduction available from the original submission.

Produced by the NASA Center for Aerospace Information (CASI)
Length to Diameter Ratio and Row Number Effects in Short Pin Fin Heat Transfer

Barbara A. Brigham and G. James Van Fossen
Lewis Research Center
Cleveland, Ohio

Prepared for the
Twenty-eighth Annual International Gas Turbine Conference
sponsored by the American Society of Mechanical Engineers
Phoenix, Arizona, March 27-31, 1983
ABSTRACT

Recently, several experiments concerning heat transfer from short pin fins have been conducted with the results indicating lower heat transfer from short pin fins than from longer pin fins found in tube banks and other similar configurations. Assessments of the effect of the number of pin rows and row geometry have also been made. It was felt that there was a need to determine the relative contribution of pin length to diameter ratio and pin row geometry on the heat transfer.

Array averaged heat transfer coefficients on pin and endwall surfaces were measured for two configurations of staggered arrays of short pin fins (length to diameter ratio of 4). One configuration contained eight streamwise rows of pins, while the other contained only four rows. Results showed that both the 8-row and the 4-row configurations for an Lp/D of 4 exhibit higher heat transfer than in similar tests on shorter pin fins (Lp/D of 1/2 and 2). It was also found that for this Lp/D ratio, the array averaged heat transfer was slightly higher with eight rows of staggered pins than with only four rows.

NOMENCLATURE

\[ h' \] average heat transfer coefficient on endwall surface
\[ h'D'/k_p \] array averaged heat transfer coefficients on pin and endwall surfaces
\[ K_a \] thermal conductivity of air
\[ K_p \] thermal conductivity of pin material
\[ L \] pin half length
\[ L_p/D \] test section streamwise length
\[ m \] pin parameter, \[ \sqrt{h'D'/k_p} \]
\[ Nu \] Nusselt number, \[ h'D'/k_a \]
\[ P \] pin perimeter, \[ \pi D \]
\[ Q \] heat dissipated in electric heater
\[ Re \] Reynolds number, \[ (w/A')D'/\mu \]
\[ S \] total heat transfer surface area
\[ T_{aw} \] adiabatic wall temperature
\[ T_r \] Eckert reference temperature
\[ T_s \] Static temperature
\[ T_w \] endwall temperature
\[ V \] flow channel volume minus pin volume
\[ w \] mass flow rate
\[ X_L \] ratio of pin spacing to pin diameter, streamwise direction
\[ X_s \] ratio of pin height to pin diameter (same as Lp/D)
For the maximum possible efficiency, modern commercial and military gas turbines operate at pressures and temperatures that require the hot section parts to be cooled. The demand for increased turbine blade reliability and durability has led to a desire for a more thorough understanding of the heat transfer and fluid flow phenomena involved in the various cooling geometries. One of the many possible cooling schemes used for turbine blades is to cast pin fins into the interior of the blade. Casting technology limits the diameter of the pin fins, requiring that they have a small length to diameter ratio as compared to the type of fins found in compact heat exchangers.

Due to a lack of data in the literature on heat transfer to short pin fins, an experimental program was initiated at NASA-Lewis Research Center with the objective of developing a better understanding of the heat transfer in pin fin passages. Initial results of the program are reported in reference 1 in which two geometries were tested. Both of these geometries were 4-row (streamwise) staggered arrays of pins set on an equilateral triangular spacing. One array had pins 2 diameters long spaced 4 diameters apart. The other array had pins 1/2 diameter long spaced 2 diameters apart. Array averaged heat transfer for these two sets of short pin fin was found to be considerably lower than for arrays containing more rows of considerably longer pins (about 8 diameters long, endwalls included, ref. 2). The data for both configurations tested in reference 1 fell on a single correlating line. In a separate test using wood pins it was found that the heat transfer coefficient on the endwall surface was approximately 35 percent lower than the heat transfer coefficient on the pin fin surface.

A second set of experiments was carried out to find out the effect of position within an array on the heat transfer to a single pin fin (ref. 3). The addition of one or more rows of fins in an inline pattern upstream of the measuring pin increased the heat transfer by about 50 percent, independent of the number of upstream rows. The addition of one, two or three rows of fins in a staggered pattern upstream of the measuring pin increased the local heat transfer 21, 64 or 98 percent respectively. It was shown that the addition of four or more rows of pins resulted in a 46 percent increase in local heat transfer. Measurement of turbulence intensity profiles in the channel, ahead of the measuring pin helped to explain the heat transfer behavior. The results of this study showed that the average channel velocity should be used as $\mu_{	ext{ref}}$ in the Reynolds number rather than the maximum velocity based on the minimum flow area. Heat transfer to the endwalls was not included.

In work carried on concurrently in Japan (ref. 4), array averaged heat transfer coefficients were measured for a 10-row (streamwise) array of staggered pin fins that were 1 diameter long. The pins were set in a square array pattern with a 3-diameter spacing in both the streamwise and transverse directions. The array averaged heat transfer results fall slightly below the correlation derived from the data of reference 1.

Another experimental effort concurrent with the present work was conducted at the Arizona State University. Results of that program have been reported in references 5 to 7. In references 5 and 6, the heat transfer was measured from pins arranged in 10-row (streamwise) staggered array patterns. In both cases, the pin length to diameter ratio was 1 with transverse by streamwise pin spacings of 2.5 by 2.5 and 2.5 by 1.5, respectively. The heat transfer to each row of pins was measured with trends similar to those of reference 3; however references 5 and 6 report average heat transfer to both pin and endwall surfaces, whereas reference 7 reports only heat transfer to the pin. This work, as well as the work done in Japan, is covered further in the discussion of results.

The work of reference 1 made clear the fact that heat transfer from arrays of short pin fins was considerably lower than that of data available for longer pins from heat exchanger work, while the work of reference 3 showed that the number of streamwise rows also had an effect on the level of heat transfer. One question remaining to be answered was which of these effects played the more dominant role in the heat transfer in short pin fins. The purpose of this paper is to present the results of a test designed to answer this question.

A test section was fabricated which had pins that were 4 diameters long. The pins were spaced 4 diameters apart in an equilateral triangular array with eight rows of pins in the streamwise direction. Heat transfer results on this model were obtained using the same inlet and flow path as in reference 1. Array average heat transfer coefficients on both the pin and endwall surfaces were obtained using this test section. The model was then cut in half, with the upstream half retained to give the configuration with only four rows in the streamwise direction. Heat transfer results are presented in the form of Nusselt numbers over a range of Reynolds numbers that cover those of interest for the turbine cooling application. The results are compared to those of references 1 and 2.

DESCRIPTION OF EXPERIMENT

Test Section and Flow Apparatus

A schematic of the rig used for the pin fin experiment is shown in Figure 1. Room air was drawn through a constant acceleration inlet, into the test section and then through either an orifice flow meter (large flow rates) or through a venturi flow meter (low flow rates). The air passes through the respective flow control valves and then on into the laboratory altitude exhaust system. Air temperatures were measured at the test section inlet, test section outlet, and flow meters. Air density at the flow meters was calculated using the ideal gas law. The data was collected and converted to engineering units using the laboratory data collection system (ref. 8).

Figure 2 shows a schematic of the two test sections used, illustrating the channel geometry. For this particular test, $A_0$ was 4, $A_1$ was 4 and $w_0$ was 0.318 cm (0.125 in.). The flow channel was made of two 0.635 cm (0.250 in.) thick copper plates in which holes were orilled for the insertion of the pins. The pins, which extended completely through the endwall plates, were then silvered into place. The dimensions of the flow channel were 3.56 cm by 12.56 in. by 1.27 cm high (3.77 in. by 5.0 in. by 0.50 in.).
The test section was heated with commercially available electric foil resistance heaters attached to the endwalls with pressure sensitive adhesive. The streamwise centerline temperatures of the endwalls were measured by thermocouples inserted in holes drilled along the edge of each endwall plate. Figure 2 illustrates how the original test section (with eight rows in the streamwise direction) was modified to give a four row configuration. The removed pin section was replaced with heated copper plates separated from the downstream edge of the four row configuration with an insulating layer of epoxy. During runs, these plates were kept at the same temperature as the test section.

**Test Procedure**

In order to account for heat losses, a set of calibrations were made with no air flow through the test section. By measuring the endwall temperatures, the ambient air temperature, and the total heat flux, the overall heat loss coefficient could be determined. This was done after the test section had reached a steady state condition, which was normally overnight. The next day, a data run was made by adjusting the power level to the heaters in order to maintain a constant plate temperature for the desired flow rate.

**Data Analysis**

The theoretical model used as the basis for the data analysis is shown in figure 3. The total heat flux for the pin and endwall surfaces is the sum of the heat flux through the endwall and the heat flux through the pin. The average heat transfer coefficients on the pin surface and endwall surface are \( h_p \) and \( h_w \) respectively. The heat lost by the pin fin surface (endwalls included) is equated to the heat lost by a plain surface with heat transfer coefficient \( h_eff \). For the purpose of this work, the average heat transfer coefficient on the pin, \( h_p \), was assumed to be equal to the heat transfer coefficient on the endwalls, \( h_w \), and is denoted by \( h \). The data analysis from here on is identical to that of reference 1 and shall be omitted.

The data was put into dimensionless form of Reynolds number and Nusselt number for purposes of comparison. The Reynolds number is defined as

\[
Re = \left( \frac{w}{\nu} \right) \frac{U'}{U} \tag{1}
\]

where

\[
U' = 4V/S \tag{2}
\]

\[
A' = \frac{V}{L} \tag{3}
\]

and \( \nu \) is the viscosity of air. The Nusselt number is defined as

\[
Nu = \frac{h'U'}{k_a} \tag{4}
\]

where \( k_a \) is the thermal conductivity of air. Both \( \nu \) and \( k_a \) were found from curve fits of data from reference 9 and were evaluated at the Eckert reference temperature (ref. 1) defined as

\[
T_r = 0.5T_w + 0.28T_0 + 0.22T_{aw} \tag{5}
\]

based on other experiments that have been done using the same type of experimental setup as was used in this work. The estimated level of uncertainty in this work is about 5 percent.

**RESULTS AND DISCUSSION**

Dimensionless heat transfer coefficients averaged over the pin and endwall surfaces were found over a range of Reynolds numbers for both the eight and four row configurations. Figure 4 is a comparison of the heat transfer data from the current work with the data of references 1 and 2 and shows some important results. First, figure 4 shows the current data plotted against the correlation derived in reference 1. It can be seen that the pin fins that were 4 diameters in length had slightly higher heat transfer coefficients than the pin fins of 1/2 and 2 diameters in length of reference 1. The second result is the array averaged heat transfer coefficients for the eight row configuration were slightly higher than the heat transfer coefficients for the four row configuration. Reference 3 showed that the heat transfer to a single pin in a staggered array of pin fins increased for the first two rows added upstream of the measuring pin. For three or more rows added upstream, the heat transfer decreased slightly and then remained constant regardless of the number of rows upstream. The same results were found in references 5 and 6. From these results it can be expected that array averaged heat transfer for an eight row array of staggered pin fins would have higher heat transfer coefficients than would a four row array, as was found in the current work.

Figure 4 also shows that the heat transfer from the 4 diameter long pin fins was lower than the heat transfer from the 8 diameters long configuration, PF-8, of reference 2. This indicates that the pin fins 1 diameters in length did not give quite as high heat transfer coefficients as some of the configurations commonly used in compact heat exchanger design. Also shown in figure 4 is the correlation for the channel with no pins used in reference 1 for comparison purposes.

Figure 4 suggests that the number of streamwise rows and the pin length to diameter ratio both have an influence on short pin fin heat transfer. However, figure 4 tends to indicate that, while the number of streamwise rows has a small effect on short pin fin heat transfer, the dominant effect on the overall level of heat transfer is the pin length to diameter ratio.

Figure 5 shows the work of Morishita, et al. (ref. 4), plotted against the correlation derived in reference 1. It can be seen that these results fall slightly below the work of reference 1. In reference 4, the heat flux was calculated from the air flow rate and the temperature difference between the inlet and outlet cooling air. Small errors or non-uniformities in temperature could lead to large errors in calculated heat flux.

Figure 6 shows the array averaged heat transfer coefficients of references 5 and 6 compared with equation 16 of reference 1. When the Nusselt and Reynolds numbers of reference 5 are converted to the definitions used herein, the data and correlation agree very well. This confirms the correlation found in reference 1 and indicates that the pin length is the dominant controlling factor in short pin fin heat transfer.

Figure 7 shows the percent increase of the Nusselt number with the plain channel correlation as the base reference, plotted against the length to diameter ratio for three different Reynolds numbers. It can be seen that for an \( L/D \) less than 2, the percent increase in the Nusselt number remains con-
stant for each Keynolds number. For an Lp/D greater than 2, the percent increase in the Nusselt number rises sharply. This rise in Nusselt number is also a function of the Keynolds number, with the percent increase in Nusselt number decreasing as Reynolds number increases. Thus at lower Keynolds numbers the effect of the length to diameter ratio is much stronger than at higher Keynolds numbers. This data was taken from figure 4, extrapolating the curves for the highest Keynolds number.

**SUMMARY OF RESULTS**

In short pin fin heat transfer over a range of Keynolds numbers applicable to the turbine cooling problem, the results of tests on a staggered array of 4 diameter long pin fins are as follows:

1. The pin length to diameter ratio is the dominant factor in the level of array averaged heat transfer coefficients for short pin fins (endwalls included).

2. The number of streamwise rows in the staggered array has a slight effect on heat transfer, with array averaged heat transfer coefficients increasing with the addition of rows in the streamwise direction.

3. For an Lp/D less than 2, the Nusselt number is a function of Keynolds number only. For an Lp/D greater than 2, the Nusselt number is a function of the length to diameter ratio as well as the Keynolds number.

**REFERENCES**


Figure 1. - Pin fin heat transfer rig.

Figure 2. - Test section schematic.
\[ Q_{\text{TOTAL}} = Q_{\text{WALL}} + Q_{\text{FIN}} \]

\[ Q_{\text{TOTAL}} = h_w (A_w - A_p) (T_w - T_{aw}) + P h^p c_p A_p \tanh (ml) (T_w - T_{aw}) \]

\[ Q_{\text{TOTAL}} = h_{\text{eff}} A_w (T_w - T_{aw}) \]

Figure 3. - Theoretical model.

Figure 4. - Current data plotted against results of refs. 1 and 2.
Figure 5. - Comparison of the heat transfer data of Morishita et. al. (ref. 4) with the correlation of ref. 1.

\[ \text{Nu}_D = 0.153 \text{Re}_D^{0.685} \text{ (ref. 1)} \]

Figure 6. - Comparison of heat transfer data from ref. 1 and array averaged data from ref. 6.
Figure 7. - Percent increase in $N_{Udp}$ from plain channel equation vs. $L_p/D$ for staggered arrays of short pin fins.