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NASA Technical Memorandum 83005

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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 L_p/D of 4 exhibit higher heat transfer than in similar tests on shorter pin fins (L_p/D of 1/2 and 2). It was also found that for this L_p/D ratio, the array averaged heat transfer was slightly higher with eight rows of staggered pins than with only four rows.

NOMENCLATURE

A'	average flow area, V/L
A_p	pin cross sectional area, $\pi D_0{}^2/4$
A_w	endwall area, $(\sqrt{3}/2)(X_t D_0)^2$
D'	characteristic length
D_0	pin diameter
h'	average heat transfer coefficient in pin fin test section
h_{ef}	effective heat transfer coefficient
h'_p	average heat transfer coefficient on pin surface
h'_w	average heat transfer coefficient on endwall surface
k_a	thermal conductivity of air
k_p	thermal conductivity of pin material
l	pin half length
L	test section streamwise length
L_p/D	pin length to diameter ratio
m	fin parameter, $\sqrt{h'P/k_p A_p}$
Nu	Nusselt number, $h'D'/k_a$
P	pin perimeter, πD_0
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 L_p/D)

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X_t ratio of pin spacing to pin diameter, transverse direction

μ viscosity of air

INTRODUCTION

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 fins 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 pins 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 pins in a staggered pattern upstream of the measuring pin increased the local heat transfer 21, 64 or 58 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 the reference velocity 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, while reference 3 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 for 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 a 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, X_t was 4, X_s was 4 and D_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 drilled for the insertion of the pins. The pins, which extended completely through the endwall plates, were then soldered into place. The dimensions of the flow channel were 9.58 cm long by 12.7 cm wide 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, an 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 in dimensionless form of Reynolds number and Nusselt number for purposes of comparison. The Reynolds number is defined as

$$Re = (w/A')U'/\mu \qquad (1)$$

where

$$U' = 4V/S \qquad (2)$$

$$A' = V/L \qquad (3)$$

and μ is the viscosity of air. The Nusselt number is defined as

$$Nu = h'U'/k_a \qquad (4)$$

where k_a is the thermal conductivity of air. Both μ and k_a were found from curve fits of data from reference 9 and were evaluated at the Eckert reference temperature (ref. 10) defined as

$$T_r = 0.5T_w + 0.28T_s + 0.22T_{aw} \qquad (5)$$

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 two 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 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-4, of reference 2. This indicates that the pin fins 4 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 definition 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_p/D less than 2, the percent increase in the Nusselt number remains con-

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stant for each Reynolds number. For an L_p/D greater than 2, the percent increase in the Nusselt number rises sharply. This r se in Nusselt number is also a function of the Reynolds number, with the percent increase in Nusselt number decreasing as Reynolds number increases. Thus at lower Reynolds numbers the effect of the length to diameter ratio is much stronger than at higher Reynolds numbers. This data was taken from figure 4, extrapolating the curves for the highest Reynolds number.

SUMMARY OF RESULTS

In short pin fin heat transfer over a range of Reynolds 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 L_p/D less than 2, the Nusselt number is a function of Reynolds number only. For an L_p/D greater than 2, the Nusselt number is a function the length to diameter ratio as well as the Reynolds number.

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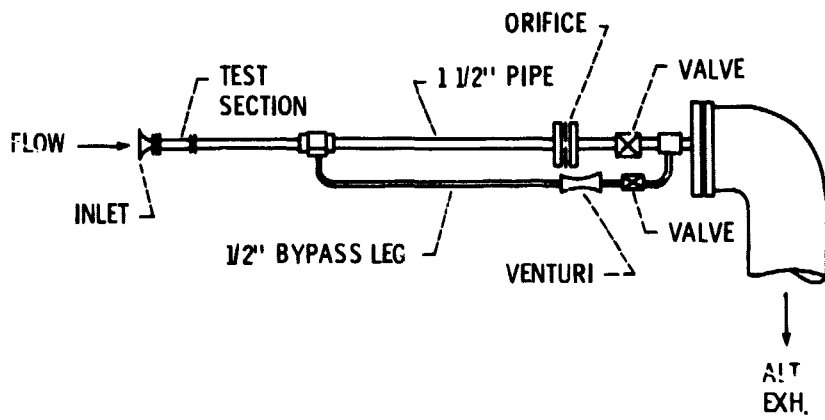
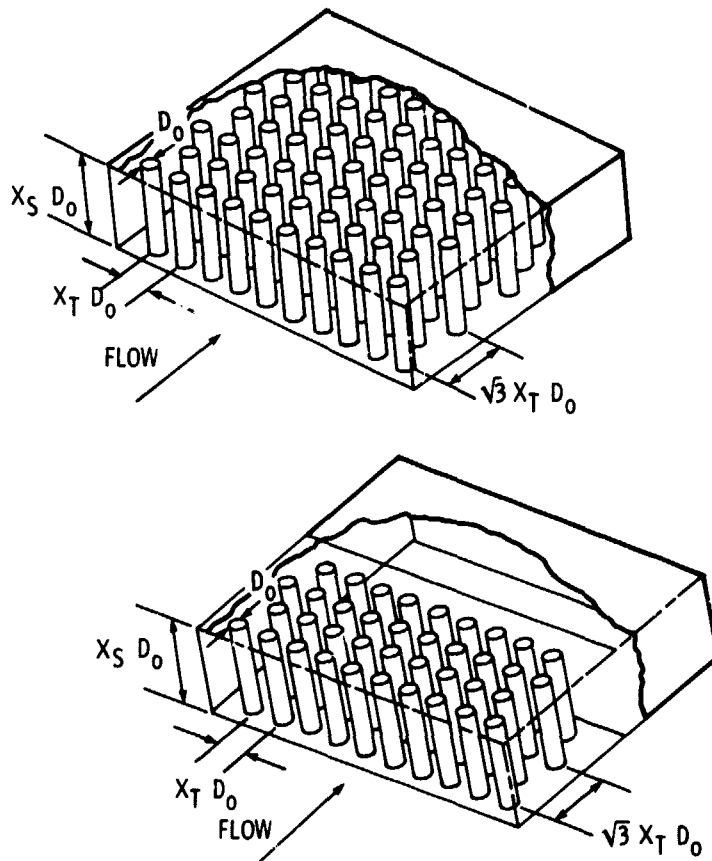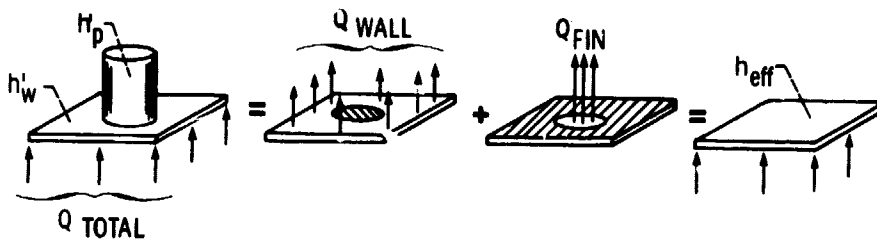Figure 1. - Pin fin heat transfer rig.

$X_S = 4, X_L = X_t = 4, \ D_0 = 0.318$ cm (0.125 in.)

Figure 2. - Test section schematic.

$$Q_{TOTAL} = Q_{WALL} + Q_{FIN}$$

$$Q_{TOTAL} = h'_w (A_w - A_p)(T_w - T_{aw}) + Ph'_p k_p A_p \tanh(ml)(T_w - T_{aw})$$

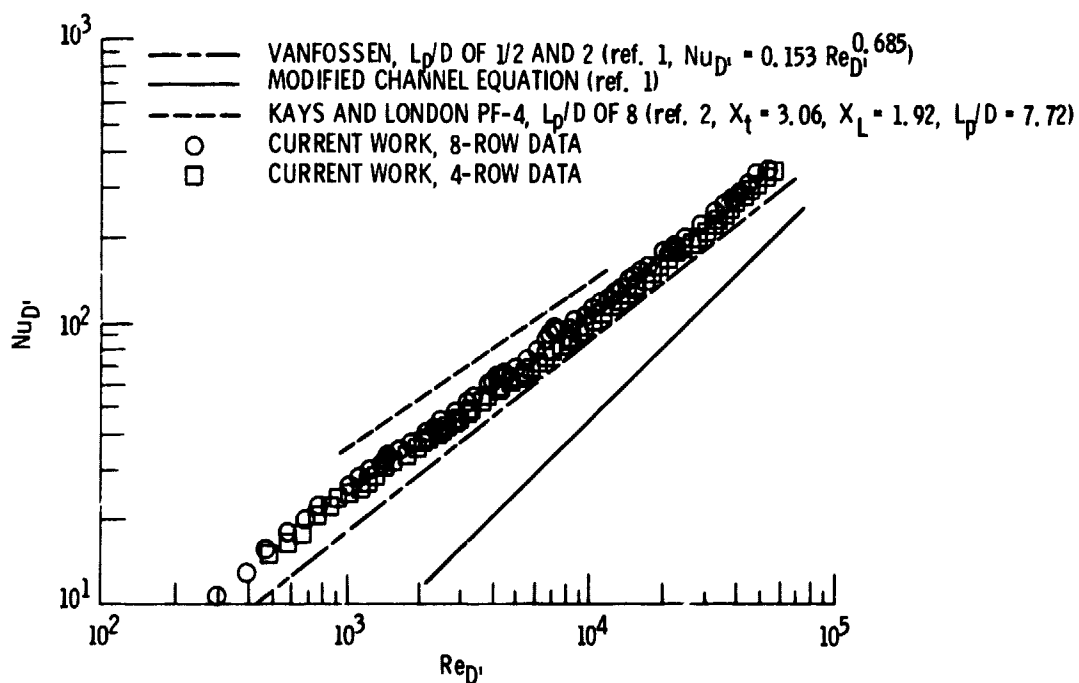$$Q_{TOTAL} = h_{eff} A_w (T_w - T_{aw})$$

Figure 3. – Theoretical model.

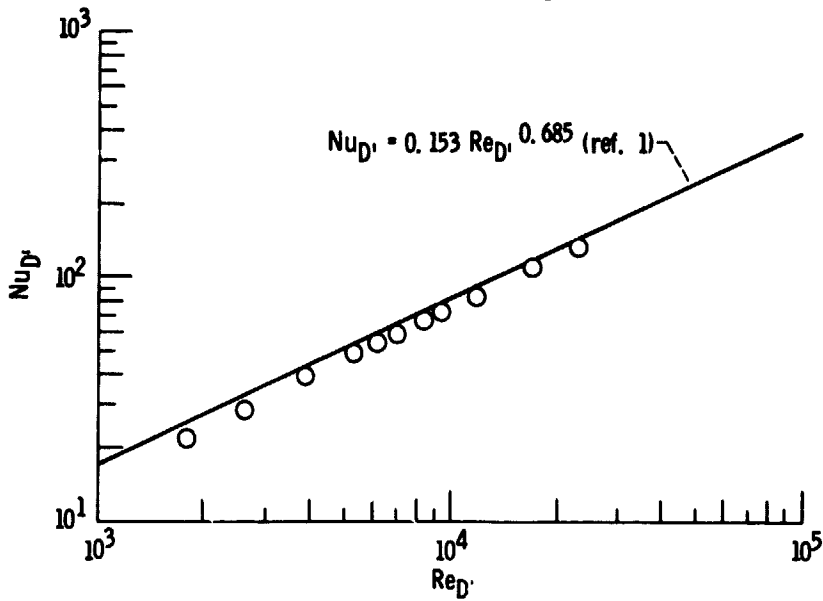Figure 4. – Current data plotted against results of refs. 1 and 2.

Figure 5. - Comparison of the heat transfer data of Morishita et. al. (ref. 4) with the correlation of ref. 1.
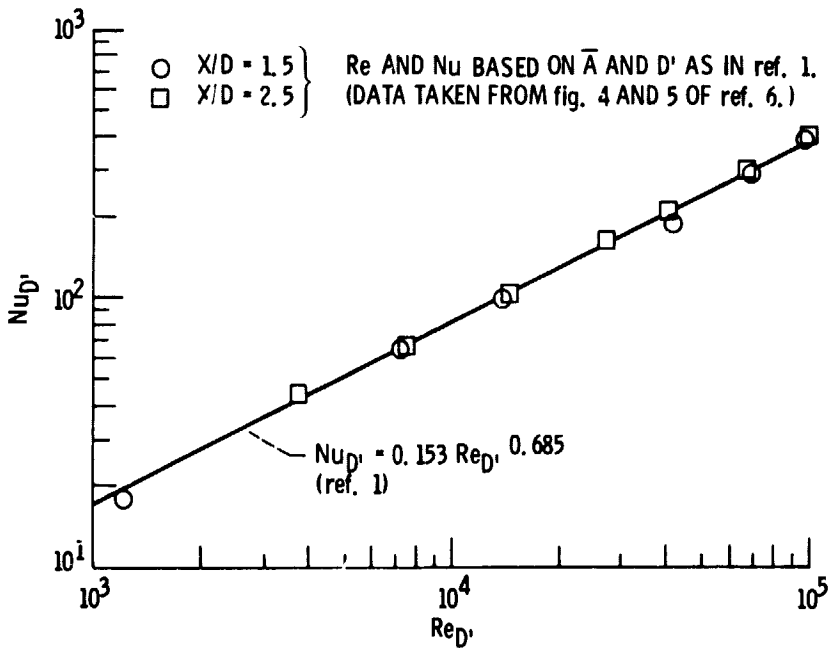

Figure 6. - Comparison of heat transfer data from ref. 1 and array averaged data from ref. 6.
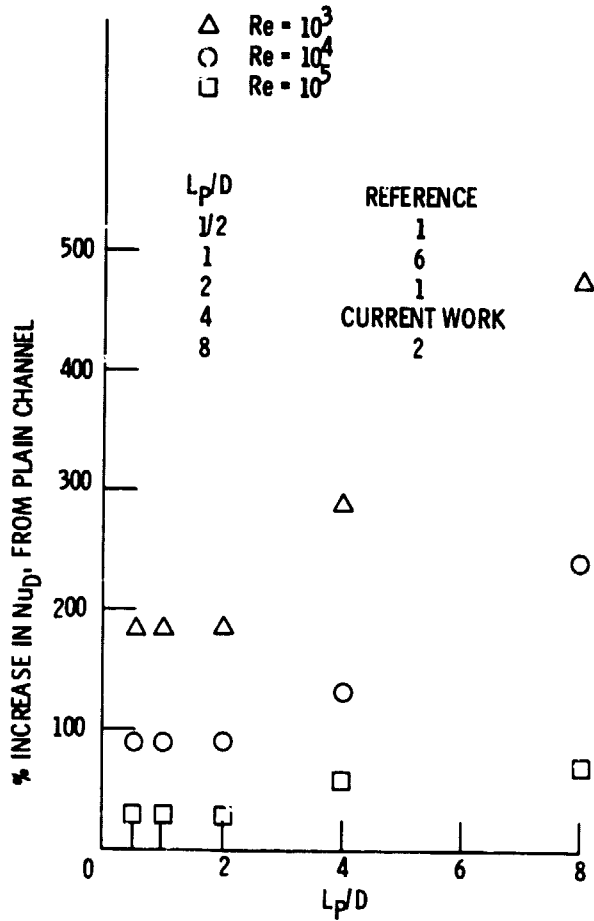
Figure 7. - Percent increase in Nu_D from plain channel equation vs. L_p/D for staggered arrays of short pin fins.