HEFAT2014
10th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
14 – 16 July 2014
Orlando, Florida

PRESSURE DROP ALONG A CHANNEL WITH MODIFIED SHORT PIN-FINS

Pretorius H.J.* Mahmood G.I. and Meyer J.P.
*Author for correspondence
Department of Mechanical and Aeronautical Engineering,
University of Pretoria,
Pretoria, 0002,
South Africa,
E-mail: johan.pretorius@deneldynamics.co.za

ABSTRACT

Short pin-fin arrays are used to enhance the heat transfer in cooling channels of gas turbine airfoils, combustor liners, electronic chips, bearing housings etc. Numerous experimental and numerical past studies ascertain the optimal spacing and length to height ratios within arrays of round short pin-fins for optimal heat transfer. However, the heat transfer enhancement in pin-finned channels is achieved at the expense of a significant pressure drop across the channel.

Augmentation of pin-fins to improve the flow and heat transfer characteristics through the channel, may however lead to further improvements. This paper presents measured pressure drop characteristics in a channel with a staggered array of round short pin-fins where each pin has two through slots. The two slots located at the two ends of each pin have an identical slot-depth to pin-diameter ratio and slot-width to pin-diameter ratio which is 0.48. The slots are oriented parallel to the mean flow direction in the pin-fin array. The pin-height to diameter ratio used is 1.28 and pitch to diameter ratio in both streamwise and pitchwise direction of the pin-fin array is 2. The array consists of 13 staggered rows of pin-fins. Pressure drop in the channel is also measured with conventional solid round pin-fins. The same array spacing, pin-diameter and pin-height was used as with the slotted pin-fin array for comparison. The Reynolds number based on the mass averaged velocity and pin hydraulic diameter ranges between 3,300 and 32,800 covering different cooling applications. The objectives of the measurements are to investigate the effect of the pin-slots on the pressure drop over the array of pin-fins and recirculation region downstream of the slotted pin in the channel. The recirculation regions are responsible for low heat transfer coefficients on the end wall in the immediate vicinity downstream of the pin-fins.

In general, the normalized pressure drop over the slotted pin-fin array decreases as Reynolds number decreases. The pressure drop over the slotted pin-fin array was also lower than the predicted pressure drop over an array of solid pin fins without slots.

The normalized pressure distribution around the slotted pin circumference at mid-height shows a stagnation region over the frontal section of the slotted pin. The normalized pressure distribution changes little between 90° and 180° over the Reynolds number range that was used; this indicates a region of local flow separation.

NOMENCLATURE

A	m^2	Channel flow area
A_{min}	m^2	Minimum flow area (between pins)
C_p		Normalized pressure coefficient
D	[m]	Pin diameter
D_h		Duct hydraulic diameter without pins
f		Friction coefficient
h	[m]	Slot height
H	[m]	Pin height, Channel height
N		Pin row number in streamwise direction
P	[Pa]	Pressure
Re_d		Reynolds number based on pin diameter
S	[m]	Pin spacing in span wise direction
T	[K]	Temperature
V	[m/s]	Mean duct velocity without pins
V_{max}	[m/s]	Mean streamwise velocity in the minimum free flow area
W	[m]	Channel width
w	[m]	Slot width
X	[m]	Pin spacing in streamwise direction
z	[m]	Height above channel floor
Special characters		
θ	[°]	Position on pin surface
μ	$N \cdot s/m^2$	Dynamic viscosity
ρ	$[kg/m^3]$	Density
$\dot{m}$	[kg/s]	Mass flow rate
Δ		Differential
Subscripts		
d		Pin diameter
max		Maximum
min		Minimum

A short pin-fin array with slotted pins as described in this paper reduces the pressure drop over an array when compared to an unslotted array with the same geometrical arrangement. Slotting of pin fins may enhance the performance of an array of short pin-fins by reducing the pressure drop across the array.

INTRODUCTION

Short pin fin arrays are used in many fields of heat transfer, where limited space is available and relatively compact heat exchangers are required. Electronic appliances and gas turbines are two sectors that extensively use short pin fins for cooling purposes [1]. For example inside a turbine blade one finds an intricate heat exchanger consisting of channels, impingement jets and pin fins. Air under high pressure is bled off from the turbine compressor and flows through the turbine blade to act as a coolant. Figure 1 shows a typical array of pin fins near the trailing edge of a turbine blade.

Numerous studies have been done to further enhance the heat transfer characteristics of short pin fins by means of augmentation [2,3,4]. The aim of this experimental study was to characterize the flow properties over a staggered array of augmented short pin fins. The purpose of the augmentation was to reduce the pressure loss over the array and improve the heat transfer.

The wake region behind a traditional pin fin is associated with a local reduction in heat transfer due to the reduced flow velocity in the wake of the pin. The pin fins used in this study were slotted at the top and the bottom of the pin fin in the stream wise direction to allow a portion of the flow to flow through the slots. Figure 2 shows the slotted pin fins used in this study. The flow through the slots created regions of flow behind the pin fins where the fluid velocity was higher than would be the case with standard pin fins.

Slots also increase the overall surface area of the pin fins, over which heat transfer can occur. The frontal area of the pin fins were also reduced which lead to a reduction in the pressure drop over the array.

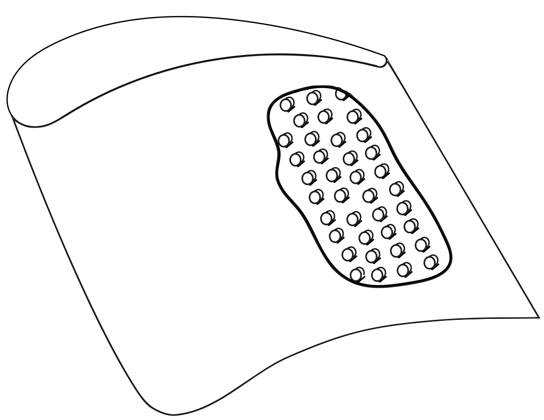

Figure 1 Diagram showing the position of pin fins in a turbine blade (adapted from [5])

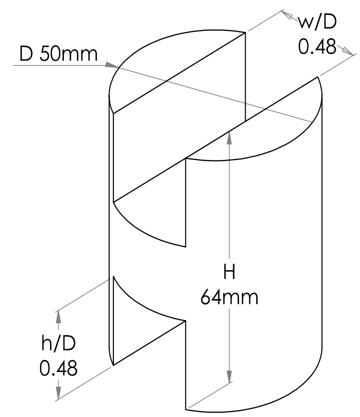

Figure 2 Detailed sketch of a Slotted Pin Fin

EXPERIMENTAL APPARATUS AND PROCEDURES

The experimental setup was configured to function as a suction wind tunnel. The experimental setup consisted of an inlet section, test section, contraction section, orifice flow meter and two 11 kW axial fans. The inlet conditions were ambient temperature and pressure. The inlet included a honeycomb section to remove rotational flow and a flow development section to ensure that the flow was fully developed before entering the test section. The test section was manufactured out of Perspex with thirteen rows of pin fins in the stream wise direction. After the test section there was a contraction section followed by an Orifice plate flow meter. Two axial fans completed the experimental setup and vent into the lab.

The data acquisition system consisted of a desktop computer, connected to an Agilent. LabView was used to record the results measured by the sensors and to control the stepper motor used to actuate the scan valve. A custom designed scan valve actuated by a stepper motor scanned through all the surface pressure ports. The scan valve was tested for leakage and found to provide an air tight seal. Current output, pressure transducers were used to reduce the susceptibility to electronic disturbances of the system. The pressure transducers were calibrated using a Setra Micro Calibrator Model 869.

An Omega PX419 differential pressure transducer was used to measure the surface pressure at multiple locations across the test section. The PX419 is temperature compensated and the standard accuracy of the transducer was given as ±0.08% of the full scale. The full scale reading of the module was 2500 Pa, the instrument uncertainty was thus ±2 Pa.

A Honeywell model FP2000 transducer was used to measure the pressure differential across the Orifice plate. The full scale reading of the unit is 6894 Pa (1 Psi), and the full scale accuracy of the unit was ±0.5%. This resulted in a measurement uncertainty of ±34.5 Pa.

The uncertainties did not include additional noise sources from amplifiers and other electronics.

The ambient pressure was measured using a Kestrel Model 4000, the accuracy of the unit is ±240 Pa. This correlated to an uncertainty of ±0.27% given the average ambient pressure of 86,5 kPa in the lab. The temperature in the lab and the test section inlet and outlet was measured with a Ttype thermocouple with an uncertainty of ±0.1°C. A DC power supply was used to supply power to the pressure transducers, and a Velmex stepper motor controller was used to interface with the stepper motor.

Test section

The pin fin spacing used in this study was 100 mm both in the streamwise and pitchwise direction as shown in Figure 3. The channel width was 500 mm and the channel height was 64 mm. The diameter of the pin fins was 50 mm and the pins were secured in position using screws on either side of the slots. Half pins not shown in Figure 3 were also used during some of the tests. The half pins were attached to the side walls in all the even number rows. This resulted in a constant minimum flow area in all of the rows effectively creating a channel with 5 pin fins per row. With the side pins removed the number of pins per row alternated between 5 and 4 on the odd and even number rows respectively. The test setup had 3 windows on the top wall allowing access to the centre pin in row 3, 7 and 11. Two pins were 3D printed with pressure taps on the surface of the pin and within the slots to measure the static pressure over the pin fin surface. Surface pressure measurements were taken in row 3, 7 and 13, to compare the development of flow through the array.

Pressure measurements were also taken on the centre line of the side wall and top wall of the array for comparative purposes. The pressure taps on the on the top wall and side wall were spaced one pin diameter apart over the length of the test section. The position of the reference pressure that was used is six pin diameters upstream of the row 1 on the centre line of the top wall.

The slots were 24 mm in depth and width and combined accounted for 36% of the frontal area of the pin fins.

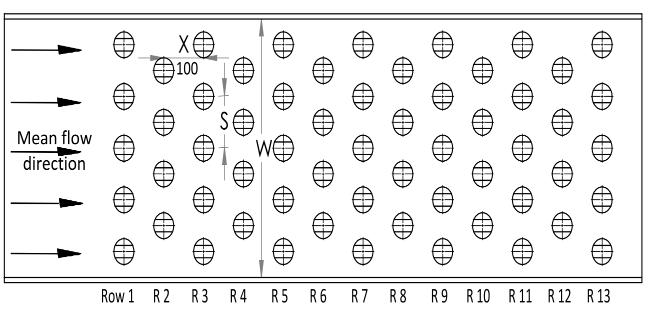

Figure 3 Layout of test section X=S=100 mm

Procedure

Final reported results are based on the average of 25 measurements per pressure port. After the scan valve moved to the next port a 10 second settling period was used to ensure that the reading have stabilized, before results were recorded. The pressure transducer signals were converted to differential pressure readings. The mass flow rate through the test section was calculated according to the ISO 5167-1 standard for orifice plate flow meters.

The Reynolds number based pin diameter was calculated using equation (1), the maximum mass averaged fluid velocity was calculated using the minimum flow area in the array, and the minimum flow area included the increased flow area due to the slots.

$$\mathrm{Re}_d = \frac{\rho V_{\max} D}{\mu} \quad (1)$$

The normalized pressure coefficient over the array was calculated using equation (2) with the reference pressure being located 6 pin diameters upstream of the first row of pins. For the pin surface pressure measurements the surface pressure at the front of the pin was used as the reference pressure.

$$C_p = \frac{\Delta P}{0.5\rho V_{\max}^2} \quad (2)$$

RESULTS AND DISCUSSION

Pressure drop over the array

The pressure drop over the array was measured along the side wall with and without side pins attached. Figure 4 shows the pressure drop over the array at various Reynolds numbers. The dips in the differential pressure were due to the location of the pressure taps in the stagnation region ahead of the pin fins on the side wall. X/D=0 was in line with the centre line of the first row of pin fins. The pressure drop over the array ranged from 4 to 184 Pa for the Reynolds number range of 3,300 to 32,800. The uncertainty of the differential pressure measurements ranged from approximately 50% at a Reynolds number of 3,300 given a 95% confidence interval calculated using the student's-t distribution and the accuracy of the transducer to 2% at a Reynolds number of 32,800. For the low Reynolds numbers the main source of uncertainty was the standard accuracy of the pressure transducers. As the Reynolds number increased and the pressure drop across the test section increased the uncertainty reduced.

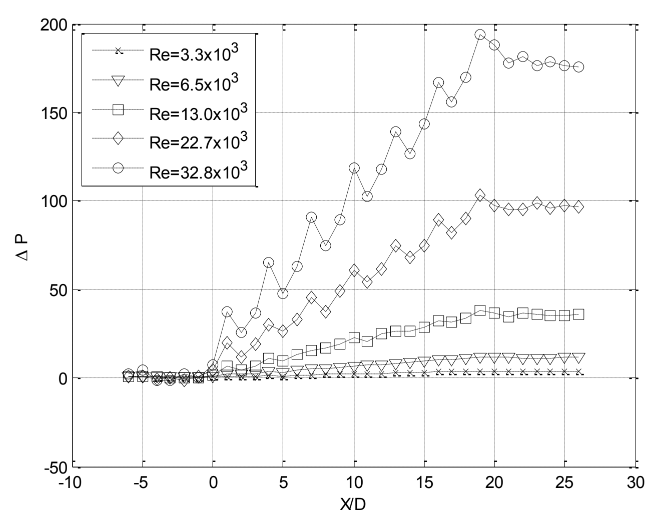

Figure 4 Pressure differential across the array for various Reynolds numbers

In general, the normalized pressure coefficient increased with decreasing Reynolds number as shown in Figure 5. At

X/D=5 there was a slope change in the results. The slope remained constant between row 4 and 13. The slope change indicated the end of the developing flow region in the array.

The friction factor coefficient results were calculated using equation (3). The differential pressure measurements used in the calculation of the friction factor were taken at X/2 upstream of the first row of pins and X/2 downstream of the last row of pins. This was the same positions used by Metzger. Figure 6 shows the friction factor results for solid unslotted pin fins and the 24 mm slotted pin fins compared to the Metzger correlations given by equations (4 & 5). These correlations have an uncertainty of ±15%.

$$f = \frac{\Delta P}{2\rho V_{\max}^2 N} \quad (3)$$

$$f = 0.317\,\mathrm{Re}^{-0.132} \; for \quad 10^3 \le \mathrm{Re} \le 10^4 \quad (4)$$

$$f = 1.76\,\mathrm{Re}^{-0.318} \; for \quad 10^4 \le \mathrm{Re} \le 10^5 \quad (5)$$

The friction factor results for solid pin fins with side pins included were within the uncertainty band of the Metzger correlations. Removal of the side pins reduced the friction factor by 15%.

The friction factor results for the array of slotted pin fins with side pins were on average 22% below the Metzger correlation and without the side pins the average friction factor was 33% below the correlation. Without side pins two bypass channels are effectively created that allow some of the flow to bypass the array, thereby reducing the total pressure drop across the array. Removing the side pin from the test setup used in this study s also resulted in the minimum flow area changing from row to row. The change in minimum flow area resulted in the mass averaged flow velocity varying by 9.4% from row to row.

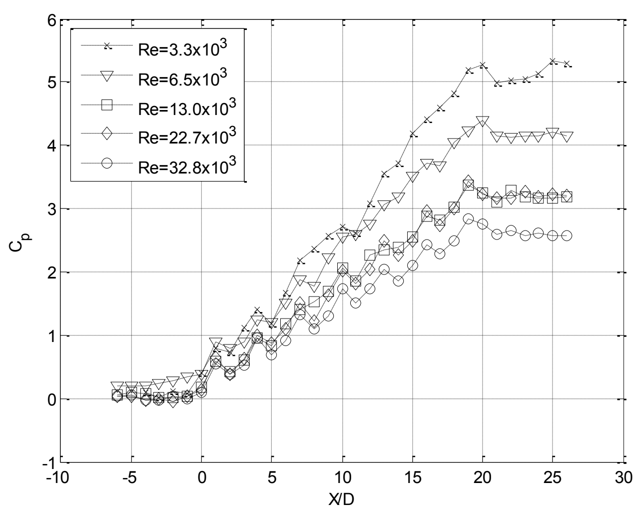

Figure 5 Normalized pressure drop along the side wall for the array of slotted pin fins with side pins attached

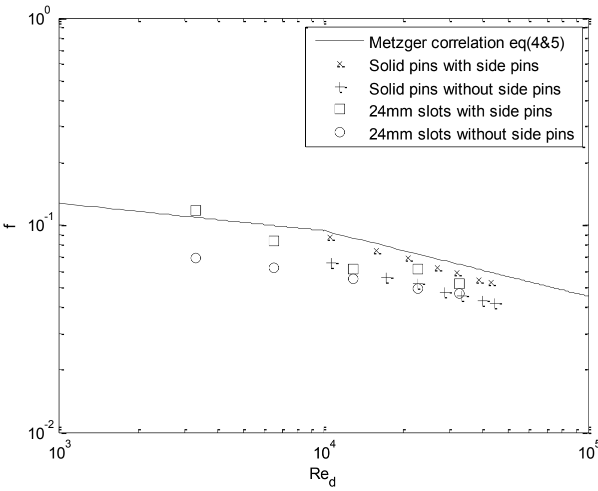

Figure 6 Friction coefficient comparisons for solid pins and slotted pins

Pin fin surface pressure

The flow around the slotted pin fins showed a steep reduction in the pressure coefficient over the entire Reynolds number range from 0° to 90° shown in Figure 7.This occurred due to the flow accelerating around the pin. In the region between 90° and 180° the pressure coefficient remained relatively constant. This indicated a region of local flow separation between 90° and 180°. The position of the separation point remained at 90° over the Reynolds number range under consideration. The location of the separation point remained unchanged from row 3 to row 11. Figure 8 shows that the normalized pressure coefficient trends remained similar independent of the location in the array, from row 3 to 11. In general the pressure coefficient around the mid height of the pin fins appeared to reduce with a reduction in Reynolds number.

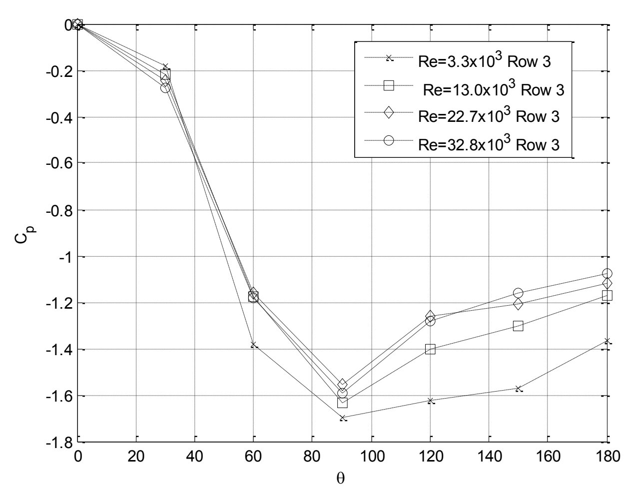

Figure 7 Normalized pressure difference at mid height (H/2) of a slotted pin fin in row 3 of the array

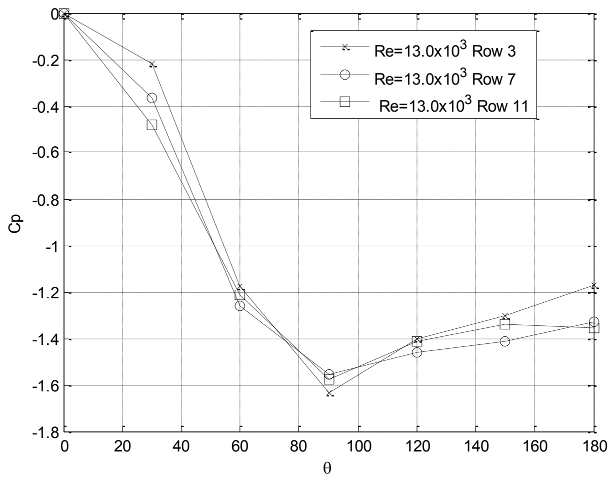

Figure 8 Normalized pressure difference at mid height (H/2) of a slotted pin fin in row 3,7 and 11 of the array

The pressure in the slots was measured on the horizontal and vertical wall of the slot at three stream wise locations on each surface. The normalized pressure coefficient in the slot was the same on both the vertical and horizontal slot wall. The variation in the stream wise direction was negligible and the normalized pressure coefficient within the slot was within the region of -1.2 for the Reynolds number range tested.

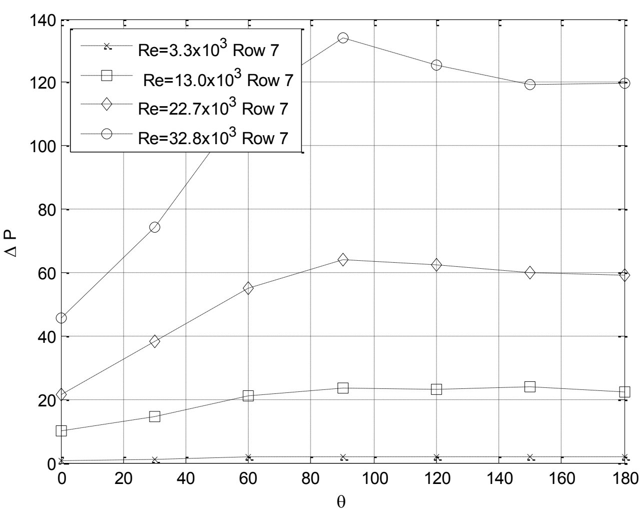

Figure 9 Differential pressure at mid height (H/2) of a slotted pin fin in row 7 of the array

The differential pressure around the mid height of a slotted pin fin in row 7 is shown in Figure 9. The reference pressure used was 6 pin diameters upstream of the first row of pin fins and hence the same as for the differential pressure measurements across the array. When compared to Figure 4 row 7 was located at X/D=10. The spikes in Figure 4 resulted because of the proximity of the pressure tap location on the side wall to the region of low pressure on the pin surface. This was verified by comparing the magnitude of the differential pressure at X/D=10 in Figure 4 to θ=90° in Figure 9, per Reynolds number.

CONCLUSION

Experimental pressure drop results indicate that slotted pin fins can reduce the friction factor over a staggered array of short pin fins while increasing the wetted surface area. Increasing the surface area available for heat transfer may contribute to higher heat transfer coefficients within the array.

For an array of slotted pin fins with an equal slot height to slot width; and a slot height to pin diameter ration of 0.48. The friction factor over the array was reduced by 22% when compared to previous studies of solid pin fins and baseline measurements.

The results also showed that the friction factor across the array is sensitive to the condition imposed on the side walls of the array. If the array does not span across the entire channel unobstructed flow passages are created that contribute to significant reductions in friction factor over the array.

Acknowledgment

Funding for the research project was partly provided by the BALLAST program of ARMSCOR of South Africa and National Research Foundation (NRF) of South Africa. Their financial assistance is sincerely appreciated.

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