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THE EFFECTS OF POROUS METAL FOAMS, ON THE THERMAL DEVELOPMENT IN A SERPENTINE PASSAGE, UNDER STATIONARY AND ROTATING CONDITIONS

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ABSTRACT

This paper presents measurements of heat transfer in a serpentine passage with porous metallic foam blocks in the straight sections, under stationary and rotating conditions. For the stationary condition, the effect of the Reynolds number is also investigated. The heat transfer measurements are obtained using the steady state liquid crystal technique using the group's rotating water flow table. The passage is of square cross-section. Porous blocks are attached to the two opposite plane sides, which under rotating conditions are also the leading and trailing sides, in a staggered arrangement. The geometry and dimensionless flow and rotation numbers are identical to those of an earlier PIV study by the authors. Only the two sides with porous blocks attached to them are heated. Flow at Reynolds numbers of 26,000 and 36,000 has been investigated at stationary conditions. A preliminary set of measurements without porous blocks resulted in Nusselt number measurements consistent with those of earlier studies. The introduction or the porous block raises heat transfer levels in the straight sections and leads to uniform heat transfer levels along the entire passage. As far as the Reynolds number effect is concerned, then Nusselt number in the ribbed passage scales according to the Dittus-Boelter correlation. Orthogonal rotation, at a rotation number of 0.32, with the axis of rotation normal to both the flow direction and the curvature axis, is found to have only a secondary effect on the thermal development, but it is shown to raise heat transfer levels along both the leading and trailing sides by 10%, in comparison to those of the stationary case.

1. INTRODUCTION

In gas turbines, turbine inlet temperatures exceed the safe operating limit of the blade material. Blade cooling technologies have thus been developed, to protect them from thermal damages.

Internal blade cooling is commonly applied at the middle part of the blade, where cooling air passes through serpentine passages with sharp U-turns and heat-transfer-enhancing ribs on opposite walls[1-4]. The use of porous metal foams instead of ribs is one promising alternative recently proposed [5-11]. The reason for this approach, is that the presence of a large surface area over a limited volume greatly facilitates the transfer of thermal energy from the walls of the cooling passage to the coolant.

The development and optimisation of effective blade cooling passages which use porous materials, requires a greater understanding of their influence and the availability of reliable RANS models. The authors' research group has been engaged in research activities on both these fronts, improving RANS models for internal passages with porous foams [12] and carrying out experimental investigations of heat and fluid flow through stationary and rotating serpentine passages with porous blocks. The flow field investigation, resulting from the use of Particle Image Velocimetry (PIV) and wall pressure measurements, has been reported in an earlier publication [13]. Here, we present the outcome of the corresponding thermal investigations, using the thermo-chromic liquid crystal (TLC) technique. More

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specifically, we have investigated the thermal characteristics through a serpentine cooling passage with a square-ended U-bend, with porous metal foam blocks attached to the passage walls. Tests have been carried out for both stationary and rotating conditions. The main objectives are to improve the understanding of the effects of porous metal foam on heat transfer development in rotating cooling passages and to generate detailed heat transfer data which can be used for the validation of CFD codes.

2. EXPERIMENTAL SETUP

As shown in Fig. 1, the rotating rig is used to conduct experiments on turbulent flow in the principal geometry of interest, square-ended bend test section. The test section is made of 10 mm thick transparent acrylic (Perspex) and mounted on the rotating table. The reason for using Perspex is the good optical access that allows use of measuring techniques, like TLC. The total length of the test section is 768mm with a constant square cross-section of 50x50 mm. Water is used as the working fluid. The reason for using water is that due to the low kinematic viscosity, ν, relative to that of air, a lower mean velocity (U_m) is needed to achieve the required value of the Reynolds number. Consequently, the rotational speed required to produce the required Rotation number ($Ro = \Omega D/ U_m$) is also lower than that for air.

For the porous roughened case, the aluminium-foam blocks are used in this study with porosity of 0.93, and its pore density is 2 PPC (pores per centimetre), made from ERG Materials and Aerospace Corporation. The porous blocks, height h=30 mm, are attached to opposite walls in staggered configuration, and the baffle spacing S/D is fixed at 1.0.

The mass flow rate is monitored through an orifice plate and the inlet temperature by using a k-type thermocouple. The two opposite walls of the passage to which the porous block are attached, are heated under uniform wall heat flux boundary conditions, using a 10 µm thick electrically heated stainless steel foil. Heat transfer measurements have been obtained using the Thermochromic Liquid crystal Technique (TLC). This technique provides an effective method for the visualisation and measurement of surface temperature. As described in our earlier publications [1,2], this method enables us to provide a mapping of the local Nusselt number distribution over the heated surfaces.

3. RESULTS

3.1 Smooth Passages

First, we focus on heat transfer through smooth passages to establish the reference levels against which to quantify the effects of the porous blocks. Figure 2 shows the normalised heat transfer distribution (the local Nusselt number divided by that of Dittus/Boelter correlation for fully developed pipe flow at the same Reynolds and Prandtl numbers), along the two heated sidewalls of the smooth passage for stationary conditions with a Reynolds number of 36,000 and Prandtl number of 5.8. Measurements along only one side of the duct are presented due to symmetry. Along the first pass, the distribution of the local Nusselt number is uniform and as the flow reaches the bend entry, the Nusselt number starts to increase. In the upstream section, the normalized Nu levels are around 1.5. Since here only two of the four duct walls are heated, the fully developed Nusselt number value is expected to be higher than that for fully developed flow in a heated pipe. In fact, the normalised Nu levels in the upstream section are consistent with those of earlier experimental [14], [16] and computational studies [17] with only two sides heated. The heat transfer levels start to increase, especially near the end wall, as the flow impinges there and separation bubbles are formed in the upper corner, shown in our earlier PIV and LDA studies,[13,14 and 15]. The normalised Nu levels reach their highest value of 2.5, over the first two diameters after the bend exit where there is flow separation. Further downstream, the Nu levels start to drop towards the upstream section values.

Figure 3 presents profiles of the side-averaged, normalised Nusselt number at two flow Reynolds numbers, 36,000 and 26,000, and at Pr of 5.8. The normalised heat transfer distribution is similar at both Reynolds numbers along the first and the second passes of the duct, confirming that the Dittus-Boelter equation provides the proper scaling for the straight smooth sections. The effect of the bend on heat transfer, however, seems to be stronger at the lower Reynolds number. This is due to the presence of corner separation, secondary flow motion and impingement, the effects of which do not scale with $Re^{0.8}$.

3.2 Porous blocks roughened passages

The effect of porous blocks on the thermal development is considered, when twelve aluminium metal foam blocks are inserted in each pass of the channel in staggered arrangement. The bend region is clear (without blocks). The flow Reynolds number is the same as in the smooth passage case.

For the stationary case, Figure 4 presents contours of the local normalised Nusselt number. The white blanks correspond to the locations of the porous blocks. Compared to the smooth duct contours of Figure 2, the Nusselt number levels are a) more uniform along the passage and b) significantly increased over the upstream section, mainly due to the enhancement in turbulence levels, observed in [13]. The heat transfer distribution along the upstream and downstream ducts with porous blocks has peaks and troughs, due to the meandering of the flow caused by the staggered porous block arrangement, discussed in [13]. The rather irregular and streaky patterns in the local Nusselt number contours are due to the irregular velocity patterns of the flow coming out of the porous blocks. Figure 5 shows a comparison of the side-averaged normalised Nusselt number profiles along the serpentine passage, with and without the porous blocks. In the upstream section the porous blocks lead to about a 30% enhancement in Nusselt number levels. Within and just after the turn, the smooth passage Nusselt number levels are similar to those without porous blocks due to the strong mixing and high turbulence levels generated by the turn [13]. Further downstream, as the bend effects diminish, the smooth passage Nu levels start to fall, while those of the passage with porous blocks remain high.

The effect of Reynolds number on heat transfer in a serpentine passage with porous blocks is demonstrated in the comparisons of the profiles of the normalised side-averaged Nusselt number at two different Reynolds numbers, shown in Figure 6. The levels of the normalised Nusselt number are similar at both Reynolds numbers. Because, as shown in the corresponding PIV study [13], the porous blocks do not cause flow separation, the resulting Nusselt number distribution still scales with $Re^{0.8}$.

Finally, attention is turned to the effect of orthogonal rotation. The axis of rotation is normal to both the main flow direction and the axis of curvature. Heat transfer measurements are presented for both the leading and trailing sides of the duct. The flow Re is again 36,000 and the rotation number 0.32. Figure 7 shows the contours of the local normalised Nusselt number along the leading and the trailing sides of the rotating passage. Though the differences in Nu levels along the two sides are not great, in the upstream pass, levels along trailing side are higher, which is consistent with the expected effects of the Coriolis force, as discussed in [3]. Nevertheless, the porous blocks dominate the thermal development, making the rotation effects secondary. A more quantitative assessment of the effects of rotation is provided by the corresponding comparisons of the profiles of the side-averaged Nusselt number, in Figure 8, in which the profile for the stationary case is also included. These comparisons reveal that along both sides, Nu levels for the rotating case are consistently higher than those for the stationary case. While for the stationary passage with porous blocks the normalised Nusselt number fluctuates around the value of 2, along both rotating sides, the normalised Nu levels vary between 2 and 2.5. Thus, for the rotation number of 0.32, Nu levels are on average 10% higher than for the stationary case.

4. CONCLUSIONS

To the authors' knowledge, this study has, for the first time, produced heat transfer measurements of the effects of the aluminium porous blocks in a serpentine cooling passage, at stationary and rotating conditions with orthogonal rotation. The main conclusions of this study are summarised below.

- For the smooth passage, the heat transfer distribution is consistent with those of earlier studies. The measured increase in heat transfer within and after the bend, is consistent with the presence of two counter-rotating vortices within the turn and flow separation downstream.

- The introduction of porous blocks, has a significant effect on the Nu distribution, causing an increase in heat transfer levels and more uniform levels over the entire passage. The heat transfer distribution along the straight passes has peaks and troughs, due to the meandering mean flow. The Nusselt number scales with the Reynolds number to the power of 0.8.

- For an orthogonally rotating passage with porous blocks, the differences between the Nusselt number levels of the leading and trailing sides are small and confined to the upstream straight section. While the presence of porous blocks dominates the thermal development, rotation increases the Nu levels on both sides. For a rotation number of 0.32 the increase is about 10%.

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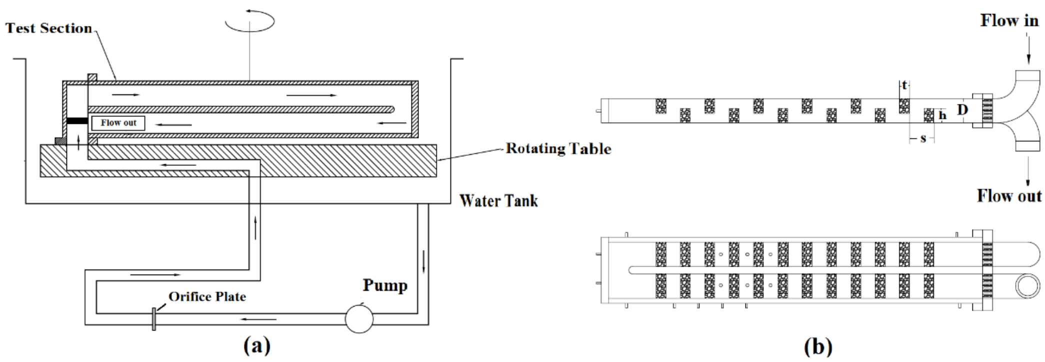

Figure 1. (a)Rotating water rig (b) Test section with porous blocks mounted.

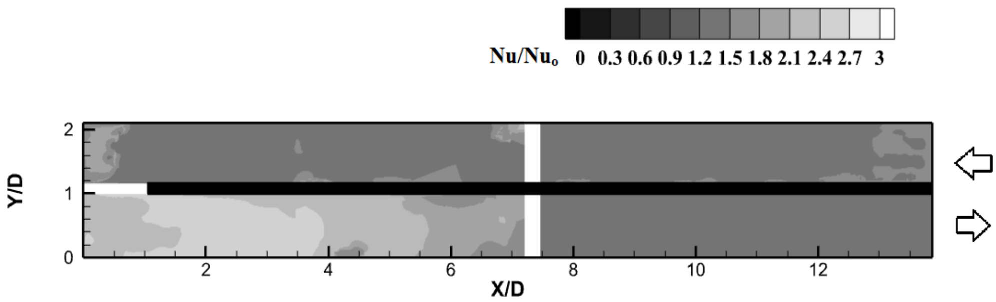

Figure 2. Normalised Nusselt number contours along the smooth duct with Re=36,000, Ro=0 and Pr=5.8.

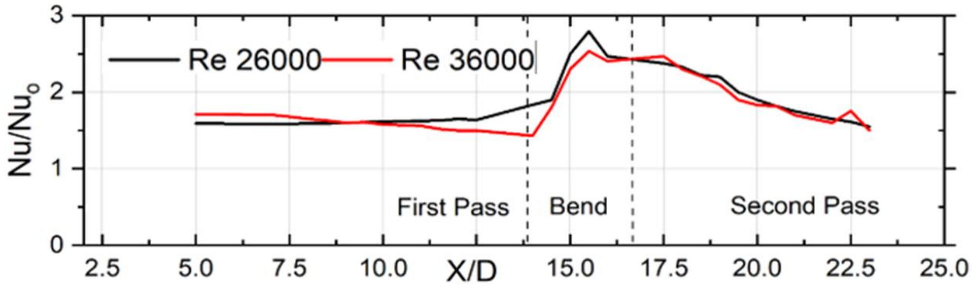

Figure 3. Normalised side-averaged Nusselt number profiles in the smooth passage. Re=26,000-36,000, Ro=0 and Pr=5.8.

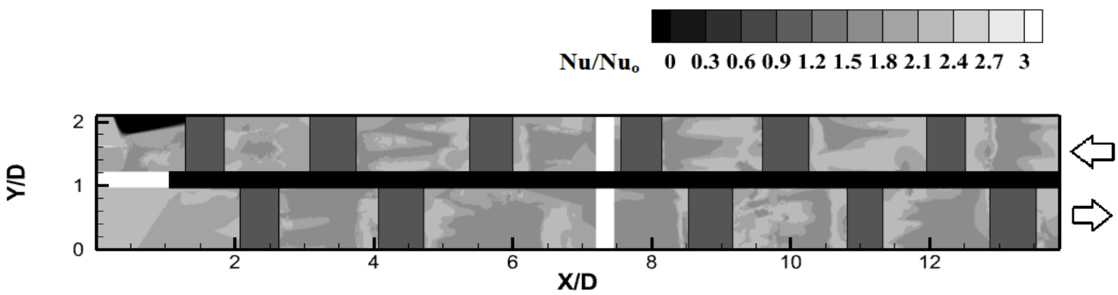

Figure 4. Normalised Nusselt number contours passage with porous blocks in the straight sections for Porosity(ε) = 0.93, S/D =1, Re = 36,000and Pr=5.8

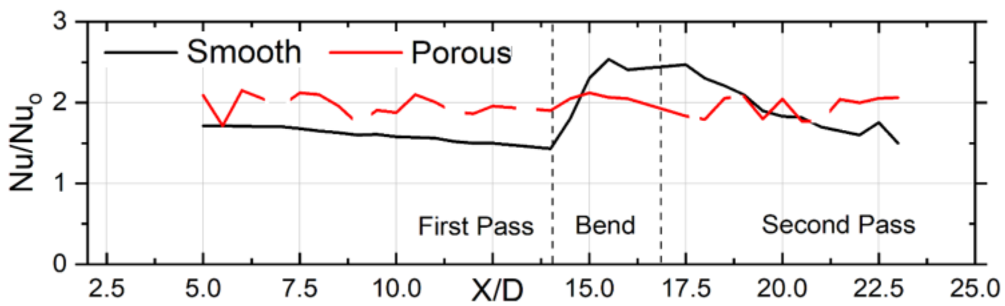

Figure 5. Comparison of the normalised side-averaged Nusselt number profiles between the smooth and a passage with porous blocks in the straight sections for *Porosity*(*ε*) = 0.93, S/D =1, Re = 36,000, Ro= 0 and Pr=5.8.

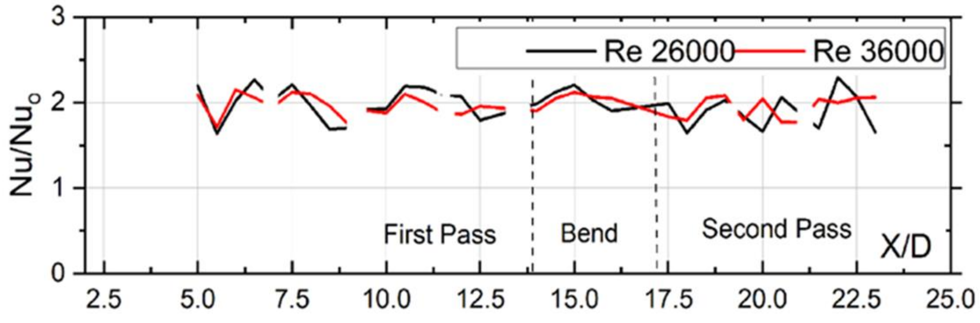

Figure 6. Comparison of the normalised side-averaged Nusselt number profiles for a passage with porous blocks in the straight sections for *Porosity*(ε) = 0.93, S/D =1, at Re of 26,000 and 36,000, Ro= 0 and Pr=5.8.

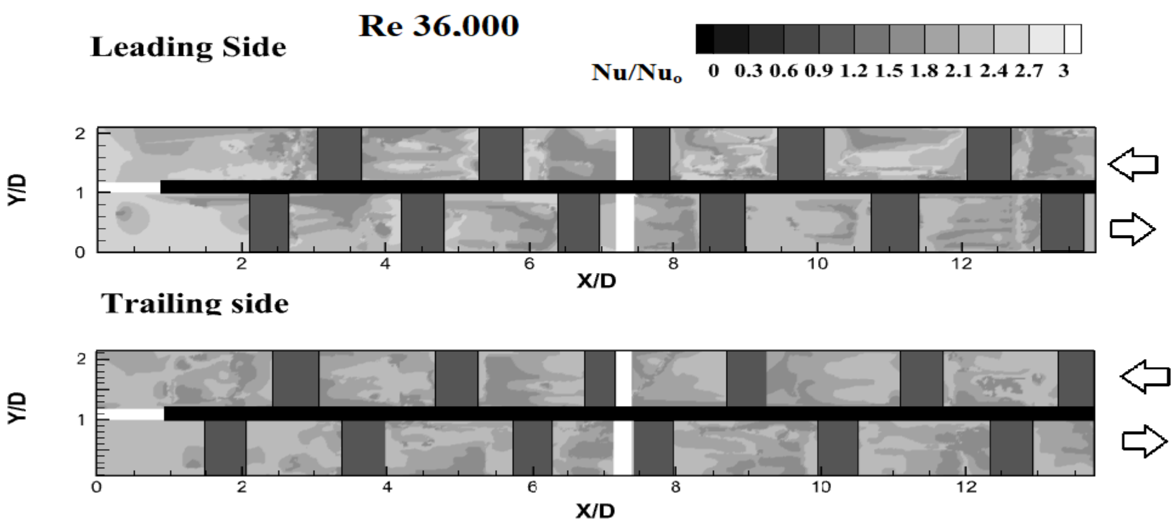

Figure 7. Normalised Nusselt number contours along both the leading and trailing sides of passage with porous blocks in the straight sections for Porosity(ε) = 0.93, S/D =1, Re = 36,000, Ro=0.32 and Pr=5.8.

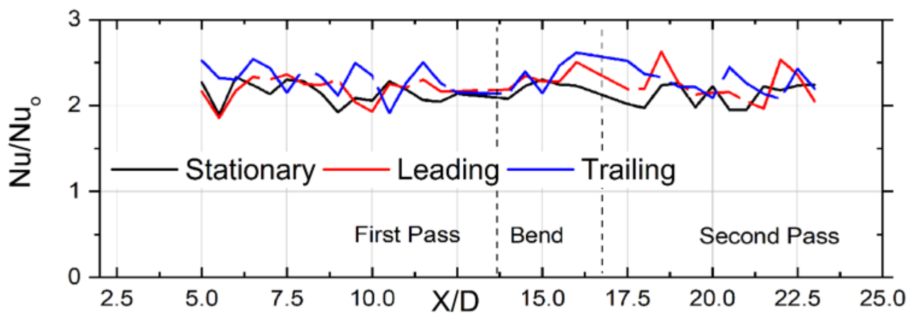

Figure 8. Normalised side-averaged Nusselt number along both the leading and trailing sides of passage with porous blocks in the straight sections for Porosity(ε) = 0.93, S/D =1, Re = 36,000, Ro= 0.32 and Pr=5.8.