NUMERICAL INVESTIGATION OF SLOT JET REVERSE IMPINGEMENT

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ABSTRACT

This work formed part of the first study to adopt a novel reverse flow technique into a double-wall cooling structure in order to achieve a higher heat transfer rate with a relatively low pressure drop. The Computational Fluid Dynamics (CFD) method has been used to study the effects of nozzle-to-target spacing distance, jet-to-jet spacing, and Reynolds number on the heat transfer rate for an array of slot impingement jets. In this study, nozzle-to-target spacing distance (H) was varied from 2 to 8 jet nozzle widths (B), jet-to-jet spacing distance (S) was also varied between 4 - 8 jet widths, and Reynolds number (Re) between 20,000 – 60,000, as shown in Figure 1. Results identified that the reverse slot jet geometry allowed for greater retention of wall jet velocity and provided an increased heat transfer surface area. Relative to a flat target, the highest overall enhancement was seen at the lowest Re range, 20,000. As H/B increased, with S/B set at 4, the effect of heat transfer against the vertical walls improved total heat transfer enhancement, increasing up to H/B = 8. When fixing H/B at 4, and varying S/B, the overall heat transfer enhancement relative to a flat target is also most pronounced at S/B of 4.

1. INTRODUCTION

Double-wall cooling structures are commonly used to enhance the heat transfer in combustor liner and aerofoil internal cooling within gas turbine engines and several other industrial applications. Within double-wall cooling schemes, impingement jets are often utilised to provide a highly targeted, and enhanced region of heat transfer. Much research has been conducted to evaluate the magnitude and distribution of heat transfer provided by impinging jet configurations, including the effect of jets impinging on a concave target, Wright et al [1], and more recently in the evaluation of the novel ‘reverse jet’ configuration, Ahmed et al [2]. This reverse jet impingement geometry was developed to reduce the impact of crossflow between subsequent jets in a row or array, and to also maximise the effective available heat transfer surface area. This is the first research to model this novel reverse target arrangement with a slot jet nozzle and was intended to begin to assess the effect of Reynolds number, H/B, and S/B on heat transfer enhancement relative to a baseline flat target case.

2. METHODS

To assess the effect of the parameters stated, an initial array of testing variations was devised, and those considered are shown in Table 1. The geometry ranges were chosen to be representative of those in a typical industrial application. When increasing H/B, the dimple surface remained identical, but we see an elongation of the vertical walls. As S/B increases, we proportionally increase the dimple radius, and therefore width of the impingement cavity.

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Case number	Reynolds number	H/B	S/B
1	20,000	4	4
2	30,000	4	4
3	40,000	4	4
4	50,000	4	4
5	60,000	4	4
6	30,000	2	4
7	30,000	6	4
8	30,000	8	4
9	30,000	4	6
10	30,000	4	8

Table 1 - Parameter variations evaluated

To achieve an understanding of the heat transfer and flow dynamics within the geometry, the commercial CFD software, ANSYS Fluent, was utilised. With the aim of getting a comprehensive interpretation for the flow field and heat transfer in the reverse jet impingement cooling structure, two-dimensional numerical computations have been carried out with use of the commercial code 'ANSYS Fluent 19.2'. A grid independence check has been performed to confirm that the results are not dependent on the mesh resolution. The thickness of the first meshing layer adjacent to the target surface is set to be 0.004 mm with expansion ratio of 1.2 to ensure that the maximum local value of the Y plus is less than 1.0. The SST k-ω turbulence model was used in all computations because of its good performance in predicting the heat transfer in jet impingement configurations at relatively low computational costs, Spring and Weigand [3], and Guo et al [4]. Coupled schemes for pressure-velocity coupling were used for better convergence.

Air was considered as the working fluid in the fluid domain, and ideal gas assumptions were made to determine its properties in calculations, the upstream jet temperature was used for the working fluid temperature, and for the slot jet, the hydraulic diameter was equal to twice the slot width ($D_h = 2B$). The model was a conjugate simulation, with aluminium as the solid, with a temperature applied at the external wall shown on Figure 1. Heat transfer data is provided as both an average Nusselt number against the constituent heat transfer surfaces, and also as an enhancement to the heat transfer. This method of displaying the data is due to the fact that although the reverse jet impingement method may have a lower magnitude of Nusselt number, it allows for a greater surface area of heat transfer to take place with the same constraints of H/B and S/B.

3. RESULTS AND DISCUSSION

All variations introduced in Table 1 were numerically evaluated, and heat transfer enhancement was evaluated by both the overall amount of heat transfer observed compared to that of a flat plate, and also as an average Nusselt number over each of the heat transfer surfaces.

Figure 2 shows the velocity contours of a typical slot jet impinging on both a flat target, and against a reverse jet target, at Re of 30,000, S/B of 6, and H/B of 6. Velocity profile of the potential core can be seen to be very similar, but with a slight extension of potential core in the reverse jet case. Where the wall jet is typically turned into a fountain upon interaction with its neighbouring jets, we of course see no jet-to-jet interaction in the reverse jet case. Instead, as the reverse jet exits the 'dimple' and continues a wall jet along the 'vertical' wall, it is seen to retain far more of its velocity during this turn, and we observe this increased velocity along the extent of the reverse jet vertical wall.

Figure 3 Shows the effect of Re on overall heat transfer enhancement for a H/B and S/D of 4, for both the dimple surface, and the reverse vertical wall surfaces relative to the heat transfer enhancement on a flat target. This data demonstrates that the reverse jet impingent scheme provides over twice the effective heat transfer when impinging against walls of identical properties, and the specified nozzle parameters. The heat transfer enhancement appears to be greatest at the lower end of the Re range, with decreasing magnitude of enhancement as Re increases. Figure 4 provides the average Nusselt number observed on each of the three heat transfer surfaces, with the flat target shown alongside the two constituent surfaces of a reverse jet target, the initial dimple, and the vertical wall. Figure 4 shows that within the Re range tested, the flat target provides very similar average Nu to the vertical wall, and far greater than the dimple surface. When considering total heat transfer however, it does not account for the greater possible surface area when using the reverse geometry.

Figure 5 shows the effect of H/B on heat transfer enhancement for a Re of 30,000 and S/B of 4. It shows that the maximum heat transfer rate enhancement is obtained at the highest H/B value tested, 8. Also, it illustrates that vertical walls can play a pivotal role in increasing the heat transfer rate by increasing the internal surface area, effectively working as internal fins. Figure 6 shows the average Nusselt number of all surfaces. The average Nu for the reverse target and the flat surfaces is very similar at H/B = 4 and 8 but it should be noted that the surface area for the reverse geometry is greater than the flat surface. At H/B = 6, we see that we may have an optimum for average Nu on all three surfaces, especially the dimple and flat walls, with the average Nu for the vertical wall changing only slightly with increasing H/B, however, it is worth noting that the vertical wall surface area at H/B = 8 is double the surface area at H/B = 4, so we also see this average applied to a far greater surface area. At H/B = 2, no vertical walls exist, only the dimple surface.

Figure 7 shows the effect of S/B on heat transfer enhancement for a Re of 30,000 and H/B of 4. We can observe that the increase S/B, and therefore increase the width of the impingement cavity, this lack of constraint, and increasing dimple surface area appears to increase the magnitude of heat transfer on the dimple surface. If we consider the vertical wall surface, however, the overall heat transfer is inversely proportional to the S/D. This may be due to the smaller diameter of the turning flow causing the boundary layer of the wall jet to be thinner. Figure 8 shows the effect of S/B on average Nu for a Re of 30,000 and H/B of 4, we can observe that an optimum S/B for the dimple surface under these parameters is at 4, although the variation within the range tested is limited. We see that there is a significant drop in vertical wall average Nu as the cavity width increases, likely due to the fact that the wall jet has further to travel around the dimple before it reaches the vertical wall. At S/B = 8, and H/B = 4, there is no vertical wall, as the dimple is proportionally larger. When considering the baseline flat wall case, however, we also see that as S/B increases we see a downward trend in average Nu over the heat transfer surface.

4. FIGURES

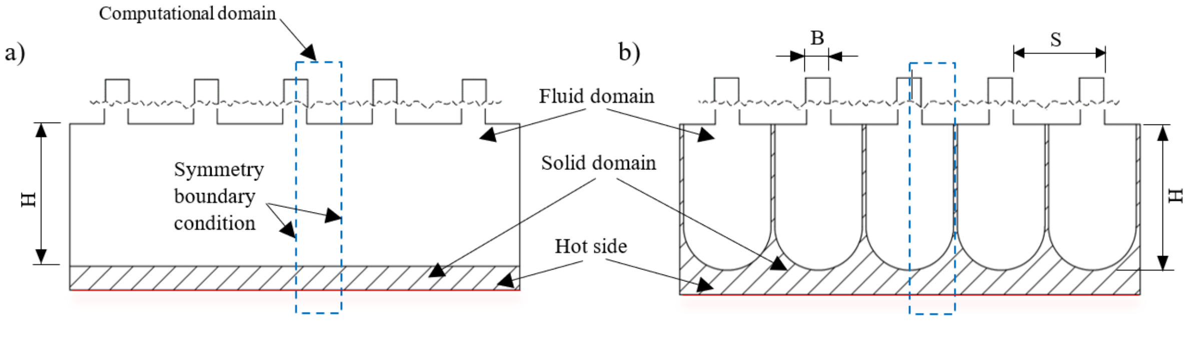

Figure 1 - The computational domains for the slot impinging jet: a) Flat target and b) Reverse target.

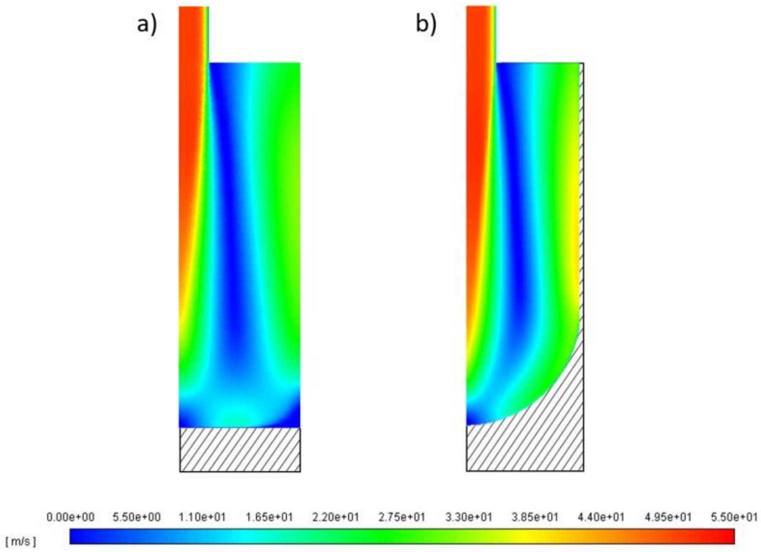

Figure 2 - Velocity contours for the slot impinging jet at H/B = 6 and Re = 30,000: a) Flat target and b) Reverse target.

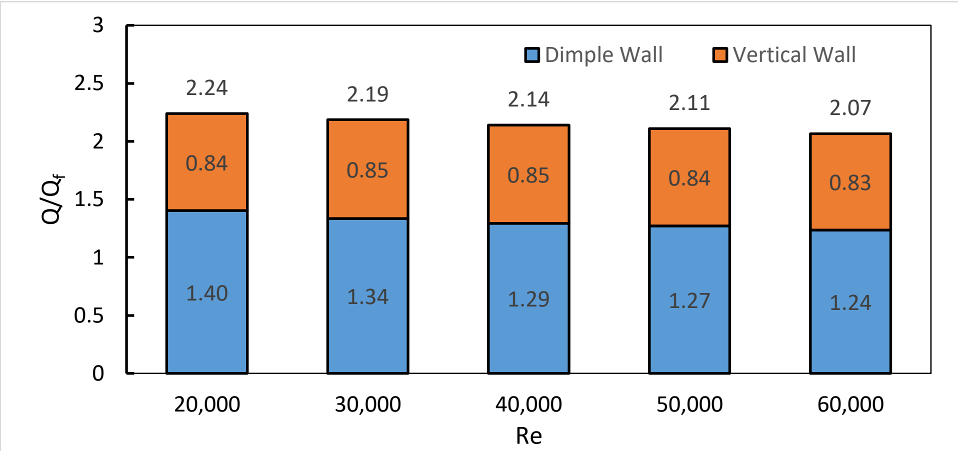

Figure 3 - Effect of Reynolds number on heat transfer enhancement for reverse jet impingement relative to flat-plate impingement at H/B = 4 and S/B = 4

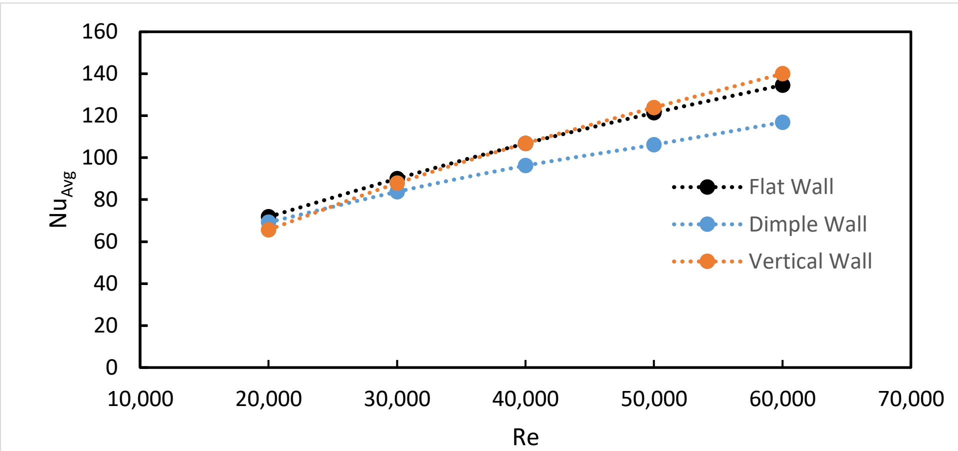

Figure 4 - Effect of Reynolds number on average Nusselt number at H/B = 4 and S/B = 4

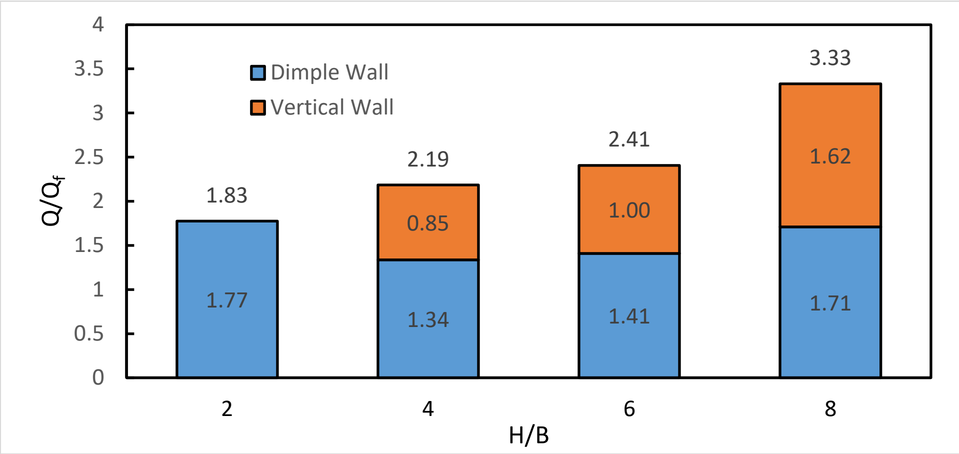

Figure 5 - Effect of H/d on heat transfer enhancement for reverse jet impingement relative to flat-plate impingement at Re = 30,000 and S/B = 4

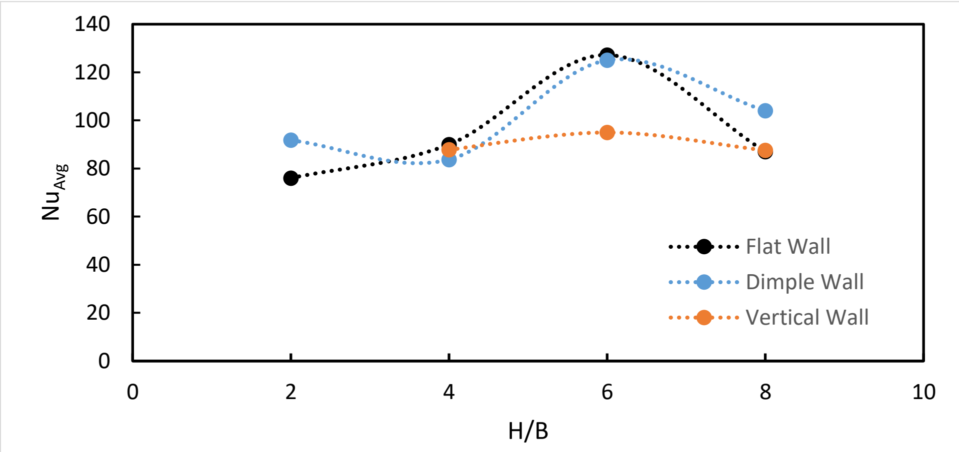

Figure 6 - Effect of H/d on Nusselt number at Re = 30,000 and S/D = 4

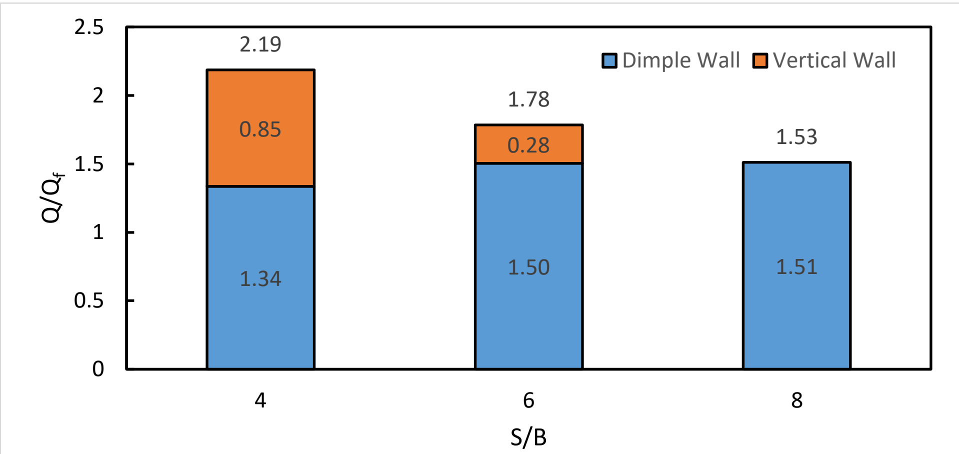

Figure 7 - Effect of S/B on heat transfer enhancement for reverse jet impingement relative to flat-plate impingement at Re = 30,000 and H/B = 4

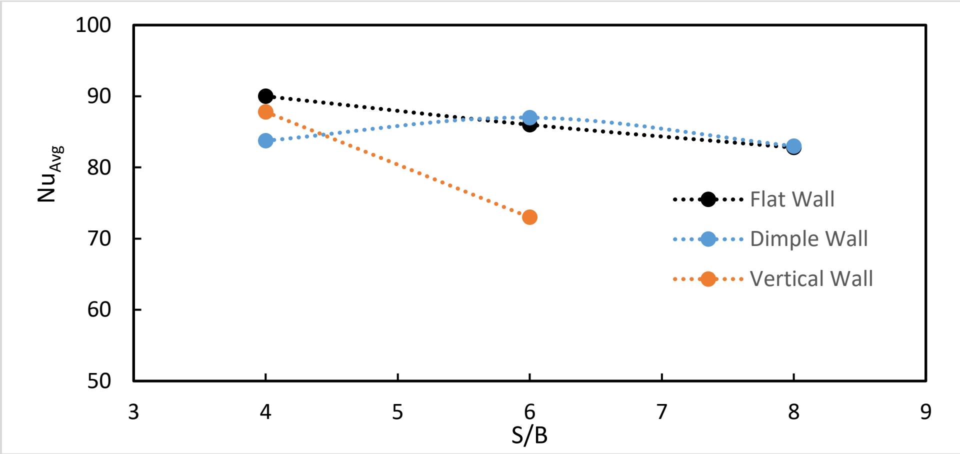

Figure 8 - Effect of S/B on Nusselt number at Re = 30,000 and H/B = 4

5. CONCLUSIONS

Results showed that the reverse slot jet geometry allowed for greater retention of wall jet velocity and provided an increased heat transfer surface area. Relative to a flat target, the highest overall enhancement was seen at the lowest Re range, 20,000. As H/B increased, with S/B set at 4, the effect of heat transfer against the vertical walls improved total heat transfer enhancement, increasing up to H/B = 8. When fixing H/B at 4, and varying S/B, the overall heat transfer enhancement relative to a flat target is also most pronounced at S/B of 4.

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