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HEAT TRANSFER ENHANCEMENT IN SINGLE IMPINGING JETS DUE TO SURFACE CAVITIES

Andrew J C King and Tilak T Chandratilleke*
* Author for correspondence
Department of Mechanical Engineering
Curtin University of Technology
GPO Box U 1987, Perth
Western Australia

E-mail: t.chandratilleke@curtin.edu.au

ABSTRACT

This paper presents an assessment of a novel technique that further enhances the heat transfer potential of a single impinging jet. The method entails a geometrical modification to the jet impingement surface wherein the jet is directed into a cylindrical cavity located coaxially beneath the jet orifice.

A numerical study is performed to examine the parametric influence on heat dissipation and flow characteristics of this modified jet impingement process. The results indicate a very significant increase in heat transfer, which is primarily dependent on cavity depth and jet Reynolds number.

INTRODUCTION

Effective removal of internally generated heat from electronic circuitry is critical for fail-safe operation of electronic devices. Product miniaturisation and increased functionality have caused internal heat generation of modern devices to surge to an unprecedented level of 10^3 W/cm^2 [1, 2]. These intense heat dissipation requirements are beyond the cooling capacity of conventional techniques that are mostly based on convective heat transfer mechanisms. The electronics industry is urgently seeking high-powered cooling and thermal enhancement solutions that would surpass current heat dissipation thresholds and meet new heat load demands.

Heat dissipation techniques incorporating air streams are traditionally at the forefront of electronic device cooling because of their simplicity and convenience. However, weak convective mechanisms of such techniques have inherent limitations and fail to fulfil intense cooling requirements of modern high-powered electronic devices. Whilst cooling modules using organic liquids or water may enhance heat dissipation, they add substantial premiums to systems in terms of increased complexity, lower reliability and higher manufacturing cost. Therefore, except for high-end applications, air-cooling still remains a very attractive proposition for heat removal provided novel enhancement methods are at the disposal of electronics industry.

Impinging air jets are highly regarded for their outstanding heat removal capabilities and are acknowledged as a viable electronic cooling option that well exceeds the performance of conventional approaches. Individual impinging jets provide an unmatched advantage for creating very intense localised cooling effect, which is very beneficial for regulating hot spots common to most electronic circuitry.

While heat and fluid flow characteristics of single jets are widely investigated, relatively fewer attempts are reported on developing effective heat transfer enhancing methods for jet impingement whereby jet heat dissipation levels can be further improved. This work presents such novel technique of high heat transfer potential.

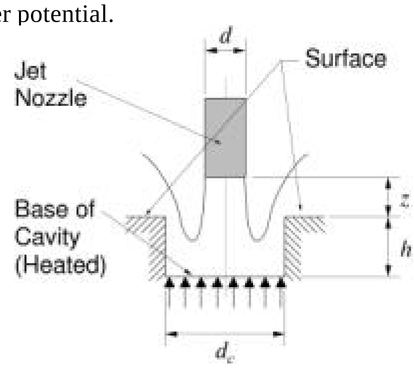

Figure 1 Diagrammatic representation of jet impingement with surface cavity.

NOMENCLATURE

d	[m]	Diameter of jet nozzle
d_c	[m]	Diameter of cavity
h	[m]	Depth of cavity
L	[m]	Net depth of cavity (distance between jet nozzle exit and base of cavity)
z	[m]	Height of jet nozzle exit above surface
Nu	[-]	Nusselt Number = $h_c d/k_f$
Re	[-]	Jet Reynolds Number = Vd/ν

METHODOLOGY

The effectiveness of the proposed cavity surface modification is evaluated through a parametric study using numerical simulation. The parameters selected are: the cavity depth, the jet-to-surface distance, and the jet Reynolds number. The investigated range for each parameter is given in Table 1.

Table 1 Parameter ranges

Parameter		Values
Jet-to-surface distance	*z*	-2, -1, 0, 1, 2
Cavity-to-jet diameter		2
Cavity depth	*h*	0, 1, 2, 3, 4
Reynolds number	Re	5000, 10000, 20000, 30000

The heat dissipation capacity of the proposed technique is assessed by evaluating the average Nusselt number $\overline{Nu}$ over only the base area of the cavity, which is given by,

$$\overline{Nu} = \frac{4\dot{q}d}{\pi d_c^2 k_f \Delta T} \qquad (1)$$

where $\dot{q}$ is the heat transfer from the cavity base. This definition of $\overline{Nu}$ permits a fair comparison between impinging surfaces with and without cavity, and the realisable thermal enhancement potential from the surface modification.

Boundary Conditions

The boundary conditions are carefully chosen to not overshadow the cavity influence on heat transfer rates. For this reason, the heat transfer to the jet fluid is allowed only through the cavity base while treating all other walls (cavity sidewalls, reference surface and jet nozzle walls) as adiabatic. The cavity base is treated as an isothermal wall with a temperature of 25 K above the jet fluid temperature. A constant velocity profile with magnitudes corresponding to the Reynolds numbers given in Table 1, is applied at the jet inlet. Neumann boundary conditions with a pressure gradient of zero are applied to the fluid exits at the top and sides of the domain.

Turbulence Modelling

For parametric studies where the flow is known to be turbulent, a variation of the two-equation *k-e* model [3] is usually employed. For impinging jet flows however, it is well reported that the *k-e* model fails to predict the flow correctly along with heat transfer rates [4]. While a number of new models have been proposed to overcome this difficulty, the v^2-f model [5] has shown very good agreement with experiment, and is generally the best choice for modelling impinging jet flows [6,7]. This model is used in the current work.

RESULTS

An appropriate length parameter called Net Cavity Depth is defined to highlight trends in results. The net cavity depth L is the sum of jet-to-surface distance z and cavity depth h, as given by Equation (2),

$$L = z + h \qquad (2)$$

In here, *z*, is considered positive when the nozzle discharges above the surface and negative when it is below the surface.

Figures 2 to 5 show the variation of $\overline{Nu}$ with the net cavity depth for Reynolds numbers 5000, 10000, 20000 and 30000.

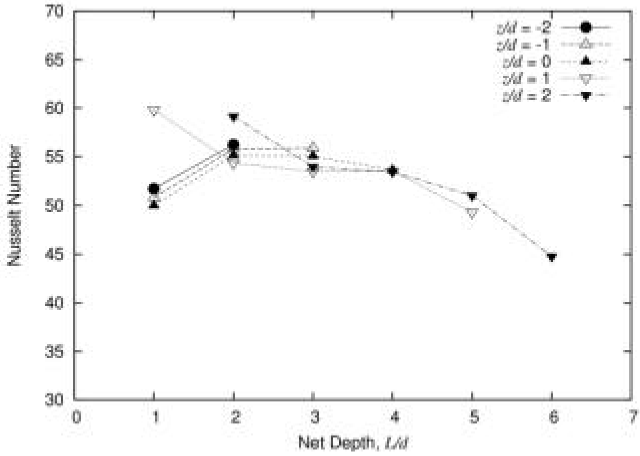

Figure 2 Average Nusselt number at base of cavity for Reynolds number of 5000

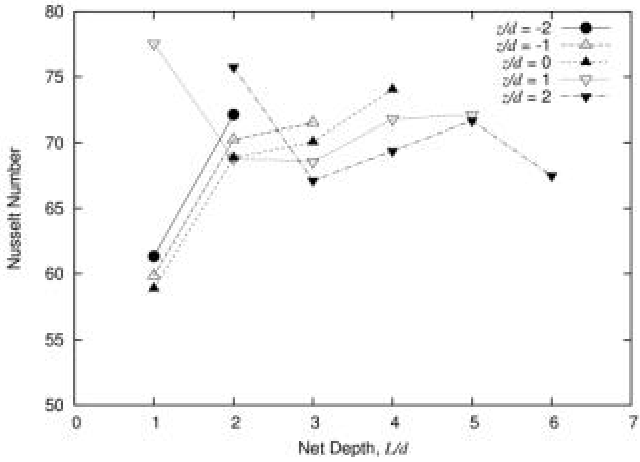

Figure 3 Average Nusselt number at base of cavity for a Reynolds number of 10000

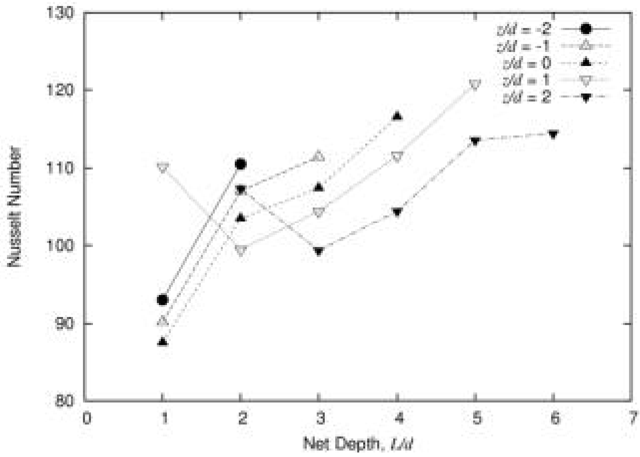

Figure 4 Average Nusselt number at base of cavity for a Reynolds number of 20000

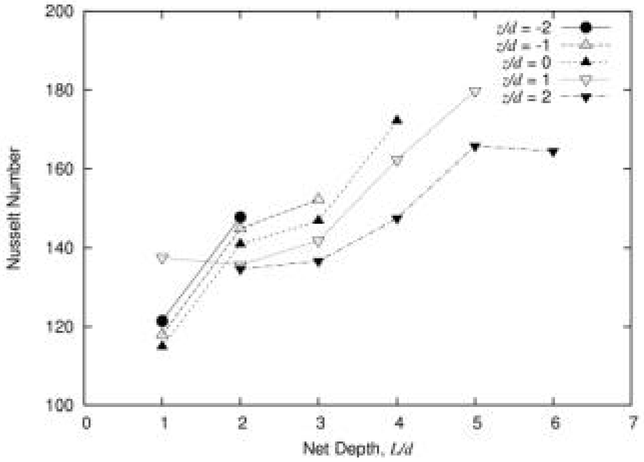

Figure 5 Average Nusselt number at base of cavity for a Reynolds number of 30000

Validation

Due to the novel nature of the proposed cavity arrangement, no experimental data is currently available for validating the numerical results. For this reason, the numerical model is only validated against the experimental data for the reference flat plate case. Figure 6 shows a comparison of the local Nusselt number for a jet-to surface distance of z/d = 2, at a Reynolds number of 23000.

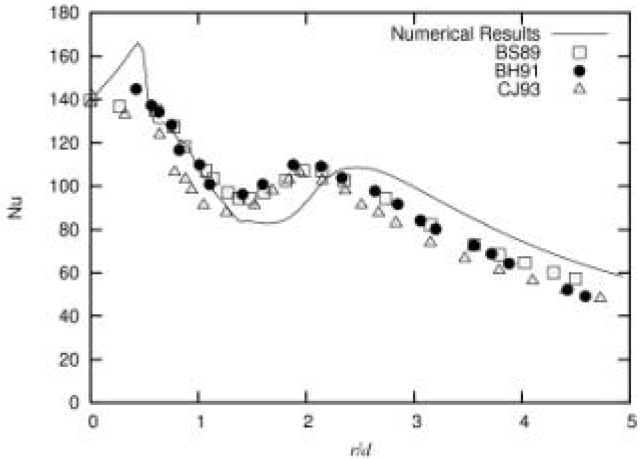

Figure 6 Variation of local Nusselt number for a flat plate at a Reynolds number of 23000.
BS89 – Baughn and Shimizu [8], BH91 – Baughn et al [9]
CJ93 – Cooper et al [10].

Trends in Results

When identifying the trends in the results depicted in Figs. 2 to 5 for the surface cavities, the data is grouped into two broad categories. The first category consists of cases where the jet exit is initially below or in-line with the level of the reference surface (z = 0), while the second category is made up of those cases where the jet exit is above the reference surface (*z* > 0).

In the first category, the presence of the cavity promotes the heat transfer rate which increases with the net cavity depth. The exception to this is in Fig. 2 at Reynolds number of 5000, where the heat transfer rate drops when the net depth exceeds 3.

In the second category where the jet exit is above the reference surface, the heat transfer initially falls when the cavity is first added to the surface, and then rises with increased net depth. The observed drop is most prominent at Reynolds numbers of 5,000, 10,000 and 20,000 while for 30,000 a negligible change in heat transfer is evident due to the cavity. In this category for Reynolds numbers 5,000 and 10,000, although the heat transfer increases after the initial drop, it never seems to exceed its magnitude corresponding to the reference (no cavity) surface. This clearly indicates the inclusion of cavities will not be beneficial for jet flows with this range of Reynolds numbers. To the contrary for Reynolds numbers of 20,000 and 30,000, the heat transfer rates well exceed those for the reference surface and significant thermal benefits are realised by adding cavities to the impingement surface.

In summary, for low Reynolds number jets (Re = 10,000) introducing a cavity to the surface leads to a reduction in heat transfer compared to a single jet impinging on a flat surface. At high Reynolds numbers (Re > 10,000), the addition of a surface cavity initially decreases heat transfer compared to a flat plate. If the cavity is subsequently deepened, it is possible to significantly enhance heat transfer rates.

CONCLUSIONS

A surface modification in the form of a cylindrical cavity situated beneath an impinging jet have been studied and proposed as an enhancement technique. The results indicate that the net-depth of the cavity is a significant factor in obtaining improved heat transfer rates from jet impingement process. In general, the heat transfer increases as the net-depth increases, although for some cases a reduction in heat transfer is observed upon the initial introduction of the cavity to the surface. However, this reduction becomes less apparent as the Reynolds number is increased. Heat transfer enhancement potential is significantly high at Reynolds numbers-above 20,000. It is envisaged that further enhancement would be possible if the cavity sidewalls are also permitted to contribute to the heat transfer process.

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