Experimental Optimisation of the Thermal Performance of Impinging Synthetic Jet Heat Sinks

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Abstract. Zero-net-mass flow synthetic jet devices offer a potential solution for energy-efficient cooling of medium power density electronic components. There remains an incomplete understanding of the interaction of these flows with extended surfaces, which prevents the wider implementation of these devices in the field. This study examines the effect of the main operating parameters on the heat transfer rate and electrical power consumption for a synthetic jet cooled heat sink. Three different heat sink geometries are tested. The results find that a modified sink with a 14x14 pin array with the central 6x6 pins removed provides superior cooling to either a fully pinned sink or flat plate. Furthermore each heat sink is found to have its own optimum jet orifice-to-sink spacing for heat transfer independent of flow conditions. The optimum heat transfer for the modified sink is $H = 34$ jet diameters. The effect of frequency on heat transfer is also studied. It is shown that heat transfer increases superlinearly with frequency at higher stroke lengths. The orientation of the impingement surface with respect to gravity has no effect on the heat transfer capabilities of the tested device. These tests are the starting point for further investigation into enhanced synthetic jet impingement surfaces. The equivalent axial fan cooled pinned heat sink (Malico Inc. MFP40-18) has a thermal resistance of 1.93K/W at a fan power consumption of 0.12W. With the modified pinned heat sink, a synthetic jet at $Re = 911$, $L_0/D = 10$, $H/D = 30$ provides a thermal resistance of 2.5K/W at the same power consumption.

1. Introduction
Recent developments of the ICT industry have led to increasing power density of electronic and photonic packages. The desire to decrease the size of computers as well as increase power means that the direction of the industry is towards greater power from smaller devices. A problem inherent in increasing the power is the heat generated from the chip unit [1]. The power increases and size reductions of electronic devices means that existing methods of cooling are becoming insufficient to meet demands. Much research is currently being carried out in the field of electronics cooling in an effort to find a cooling technology which can fulfil the cooling needs. A large degree of what is being researched and investigated involves jet flow impingement in various forms.

Synthetic jet flow is named so as the fluid motion is ‘synthesized from the ambient fluid. As such the system is also known as a Zero-Net-Mass flux device. Synthetic jets have been studied in depth for many decades, in particular as flow alteration devices, such as for stall control [2]. More recently in response to the increasing power density of chips the devices have been investigated as heat dissipation devices.
The basis of operation is to create vortical flow which will dissipate the heat from the heated surface to the ambient. The vortical flow is a series of repeated vortices ejected individually through an orifice. The flow is typically generated suction and ejection of fluid from a small cavity using a diaphragm [3]. The pulsating nature of synthetic jet flows increases entrainment of ambient fluid in comparison to a continuous jet, which in turn provides a cooling performance, comparable to continuous impinging jets, yet without the need for a pressurized fluid supply [4]. A major advantage of synthetic jets over other jet flows is there is no need for external pressurized fluid making such devices more spatially efficient and less expensive. Defining parameters of synthetic jets are the Reynolds number \( \text{Re} = \frac{U_0 D}{\nu} \) and the stroke length \( L_0/D \). The stroke length is the displacement of a slug of fluid from the orifice during the ejection phase of the cycle [5], in this case made non-dimensional by the diameter of the hole. The velocity of fluid through the system is taken as the frequency at which the stroke length is repeated such that

\[
U_0 = f \cdot L_0 = f \cdot \frac{1}{\int_{t=0}^{t_{max}} u_0(t) dt}
\]

where \( u_0(t) \) is the area-averaged orifice velocity. The stroke length determines the flow characteristics, particularly at low values of \( L_0/D \). Vortex detachment from an axisymmetric sharp-edged orifice will not occur unless \( L_0/D > 0.5 \)[5]. The flow field of impinging synthetic jets is further characterised by the orifice-to-surface distance \( H \). This spacing parameter is typically assigned a non-dimensional form relative to the orifice diameter \( H/D \). The flow characteristics of impingement are heavily dependent on the spacing and studies have shown an optimal value exists within each system in order to achieve maximum heat transfer [6-9].

Although the potential for synthetic jets as cooling devices has been researched there exist few practical applications in existence today. The reason behind this is due to a lack of complete understanding of the best operating conditions of a synthetic jet system. Research by Gillespie et al. [9] showed substantial enhancement in heat transfer using a zero net mass transfer device due to the strong mixing characteristics. Furthermore, studies have shown a wide range of spacing (4 < \( H/D \) < 11) at which the heat transfer is maximised [9, 10]. However, the long-term goal of implementing synthetic jets in small electronic devices is to further decrease the size of devices. In this light research such as that carried out by Valiorgue et al. [11] is prevalent as it investigates heat transfer characteristics in small spacing. Particle image velocimetry was used to show how varying parameters such as the stroke length and the Reynolds number dramatically affected the flow and heat transfer characteristics. This study highlights how the ability for a synthetic jet actuator to dissipate heat is largely dependent on the spacing (\( H/D \)). Furthermore it provides visual data which contrasts well with data provided ten years earlier by Gharib et al. [12] to show the difference in fully develop vortices with entrainment and vortices which recirculate due to lack of entrainment. A significant flaw of synthetic jets is highlighted in this research, which is recirculation. The suction of previously ejected fluid means that the temperature of air within the cavity of the actuator will increase and thus have inferior cooling potential.

Persoons [13] developed a model to design and operate synthetic jet actuators by coupling the mechanical and fluidic model. Experimentation within the same paper verified that for rectangular and circular orifices the model holds true.

Evidently the heat transfer capabilities of a synthetic jet system are heavily dependent on the geometric parameters which define it. Chaudhari et al. [7] investigated the effect of geometry by empirical means to qualify how the flow produced by different shaped orifices affected the heat transfer away from a heated surface. Research obtained interesting results which clarified that at larger spacing a circular orifice is preferred whereas at smaller spacing a rectangular orifice provides superior cooling. This mirrors studies which visualise flow from slot orifices and show that the long vortices generated along the length of the slot curl-up and collapse on themselves [4, 14]. The reason for this transition to turbulence lies in the generation of counter rotating stream-wise vortex pairs that wrap around the primary vortices. The onset of the counter rotating vortices is caused by instability.
and small scale motion of the primary vortices[4]. The key to maximising heat transfer using synthetic jets is to correctly design the orifice and system which produces a flow which achieves strong entrainment of ambient fluid and has vigorous mixing near the impingement surface[15].

Although much research has been carried out on the effect of orifice geometry, the understanding of the effect of heat sink geometry on heat transfer characteristics is incomplete. The current study aims to address this lack of knowledge and quantifiably compare the heat transfer achievable with different heat sink configurations. The research investigates the heat transfer as a function of spacing (H/D) with each sink. The study also looks at how the heat transfer is affected by the frequency under the circumstances of constant Reynolds number and constant stroke length.

2. Experimentation

2.1. Test Facility

The design of the system is inspired by the set-up and testing carried out by Chaudhari et al. [6, 7, 16]. The experimental rig allows easy spacing adjustment of the jet actuator using a slider mechanism. The jet actuation is produced by the oscillatory motion of the diaphragm of a Visaton BF32-8Ω speaker. The cavity is cylindrical aluminium which fits around the perimeter of the diaphragm with 0.5mm clearance to allow unrestricted motion of the diaphragm. The cavity wall has a height of 10mm resulting in a total cavity volume of 7068mm$^3$. The orifice plate is located directly opposite the speaker. The cylindrical orifice is constructed from nylon of 5mm thickness. The orifice hole is 2mm diameter. For the purposes of comparison tests are carried out with a fan cooling device impinging on the same heat sinks. The fan is a Sunon Maglev GM0504PEV1-8, mounted directly on top of the sink being tested (H = 0).

The jet produced impinges onto a pin fin heat sink (Malico Inc. MPF40-18). This sink is an array of 14x14 fins each 8mm high, diameter 1mm. The sink is anodized aluminium. The system is designed to be modular for easy changing of heat sinks. The heating apparatus is displayed in figure 1. A heater plate is adhered to the underside of a copper block of 7mm thickness. The copper block sits within an insulation block which is insulated on all but one side. The exposed side, measuring 40mmx40mm, faces the synthetic jet. On the exposed side a heat sink of selected geometry will be placed. To ensure good thermal contact a thermal paste is used which minimizes any contact resistance. All heat conducted through the copper block to the heat sink side is assumed to dissipate from the heat sink surface (i.e. conduction resistance of heat sink base is negligible). A constant power supply to the insulated face of the copper block is maintained at 2.6W.

Figure 1. Schematic diagram of the test apparatus
2.2. Synthetic jet operating point

The focus of this study is primarily on the heat transfer characteristics of the system. As such the measurement devices in operation are focused on temperature monitoring more so than flow characterisation or visualisation. The synthetic jet is operated using LabView interface and National Instruments CompactDAQ System designed according to a model derived by Persoons [13,17]. A G.R.A.S. 40BH microphone and amplifier are used for pressure measurement in the cavity. Having set the operating frequency and amplitude the program takes in the pressure measurement and by accounting for the size of the cavity and nature of the orifice exit calculates critical flow data such as the stroke length, area-averaged orifice velocity and Reynolds number. The calculations carried out by the program implemented are those derived by Persoons and O’Donovan to estimate jet velocity from the cavity pressure [17].

2.3. Convective heat transfer measurements

The method of calculating the heat transferred from the surface of the heat sink is described below. The heat input to the system is equal to the heat dissipated through conduction, radiation and the convective heat transfer away from the surface is obtainable. The heat dissipated by convection is taken as the product of the heat transfer coefficient and the difference in temperature between the surface and ambient. In the field of synthetic jets it is more appropriate to use the difference between the surface and the cavity temperature:

\[ q_{\text{conv}} = q_{\text{gen}} - q_{\text{cond}} - q_{\text{rad}} = h(T_{\text{surface}} - T_{\text{jet}}) \]  

(2)

As different heat sink geometries will result in varying natural convection capabilities it is necessary to distinguish the forced convection from the total achievable convection. It is assumed that the fin efficiency is 100% so that the base plate temperature is an accurate approximation of the true surface temperature, to appropriately and quantitatively compare how effectively a synthetic jet transfers heat away from a sink due to forced convection it is necessary to calculate the heat transfer from each sink through forced convection only. The Nusselt number for forced convection is found by separating the Nusselt number due to natural convection from the mixed convection Nusselt number [18]:

\[ Nu_{\text{forced}} = \left( Nu_{\text{mix}}^3 - Nu_{\text{nat}}^3 \right)^{1/3} \]  

(3)

where both \( Nu_{\text{mix}} \) and \( Nu_{\text{nat}} \) are found empirically. Testing for natural convection is carried out with the jet actuator in position turned off at a variety of heights. The need to carry out testing in this manner is due to the effect spacing has on radiation and possible restriction of flow away from the surface in close spacing. By obtaining a curve for \( Nu_{\text{nat}} \) against Ra and H/D equation (3) can be used. The correlation of natural Nusselt number to Rayleigh number is found as:

- Case A (Flat Surface) \( Nu = (2.547 - 1.538/(H/D))Ra^{1/4} \)
- Case B (14x14 Pinned Sink) \( Nu = (2.886 - 1.029/(H/D))Ra^{1/4} \)
- Case C (14x14 - 6x6 Pinned Sink) \( Nu = (2.547 - 1.538/(H/D))Ra^{1/4} \)

The length scale used for \( Nu_{\text{mix}} \) and \( Nu_{\text{nat}} \) is the length of the heat sink base. However, for analysis of synthetic jets a more fitting length scale is the orifice geometry. In order to convert \( Nu_{\text{forced}} \) to \( Nu_{\text{jet}} \) the following equation must be implemented:

\[ Nu = Nu_{\text{forced}} \frac{D}{L_{ref}} \frac{(T_{\text{surface}} - T_{\text{ambient}})}{(T_{\text{surface}} - T_{\text{jet}})} \]  

(4)

where \( L_{\text{ref}} \) is the length of the side of the heat sink base. This can now be used to find a synthetic jet heat transfer coefficient.
The corresponding thermal resistance is

\[ R_{th} = \frac{1}{hA} \]  

where A is the heat sink base area.

2.4. Energy efficiency

Efficiency of synthetic jets is still undefined due to the different applications in different fields. For the purposes of fluid motion the efficiency is typically taken as the ratio of the fluidic power to the electrical actuation power \[13\]. However, for the purposes of this study it is more appropriate to compare the heat transfer to the power input. This quantity is known as the effectiveness, \( \varepsilon \), such that

\[ \varepsilon = \frac{q_{\text{forced}}}{P_{\text{electric}}} \]  

3. Results & Discussion

3.1. Effect of jet orifice-to-impingement surface spacing

Figure 2 shows how heat transfer changes with jet to surface spacing at a given jet operating condition (Re, \( L_0/D \)). The heat transfer coefficient has positive correlation with the non-dimensional spacing parameter such that a maximum heat transfer coefficient is reached at a given spacing \( H/D \). Although the optimum height for heat transfer varies depending on the heat sink geometry it is clear that an optimum spacing exists for each sink. In addition to this the results show that the optimum spacing is a function of sink geometry independent of Reynolds number. This is evidenced by comparison of figure 2a and figure 2b. Although the Nusselt number is drastically reduced with reduced Reynolds number the optimum heat transfer relative to spacing occurs at the same point for each sink. Ignoring minor discrepancies which exist between the two data the trend remains very similar. The pinned sinks achieve optimal heat transfer at a larger spacing than the flat plate. In both graphs the Nusselt number of all three heat sinks is shown to increase to a point where the spacing has less effect as it changes. As stated previously there exists an optimum operating spacing for maximising heat transfer.

Interestingly far superior heat transfer is achieved from a heat sink which incorporates pins in its geometry. The figures shown above consist of forced convection only which indicates that the flow interaction between vortices and pins achieves cooling better than vortices impinging on a flat plate. The height, \( H \) is measured from the point of flow exit to the point of first impingement. In the case of a pinned sink this is from the top of the pins to the bottom of the orifice plate.

Optimum heat transfer is achieved at a much smaller spacing for flat plate (\( H/D = 20 \)) than the optimum spacing for a pinned heat sink (\( H/D = 34 \)). A modified sink was tested in conjunction with the aforementioned sinks. This sink consisted of an array of 14x14 pins with the central 6x6 pins removed. This sink performed better over the entire range than either the flat plate or the fully pinned heat sink. It can be postulated that the existence of pins in the central portion of the plate reduce the flow momentum and impingement. Furthermore, it is possible that the fluid-pin interaction reduced the flow significantly that recirculation occurs for small spacing. For both Reynolds number tests the Nusselt number increases dramatically over the range \( 15 < H/D < 20 \), for the pinned heat sink. Evidenced by the fact that the flat plate outperforms the fully pinned sink at smaller spacing before a drastic increase, it appears that recirculation occurs. When these pins are removed the best of both is achieved. Heat transfer occurs on impingement and as the flow spreads outwards the interaction provides further cooling which is not obtainable with a solely flat plate.
Figure 2. Average Nusselt number as a function of nozzle-to-surface spacing $H/D$ for three different heat sink geometries: (A) flat surface, (B) 14x14 Pins, (C) 14x14 array with central 6x6 pins removed (I. $Re = 1600$, $L_0/D = 12$ and II. $Re = 800$, $L_0/D = 6$)

| Case | Sink | Re   | Optimum $H/D$ | $Nu$  | $h$ [W/(m²K)] | $R_{th}$ [K/W] |
|------|------|------|---------------|-------|---------------|----------------|
| A    | Flat Plate | 1600 | 20            | 7.62  | 86.6          | 7.20           |
|      |       | 1200 | 15            | 6.37  | 77.4          | 8.09           |
|      |       | 800  | 20            | 4.76  | 60.5          | 10.36          |
| B    | Pinned | 1600 | 30            | 14.13 | 151.8         | 4.10           |
|      |       | 1200 | 30            | 10.17 | 115.3         | 5.37           |
|      |       | 800  | 30            | 6.15  | 75.3          | 8.32           |
| C    | Modified | 1600 | 32            | 14.10 | 147.7         | 4.23           |
|      |       | 1200 | 35            | 10.70 | 117.1         | 5.30           |
|      |       | 800  | 30            | 6.31  | 76.1          | 8.20           |

Table 1. Heat transfer at optimum spacing for three different sink geometries at three different Reynolds numbers

| Case | Sink       | $Nu$      | $R^2$     |
|------|------------|-----------|-----------|
| A    | Flat Plate | $3.13 \left( \frac{1 + 3.069 (H/D)^2}{1 + (H/D)^{2.249}} \right)$ | 0.95 |
| B    | Pinned     | $5.04 \left( \frac{1 + 4.343 (H/D)^2}{1 + (H/D)^{4.254}} \right)$ | 0.991 |
| C    | Modified   | $5.935 \left( \frac{1 + 3.69 (H/D)^2}{1 + (H/D)^{3.043}} \right)$ | 0.981 |

Table 2. Average Nusselt number relationship with spacing for three different sink geometries ($Re = 1600$, $L_0/D = 12$)
3.2. Effect of synthetic jet actuation frequency

Testing carried out in the course of this research examined how heat transfer corresponds with changes in frequency. A relationship is found which correlates the the Nusselt number with the Reynolds number. In this case the stroke length is kept constant at a specified number and the Nusselt number is normalised by the Reynolds number. The results show the manner in which the heat transfer increases with frequency. At a smaller stroke length the heat transfer is less susceptible to large change with frequency and thus the correlation between the heat transfer and frequency is linear. At higher stroke lengths each change in frequency affects the flow to a greater extent and the heat transfer increases exponentially with increase in frequency. This increase suggests that to maximise heat transfer in any system frequency should be maximised to a point where the power supplied remains safe for the actuator device.

The increase in frequency results in improved heat transfer. The improvement brought about by the increased frequency is dependent on the stroke length at which the system is operating.

| Case | Sink     | L_o/D | H/D | Nu              | R^2   |
|------|----------|-------|-----|-----------------|-------|
| A    | Flat Plate | 10.1  | 30  | 1.29×10^2 Re^{0.865} | 0.988 |
| B    | Pinned   | 10.1  | 20  | 5.13×10^4 Re^{1.425} | 0.979 |
| C    | Modified | 10.1  | 30  | 1.44×10^4 Re^{1.751} | 0.971 |

Testing carried out to investigate the relationship between heat transfer and frequency whilst maintaining the Reynolds number constant. The results obtained are shown in figure 4. As can be seen the derived equations for the case of pinned sink and flat surface hold true. As frequency is increased the heat transfer relative to the Reynolds number remains constant. Discrepancies at low frequencies are omitted as the power supplied from the loudspeaker was insufficient to produce flow superior to natural convection. Similarly at high frequencies inconsistencies occurred due to overheating of the loudspeaker where the cavity temperature increased due to the temperature of the speaker. As such any data points requiring a power supply greater than 1.5W are omitted.

3.3. Effect of heat sink orientation with respect to gravity

In an investigation into the effects of gravity on cooling the testing for spacing is repeated with the sink in a vertical orientation. The heat sink is perpendicular to the ground and the jet impinges on its surface normally. Figure 4 compares the data obtained from both the horizontal and vertical orientations.

In order to correctly analyse how the orientation affected the heat transfer only the forced convection components of heat transfer are presented in the above data. Although convention would suggest that a vertical plate would achieve better heat dissipation the data obtained does not support this. Minor improvement occurs at some spacing but accounting for margin of error this change is insignificant. The results show also that the orientation has little effect on the optimum spacing. The optimum spacing is maintained for each heat sink irrespective of orientation.
Figure 3. Average Nusselt number as a function of nozzle-to-surface spacing H/D for three different heat sink geometries: (a) flat surface, (b) 14x14 Pins, (c) 14x14 array with central 6x6 pins removed. Re=1600, f =250Hz. (I. Horizontal sink base, II. Vertical sink base)

Figure 4. I. Thermal resistance as a function of Reynolds for three different heat sink geometries for fan created flow. II. Heat transfer coefficient as a function of Power for fan created flow; (A) Flat plate, (B) 14x14 Pin array, (C) 14x14 Pin array with central 6x6 Pins removed

Figure 5. I. Electric power as a function of frequency for three different heat sink geometries (L_o/D = 10). II. Heat transfer coefficient as a function of Power (L_o/D = 10); (A) Flat plate (H/D = 20), (B) 14x14 Pin array (H/D = 30), (C) 14x14 Pin array with central 6x6 Pins removed(H/D = 30).
3.4. Power consumption
A comparison of the data in figure 4 and figure 5 shows that under certain conditions synthetic jet cooling devices outperform traditional fan cooling devices. Of particular interest is the relationship between heat transfer coefficient and input power to the cooling device. As previously stated, the modified pinned heat sink operates most effectively in the case of synthetic jet flows. However for a fan flow a pinned heat sink achieves the highest heat transfer coefficient. As expected as the turbulence of the flow increases the thermal resistance reduces. Although the synthetic jet flow has a much higher thermal resistance at low frequencies the rate of reduction is much greater and thus at greater operating frequencies the thermal resistance is comparable to that of the fan produced flow. Figure 4 II shows that the pinned heat sink has the maximum heat transfer coefficient for any fan power input. This heat transfer coefficient is almost twice as great as h for and equivalent power input to the jet actuator for the same sink. However, the implementation of the modified sink with jet actuation greatly improves the heat transfer coefficient over any of the sinks in use with a fan flow.

4. Conclusion
The study has investigated some of the main parameters pertaining to the heat transfer attainable with impinging synthetic jets on conventional extended surface heat sinks. Experimentation carried out on three different heat sink geometries has shown that any sink implemented will alter the heat transfer relative to its specific geometry independent of flow conditions. In addition to this each case has a unique optimum spacing between the jet actuator and sink where heat transfer is maximized for the given configuration. The study examined the heat transfer from sinks at a variety of flow conditions (in terms of jet Reynolds number 500<Re<2,500 and dimensionless stroke length 5 < L_0/D < 12) and found that the manner in which heat dissipation changed with spacing is repeated.

The comparison of the performance of different heat sink geometries in combination with the same synthetic jet flow shows that at any spacing a modified heat sink with pins surrounding a flat impingement zone provides superior cooling to either a flat surface or a fully pinned sink. The study finds that more research should carried out in the field of impingement surfaces for synthetic jets as flat surface impingement is inferior to pinned array sinks, particularly at larger spacing (H/D > 10).

Orientation of heat sinks in the vertical plane showed no increase in heat transfer capabilities over a sink lying in the horizontal plane. The orientation in the vertical plane further shows that the heat sink geometry affects heat transfer independent of flow conditions.

Investigation into the effects of frequency on the heat transfer shows that at large stroke lengths a change to frequency will have a greater effect on the heat dissipated from the surface. However, at smaller stroke lengths the effect of frequency is decreased.

The analysis of heat transfer coefficient in relation to power consumption indicates that by using the best available heat sink at optimum spacing a synthetic jet flow can enhance cooling.

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