Heat transfer characteristics of a pair of impinging synthetic jets: Effect of orifice spacing and impingement distance

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Abstract. Recent research has shown that an impinging synthetic jet attains local heat transfer rates that rival those of a steady impinging jet. A single synthetic jet still requires a buoyancy-driven or forced cross flow to avoid recirculation of heated fluid. However, multiple adjacent synthetic jets exhibit a fluidic interaction that results in flow vectoring towards the direction of the jet leading in phase. Previous results have shown a significant heat transfer enhancement and the establishment of a jet-induced cross-flow, which could eliminate the need for an external forced cross-flow. This paper presents the findings of an experimental study to optimize the orifice-to-impingement distance \( H \) and orifice centre-to-centre separation distance \( S \). The convective heat transfer rate is determined on an electrically heated metal foil using thermal imaging. The jets are driven by a pair of adjacent jet actuators forcing air through two rectangular slot orifices of width \( D = 1.65 \text{mm} \) and aspect ratio \( \alpha = 27:1 \). The jets are maintained at a constant Reynolds number and stroke length \( (Re = 600, L_0/D = 29) \). For the parameter range considered, an optimum setting of \( H/D = 24 \) and \( S/D = 3 \) operated within a phase difference region of \( 60^\circ < \Phi < 120^\circ \) gives the best cooling performance.

1. Introduction

Recently, synthetic jets have been researched as an alternative to conventional steady jets for cooling electronics [1–3]. A key advantage lies in the nature of the synthetic jet in that it has zero net mass-flux (ZNMF) since it recycles ambient fluid. This eliminates the need for an external pressurized fluid supply, making it more spatially efficient and less expensive.

The basic concept of a synthetic jet has been described concisely as a time-averaged fluid motion generated by sufficient strong oscillatory flow at a sudden expansion [4]. The oscillatory flow is typically set up by an oscillating diaphragm which periodically forces fluid into and out of a cavity through an orifice. During each displacement of the diaphragm there is a repeated increase and decrease of pressure in the cavity. This causes cyclic ejection and suction of fluid from the cavity and hence forms a train of vortices. For a given orifice, the flow field of a free synthetic jet is governed by the Reynolds number \( Re = \rho U_0 D/\mu \) (\( D \) is the slot width) and stroke length \( L_0 \). The stroke length is defined as the distance that a slug of fluid travels away from the orifice during the ejection portion of the cycle or period [5]. Based on the average ejection velocity \( U_0 \), it is typically calculated as [4]

\[
L_0 = \int_0^{t/2} u_0(t) \, dt = \frac{2U_0}{f} \quad (1)
\]

where \( u_0(t) \) is the area-averaged orifice velocity and \( f \) is the driving frequency.
For impinging synthetic jets, the flow field is further characterized by the orifice-to-impingement distance $H$ (or non-dimensionally, $H/D$). Research suggests that an optimal $H/D$ value exists where the vortex pair impinges the surface with maximum heat transfer [6,7]. If the value of $H/D$ is too high, the flow reduces in intensity at the impingement surface since the vortices lose coherence and decay into turbulence. If it is too small, the jet is not fully developed, does not spread sufficiently and turbulent mixing is not fully established [6,7].

The distance travelled by the vortex pairs of the jet depends on the volume of fluid ejected in a cycle, i.e. the stroke length [1,7,8]. Valiorgue et al. [9] related this to the orifice-to-impingement distance remarking that there exists a critical value, $L_0/H \cong 2.5$, below which the heat transfer is significantly affected by stroke length effects and above which it is independent of stroke length. These findings have been extended by Persoons et al. [10] to cover a wider range of stroke lengths and orifice-to-surface distance. The author demonstrated that a single axisymmetric impinging synthetic jet can attain local convective heat transfer rates comparable to those of a steady impinging jet.

The flow phenomena in interaction between adjacent synthetic jets have been researched by Smith et al. and Luo et al. [11,12]. Using particle image velocimetry (PIV) to measure the velocity field near two adjacent synthetic jet orifices, both authors reported an enhancing or enlarging effect on the overall jet when the two jets are in phase (i.e. the cycles of each jet cavity are in phase). This effect is due to attraction and merging of adjacent vortex pairs. Smith et al. [11] also reported that the flow rate is twice that of a single jet since each jet entrains fluid from one side. Other enhancements include more coherency and higher velocity further downstream compared to a single jet [11].

Phase difference $\Phi$ between the jet actuators causes tilting of the overall jet field in the direction of the jet leading in phase. The vortex pairs, in this case, do not merge or coalesce but instead experience a vectoring mechanism which Luo et al. [12] call ‘attract-impact causing deflection’. The vortex pair lagging in phase is attracted towards the leading vortex pair, impacting it and subsequently deflecting it. This causes the combined jet to be tilted in the direction of the jet leading in phase.

This vectored flow has been shown by Persoons et al. [13] to enhance the cooling performance of impinging synthetic jets by inducing a cross-flow which supplies fresh cooling fluid. Combining PIV measurements (Figure 1) of the flow field and thermal images of a heated foil impinged by a pair of adjacent synthetic jets, an optimal phase difference range of $90^\circ < \Phi < 120^\circ$ was found for constant values of $Re = 600$ and $L_0/D = 29$. Larger values of $\Phi$ demonstrated over-vectoring and a resulting loss in vortex strength at impingement. The authors remarked that in this optimal range, the jet sets up a cross-flow which draws in fresh cooling fluid while maintaining good vortex strength and hence mixing, resulting in the highest convective heat transfer at impingement.

The current work aims to investigate the interaction between adjacent synthetic jets at the same test conditions ($Re = 600$ and $L_0/D = 29$) and to optimize their convective heat transfer as a function of the jet position relative to the impingement surface ($H/D$). This work also investigates the orifice centre-to-centre distance $S$ (or non-dimensionally, $S/D$) for optimal heat transfer results.

### 2. Experimental Method

#### 2.1. Impinging Synthetic Jet Test Facility

The schematic in Figure 2 shows the test rig. A pair of Visaton FR 8 speakers is used to generate the oscillatory motion that forms the synthetic jet. Each speaker is mounted to a rectangular casing which is separated by a thin (1 mm thick) rectangular sheet of aluminium. This makes up the two jet cavities (#1 & #2). The jet cavities sit on a pair of rectangular orifice plates machined from polymethyl methacrylate (PMMA) of thickness $L = 10$ mm, each with a slot of width $D = 1.65$ mm and 44.5 mm long (aspect ratio $\alpha = 27:1$). The $H/D$ and $S/D$ values were adjusted by placing PMMA spacers between the orifice plates and impingement foil.
Fluid compressibility effects can make it difficult to accurately estimate the velocity and hence Reynolds number in a synthetic jet flow, particularly when two adjacent synthetic jets are causing interfering pressure fields [14]. In this work a relationship between the jet velocity and cavity pressure derived by Persoons and O’Donovan is used as a method of estimating the jet velocity [15]. The pressure amplitude $p^*$ is measured by placing a high pressure microphone (G.R.A.S. 40BH, 0.5mV/Pa) in one of the cavities. The model then relates this to the jet velocity amplitude $U_m^*$:

$$\frac{\rho a U_m^*}{p^*} = \left(\frac{2V_c}{AL'}\right)^{1/2} \left(\left(\frac{f}{f_0}\right)^2 + \left(\frac{f}{f_0}\right)^4 + \left(\frac{KV_c}{A'L'}\frac{p^*}{\rho a^2}\right)^{1/2}\right)^{1/2}$$

(2)

where $L'$ is the effective orifice length, $a$ is the speed of sound, $f_0$ is the Helmholtz resonance frequency ($= a/(2\pi L') A'L'/V_c^{1/2}$), $V_c$ is the cavity volume and $A'$ is the orifice cross sectional area. $Re$ and $L_o$ are calculated from the time-averaged ejection velocity for sine wave excitation $U_0 = U_m^*/(\pi/2)$.

2.2. Convective Heat Transfer Coefficient

The impingement surface consists of a rectangular (0.15m x 0.145m) constantan foil (Goodfellow) which is heated up with an electrical current. The foil is sufficiently thin ($t_f = 10 \mu$m) to be considered a constant heat flux boundary. The foil is bonded with an electrically conductive adhesive to two copper bars on either side and is held taut with a spring-loaded tensioning mechanism.

The underside of the foil is sprayed with a matte black paint of high emissivity. A thermal imaging camera (FLIR Systems Thermovision A40M) reads the temperature distribution across the foil, with a spatial resolution of 0.4 mm/pixel. Thermocouples (type K) are used to record air temperatures at different points. The convective heat flux $q''$ [W/m$^2$] is determined from the electrical power to the foil $q''_{ohm}$ and corrected for heat losses due to: radiation $q''_{rad}$ from the foil surfaces (top and bottom); natural convection $q''_{cnv}$ from the bottom of the foil and; spreading of heat across the foil from lateral conduction $q''_{cnd}$. This leads to the following expression:

$$q'' = q''_{ohm} - q''_{rad} - q''_{cnv,bot} + q''_{cnd} = h(T - T_{jet})$$

(3)
$T$ is the local foil temperature measured by the thermal imaging camera and $T_{jet}$ is the air temperature inside the jet cavity. This expression is used to calculate the local heat transfer coefficient $h$.

2.2.1. Calibration and Uncertainty. Since the accuracy of $h$ is dependent on the accuracy of the temperature difference measurement ($T - T_{jet}$), both the camera and the jet thermocouple have been calibrated to the same sensor, a pre-calibrated T-type thermocouple, in order to eliminate any bias error between the temperature measurements. By applying the principle of propagation of errors, the estimated uncertainty of the heat transfer coefficient, $h$, is given by:

$$\frac{\Delta h}{h} = \sqrt{\left(\frac{\Delta q''}{q''}\right)^2 + \left(\frac{\Delta T^2 + \Delta T_{jet}^2}{(T - T_{jet})^2}\right)}$$

(4)

The relative convective heat flux uncertainty $\Delta q''/q''$ has been estimated to be around 7%. $\Delta T$ and $\Delta T_{jet}$ are the uncertainty values for the camera and thermocouple inside the jet, respectively, which were estimated to be 0.05°C and 0.1°C. The temperature difference $T - T_{jet}$ is taken as the minimum temperature difference found in the stagnation zone. This ensures a conservative estimate since further from the stagnation region, this temperature difference increases and hence the uncertainty decreases.

3. Results and Discussion

Table 1 summarizes the results. The maximum value and the spatial average of the local heat transfer coefficient, $h_{max}$ and $h_{avg}$ respectively, are determined along the centreline ($y = 0$) across the stagnation zone, as well as the corresponding Nusselt numbers ($Nu_{max}$ and $Nu_{avg}$) where $Nu = hD/k$ (where $k$ is the thermal conductivity of the air, based on the mean film temperature). These values are tabulated with the orifice-impingement distance ($H/D$) and the orifice centre-to-centre distance ($S/D$) for four incremental phase differences between the jets. Also included in Table 1 is an uncertainty estimate ($\Delta h/h$) for each measurement. These tests were performed at a constant Reynolds number and dimensionless stroke length of $Re = 600$ and $L_0/D = 29$, respectively. A constant power supply to the foil of 16W was maintained and the jets operated at a frequency of $f = 57$ Hz.

Figure 3. Heat Transfer Coefficient Distributions for two jets running in phase for $Re = 600$, $L_0/D = 29$ and $S=3D$ and (a) $H=6D$, (b) $H=12D$ and (c) $H=24D$
Table 1 - Heat transfer characteristics for a pair of interacting impinging synthetic jets at $L_0 = 29D$ and $Re = 600$

| $H/D$ | $S/D$ | $\Phi$ (°) | $h_{avg}$ (W/m$^2$K) | $h_{max}$ (W/m$^2$K) | $Nu_{avg}$ | $Nu_{max}$ | $\Delta h/h$ (%) |
|-------|-------|------------|------------------------|------------------------|------------|------------|------------------|
| 6     | 3     | 0          | 123.7                  | 264.4                  | 7.8        | 16.8       | 7.8              |
| 6     | 3     | 60         | 124.5                  | 272.5                  | 7.9        | 17.2       | 8                |
| 6     | 3     | 120        | 121.6                  | 313.8                  | 7.7        | 19.8       | 8.1              |
| 6     | 3     | 180        | 126.6                  | 328.4                  | 8          | 20.6       | 8.4              |
| 24    | 3     | 0          | 145.3                  | 209                    | 9.1        | 13.1       | 7.5              |
| 24    | 3     | 60         | 148.9                  | 234.7                  | 9.3        | 14.6       | 7.7              |
| 24    | 3     | 120        | 131.6                  | 210.8                  | 8.2        | 13.1       | 7.5              |
| 24    | 3     | 180        | 116.3                  | 200.1                  | 7.2        | 12.4       | 7.5              |
| 12    | 3     | 0          | 130.9                  | 198.2                  | 8.2        | 12.5       | 7.7              |
| 12    | 3     | 60         | 134.7                  | 233.2                  | 8.5        | 14.6       | 7.7              |
| 12    | 3     | 120        | 130.9                  | 280.7                  | 8.2        | 17.6       | 7.9              |
| 12    | 3     | 180        | 130.8                  | 266.5                  | 7.2        | 13.7       | 7.8              |
| 12    | 4.5   | 0          | 136.2                  | 218.3                  | 8.5        | 13.7       | 7.7              |
| 12    | 4.5   | 60         | 126.7                  | 234.4                  | 7.9        | 14.7       | 7.7              |
| 12    | 4.5   | 120        | 130.4                  | 263.8                  | 8.2        | 16.6       | 7.8              |
| 12    | 4.5   | 180        | 130.8                  | 266.5                  | 8.2        | 16.7       | 7.8              |
| 12    | 6.3   | 0          | 145.4                  | 270.4                  | 9.1        | 17         | 8                |
| 12    | 6.3   | 60         | 131.7                  | 248.3                  | 8.3        | 15.6       | 7.7              |
| 12    | 6.3   | 120        | 131.9                  | 261.1                  | 8.3        | 16.4       | 7.8              |
| 12    | 6.3   | 180        | 134.4                  | 262.2                  | 8.4        | 16.4       | 7.8              |
| 12    | 12.4  | 0          | 125.7                  | 232.3                  | 7.9        | 14.6       | 7.6              |
| 12    | 12.4  | 60         | 130.5                  | 231.9                  | 8.2        | 14.6       | 7.6              |
| 12    | 12.4  | 120        | 139.2                  | 255.4                  | 8.7        | 16.1       | 7.8              |
| 12    | 12.4  | 180        | 131                    | 218.5                  | 8.2        | 13.7       | 7.5              |

3.1 Effect of Jet-Surface Distance, $H/D$, on Convective Heat Transfer

By comparing the $H/D$ settings for the smallest value of $S/D = 3$, a trade-off is observed between $h_{avg}$ and $h_{max}$. For the most confined flow of $H/D = 6$, the phase-averaged $h_{avg}$ values are 12% lower than those at $H/D = 24$. Conversely, the phase-averaged $h_{max}$ values for $H/D = 6$ are 23% higher than those of $H/D = 24$. The intermediate setting of $H/D = 12$ displays intermediate values of $h_{avg}$ and $h_{max}$.

This trend is best-explained by Figure 3 which shows the distribution of $h$ across the viewing area of the thermal imaging camera. The stagnation zone has a much wider and rounder profile for $H/D = 24$ than it does for $H/D = 6$. This suggests that for $H/D = 24$, the vortex pairs have fully merged and developed into a single wider jet. At $H/D = 6$, the vortices impinge the surface individually, as demonstrated by the twin peaks (Figure 3a) in the stagnation zone. The high vortex strength at impingement at this low $H/D$ is possibly what causes such large $h_{max}$ values. This is consistent with the theory for a single synthetic jet [1,6,7], that a trade-off exists between the mixing mechanisms of a coherent vortex pair at small jet-surface distance and the spreading, vortex merging and turbulent mixing seen at large jet-surface distances. Figure 3b also illustrates a balance between $h_{avg}$ and $h_{max}$ for $H/D = 12$. This suggests that the jet is in a transitional stage where both coherent vortex mixing and turbulent mixing are taking place.
For \( H/D = 24 \) and \( S/D = 3 \), Persoons et al. [13] have demonstrated the effect of the phase difference \( \Phi \) on the flow field using PIV, which revealed vectoring of the flow in the direction of the jet leading in phase. For intermediate phase differences (typically \( 60^\circ < \Phi < 120^\circ \)), a strong impinging jet is formed while also inducing a cross-stream flow with an average velocity of about \( 0.1U_0 \). Figure 4a presents the \( h_{\text{max}} \) values as a function of \( \Phi \) showing an enhancement in the stagnation heat transfer rate as the phase difference is increased. The \( h_{\text{max}} \) value is also shown for a single jet at \( H/D = 6 \). The high heat transfer enhancement in Figure 4a for \( H/D = 24 \) at \( \Phi = 75^\circ \) (\( h_{\text{max}} = 262 \text{ W/m}^2\text{K} \)) is reminiscent of those PIV measurements [13]. Interestingly, for \( H/D = 6 \) there is a substantial enhancement for large phase differences \( (120^\circ < \Phi < 180^\circ) \), peaking at \( \Phi = 165^\circ \) to a value of \( h_{\text{max}} = 343 \text{ W/m}^2\text{K} \). It is possible that a larger phase difference is required to induce a cross flow at \( H/D = 6 \) because of the confined space. It is also possible that this cross flow is much stronger since the peak \( h_{\text{max}} \) value is 24% higher than that of \( H/D = 24 \). The results for \( H/D = 12 \) also show significant \( h_{\text{max}} \) enhancement in the region of \( (75^\circ < \Phi < 135^\circ) \) peaking at \( \Phi = 135^\circ \) to a value of \( h_{\text{max}} = 290 \text{ W/m}^2\text{K} \). Thus, the phase difference at which maximum heat transfer occurs seems to vary inversely with \( H/D \) (\( \Phi_{\text{max}} = 165^\circ, 135^\circ, 75^\circ \) respectively for \( H/D = 6, 12, 24 \)).

![Image](image.png)

**Figure 4.** Local maximum convective heat transfer \( h_{\text{max}} \) as a function of phase difference for \( Re = 600, L_0/D = 29 \) (a) \( S/D = 3, H/D = 6, 12, 24 \) and for a single jet at \( H/D = 6 \); and (b) \( H/D = 12, S/D = 3, 4.5, 6 \) and 12

### 3.2 Effect of Orifice centre-to-centre distance, S/D, on Convective Heat Transfer

Comparison of the heat transfer characteristics as a function of \( S/D \) for a fixed \( H/D = 12 \) (see Table 1), suggests little variation in the average heat transfer coefficient, with no clear indication of an optimum setting for \( S/D \). However, the \( h_{\text{max}} \) values show the largest enhancement with phase difference for the smallest spacing of \( S/D = 3 \). This is illustrated more clearly in Figure 4b which presents the \( h_{\text{max}} \) values as a function of \( \Phi \). The region of optimum phase difference difference \( (75^\circ < \Phi < 135^\circ) \) appears to be significant or pronounced for \( S/D = 3 \). It then appears to ‘flatten out’ as the spacing is increased, showing little or no variation in \( h_{\text{max}} \) with \( \Phi \). This suggests that the effect of vectoring is more dominant when the vortex pairs are ejected with less distance between them. However, for phase differences outside this optimum range, the lowest spacing displays inferior stagnation heat transfer.

The profiles of \( h \) along the x direction of the foil for different \( S/D \) settings are shown in Figure 5. It can be seen in Figure 5a from the double peaks that for jets running in phase (\( \Phi = 0^\circ \)), there is little or no merging of the adjacent vortex pairs, except for the smallest orifice centre-to-centre...
distance of $S/D = 3$ which appears to reduce stagnation zone heat transfer. At $\Phi = 75^\circ$ (Figure 5b) there is clear merging of adjacent jets except at $S/D = 12$ which is possibly too large an orifice centre-to-centre distance for effective adjacent jet interaction. The lowest orifice centre-to-centre distance of $S/D = 3$ now displays the best stagnation heat transfer suggesting that the induced-cross flow caused by jet vectoring [13] is greater, while still maintaining strong impingement.

![Figure 5](image)

**Figure 5.** Profiles of the surface heat transfer coefficient for $Re = 600$, $L_o/D = 29$ and $H/D = 12$ (a) $\Phi = 0^\circ$ and (b) $\Phi = 75^\circ$

### 3.3 Discussion of Optimal Performance

The results have shown how the impingement cooling varies with jet-surface distance $H/D$ and orifice centre-to-centre distance $S/D$ and as a function of the phase difference $\Phi$, between the actuators.

The largest $H/D$ value of 24 displays a $h_{\text{avg}}$ value 20% and 15% higher than that of $H/D = 6$ and $H/D = 12$, respectively, when operated at optimum phase difference. Therefore the largest value of $H/D = 24$ operated within a phase difference region of $60^\circ < \Phi < 120^\circ$ gives the best cooling performance. The lowest $H/D$ value of 6 displays a $h_{\text{max}}$ value 24% and 15% higher than that of $H/D = 24$ and $H/D = 12$, respectively, when operated at optimum phase difference. This suggests that for impingement cooling scenarios where the impingement area is very small, a value of $H/D = 6$ operated within a phase difference region of $150^\circ < \Phi < 180^\circ$ gives the best cooling performance.

For a $H/D$ value of 12, when operated at an optimum phase, the smallest spacing value of $S/D = 3$ gives a $h_{\text{max}}$ value 9% greater than each of the other settings ($S/D = 4.5$, 6 and 12) which appear to be enhanced very little by phase difference. The $h_{\text{avg}}$ values for each value of $S/D$ are roughly the same with $S/D = 6$ higher by around 5% but, given the uncertainty of 7 - 8% in the $h$ values, this is not significant. Therefore the smallest value of $S/D = 3$ when operated within a phase difference region of $75^\circ < \Phi < 135^\circ$ gives the best cooling performance.

### 4. Conclusion

This study has investigated the convective heat transfer performance of two impinging synthetic jets. Thermal imaging of an electrically heated foil has been performed for different orifice-to-impingement surface ($6 \leq H/D \leq 24$) and orifice centre-to-centre distances ($3 \leq S/D \leq 12$). The cooling performance has been characterized in terms of the local maximum and average heat transfer coefficient across the stagnation zone of the jet flow. The study is carried out for a constant Reynolds number and dimensionless stroke length of 600 and 29, respectively, for a pair of jets each with a rectangular orifice of aspect ratio $a = 27:1$.

Increasing the jet-orifice-to-impingement surface distance from $H/D = 6$ to $H/D = 24$ produces a wider jet with a higher average heat transfer coefficient across the stagnation zone but resulting in a lower maximum heat transfer coefficient. Increasing the phase difference between the
jets from $\Phi = 0^\circ$ to $\Phi = 180^\circ$ has an enhancing effect on the impingement cooling caused by an induced-cross flow drawing in fresh air [13], particularly at the lowest jet orifice-to-impingement surface distance tested ($H/D = 6$). Increasing the spacing between the adjacent jets reduces the adjacent jet interaction and hence cooling enhancement with phase difference. The results of this study suggest that an optimum setting of $H/D = 24$ and $S/D = 3$ operated within a phase difference region of $60^\circ < \Phi < 120^\circ$ gives the best cooling performance.

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