The effects of stroke length and Reynolds number on heat transfer to a ducted confined and semi-confined synthetic air jet

D.I.Rylatt*, T.S.O'Donovan
Heriot Watt University Edinburgh

Email: dir1@hw.ac.uk

Abstract. Heat transfer to three configurations of ducted jet and un-ducted semi-confined jets is investigated experimentally. The influence of the jet operating parameters, stroke length ($L_0/D$) and Reynolds ($Re$) number on the heat transferred to the jet is of particular interest. Heat transfer distributions to the jet are reported at $H/D = 1$ for a range of experimental parameters $Re$ (1000 to 4000) and $L_0/D$ (5 to 20). Secondary and tertiary peaks are discernable in the heat transfer distributions across the range of parameters tested. It is shown that for a fixed $Re$ varying the $L_0/D$ has little effect on the magnitude of the stagnation region heat transfer but does effect the position and magnitude of the secondary and tertiary peaks in the heat transfer distribution. It is also shown that for a fixed $L_0/D$ increasing the $Re$ has a significant effect on the magnitude of the stagnation region heat transfer but has little impact on the position of the secondary and tertiary peaks in the heat transfer distributions. Ducting is added to the configuration to improve heat transfer by drawing cold air from a remote location into the jet flow. Ducting is shown to increase stagnation region and area averaged heat transfer across the range of jet parameters tested when compared with an un-ducted jets of equal confinement. Increasing the stroke length from $L_0/D = 5$ to 20 for a Reynolds number of 2000 reduces the enhancement in stagnation region heat transfer provided by the ducting from 35% to 10%; the area averaged heat transfer provided by the ducting also changes from a 42% to a 21% enhancement. This is shown to be partly due to relative magnitude of the peaks in heat transfer outwith the stagnation region; at low stroke lengths, the difference in the magnitude of these peaks is large and reduces with increasing $L_0/D$. It is also shown that as $L_0/D$ is increased the stagnation region heat transfer to the un-ducted jets increases while for the ducted jets stagnation region heat transfer decreases with increasing $L_0/D$. Increasing Reynolds number from 1000 to 4000 for a stroke length from $L_0/D = 10$ increases the increase in stagnation region heat transfer provided by the ducting from 10% to over 50% and increases the increase in area averaged heat transfer provided by the ducting from 15% to 45%. This is shown to be primarily due to the magnitude of the stagnation region heat transfer. While the heat transfer increases with $Re$ for all configurations of jet the increase is much more significant for the ducted jets.
1. Introduction
Due to the high stagnation region and area averaged heat transfer coefficients jet impingement can achieve it has, for many years, been used in industrial applications. Synthetic air jets are a promising new technology which has been shown by both Smith and Swift [1] and Pavlova and Amitay [2] to be capable of providing more than twice the cooling of continuous air jet impingement for a similar range of Reynolds numbers.

The processing power of computers has increased greatly in recent years leading to higher demands on thermal management systems. This increase in processing power has been accompanied by a steady reduction in size and the available space in computers due to the desire for increased functionality in the same or smaller housing. Kercher et al. [4] have shown that a synthetic air jet using the same power as a standard commercial fan can provide more than twice the cooling. The current CPU Thermal Design Power (TDP) for an Intel® Core™ i7-860 is 95W and the TDP of the Pentium 3 circa 1999 was around 35W. While this shows that Gunther’s [5] prediction in 2001 of a twofold growth in the TDP every two years was not realised, it is clear that the fan-fin heat exchangers will not meet the growing cooling requirement of electronic component. New technologies, such as synthetic air jets have the potential to meet this requirement. Chaudhari et al [6] show that a synthetic jet has a maximum heat transfer performance at H/D = 10. However when the jet is confined to H/D = 3 this is reduced by 80%. Persoons et al [7] present heat transfer data for an axisymmetric jet similar to that used in this study; and report a reduction in cooling efficacy when the jet operates below its optimal spacing of 3.4 jet diameters. Smith and Glezer [8] and Smith and Swift [3] attribute the high rates of heat transfer achieved by synthetic jets to high rates of growth in both the jet column width and volume flux due to entrainment. This is limited at lower nozzle to impingement surface spacings. In addition to this, at low H/D, the fluid between the heated impingement surface and the jet orifice plate is highly confined and is re-circulated into the jet flow. The elevated temperature of this fluid above that of the ambient air reduces the temperature difference between the synthetic air jet and the impingement surface and therefore results in substantially reducing cooling performance. Several studies have investigating the effects of semi-confinement on the heat transfer to steady impinging jets [9-14]. Youn et al [13] investigated the effects of semi-confinement of a microscale steady jet for a wide range of confining plate lengths, L_op (3 to 48 jet diameters) and nozzle to impingement surface spacings, H (1 to 20 jet diameters). Overall, confinement is shown to reduce the heat transfer. At low nozzle to impingement surface spacings, even relatively minor semi-confinement (L_op/D=6) can reduce the stagnation point heat transfer by more than 50%. In general, at low H/D however, further increasing the confinement does not lead to substantial further reductions in the surface heat transfer, but it does affect the height at which the effect of the increased plate length will be overcome. Herein lies one significant difference between steady and synthetic air jet impingement heat transfer. Chaudhari et al [6] investigated the effect of increasing the semi-confinement of a synthetic jet from L_op/D = 7.86 to 13.75 and report a 50% reduction in heat transfer for the more confined jet across the full range of nozzle to impingement surface spacings tested (H/D=1 to 14). Therefore, it is clear that heat transfer to semi-confined impinging synthetic jets is significantly different from steady jets, which only display the effects of increased confinement at low spacings. The motivation for this work is to ameliorate the effects of confinement on heat transfer to a synthetic jet at low H/D by adding ducting to the orifice plate of the jet. This allows air to be drawn into the jet from some remote location. Rylatt and O’Donovan [14] have shown that ducting can significantly increase the heat transfer in both the stagnation region and on an area averaged basis. This paper focuses on the effect of the stroke length and Reynolds number of the jet on the efficacy of the ducting.

2. Experimental setup;
A schematic of the experimental setup is presented in figure 1: the rig consists of two main components: 1) the synthetic air jet assembly, and 2) a heated impingement surface. The jet is mounted directly above the heated surface. As can be seen in figure 1, the synthetic air jet consists of a cavity with an acoustic speaker operating as the jet diaphragm on one end and the jet orifice opposite it. A detailed description of the experimental rig used in this investigation is shown by Rylatt and O’Donovan [14].
Heat transfer data are presented in the form of the time-average and time-varying Nusselt number which uses the diameter of the jet to normalise the surface convective heat transfer coefficient; this is calculated using equations 1 to 3. Equation (2) is based on the assumption that heat generated within the foil is conducted in 1 dimensional through the thickness of the foil before convecting to the impinging jet. However, Patil and Narayanan [15], who also investigated the application of the thin foil infrared thermography technique for heat transfer measurements in convective flows, found that “lateral conduction heat flux rate contributed a significant portion to the net heat entering the control volume”. Therefore, in order to accurately assess the local heat transfer taking place from the foil to the impinging jet, lateral conduction must be considered.

\[
Nu = \frac{hD}{k} \quad (1)
\]

\[
h = \frac{q_{gen}^*}{(T_{surf} - T_{jet})} \quad (2)
\]

\[
q_{gen}^* = \frac{VI}{A_{surf}} \quad (3)
\]

A correction for lateral conduction as used by Golobic et al [16] was used in this study to calculate the mean and time-varying surface heat transfer coefficient. As with the rig description a more detailed explanation of this process has been presented by Rylatt and O’Donovan [14]. The Reynolds number, \( Re \) is defined using the velocity scale propose by Smith and Glezer [8] shown in equation 4:

\[
U_0 = \frac{L_0}{\tau} \quad (4)
\]

Where \( L_0 \) is the stroke length, given by equation 5:

\[
L_0 = \int_0^{t/2} u_0(t)dt \quad (5)
\]

The synthetic jet Reynolds number, \( Re \), may be calculated as in equation 6:
In this investigation the jet outlet velocity is not measured directly rather the time varying relative pressure of the cavity is measured and the jet exit velocity calculated via the method proposed by Persoons and O’Donovan [17]. By using the conservation of momentum in the orifice, the time varying jet exit velocity can be calculated as follows:

\[ m \frac{dU}{dt} + F_D(U_0) = pA \]

where \( F_D(U_0) \) represents a damping force, and \( m = \rho A L' \) is the mass of gas in the orifice. The effective length \( L \) is the sum of the geometric length \( L \) and end corrections \( L' - L = 2\beta D \), where \( \beta = 0.425 \) [18] for a sharp-edged circular orifice. The uncertainty in the Nusselt number is calculated using the method outlined by the ASME [19] and found to be less than 3%.

3. Results and Discussion
Heat transfer to a ducted and un-ducted synthetic jet is presented in this section. Also assessed is the influence of confinement on heat transferred to the jet. Various configurations of the jet were compared over a range of Reynolds numbers (1000 to 4000) and stroke lengths (5 to 20 \( L_0/D \)) at a jet impingement surface spacing of \( H/D = 1 \). The ducting added to the jet is a 1.5 mm thick steel plate measuring 20 mm by 20 mm and was attached at a distance 1 mm from the orifice plate of the jet. Three different duct configurations are studied, each with a hole in the centre of the plate which is centred on the jet orifice; the holes were 6, 8 and 10 mm in diameter or 1.2, 1.6 and 2 times the jet diameter.

To hold the Reynolds number constant and vary the stroke length the amplitude and frequency of the excitation signal to the jet diaphragm is varied. Figure 2 shows the excitation frequency variation with Reynolds number at \( H/D = 1 \). The constant Reynolds number jets used in this investigation has a Reynolds number \( Re = 2000 \). With increasing stroke length the excitation frequency of the jet is reduced which increases the period of the jets expulsion and suction strokes, generating a jet of constant velocity that expels a larger volume of air in a single cycle. The stroke lengths tested in this investigation were 5, 10, 15 and 20 which had corresponding excitation frequencies of 129, 65, 43 and 33 at \( H/D = 1 \).

In order to hold the stroke length constant and vary the Reynolds number, the amplitude and frequency of the excitation signal to the jet diaphragm is varied. Figure 3 shows that the excitation frequency varies linearly with Reynolds number at \( H/D = 1 \). The constant stroke length jets used in this investigation have a stroke length \( L_0/D = 10 \). As the stroke length is held constant the Reynolds number is varied by changing the frequency which reduces the time period in which the same volume of fluid passes through the jet orifice on both the expulsion and suction stroke of the jet cycle. This has
the effect of increasing the fluid velocity and therefore the Reynolds number. The Reynolds numbers
tested in this investigation were 1000, 2000, 3000 and 4000 which had corresponding excitation
frequencies of 33, 43, 65 and 129 at $H/D=1$.

3.1 Constant Reynolds number

Figure 4 shows the time averaged heat transfer to the three configurations of ducted jet and the jet
confined by $L_{op}/D = 18$ and 40. At $L_0/D = 5$, $Re = 2000$ (figure 4(a)) it can be seen that heat transfer to the
more confined jet ($L_{op}/D = 40$) is lower than that of the less confined jet ($L_{op}/D = 18$). This is most
noticeable in the stagnation region and the region where peaks in the heat transfer distributions occur,
$r/D \approx 3$. It can also be seen that heat transfer to all there configurations of ducted jet is higher in
magnitude than that of the two jets confined by $L_{op}/D = 18$ and 40. It can also be seen that the peaks in
heat transfer that occur outwith the stagnation region for all configurations of jet occur are closer to
the stagnation region when ducting has been added to the jet; the peaks occur at $r/D \approx 2$ for the ducted
jets and at $r/D \approx 3$ for the un-ducted jets. It is apparent that as $L_0/D$ is increased the difference in
magnitude of the peaks outwith the stagnation region between the ducted and un-ducted jets reduces.
For the jet setting of $L_0/D = 10$, $Re = 2000$ (figure 4(b)) it is again noticeable that heat transfer to the
more confined jet ($L_{op}/D = 40$) is lower than that of the less confined jet ($L_{op}/D = 18$). This is most
noticeable in the stagnation region and the region where peaks in the heat transfer distributions occur,
$r/D \approx 3$. Again it is observed that heat transfer to all three configurations of ducted jet is higher in
magnitude than that observed for the confined jets. When comparing figures 4(a) and 4(b) it can be seen
that the increase in heat transfer provided by ducting the jet when compared with the jets semi-
confined to $L_{op}/D = 18$ and 40 is lower for the higher stroke length jet.

Comparing the performance of the jet confined by a plate, $L_{op}/D = 18$ to that confined by $L_{op}/D = 40$
across the range of stroke lengths tested, it is seen that heat transfer to the jet begins to converge both
in the stagnation region and in the areas where peaks occur outwith the stagnation region, with
increasing stroke length. Comparing the performance of the three different configurations of ducted jet
across the range of stroke lengths tested it can be seen that the increase in stagnation heat transfer
offered by the ducting reduces with increasing stroke length. It can also be seen that the level of heat
transfer provided by the three different configurations of ducted jet varies little across the range of
stroke lengths tested. It is observed that at $L_0/D =5$ (figure 4(a)) the peaks in heat transfer outwith the
stagnation region occur in close proximity but vary in magnitude for the different ducts significantly;
as $L_0/D$ is increased, the peaks become similar in magnitude. It can be seen that the peaks in heat
transfer outwith the stagnation region for the un-ducted jets very little in position across the range of
stroke lengths tested ($r/D \approx 3$ to 3.5) whereas the ducted jets vary more ($r/D \approx 2$ to 3).

Heat transfer to the jet in two regions is quantitatively accessed; these regions are 1) the stagnation
region which extended over an area $1D$ in diameter centred around the stagnation point, to account for
local minima at the stagnation point and 2) the area averaged heat transfer which is accessed over a
circular area of diameter $14D$. Stagnation region heat transfer is an indication of the maximum heat
transfer rate the jet can provide and is used in many investigations [7, 13, 20-22] to assess a jets
performance in both steady and synthetic jet studies. Area averaged heat transfer gives an indication of
the practical heat transfer rate available to the jet in real world applications; this area is selected
because it is comparable with the size of commonly used CPU heat exchangers [23].

Figures 5(a) and 5(b) show how heat transfer to the various jet configurations varies with $L_0/D$ at $Re = 2000$
in the stagnation region and on an area averaged basis respectively. By comparing the
performance of the un-ducted jet with the ducted jet in the stagnation region (figure 5(a)), it is noticeable that $Nu_{stag}$
for the ducted jets decrease with increasing stroke length while the opposite is
true of the un-ducted jets.

For the two differing levels of confinement it can be seen that the stagnation region heat transfer to the
less confined jet is higher in magnitude across the range of stroke lengths tested, but that as $L_0/D$ is
increased the difference in their performance reduces. It is clear that the two larger diameter ducts
perform best across the range of stroke lengths tested, it is also clear that the $1.6D$ ducting performs best at $L_0/D = 5$ and 10 while the $2D$ ducting performs best at $L_0/D = 15$ and 20.
4: Heat transfer distributions for ducted and un-ducted jets for fixed Reynolds number of \( Re = 2000 \), \( H/D = 1 \),
\( (L_0/D = 5, f_e = 129 \text{ Hz}), (L_0/D = 10, f_e = 65 \text{ Hz}), (L_0/D = 15, f_e = 43 \text{ Hz}), (L_0/D = 20, f_e = 33 \text{ Hz}) \).

5: stagnation region, area averaged heat transfer and percentage increase in heat transfer for ducted and un-ducted jets
\( H/D = 1, Re = 2000 \) and \( (L_0/D = 5, f_e = 129 \text{ Hz}), (L_0/D = 10, f_e = 65 \text{ Hz}), (L_0/D = 15, f_e = 43 \text{ Hz}), (L_0/D = 20, f_e = 33 \text{ Hz}) \).
Figure 5(b) shows the percentage increase in heat transfer when compared to the more confined jet of $L_0/D = 40 $; this level of confinement has been selected as it is the same level of confinement provided by the duct. As can be seen in Figures 5(c) the 1.6 $D$ ducting is most effecting at $L_0/D = 5 $, providing a 34% increase in heat transfer; the efficacy of the ducting reduces with increasing $L_0/D$ and at $L_0/D = 20 $ the 2$ D$ ducting is only increasing the stagnation region heat transfer by 10%. Figure 5(b) shows the area average heat transfer provide by all jet configurations across the range of stroke lengths tested. Again it is observed that the jet semi-confined to $L_0/D = 18 $ delivers higher rates of area averaged heat transfer across the range of stroke lengths tested than the more confined jet of $L_0/D = 40 $. It is apparent that the difference in magnitude of the area averaged heat transfer delivered for the two different levels of confinement is less than is observed for stagnation region heat transfer and by $L_0/D = 20 $ both levels of confinement deliver similar heat transfer rates. This is due the secondary peaks been similar in magnitude for both levels of confinement across the range of stroke lengths tested with the difference in the magnitude of the peaks reducing with increasing stroke length. It is noticeable that unlike the stagnation region, the area averaged heat transfer to the ducted jets increases with stroke length. This is due the position of the peaks in heat transfer outwith the stagnation region. As $L_0/D$ increases these peaks reduce in magnitude but increase in radial position from the stagnation point and therefore act over a larger surface area and increase the area averaged heat transfer. It is clear that the 1.6 and 2$ D$ ducting provide the greatest increase in area averaged heat transfer across the range of stroke lengths tested with the 2$ D$ ducting performing best at all $L_0/D$ except $L_0/D = 10 $. Figure 5(b) shows the percentage increase in area averaged heat transfer achieved by adding the ducting to the jet when compared with the un-ducted jet confined to $L_0/D = 40 $. It is noticeable that the efficacy of the ducting varies differently on an area averaged basis when compared with the stagnation region. The effectiveness of the ducting reduces almost linearly with increasing $L_0/D$ in the stagnation region whereas on an area averaged basis the efficacy of the ducting reduces significantly from $L_0/D = 5$ to 10 but then remains relatively constant for the rest of the stroke lengths tested. This is due to the peaks outwith the stagnation region acting over a larger area with increasing $L_0/D$; this would seem to not only compensate for their reduction in magnitude but also the reduction in the efficacy of the ducting in the stagnation region with increasing stroke length. It is clear that the 2$ D$ ducting provides the greatest increase in heat transfer cross the range of stroke lengths tested except at $L_0/D = 10 $ where the 1.6 $D$ ducting performs better. The 2$ D$ ducting increase area averaged heat transfer by 41% at $L_0/D = 5 $ and delivers 215 at $L_0/D = 20 $.  

**Constant stroke length**

Figure 6 shows the time averaged heat transfer to the three configurations of ducted jet and the jet semi-confined to $L_0/D = 18 $ and 40 for a range of Reynolds when the stroke length is held constant at $L_0/D = 10 $. Comparing the jets with differing levels of semi-confinement across the range of Reynolds numbers tested it can be seen that the less confined jet ($L_0/D = 18 $) delivers higher rates of stagnation region heat transfer but that the peaks in heat transfer outwith the stagnation region are of similar magnitude for both levels of confinement and occur at the same radial location at each Reynolds number. As with the constant Reynolds number jets radial location of the peaks in the heat transfer outwith the stagnation region for the undusted jets vary little across the range of Reynolds numbers tested ($r/D \approx 3 $ to 3.8). It is noticeable that the peaks in heat transfer outwith the stagnation region for the ducted jets vary little across the range of stroke lengths tested in figure 6 occurring at $r/D \approx 2.5 $, comparing this with the constant Reynolds number jets shown in figure 4, it is clear from this that it is the variation in stroke length which influences the location of these peaks. It is observed that as the Reynolds number is increased the difference in the heat transfer delivered by the three ducted configurations of jets becomes more distinct. In figure 4 it was observed that the difference in heat transfer delivered by the three different configurations of ducting reduces with increasing stroke length. It is also noticeable that magnitude of the heat transfer varies more when the Reynolds number if varied for a fixed stroke length than when the stroke length is varied for a fixed Reynolds number. From this we can see that the stroke length of a synthetic jet influences the magnitude and location of peaks in the heat transfer outwith the stagnation region whereas the Reynolds number of a synthetic jet influences the magnitude of the heat transfer to the jet.
6: heat transfer distributions for ducted and un-ducted jets for fixed stroke length of $L_0/D = 10$, $H/D = 1$ ($Re = 1000$, $f_e = 33$ Hz), ($Re = 2000$, $f_e = 65$ Hz), ($Re = 3000$, $f_e = 96$ Hz), and ($Re = 4000$, $f_e = 129$ Hz),

7: stagnation region, area averaged heat transfer and percentage increase in heat transfer for ducted and un-ducted jets $H/D = 1$, $L_0/D = 10$, ($Re = 1000$, $f_e = 33$ Hz), ($Re = 2000$, $f_e = 65$ Hz), ($Re = 3000$, $f_e = 96$ Hz), and ($Re = 4000$, $f_e = 129$ Hz),
Figures 7(a) and 7(b) show how heat transfer to the various jet configurations varies with $L_0/D$ at $Re = 2000$ in the stagnation region and on an area averaged basis respectively. By comparing the performance of the un-ducted jet with the ducted jet in the stagnation region (figure 7(a)), it is noticeable that $Nu_{stg}$ for the ducted jets increase with increasing Reynolds number.

For the two differing levels of confinement it can be seen that the stagnation region heat transfer to the less confined jet is higher in magnitude across the range of stroke lengths tested, but that as the Reynolds number is increased the difference in their performance increases showing that the effects of semi-confinement is more significant with increasing Reynolds number. It is clear that the two larger diameter ducts perform best across the range of stroke lengths tested, the $1.6D$ and $2D$ ducting performing identically at $Re = 1000$ and $2000$ with the $2D$ ducting performs better at $Re = 3000$ and $4000$. Figure 7(c) shows the percentage increase in heat transfer when compared to the more confined jet of $L_0/D = 40$; this level of confinement has been selected as it is the same level of confinement provided by the duct. As can be seen in Figures 7(c) the $1.6D$ and $2D$ ducting is most effecting at $Re = 1000$ providing a $10\%$ increase in heat transfer; the efficacy of the ducting increases with increasing $Re$ and at $Re = 4000$ the $2D$ ducting increases the stagnation region heat transfer by $53\%$.

Figure 7(b) shows the area average heat transfer provide by all jet configurations across the range of stroke lengths tested. Again it is observed that the jet semi-confined to $L_0/D = 18$ delivers higher rates of area averaged heat transfer across the range of stroke lengths tested than the more confined jet of $L_0/D = 40$. It is apparent that the difference in magnitude of the area averaged heat transfer delivered for the two different levels of confinement is less than is observed for stagnation region heat transfer. This is due the secondary peaks been similar in magnitude for both levels of confinement across the range of stroke lengths tested with the difference in the magnitude of the peaks reducing with increasing stroke length. It is clear that the $1.6$ and $2D$ ducting provide the greatest increase in area averaged heat transfer across the range of Reynolds numbers tested with the $1.6D$ ducting performing best at all $Re = 1000$ and $2000$ and the $2D$ ducting performing best $Re = 3000$ and $4000$. It is Figure 7(d) shows the percentage increase in area averaged heat transfer achieved by adding the ducting to the jet when compared with the un-ducted jet confined to $L_0/D = 40$. It is shown that $1.6D$ ducting performs best at $Re = 1000$ and $2000$ offering $15$ and $24\%$ increase in area averaged heat transfer respectively. The $2D$ ducting performs best at $Re = 1000$ and $2000$ increasing area averaged heat transfer by $35\%$ and $45\%$ respectively.

4. Conclusion

Increasing confinement has been shown to reduce heat transfer to a synthetic air jet. The reduction in heat transfer caused by increased semi confinement is most significant in the stagnation region, and this is most pronounces for low stroke lengths for a fixed Reynolds number and high Reynolds numbers for a fixed stroke length.

Ducting is shown to increase stagnation region and area averaged heat transfer across the range of jet parameters tested when compared with an un-ducted jet of equal confinement. Increasing the stroke length from $L_0/D = 5$ to $20$ for a Reynolds number of $2000$ reduces the enhancement in stagnation region heat transfer provided by the ducting from $35\%$ to $10\%$ and reduces the enhancement in area averaged heat transfer provided by the ducting from $42\%$ to $21\%$. This is shown to be partly due to relative magnitude of the peaks in heat transfer outwith the stagnation region; at low stroke lengths the difference in the magnitude of these peaks is large and reduces with increasing $L_0/D$. It is also shown that as $L_0/D$ is increased, the stagnation region heat transfer to the un-ducted jets increases while for the ducted jets the stagnation region heat transfer decreases with increasing $L_0/D$. Increasing Reynolds number from $1000$ to $4000$ for a stroke length from $L_0/D = 10$ increases the enhancement in stagnation region heat transfer provided by the ducting from $10\%$ to over $50\%$ and increases the enhancement in area averaged heat transfer provided by the ducting from $15\%$ to $45\%$. This is shown to be primarily due to the magnitude of the stagnation region heat transfer, while it increases with $Re$ for all configurations of jet the increase is much more significant for the ducted jets.
5. References

[1] Smith, B.L. and G.W. Swift, Synthetic jets at large Reynolds number and comparison to continuous jets, in 31st AIAA Fluid Dynamics Conference and Exhibit. 2001. Anaheim.

[2] Pavlova, A. and M. Amitay, Electronic cooling using synthetic jet impingement. ASME Journal of Heat Transfer, 2006. 128: p. 897 - 907.

[3] Smith, B.L. and G.W. Swift, A comparison between synthetic jets and continuous jets. Experiments in Fluids, 2003. 34: p. 467 - 472.

[4] Kercher, D.S., et al., Microjet cooling devices for thermal management of electronics. IEEE Transactions on Components and Packaging Technologies, 2003. 26: p. 359 - 366.

[5] S. Gunther, F.B., D. M. Carmean, J. C. Hall, Managing the Impact of Increasing Microprocessor Power Consumption Intel Technology Journal, 2001. Vol. 5, (No. 1).

[6] Chaudhari, M., B. Puranik, and A. Agrawal, Heat transfer characteristics of synthetic jet impingement cooling. International Journal of Heat and Mass Transfer, 2010. 53(5–6): p. 1057-1069.

[7] Persoons, T., A. McGuinn, and D.B. Murray, A general correlation for the stagnation point Nusselt number of an axisymmetric impinging synthetic jet. International Journal of Heat and Mass Transfer, 2011. 54(17–18): p. 3900-3908.

[8] Smith, B.L. and A. Glezer, The formation and evolution of synthetic jets. Physics of Fluids, 1998. 10: p. 2281 - 2297.

[9] Ashforth-Frost, S. and K. Jambunathan, Effect of nozzle geometry and semi-confinement on the potential core of a turbulent axisymmetric free jet. International Communications in Heat and Mass Transfer, 1996. 23: p. 155 - 162.

[10] Garimella, S.V. and R.A. Rice, Confined and submerged liquid jet impingement heat transfer. ASME Journal of Heat Transfer, 1995. 117: p. 871 - 877.

[11] Colucci, D.W. and R. Viskanta, Effect of nozzle geometry on local convective heat transfer to a confined impinging air jet. Experimental Thermal and Fluid Science, 1996. 13: p. 71 - 80.

[12] Gao, N. and D. Ewing, Investigation of the effect of confinement on the heat transfer to round impinging jets exiting a long pipe. International Journal of Heat and Fluid Flow, 2006. 27(1): p. 33-41.

[13] Youn, Y.J., K. Choo, and S.J. Kim, Effect of confinement on heat transfer characteristics of a microscale impinging jet. International Journal of Heat and Mass Transfer, 2011. 54(1-3): p. 366-373.

[14] Rylatt, D.I. and T.S. O'Donovan, Heat transfer enhancement to a confined impinging synthetic air jet. Applied Thermal Engineering, 2013. 51(1–2): p. 468-475.

[15] Patil, V.A. and V. Narayanan, Application of heated-thin-foil thermography technique to external convective microscale flows. Measurement Science and Technology, 2005. 16: p. 472-476.

[16] Golobic, I., et al., Experimental determination of transient wall temperature distributions close to growing vapor bubbles. Heat and Mass Transfer, 2007.

[17] Persoons, T. and T.S. O'Donovan, A pressure-based estimate of synthetic jet velocity. Physics of Fluids, 2007. 19.

[18] Beranek, L.L., Acoustics. Physics Today, 1955. 8(10): p. 27-28.

[19] Policy on reporting uncertainties in experimental measurements and results. ASME Journal of Heat Transfer Policy.

[20] Hoogendoorn, C.J., The effect of turbulence on heat transfer at a stagnation point. International Journal of Heat and Mass Transfer, 1977. 20: p. 1333 - 1338.

[21] Shadlesky, P.S., Stagnation point heat transfer for jet impingement to a plane surface. AIAA Journal, 1983. 21: p. 1214 - 1215.

[22] Choo, K.S., et al., Heat transfer characteristics of a micro-scale impinging slot jet. International Journal of Heat and Mass Transfer, 2009. 52(13-14): p. 3169-3175.

[23] Saini, M. and R.L. Webb, Heat rejection limits of air cooled plane fin heat sinks for computer cooling. IEEE Transactions on Components and Packaging Technologies, 2003. 26(1): p. 71-79.