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An experimental study of a multi-orifice synthetic jet with application to cooling of compact devices

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ABSTRACT
In the present work, the flow characteristics of free and impinging synthetic jets formed from a single cavity but with a different number of orifices are investigated using the particle image velocimetry technique. A synthetic jet is a zero-net mass-flux device that formed by periodic ejection and suction of fluid through a small opening. The flow characteristics are used to explain the heat transfer results in the literature. The number of satellite orifices, distance between orifices, and location of the surface from orifices are varied to observe the phenomenon of jet interaction and its influence on flow recirculation. The results show that interaction between adjacent jets causes the diversified flow characteristics of the multiorifice jet with respect to a single orifice jet. The multiorifice jet exhibits significantly higher entrainment and mixing compared to its equivalent diameter of a single orifice jet. Moreover, results from the impinging jet corroborate that the impingement of vortex pairs, their breakdown, and formation of a strong wall jet are responsible for a high heat transfer coefficient for the case of larger surface spacing ($z/d ≥ 8$; normalized distance between orifice and surface). However, for impingement at small surface spacing ($z/d ≤ 2$), change in recirculation behavior with different orifice configurations can lead to a variation in the heat transfer coefficient. For the large and medium surface spacing ($2 < z/d < 8$), the center orifice gets sufficient time to develop, which draws the satellite and impingement jets as a single orifice jet. In the case of large surface spacing, the wall jet vortices are not attracted during the suction stroke. While for medium surface spacing, wall jet vortices are closer to the orifice and easily entrained during the suction stroke and form a recirculating region. In the case of small surface spacing, the center jet does not get sufficient time, and hence both center and satellite jets impinge separately. It was found that the center jet having higher ejection velocity impinges earlier than the satellite jet and the wall jet vortex formed from the center orifice drives along a wall jet. Therefore, recirculation is absent for smaller surface spacing, and fresh fluid is sucked and impinges on the surface in each cycle. This study expands the current knowledge of multiorifice free and impinging jets and establishes a relationship between heat transfer and fluid flow, which eventually facilitates an efficient and effective heat transfer system for compact devices.

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I. INTRODUCTION
Air cooling is preferred in many heat transfer applications due to its availability and low cost. Generally, air is flowed over different fin geometries by using a fan to achieve high heat transfer. However, this approach can make the system bulky, which is undesirable. A synthetic jet is compact and has huge potential for localized cooling applications. It can be created by the oscillation of a speaker diaphragm inside a cavity, which gives rise to periodic suction and blowing, causing the formation of vortices at the exit of the orifice. The synthetic jet is characterized by the formation of a pair of counter-rotating vortices during the ejection cycle, which travels in the axial direction, resulting in mass entrainment and mixing. Due to its wide applications, the fluid flow and heat transfer characteristics of a single orifice synthetic jet have received considerable attention both experimentally and numerically. The studies based on a single orifice synthetic jet have focused on the effect of several operational [effect of surface spacing, Reynolds number (Re), and frequency of excitation (f)] and geometrical parameters (orifice shape/thickness and cavity...
geometry) on the heat transfer rate.\textsuperscript{9,12,17,29} The acoustic aspect of the synthetic jet has been reported by Bhapkar et al.,\textsuperscript{4} who found an increase in the noise level with an increase in the orifice diameter. The orifice diameter is a highly influential parameter compared to other geometrical parameters and decides the turbulence generation properties of the synthetic jet in its near field and eventually controls the mixing and heat transfer rates.\textsuperscript{9}

The earlier investigations have also compared the cooling performance of synthetic jet with a continuous (steady) jet, considering similar development in the downstream.\textsuperscript{10} Pavlova and Amitay\textsuperscript{26} found that the synthetic jet is more efficient than a steady jet and produces three times higher heat transfer rate for the same average Reynolds number. Their particle image velocimetry (PIV) measurements showed that enhancement in the turbulence level is due to the impingement of large-scale coherent structures on the surface, which is responsible for a high heat transfer rate. Gillespie et al.\textsuperscript{11} also observed that the synthetic jet provides a higher heat transfer rate than a steady jet at an equivalent Reynolds number. However, in both these studies, the maximum Reynolds number was less than 740 and the flow was in the laminar regime. Therefore, lower performance of the continuous jet is expected owing to the absence of large scale mixing. Chaudhari et al.\textsuperscript{2} compared the heat transfer results of steady and synthetic jets at a comparatively large Reynolds number (Re $\approx$ 4000). They reported that for small surface spacing ($z/d < 4$), the steady jet provided a higher heat transfer rate than the synthetic jet and had a comparable value for larger surface spacing ($z/d \geq 5$). However, in the near jet field, the synthetic jet provided greater entrainment and turbulence level than its counterpart steady jet.\textsuperscript{3,28} Accordingly, a higher heat transfer rate is expected with the synthetic jet as compared to the steady jet. It was noted that for small surface spacing, the flow recirculation of the same fluid and ingested heated air is responsible for the diminished heat transfer rate.\textsuperscript{3,10} Furthermore, Ghaffari et al.\textsuperscript{10} observed in their PIV study that for small surface spacing, the impinging vortices are close enough to the orifice. Hence, they recirculate the hot air, reducing the temperature difference between surface and fluid, thereby reducing the heat transfer. For small surface spacing, the effect of confinement level and stroke length of the synthetic jet on the heat transfer rate has been investigated by Pavlova and Amitay,\textsuperscript{26} Valiorgue et al. (2009),\textsuperscript{30} and Greco et al.\textsuperscript{15} These studies reported that the heat transfer rate is very sensitive to the level of confinement and stroke length.

The above studies suggest that the location of the surface from the synthetic jet orifice plays a significant role in determining the variation in heat transfer and found that the performance of the synthetic jet degrades at smaller surface spacing due to suction and ejection of the same fluid. However, it is paramount to locate the synthetic jet at smaller surface spacing for compact electronic packaging. In this direction, Bhapkar et al.\textsuperscript{4} studied the effect of different cavity shapes on the ensuing heat transfer rate. They reported that the selection of proper orifice cavity shape mitigates the effect of confinement at a small surface spacing ($z/d < 2$). They suggested that the shape of the cavity regulates the release of trapped hot air, thereby influencing the heat transfer rate. Further advances in synthetic jet technology introduced the idea of the multiple orifice synthetic jet, which proved to have superior performance to that of the single orifice synthetic jet particularly for small surface spacing, where the proper orifice configuration mitigates the effect of confinement.\textsuperscript{4,12,15,25}

Chaudhari et al.\textsuperscript{7} found that a multiorifice synthetic jet exhibits a 30% higher heat transfer rate compared to a single orifice synthetic jet for the same input power. Furthermore, they reported that the orifice configuration plays a critical role in the alteration of heat transfer for small surface spacing ($z/d \leq 2$) and may be highly useful for small-scale electronics. Mangate et al.\textsuperscript{17} found that a multiorifice jet with optimal orifice shape can even provide a 75% higher heat transfer rate than a single orifice synthetic jet. Tan and Zhang\textsuperscript{7} measured the heat transfer characteristics of three different configurations of the multiorifice jet. They observed that the distance between orifices significantly affects the heat transfer rate. The coefficient of heat transfer was higher than a single hole orifice only for moderate spacing (pitch to diameter ratio, s/d = 2) between the holes. Greco et al.\textsuperscript{31,32} visualized the flow characteristics of a twin synthetic jet placed close to each other and separated by 1.1d. They observed that twin jets exhibit a strong interaction between the two jets and a higher turbulence level, and hence, an enhancement in the heat transfer rate. Watson et al.\textsuperscript{33} visualized the flow of interacting multiple synthetic jets placed at different distances. It was found that when the distance between jets is sufficiently large, they do not interact with each other and flow separately. However, when the distance is reduced, adjacent vortices of the jets interact with each other resulting in the cancelation of vorticity. When the distance is further reduced, the jets interact in the near field itself to form a single jet. They also noticed the appearance of bigger vortex rings with multiple jets compared to a single jet.

The above studies suggested that the multiorifice synthetic jet offers several advantages over a single orifice synthetic jet considering its higher heat transfer rate at small surface spacing. The flow characteristics of a multiorifice synthetic jet created from the single cavity but with different orifices have not been investigated in detail. There are few studies based on a multiple orifice configuration, which have investigated the flow mechanics of synthetic jets. However, in a multiple orifice jet, the flow characteristics of each jet is independent of each other. While in the multiorifice jet, the flow characteristics of satellite and center orifices are highly interdependent. Therefore, it is interesting to examine the flow physics of multiorifice jets. A detailed understanding of the fluid mechanics of a multiorifice synthetic jet will allow the optimization of the operating condition and geometrical parameters for making an efficient and effective heat transfer system. In this context, the present work investigates the flow behavior of free and impinging multiorifice synthetic jets with the help of the particle image velocimetry (PIV) technique.

II. EXPERIMENTAL SETUP AND DATA PROCESSING

This section provides the details of the particle image velocimetry measurement setup, along with relevant details for synthetic jet assembly, definition of parameters, data processing, and uncertainty in the measurements.

A. Measurement details

The PIV measurement was carried out inside a transparent acrylic chamber (size $50 \times 50 \times 50$ cm$^3$) filled with seeding particles (oil particles; diameter $= 2–4$ μm$^3$). The experimental setup used in the present study is shown in Fig. 1. The air jet is produced
using an acoustic speaker assembly [Fig. 1(b)] mounted on top of the chamber. The oscillations of the speaker diaphragm are controlled by a function generator,\textsuperscript{2,5} which provides a sinusoidal power input of prescribed frequency to the speaker. The amplitude of the diaphragm 4.5 V ($V_{\text{rms}}$) was kept constant during the measurement. The frequency of oscillations was maintained at 125 Hz, which is the resonance frequency of the speaker to provide the best performance. The resonance frequency was determined through hot-wire anemometry (HWA) (TSI, IFA300) based measurements. For this, a tungsten platinum-coated single-wire probe (Model 1210-T1.5; diameter of wire 3.8 μm, length of the sensing element 1.27 mm, and coefficient of resistance of 0.0042/C) was employed. Other details about velocity measurement using hot-wire are provided elsewhere.\textsuperscript{5,2}

Various configurations of orifice plates such as $5 \times 3 \times 2$, $5 \times 3 \times 4$, and $5 \times 3 \times 8$ are used for the formation of the multi-orifice jet. In configuration $5 \times 3 \times 2$ [Fig. 1(c)], 5 mm and 3 mm represent the diameter of the center and surrounding satellite orifices, respectively, while 2 is the number of satellite orifices. In addition, 5 and 8 mm single orifices in the synthetic jet are employed as baseline cases. The idea behind these choices was to have either the same size of the center orifice or approximately the same open area.

The seeding particles are produced by a particle generator shown in Fig. 2. Pressurized air (overpressure up to 1.5 bars concerning output pressure) is supplied through two/four tubes (ID = 5 mm), depending upon the seeding requirement. The seeding particle density is controlled by adding additional air into the container through an opening provided at the top. Additional details of the oil particle generator fabricated for seeding of air are available in Ref. 18.

The particles in the measurement tank are illuminated by twin pulsed Nd:YAG lasers, while a CCD camera is placed perpendicular to the plane of the laser sheet.\textsuperscript{27} The CCD camera (1024 × 1392 pixels 12-bit CCD; PCO Pixi, Germany) receives the scattered light; thereby, the images of particles are captured. The wavelength of the laser beam is 532 nm, and the duration of the laser pulses is 7.7 ns and 6.8 ns.\textsuperscript{27} The laser and CCD are synchronized with a synchronizer that provides appropriate timing for triggering them. The synchronizer unit controls the parameters such as pulse frequency and the time delay between the laser pulses. The experiments are carried out in a dark room to minimize the effect of ambient light.

The local velocity is determined within the imaging region by calculating the particle displacement in an interrogation spot over the time delay period. The time interval between a pair of laser pulses is set at 0.09 ms according to the flow velocity. The total 360 pairs of images are recorded at a sampling rate of 5 Hz in the present study. The recorded images are analyzed by two-step cross-correlation analysis, starting from the first step with an interrogation window size of $64 \times 64$ pixel$^2$ to the second interrogation size of $32 \times 32$ pixel$^2$. A standard 50% horizontal and vertical overlap is employed between successive interrogation regions. Less than 5% of the total vectors were detected as incorrect; these bad vectors were corrected by using a bilinear interpolation scheme.
The magnification of the recorded images is 0.079 mm/pixel, which yields a vector spacing of 1.26 mm.

**B. Data processing**

The velocity measurement method using HWA is adopted from Refs. 2 and 5. The centerline velocity ($U_0$) at the nozzle exit is found to be 8.4 m/s, and the corresponding Reynolds number ($Re = \frac{U_0 d}{\nu}$) is 2700 (here $d$ is the orifice diameter and $\nu$ is the kinematic viscosity of air). The other governing parameters of the synthetic jet are the following: stroke length, $L_0 = \frac{U_0 f}{f} = 0.07$ m (where $f$ is the excitation frequency, 125 Hz), Strouhal number ($Sr = \frac{f d}{U_0} = \frac{d}{L_0} = 0.075$), and Stokes number ($S = \frac{d}{\sqrt{2 \pi f \nu}} = 35$).

As discussed in Ref. 34, the uncertainty in the velocity measurement is determined by considering four main factors identified by Wang et al.19 and Raffel et al.22 as follows: uncertainty related to the equipment, particle motion, sampling error, and data processing. The individual uncertainty determined from each factor is combined through the uncertainty propagation method. The combined experimental uncertainty in the measurement of mean velocity is about ±5.6%.

**III. RESULTS AND DISCUSSION**

The results obtained from our experimental study are delineated into different sections. Section III A discusses the interaction between jets. Sections III B and III C provide detailed information on the fundamental aspects of the multiorifice impinging jet with emphasis on the influence of the satellite orifice configuration on the jet flow behavior. Following these results, Sec. IV presents a discussion on recirculation and availability of fresh air for different configurations. In Sec. V, we relate the present flow results to the heat transfer behavior reported in the literature.

**A. Flow dynamics of a free jet**

In this section, the nature of interaction between free jets for a multiorifice configuration is studied; these jets emerge out from the same cavity and are separated by different distances. Figure 3 shows the mean velocity vector field superimposed with vorticity contour for three different configurations. The vorticity (w) is normalized with the orifice diameter ($d = 5$ mm) and their respective maximum centerline velocity ($V_m$). In this figure, five out of six streamwise vectors have been skipped at regular intervals for clarity. For the first configuration [Fig. 3(a)], jets come out through two orifices each of diameter 5 mm and the distance between them is 20 mm. In this case, the jets originate as single orifice jets; their interaction can be seen through merging of their vorticity contours. This figure also shows that in the near jet region (before the jet interaction), the maximum axial velocity occurs at the geometrical center of each orifice. Further downstream, the outer shear layer decelerates due to the quiescent surrounding. While the fluid lying between the inner shear layers accelerates due to the development of counter-rotating vortex pairs between the orifices. Thus, the gradual development of both inner and outer shear layers occurs, and at certain downstream distance ($z/d < 12$), velocity in the inner region between the orifices becomes maximum. Thereafter, multiple nature of the jet is transformed into a single combined jet. A self-similar behavior is observed as the free jet develops further downstream.23,36

The case when the distance between the orifices reduces to 12 mm is shown in Fig. 3(b). Here, the inner shear layers interact without traveling much distance downstream, and multiple jets appear as a single jet emerging out of the orifices. The axial velocity becomes maximum at a location between the two orifices before the axial distance of $z/d < 4$ [Fig. 3(b)]. Watson et al.11 in their smoke wire visualization based study also observed a similar jet interaction phenomenon for multiple orifice jets. The streamlines shown in
this figure reveal the mutual suction phenomenon of adjoining jets. The upward direction of streamlines for z/d < 5 exhibits the effect of suction stroke experienced by the flow. For z/d > 5, the downward inclination of streamlines shows that ambient fluid is entrained by the free jet.

The jets emerging from satellite and center orifices in the 5 × 3 × 4 configuration with 6 mm pitch circle radius (PCR) for satellite orifices are shown in Fig. 3(c). This figure shows that for sufficiently close orifices, jet interaction occurs very close to the jet exit location owing to the cancelation of oppositely signed vorticity in the inner shear layers and subsequently emerges as a single strong jet. It was observed that different orifice configurations have distinct ability of mixing the surrounding fluid.

The entrainment characteristics of four configurations are quantified by evaluating the variation of mass flow rate with the axial direction as shown in Fig. 4. The procedure for calculating the mass flow rate is adapted from Ref. 36. The mass flow rate is normalized by fluid density, equivalent orifice diameter, and the respective maximum centerline velocity. The equivalent diameter for the 5 × 3 × 4 configuration is 7.81 mm. The 8 mm single orifice is considered as a baseline case for the purpose of comparison. The multiorifice configuration jet shows a higher entrainment rate in the near field than the single orifice jets. This figure shows that the mass flow rate for the 5 mm orifice is least among all the configurations. The configuration 5 × 3 × 8 shows a higher mass flow rate up to z/d = 12 compared to the other configurations. The possible explanation for such differing behavior is due to the local distribution of the satellite orifices. The fluid viscosity limits the amount of fluid drawn by the diaphragm during the suction stroke; one single hole therefore cannot pull the fluid beyond a certain downstream region. However, with an increase in the number of orifices, fluid can be drawn from several local regions; hence, an increase in the quantity of flow is observed.

FIG. 3. Mean velocity vector field superimposed with vorticity contour for various multiorifice configurations: (a) Two orifices separated at 20 mm apart, (b) two orifices separated at 12 mm apart, and (c) 5 × 3 × 4 configuration (PCR = 6 mm).

FIG. 4. Variation of normalized mass flow rate with axial distance, where m* represents the normalized mass flow rate.
B. Flow dynamics of an impinging jet

In this section, the flow characteristics of an impinging jet of various surface spacings are reported. The location of the surface from the orifice determines the jet confinement level, and hence the availability of fresh air during a suction phase of the synthetic jet. The presence/absence of fresh air eventually leads to the variation in heat transfer behavior of the impinging jet. Furthermore, the effect of a different configuration of the satellite orifice on the availability of fresh air for impingement is also described.

Figure 5 presents the instantaneous velocity vector field of an impinging jet \((z/d = 10)\) of the \(5 \times 3 \times 8\) configuration. In this figure, velocity is superimposed with vorticity contours. Near the jet orifice, the direction of velocity vectors exhibits the flow characteristics during the ejection and suction phases of the synthetic jet. The fluid is coming out from the cavity in Fig. 5(a), while fluid is going into the cavity in Fig. 5(b). The exact position of the diaphragm is not being tracked in these measurements owing to the relatively high excitation frequency. However, the captured images are selected in such a manner that they exhibit the suction and ejection cycles. Figure 5(a) shows that during the ejection phase of the speaker, a pair of vortices evolves out from the orifices. The primary vortices are known for their large scale mixing ability between the jet and surrounding fluids. These vortices are convected downstream and approach toward the surface. The developing vortices do not lose their coherence until they strike the plate. The traces of vortices having impinged in the previous cycle can be seen on the surface as moving out along the radial direction. The smaller structures move along the plate causing the formation of a wall jet along the surface. The wall jet vortices dissipate into smaller vortices, thereby promoting small scale mixing, and are responsible for the enhancement in the heat transfer rate from the hot plate. Thus, the wall jet would act as a heat carrier, which takes away the heat while moving radially outward along the hot plate.

Figure 5(b) shows that during the suction phase of the jet, ambient fluid is sucked inside the speaker cavity. Note that in this phase, fresh fluid (not associated with the previous cycle fluid) is being sucked into the speaker cavity. The ability to draw fresh fluid is because the location of the impingement plate is sufficiently far from the orifice. Thus, the impingement plate is struck by at least a part of the fresh fluid in each cycle, and recirculation of fluid after impingement on the surface is absent. The impingement of fresh (cold) fluid has a higher ability to enhance the heat transfer rate compared to the same recirculating (hot) fluid. It was observed that for larger surface spacing \((z/d > 8)\), other configurations \((5 \times 3 \times 4 \text{ and } 5 \times 3 \times 2)\) also behave in a similar fashion, i.e., they exhibit the absence of recirculating fluid; hence, these cases are not being discussed separately.

In contrast, for small surface spacing, the availability of fresh air for different configurations is an issue as elucidated later in this section with the help of the mean velocity field.

The instantaneous velocity vector fields shown in Fig. 5 also provide some insight into the flow separation and reattachment phenomenon, which occurs along the surface and is further connected to the local heat transfer distribution. In Figs. 5(a) and 5(b), close to the surface and in the proximity of the radial location of \(r/d = \pm 5\), velocity vectors are seen to move away from the surface, which elucidates the separation of flow and reattachment in its immediate downstream region. The flow separation and reattachment phenomenon is associated with the development of primary structures. As discussed in Refs. 34 and 35, during certain stages of the primary structure flow separation, the formation of a secondary vortex structure arises along the wall jet, which becomes responsible for peaks in the Nusselt number distribution. Figure 6 shows local peaks in the rms velocity (both streamwise and cross streamwise) for the \(5 \times 3 \times 2\) configuration. The rms velocity is measured at 0.5 d above the surface and is normalized with the maximum streamwise velocity along the same characteristic line \((r/d)\). It is interesting to observe that peaks in rms velocity also occur in the close proximity of the radial location of flow separation/reattachment \((r/d = \pm 5)\). Thus, the distribution of rms velocity reinforces the variation in Nusselt number along the radial direction.

The flow behavior for medium and small surface spacings is described using the averaged velocity vector field shown in Figs. 7–11. For clarity of these figures, one streamwise and one crosswise vectors are skipped in Figs. 7–9. Figure 7 shows the mean velocity field superimposed with vorticity contours for a medium surface spacing \((z/d = 6)\) and \(5 \times 3 \times 4\) configuration. This figure illustrates that on each side of the issuing jet, a large vortical structure (rotating fluid) occurs even in the time-averaged field. This whirling fluid communicates with both the impinging surface and the orifices. A possible explanation for this phenomenon is the orientation of the vortical structure and proximity of flow to the orifice during the suction stroke. As the orifices and surface come close enough to
FIG. 6. Variation of normal stresses with the radial direction measured at a distance 0.5 d from the surface for the 5 × 3 × 2 configuration.

FIG. 7. Flow recirculation shown by the mean vector field superimposed with vorticity contour for the 5 × 3 × 4 configuration at z/d = 6. Note that only one in two vectors is shown in both radial and axial directions for clarity.

FIG. 8. Mean vector field superimposed with vorticity contour for the 5 × 3 × 8 configuration at z/d = 5. Note that only one in two vectors is shown both in axial and radial directions for clarity.

FIG. 9. Mean vector field superimposed with vorticity contour for the 5 × 3 × 2 configuration at z/d = 4. Note that only one in two vectors is shown both in axial and radial directions for clarity.

FIG. 10. Mean vector field superimposed with vorticity contour for the 5 × 3 × 4 configuration at z/d = 2.5. Note that only one in two vectors is shown in the radial direction for clarity.

For a single orifice jet, Bhapkar et al. observed an enhanced cavity temperature for small surface spacing (z/d ≤ 2) compared to larger surface spacing. This way during the suction phase, a fraction of hot air is sucked back, which limits the availability of fresh air in the next cycle. For other configurations, a similar recirculation phenomenon is also observed. Therefore, results for only one configuration are being presented for each surface spacing (Figs. 7–10).

Figure 8 shows that a decrease in the impingement distance to z/d = 5 increases the amount of recirculating air and decreases the availability of fresh fluid. A further decrease in surface spacing to z/d = 4 and z/d = 2.5 in Figs. 9 and 10, respectively, makes the motion of fresh fluid imperceptible. Observe from Figs. 7–10 that with a decrease in the surface spacing, the size of the recirculating structure decreases and its center moves toward the jet. This phenomenon suggests that with a decrease in the surface spacing between the orifice and impinging plate, the wall jet reaches closer to the orifice and the influence of suction on wall jet fluid increases; hence, vortices are being sucked back during their early stages itself.

Figure 11 illustrates the velocity field for the smallest surface spacing (z/d = 1.5). At this spacing, flow however does not follow the above-mentioned trend. This figure shows that despite flow recirculation, a large amount of fresh air is nonetheless drawn toward the orifice during the suction stroke. Although some recirculation occurs, fresh air would help to cool the plate so the high heat transfer is expected in this position. A high Nusselt number is indeed observed for this spacing, as discussed further in Sec. V.
FIG. 12. Variations of radial velocity for different configurations and surface spacing.

The strength of the air sucked back for each configuration is quantified in Fig. 12. For this, the radial velocity is plotted at 0.5\(d\) above the surface. A positive value of radial velocity indicates an outward moving flow. For better understanding and to aid comparison of the results, the velocity at two surface spacings (\(z/d\) values) is plotted in this figure. At larger surface spacing (\(z/d = 10\)), the direction of the velocity vector elucidates that after impingement, the flow moves radially outward on either side of the plate. While at a smaller surface spacing (\(z/d = 1.5\)), the flow pattern was found to be exactly opposite of this (i.e., the fluid moves radially inward), for each of the three configurations (5\(\times\)3\(\times\)8, 5\(\times\)3\(\times\)4, and 5\(\times\)3\(\times\)2). The negative velocity in the positive \(r/D\) direction and positive velocity in the negative \(r/D\) direction clearly show the inward flow. Note that for clarity, the radial velocity for configuration 5\(\times\)3\(\times\)4 has been not included in this figure. For the 5\(\times\)3\(\times\)8 case, the radial velocity is close to zero after \(r/d = \pm 6.5\), which indicates that very little fresh fluid is available in this case. However, for 5\(\times\)3\(\times\)2 and 5\(\times\)3\(\times\)4 configurations, air still moves radially inward up to \(r/d = 10\), which indicates a large entrainment of fresh air. Note that to calculate the volume of entrained air, the radial velocity has to be multiplied by area, where the area grows linearly with radial distance \(r\). The result suggests that for small surface spacing, the amount of fresh air impinging on the surface depends on the number of satellite orifices.

C. Effect of PCR on flow recirculation of an impinging jet

As reported in Sec. II A, to alter the PCR (pitch circle radius), different top plates with different distances of the satellite orifice from the center orifice were employed. The effect of variation of PCR on the flow characteristics for the 5\(\times\)3\(\times\)4 configuration is shown in Fig. 13. This figure also illustrates the influence of PCR on flow recirculation for different surface spacings.

Figure 13(a) illustrates that flow recirculation is missing at small surface spacing \(z/d = 1.5\). While for the same configuration and PCR, as the distance of the plate is increased from \(z/d = 1.5\) to \(z/d = 4\), recirculation is observed [Fig. 13(b)]. Figure 13(c) shows the mean velocity field for the same surface spacing as in Fig. 13(b) but with an increase in PCR to 10 mm. This figure shows the separate jets emerging out from the center and satellite orifices and their individual impingement on the surface. Figure 13(c) illustrates that with an increase in PCR from 8 mm to 10 mm, more fresh air is sucked back compared to the 8 mm PCR case. This is evident from the movement of the center of the recirculating region toward the free jet in Fig. 13(c).

Thus, the recirculating nature of the impinging multiorifice jet depends upon both the surface spacing and PCR. The results suggest that for small surface spacing with an increase in PCR, the satellite jets do not mix with the center orifice, and the jets rather maintain their individual identities. While for medium surface spacing, the satellite jet is attracted toward the center orifice and it forms a single jet before impingement.

IV. DISCUSSION ON FLOW RECIRCULATION AND AVAILABILITY OF FRESH AIR

The recirculation and availability of fresh fluid for the medium and small surfaces are explained through a schematic diagram in Fig. 14. The schematics are drawn by observing the temporal behavior of the jet through smoke wire visualization (not shown) and instantaneous velocity fields.

Figure 14(a) presents the schematic for medium surface spacing. In this case, during the ejection stroke, the fluid is discharged from both the center and satellite orifices. However, after a certain
downstream distance from the orifice, the center orifice jet having higher momentum pulls the satellite orifice jet to form a single large size vortical structure. This primary vortex gets convected downstream and reaches to the surface and gets deflected without losing its original nature (shown by the dotted line). During the suction stroke, the wall jet vortex structure is easily pulled toward the orifice owing to its proximity to the orifice. Furthermore, the direction of its motion also helps to be pulled during the suction stroke (shown by the dotted arrows). This is how the same fluid starts to recirculate in the gap between the two plates. As the distance between the orifice and surface decreases (up to $z/d = 2$), the wall jet vortex reaches close to the orifice and blocks the motion of fresh air toward the orifices; hence, the availability of the fresh fluid decreases.

For small surface spacing ($z/d < 2$), the center jet does not get sufficient time to develop; hence, the satellite jet is not pulled and all the jets impinge separately on the surface [Fig. 14(b)]. The area average velocity at the center orifice is around 3 m/s, while the time duration of the ejection stroke is 4 ms for 125 Hz actuation frequency. The total distance traveled during the ejection stroke by the center orifice jet is roughly 12 mm, which is more than 2 d (10 mm), i.e., the center jet gets enough time to convert into a wall jet before starting the suction stroke. Chaudhari et al. observed that all orifices suck and eject fluid at the same time, but the velocity of the satellite orifice was 33% lesser than the center orifice. Thus, the jet fluid coming out of the satellite orifices takes approximately 50% more time to travel the same distance. The satellite jets therefore do not have enough time to form a wall jet. The vortex of the wall jet formed from the center orifice intercepts the satellite jet and drives the satellite jet fluid along the wall jet [Fig. 14(b)]. Note the absence of recirculation in this case, while fresh fluid continues to be sucked in each cycle.

V. CONNECTION WITH HEAT TRANSFER RESULTS

The variation in heat transfer with surface spacing and for different orifice configurations is shown in Fig. 15 (taken from Ref. 5). It was observed that the multiple orifice configurations have a higher heat transfer coefficient compared to a single orifice configuration (both 5 mm and 8 mm). For a single orifice and $5 \times 3 \times 8$ configuration, the heat transfer coefficient increases gradually, reaches to a maximum value, and starts decreasing. However, the heat transfer coefficient for multiorifice configurations ($5 \times 3 \times 1$, $5 \times 3 \times 2$, and $5 \times 3 \times 4$) exhibits two local maxima, one at lower surface spacing ($z/d = 2$) and another at larger spacing. A possible explanation for such differing behavior in heat transfer is proposed in three parts using the present findings: variation of Nusselt number with surface spacing, variation in the magnitude with different configurations, and peaks in Nusselt number.

For a single orifice, the justification for the highest heat transfer rate at a particular surface spacing can be explained by change

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**Figure 14.** Schematic representation of the development of the recirculation pattern in the gap between the orifice and impingement plates for (a) medium surface spacing ($2 \leq z/d < 8$) and (b) small surface spacing ($z/d < 2$).

**Figure 15.** Variation of the average Nusselt number for different orifice configurations. Reprinted with permission from Chaudhari et al., Int. J. Heat Mass Transfer 54(9-10), 2056–2065 (2011). Copyright 2011 Elsevier.
in jet momentum and vortex strength as a function of surface spacing. Ghaffari et al.\textsuperscript{34,35,31} reported that for small surface spacing owing to the reduced growth of vortices, they do not impinge on the surface with their peak strength. In addition, the impingement of primary vortices on the surface increases the axial velocity (normal velocity), which diffuses to the impinging surface and thereby enhances the momentum transfer to the surface.\textsuperscript{30} With an increment in the surface spacing beyond a certain limit (z/d $\approx$ 8–10), a primary vortex ring starts to disintegrate into smaller ring vortices; also the jet loses its momentum, and hence a decrease in heat transfer rate occurs.

The heat transfer coefficient shown in this figure exhibits distinct values for different configurations even for the same surface spacing. The higher mass flow rate (Fig. 4) and impingement of larger vortical structures in the multiorifice configuration, compared to a single orifice jet, lead to a correspondingly higher heat transfer rate. The heat transfer result for small surface spacing highly depends on the impingement of the vortical structure and jet entrainment. A head-on collision of counter-rotating vortices on the surface leads to the surface removal action and an enhancement in the heat transfer rate. The 5 $\times$ 3 $\times$ 1 configuration having a single satellite orifice entrains less air compared to the 5 $\times$ 3 $\times$ 2 configuration. In addition, with an increase in satellite orifice (5 $\times$ 3 $\times$ 2), the jet momentum increases symmetrically and an increase in the heat transfer rate occurs compared to the 5 $\times$ 3 $\times$ 1 configuration. For the center orifice with the four satellite orifice case, the shear layer of the center orifice is shared by the four satellite orifice. Hence, the strength of the center jet is reduced, which leads to a reduction in the first peak of Nusselt number, compared to the 5 $\times$ 3 $\times$ 2 configuration. The heat transfer result for small surface spacing beyond z/d $> 2$, fluid recirculation starts, which results in the availability of negligible or very less fresh fluid and low heat transfer coefficient. With a further increase in surface spacing, the availability of fresh air increases and also vortices impinge on the surface with higher momentum, leading to a higher (and the second peak) Nusselt number.

The peaks in Nusselt number observed in Fig. 15 at z/d $= 2$ are further explained with the help of the availability of fresh air [Fig. 14(b)]. Figure 15 shows that for all configurations, with an increase in surface spacing, the average Nusselt value increases until z/d $= 2$. With a further increase in the surface spacing, a fall in the heat transfer rate can be observed (except for 5 $\times$ 3 $\times$ 8 and single orifice jet cases). With the increase in surface spacing beyond z/d $> 2$, fluid recirculation starts, which results in the availability of negligible or very less fresh fluid and low heat transfer coefficient. With a further increase in surface spacing, the availability of fresh air increases and also vortices impinge on the surface with higher momentum, leading to a higher (and the second peak) Nusselt number.

The effect of PCR on the heat transfer rate adopted from the study of Chaudhari et al.\textsuperscript{7} is shown in Fig. 16. The dimension of the heated plate of 40 $\times$ 40 mm\textsuperscript{2} was employed. It was observed that with an increase in PCR, the location of peaks in the average Nusselt number is shifted to larger surface spacing. As discussed in Sec. IV, the heat transfer rate increases with an increase in the surface spacing up to z/d $= 8–10$ due to optimal vortex strength, jet velocity, and turbulence. We also observed in Sec. III C that for a PCR value of 6 mm, the availability of fresh fluid is no longer limited up to z/d $= 2$, while with an increase in PCR, the critical limit of surface spacing for the availability of fresh air is shifted to a larger value of surface spacing.

The present velocity measurements therefore correlate well with the previous heat transfer results and help explain the somewhat unexpected trend in Nusselt number observed by Chaudhari et al.\textsuperscript{7} for the multiorifice configuration. The local peak in Nusselt number at the small spacing suggests that the synthetic jet can be effectively utilized for cooling of compact devices.

**VI. CONCLUSIONS**

The present experimental study focuses on the interaction of the multiorifice synthetic jet and describes its consequences on the heat transfer study reported in the literature. The results show that the fluid dynamics of the satellite and center orifice jets is highly dependent upon the orifice configuration, pitch to circle radius (PCR), and downstream interaction between the vortices. It was observed that larger distance between the orifices leads to the inner vortex rolls toward the center of the jet and the inner flow appears as a cone shape, while for sufficiently close orifices, the emerging jet appears as a single jet. It was also observed that with an increase in the number of satellite orifices, jet entrainment increases with axial distance and a strong wall jet is produced. Therefore, for large surface spacing (z/d $= 8–10$), the formation of a strong wall jet, which carries away a large amount of heat from the plate, and absence of flow recirculation are the causes for the higher heat transfer rate. As the distance between the plate and orifice reduces, the recirculation of hot air occurs resulting in a drop of heat transfer coefficient.

For impingement at a small surface spacing (z/d $\leq 2$), the appearance of peaks in heat transfer for different orifice configurations is explained by three different mechanisms. These are the following: interaction of the inner shear layers, vortex orientation,
and availability of fresh air. For small spacing (z/d < 2), the center jets do not get sufficient time to develop. Therefore, the satellite jet is not pulled by the center jet, and the jets impinge separately on the surface. In this situation, the wall jet vortex produced from the center jet is aided by flow from the satellite orifices and the flow drives away along the wall jet. Large-scale recirculation of flow is absent in this case, and fresh fluid is sucked in each cycle, resulting in a local peak in Nusselt number. The difference in the value of the peak Nusselt number for different configurations is mainly due to the variation in the amount of fresh air available for cooling the surface.

The effect of PCR on the flow recirculation is also investigated. The mean velocity vector shows that with an increase in PCR, the absence of flow recirculation occurs for both small and medium surface spacings. It was observed that with an increase in PCR, the distance between the satellite and center orifice jets increases. Hence, jets issuing from the satellite and center orifices behave as a multi-jet. The present investigation is able to predict the nature of the local heat transfer distribution. These results provide useful insights into occurrence of local peaks observed with multiple synthetic jet configurations and are therefore useful in the design of compact synthetic jet based cooling devices.

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