Non-stationary convective heat transfer in an air synthetic impinging jet. Experiment and numerical simulation

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An unsteady local heat transfer in an air synthetic non-steady-state jet impingement onto a flat plate with a variation of the Reynolds number, nozzle-to-plate distance and pulses frequency is experimentally and numerically studied. Measurements of the averaged and pulsating heat transfer at the stagnation point are conducted using a heat flux sensor. The axisymmetric URANS method and the Reynolds stress model are used for numerical simulations. For local values of heat transfer, zones with the maximum instantaneous value of heat flux and heat transfer coefficient are identified. The heat transfer increases at relatively low nozzle-to-plate distances ($H/d \leq 4$). The heat transfer decreases at high distance from the orifice and target surface. An increase in the Reynolds number causes reduction of heat transfer.

1. Introduction

Impinging jets (IJ) are widely used in many engineering applications due to the extremely high values of heat transfer at the stagnation point and high values of removing heat. IJ cooling requires fans or other additional devises, which can be impossible in various practical situations. Synthetic jets do not require these additional devises associated with traditional continuous or pulsed impinging jets. A synthetic impinging jet (SIJ) is a fluidic device and it is one of the types of active control of mean and fluctuational flow, turbulence and heat and mass transfer in a jet impingement [1–3].

A synthetic jet produces zero-net-mass-flux by the “blowing” out of fluid and “suction” it in to out across an orifice [1–3]. The synthetic jet actuator works very well for the near-resonance conditions [4,5]. The “blowing” and “suction” processes can be created by piezoelectric, electromagnetic, acoustic, and mechanical actuator devices. The “blowing” –“suction” of the fluid occurs without the injection of a mass flux, it allows to create a compact device for local cooling. This is the main difference between the SIJ and well-known classic continuous or pulsed jet impingement cooling systems that use a fan to move and control the flow of a coolant. This makes SIJ attractive for heat transfer enhancement in applications, for instance, in microelectronics [6] and various devices [7–9].

To the best of the author’s knowledge, in the last 15 years a number of the experimental [7–9] and numerical [10,11] studies of the flow structure and heat transfer in SIJ have been carried out. Note that numerical predictions [10,11] were performed using the URANS approach and Menter’s $k–\omega$ SST [12] on the commercial CFD package Fluent using. Additionally, the $v^2–f$ turbulence model [13] is used in [11].
The flow structure and mass transfer in a round synthetic, hybrid SIJ (mass flow rate is not equal to zero per cycle), pulsed and stationary impinging jets were measured in [14,15]. Hot-wire experiments showed the hybrid synthetic jet exhibits a higher extrusion volume flow rate than the ordinary synthetic jet. The breakdown and coalescence of the vortical train is much faster at large amplitudes of flow oscillations. The mass transfer rate to the impingement wall is then much higher.

The aim of the present work is to conduct the experimental and numerical study of the local averaged and unsteady stagnation heat transfer in the SIJ.

2. Experimental setup

![Figure 1](image)

**Figure 1.** Schematic of the experiment: (1) acoustic speaker, (2) flat plate with orifice, (3) copper flat plate, (4) electric heater, and (5) heat-flow sensor.

A schematic of the experiment is shown in figure 1. The experimental setup includes: a low-frequency signal generator G3-102, a loudspeaker 10 GD-30 (1), and an orifice plate thickness was 1.2 mm with a hole with sharp edges and a diameter \( d = 7 \) mm (2), a heated copper plate (3), an electric heater (4), heat-flow film sensor (HFFS) (5), thermocouples, hot-wire anemometer DISA 55M for recording characteristics at the initial section of the jet. Heat-transfer section (3) was a copper plate 190 mm in diameter and 50 mm thick. Area (3) was heated using electric heater (4) with the boundary condition \( T_W = \text{const} = 50-60^\circ\text{C} \). Miniature heat-flow film sensors (5) (HFFS) \( 2 \times 2 \times 0.2 \) mm in size were glued to the plate surface [16]. The HFFSs were connected to an automatic information-gathering system through an amplifier and an analog-digital converter (L-Card Е14-140). This device allows one to measure the instantaneous value of heat flow \( q' \) in the frequency range up to 3 kHz and calculate the average \( Q \) and rms \( q \) values of heat flow based on the time series (the sample size is \( 1 \times 10^4 \)). The local average values of the heat transfer coefficient \( \alpha \) were determined from the heat flux \( Q \) and the temperature difference between the surface of the heated wall \( T_W \) and the air jet in the initial section \( T_0 \). The plate \( T_W \) and jet \( T_0 \) temperatures were measured by chromel-copel thermocouple wires 0.2 mm in diameter, and the thermopower was recorded by a Sch-301-1 millivoltmeter.

3. Mathematical model

The flow is treated as a non-steady-state, incompressible and axisymmetrical and described by continuity, two-momentum and energy equations. The system of URANS axisymmetric equations has been shown in [17]. The Reynolds stress (second-moment closure) model for the single-phase flow consists of a system of equations for the second moments in the single-phase impinging jet and an equation for turbulent kinetic energy dissipation [18]. The “in-house” code is employed to study a turbulent air non-steady-steady SIJ. It was previously developed and validated against experimental results of other works in the previous study [17]. The validation analysis has been performed as well as for mean and fluctuational and heat transfer in a steady-state and pulsed turbulent gas jet impingement.

During the half-period of “blowing” \( T/2 \), the flow with the mean-mass velocity \( U_0 \) flowing out from the cavity, and during the half-period of “suction” \( T/2 \), the gas flow with the same mean-mass velocity is flowing return into the cavity (“suction” time period). Therefore, the total mass flow rate for the
period $T$ is equal to zero. The sinusoidal time-dependent velocity profile is assumed at the orifice outlet cross-section.

$$U_0(t) = U_{\text{max}} \sin(2\pi ft).$$

The average flow velocity $U_0$ over the half-period of "injection" of the synthetic jet, averaged over the outlet area, the Reynolds number $Re_0$ and stroke length of SIJ $L_0$ are determined by the relations [1]:

$$U_0 = \frac{1}{T} \int_0^{T/2} U(t) dt, \quad Re_0 = U_0 d / \nu, \quad L_0 = U_0 / f,$$

where $T$ is the total period of the "injection-suction" cycle, $\nu$ is the kinematic viscosity of the air, and $f$ is the frequency of pulsation.

A sketch of the development of an impinging synthetic jet is shown in figure 2. All predictions are conducted for air at atmospheric pressure. The diameter of the sharp-edged orifice is $d = 7$ mm. The Reynolds number of the jet varied in the range $Re_0 = U_0 d / \nu = (0.5–5) \times 10^3$.

Here $U_0$ is the average velocity for a half-period of "blowing" of a synthetic jet and $U_{\text{max}}$ is the maximal velocity on the period. The wall temperature was constant and equal to $T_W = 323$ K. The initial gas temperatures in the actuator cavity and in the surrounding flooded space were equal to $T_1 = T_b = 293$ K and coincided with the temperature in the quiescent medium and in the actuator cavity. The nozzle-to-plate distance varies in the range $H/d = 1–6$. Numerical simulations are performed for continuous jet ($f = 0$ Hz) and pulsation frequencies $f = 1–200$ Hz. The Strouhal number, determined from the diameter of the outlet, varied within the range $St = f d / U_0 = 0.002–0.2$.

![Figure 2. Sketch of the computational domain.](image)

The computational domain is a cylinder of $10H$ in the radial direction and height $H$ (see figure 2). The computational grid, non-uniform in both axial and radial directions. The synthetic jet “actuator” in our numerical predictions is used for the generation of flow oscillations. The synthetic flow is organized in the cavity and their inner diameter is equal to the diameter of impinging surface ($D = 190$ mm). The height of the cavity is $8H$, and at this distance the “diaphragm” is set. The “diaphragm” forms the synthetic jet of the "blowing-suction" type. The moving type of the “diaphragm” is assumed to be sinusoidal and the zero-net-mass-flux at the cycle. The distance from the impinging surface to the confined surface is 10 mm. The time step is equal to $\Delta t = 10 \mu s$.

4. Experimental and numerical results. Their discussion

4.1. Experimental results

At the first stage of the experiments, using dynamic and thermal measurements, the characteristics of the synthetic jet were determined depending on the operating parameters. The maximum jet velocity during the period in the initial section of the jet is practically independent of the orifice diameter, and is mainly determined by the amplitude and frequency of the signal supplied to the loudspeaker. The range of the obtained maximum velocities at the nozzle outlet was $U_{\text{max}} = 1.5–7$ m/s, while the Reynolds number was $Re_0 = 700–3800$. Figure 3 shows a typical time series of heat flux pulsations on
a copper plate at the stagnation point (the point of intersection of the jet axis and the plate surface). Experimental parameters: orifice diameter \( d = 7 \text{ mm} \), distance to the heated surface \( H/d = 2 \), pulsation frequency \( f = 90 \text{ Hz} \), Reynolds number was \( Re_0 = 3150 \). As can be seen from the figure 3, the change in the instantaneous value of the heat flux over time looks like a periodic function with a period of 0.011 s and varies within \( q' = 2360-3780 \text{ W/m}^2 \). In this case, the average value of the heat flux is \( Q = 2840 \text{ W/m}^2 \). The difference between the presented signal and the harmonic function is due to the synthetic jet source design [1].

![Figure 3. Time series of the heat flux density at the plate stagnation point \( Re_0 = 3150 \), \( f = 90 \text{ Hz} \), \( d = 7 \text{ mm} \), \( h/d = 2 \).](image)

![Figure 4. Power spectrum of the heat flux density at the stagnation point of the plate \( Re_0 = 3150 \), \( f = 90 \text{ Hz} \), \( d = 7 \text{ mm} \), \( h/d = 2 \).](image)

The next figure 4 shows the power spectrum of the heat flux pulsations at the stagnation point \( b \) for the frequency \( f = 90 \text{ Hz} \) and the Reynolds number \( Re_0 = 3150 \). As can be seen from the figure 4, the spectrum contains the fundamental frequency \( f = 90 \text{ Hz} \) and its multiple harmonics \( 2f, 3f \) and \( 4f \). At ones, the spectral power of multiple harmonics decreases with increasing harmonic number. In the literature on impact jets, there are two points of view regarding spectral analysis. Thus, the authors of [19], when superimposed oscillations were applied to heat transfer in an impinging stationary jet, experimentally found a subharmonic \( f/2 \) (the frequency is half the fundamental one). A similar result with the presence of a subharmonic was obtained by the authors of [20] in the numerical simulation of heat transfer for a synthetic impact jet on a flat plate. At the same time, the frequency \( 2f \) was obtained in experiments [21] for an impact stationary jet with perturbation. Thus, this issue requires additional research.

4.2. Numerical and experimental results
The results of comparisons of the results of authors’ numerical simulations of heat transfer at the stagnation point of SIJ with the data of authors’ measurements and semi-empirical correlation [7] are shown in figure 5 with the variation of the Reynolds number \( Re_0 = 1900–3800 \). It should be noted authors [7] used another definition of characteristic velocity: \( U_0 = 2fL_0 \). We used more common definition \( U_0 = fL_0 \) and therefore \( Re_0 = 2^*Re_0 \). The semi-empirical correlations of [7] is shown in the figure 5 and it generalized the stagnation heat transfer depending on the dimensionless stroke length \( L_0/H \) (1):

\[
\begin{align*}
\text{Nu}_0 = \left( 0.19 + 0.532 \times \frac{L_0}{H} \right) \times 2^{0.32} = 0.237 + 0.668 \times \frac{L_0}{H}, \quad &L_0 / H \leq 2.5, \\
\text{Nu}_0 = 1.52 \times 2^{0.32} = 1.9, \quad &L_0 / H > 2.5.
\end{align*}
\]
It can be seen that stroke length in the region of small \( L_0/d \leq 5 \) affects heat transfer at the stagnation point of SIJ. At large \( L_0/H \), this influence decreases, and the synthetic jet acquires the patterns of development of pulsed impinging jet (see Figure 5). These conclusions are in qualitative agreement with the data [7,8]. Our measurements and numerical calculations are characterized by a significant excess of the heat transfer at the stagnation point in comparison with the correlation data [7] at small values.

Figure 5. The distributions of Nusselt numbers at the stagnation point vs. dimensionless stroke length for the round synthetic jet impingement. Re\(_0\) = 1900–3800, \( H/d = 2 \), \( d = 7 \) mm, \( f = 18–460 \) Hz, \( U_0 = 4.3–7 \) m/s, \( T_W = 323 \) K, \( T_0 = 293 \) K, \( L_0/H = 0.75–17.1 \). Line 1 is the semi-empirical correlations of [7, 2, 3] and 2 and 3 are authors’ measurements and RSM predictions respectively.

Conclusion
The unsteady and time-averaged heat transfer in the air synthetic non-steady-state jet impingement onto a flat plate with a variation of the Reynolds number, nozzle-to-plate distance and pulses frequency is experimentally and numerically studied. Measurements of the averaged and pulsating heat transfer at the stagnation point are conducted using the heat flux sensor. The axisymmetric URANS method and the Reynolds stress model are used for numerical simulations.

For the small nozzle-to-target distances the stagnation Nusselt number increases. With a further increase in the distance, a decrease in the intensity of heat transfer is obtained. A comparison between our measured and predicted results on heat transfer both at the stagnation point in the synthetic impinging jet with the results of other works is performed. The satisfactory quantitatively agreement is obtained.

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References
[1] Glezer A and Amitay M 2002 *Annu. Rev Fluid Mech.* 34 503.
[2] Holman R, Utturkar Y, Mittal R, Smith B and Cattafesta L 2005 *AIAA J.* 43 2110.
[3] Krishan G, Aw KC and Sharma RN 2019 *Appl. Thermal Eng.* 149 1305.
[4] Sharma RN 2007 *AIAA J.* 45 1841.
[5] Trávníček Z, Fedorchenko AI, and Wang AB 2005 *Sensors and Actuators A: Physical* 120 232.
[6] Pavlova A and Amitay M 2006 *ASME J. Heat Transfer* 128 897.
[7] Valiorgue P, Persoons T, McGuinn A and Murray DB 2009 *Exp. Therm. Fluid Sci.* 33 597.
[8] Persoons T, McGuinn A, and Murray DB 2011 *Int. J. Heat Mass Transfer* 54 3900.
[9] Gil P, Wilk J, Smusz R and Galek R 2020 *Int. J. Heat Mass Transfer* 160 120147.
[10] Bazdidi-Tehrani F, Karami M and Jahromi M 2011 *Heat Mass Transfer* 47 1363.
[11] Hatami M, Bazdidi-Tehrani F, Abouata A and Mohammadi-Ahmar A 2018 *Int. J. Thermal Sci.* **127** 41.

[12] Menter FR 1994 *AIAA J.* **32** 1598.

[13] Parneix S, Durbin PA and Behnia M 1998 *Flow, Turbulence Combust.* **60** 19.

[14] Trávníček Z and Tesař V 2003 *Int. J. Heat Mass Transfer* **46** 3291.

[15] Trávníček Z, Vit T and Tesař V 2006 *Phys. Fluids* **18**, 081701.

[16] Mityakov AV, Sapozhnikov SZ, Mityakov VY, Snarskii AA, Zhenirovsky MI and Pyrhonen JJ 2012 *Sens. Actuators A.* **176**, 1.

[17] Pakhomov MA and Terekhov VI 2015 *Int. J. Thermal Sci.* **87** 85.

[18] Craft TJ and Launder BE 1992 *AIAA J.* **30** 2970.

[19] Silva LA and Ortega A 2013 *ASME J Heat Transfer* **135** 082201.

[20] Liu T and Sullivan JP 1996 *Int. J. Heat Mass Transfer* **39** 3695.

[21] Alekseenko SV, Markovich DM and Semenov VI 1997 *Exp Heat Transfer, Fluid Mech. and Thermodyn.* (Edizioni ETS) p 1815.