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Influence of pulses frequency on forced convection in nonstationary jets inpingement

© M.A. Pakhomov, V.I. Terekhov

Kutateladze Institute of Thermophysics, Siberian Branch, Russian Academy of Sciences, Novosibirsk, Russia
E-mail: pakhomov@ngs.ru, terekhov@itp.nsc.ru

Received July 6, 2023
Revised August 4, 2023
Accepted August 4, 2023

The authors have modeled axisymmetric nonstationary pulsed and synthetic impinging jets and compared the transport processes in such flows with varying pulses frequency. In unsteady impinging jets, the friction on the wall is higher than the corresponding value for a stationary impact jet (up to 25%). It is found that, at the Strouhal number $\mathrm{St} = 0.08$, the time averaged heat transfer at the stagnation point of unsteady impinging jets is the most important. The ranges of values of the Strouhal numbers are shown, where the use of non-stationary impinging jets gives an advantage in heat transfer intensification, and where its deterioration is observed.

Keywords: impinging pulsed and synthetic jets, numerical simulation, transport processes, heat transfer enhancement.

DOI: 10.61011/TPL.2023.10.57052.19675

Impinging jets are characterized by significantly higher values of the local heat transfer coefficient at the stagnation point than common internal flows [1,2]. Such flows are used widely in practice for cooling of thermally-loaded components of power-generating equipment [3], pulsed flow control [4], etc.

The use of synthetic and pulsed impinging jets (SIJs and PIJs, respectively) is one of the ways to exercise active flow and heat transfer control [5–8]. SIJs are produced by periodic oscillations of a surface inside a certain chamber (volume). This causes the formation of an „injection“−„suction“ flow [5]. A flow of gas with mass-average velocity U_0 averaged over time and the orifice section is ejected outward from the chamber within the „injection“ half-period and enters the chamber with the same velocity within the „suction“ half-period. Thus, the gas rate within a full period is zero in such flows [5]. There is then no need to use a fan or a compressor, which makes it possible to construct compact devices for local heat transfer enhancement in various types of equipment [5].

The advantages of synthetic or pulsed impinging jets over a stationary flow in terms of friction on the wall and heat transfer enhancement have not been studied yet. Therefore, the primary goal of the present study is to compare the options for control over the processes of momentum and heat transfer in unsteady impinging jets using the example of axially symmetric unsteady synthetic and pulsed impinging jets.

The diagram of propagation of a synthetic impinging jet and the shapes of velocity profiles for a PIJ, an SIJ, and a stationary impinging jet are shown in Figs. 1, *a* and *b*, respectively, where U_m is the maximum flow velocity on the jet axis in a cycle and the oscillation period is plotted along the abscissa axis. Diagrams for stationary and pulsed impinging jets are similar to the one presented in Fig. 1, *a*; the only difference is that a diaphragm is not needed in these cases. The diameter of the outlet orifice was $d = 7$ mm. The mass-average gas (air) flow velocity averaged over time at the tube end varied within the range of $U_0 = 3-10$ m/s; the corresponding Reynolds number of the jet was $\mathrm{Re} = U_0 d/\nu = (3-5)\cdot 10^3$. Wall temperatures were $T_W = \mathrm{const} = 323$ K. The initial temperatures of gas in the chamber and in the surrounding submerged environment were $T_1 = T_e = 293$ K. The distance from the orifice exit section to the impinging surface was $H/d = 2$. Calculations were performed for frequencies $f = 5-300$ Hz of application of a momentum flow. The Strouhal number was $\mathrm{St} = fd/U_0 = 0.001-0.2$. The thickness of a thermally insulated wall with the orifice through which a jet was injected was $b = 2$ mm. All numerical calculations were performed at fixed mass flow rates in stationary G_{st}, pulsed G_{PIJ}, and synthetic G_{SIJ} impinging jets: $G_{st} = G_{\mathrm{PIJ}} = G_{\mathrm{SIJ}} \approx 0.16$ g/s.

The URANS equation system [8] and the model of transfer of Reynolds stresses [9] were used to simulate the motion and heat transfer in an unsteady axially symmetric jet. The finite volume method was used to obtain a numerical solution with QUICK (for convective terms) and central differences (for diffusion terms) of second-order accuracy. Pressure and velocity fields were corrected in accordance with the SIMPLEC technique. The grid condensed toward all solid surfaces in such a way that the maximum distance from the wall to the center of the first calculation node was $y_+ = yU_*/\nu \leqslant 0.8$. Here, y is the distance from the wall along the normal to the surface, U_* is the friction velocity, and ν is the kinematic viscosity. Calculations for all three types of impinging jets were performed on the grid of 200×256 control volumes.

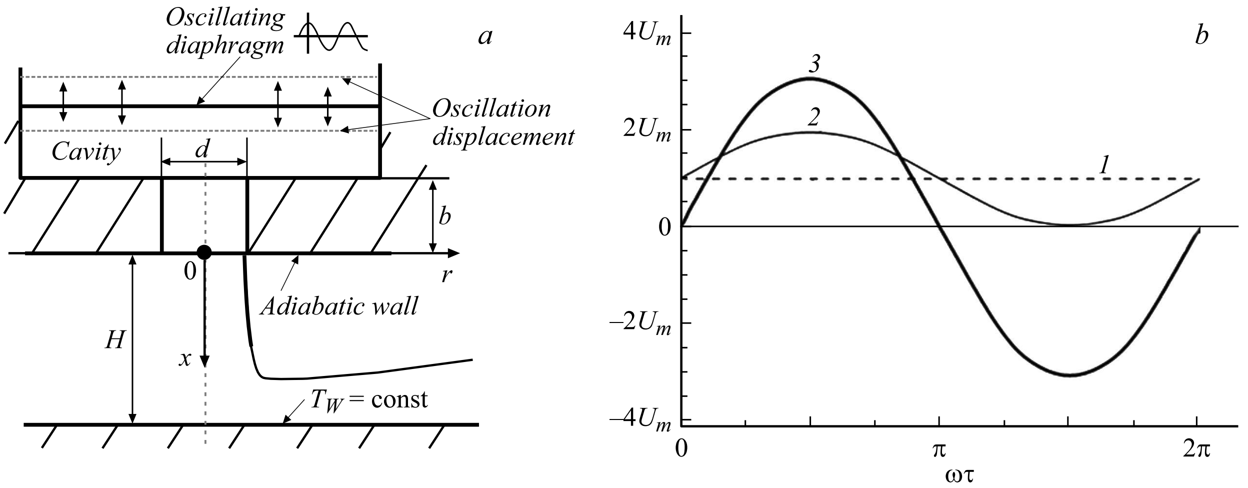

Figure 1. *a* — Diagram of the calculation domain for a synthetic impinging jet. *b* — Velocity distribution in stationary and oscillating flows at different time points within the oscillation period: stationary (*1*), pulsed (*2*), and synthetic (*3*) impinging jets.

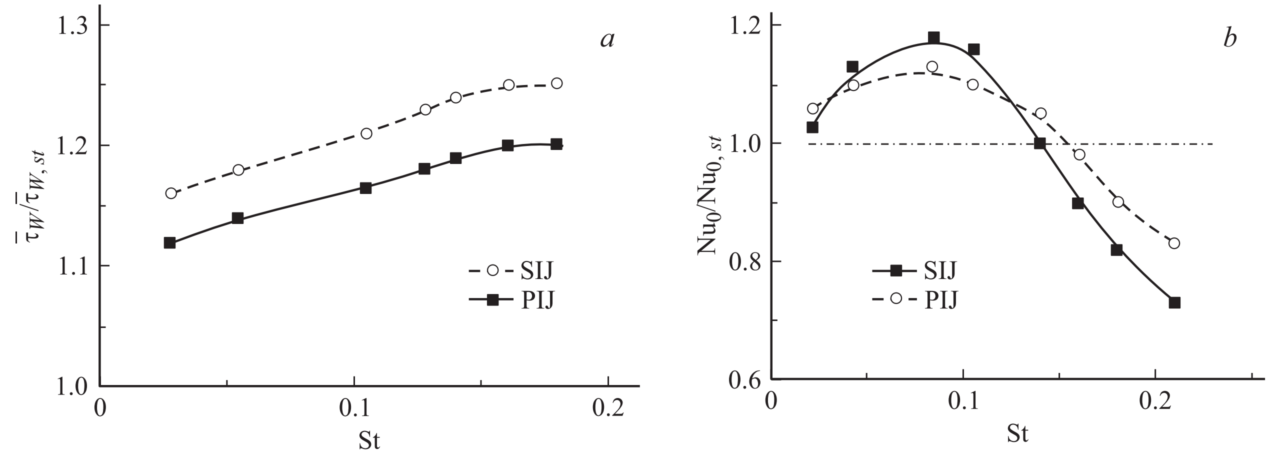

Figure 2. Effect of Strouhal number St on the ratio of averaged friction on the wall (*a*) and on the parameter of heat transfer enhancement at the stagnation point (*b*). $G_{st} = G_{\text{PIJ}} = G_{\text{SIJ}} \approx 0.16$ g/s, $\text{Re}_0 = 5000$, $U_0 = 10.75$ m/s, $L_0/d = 25$, and $f = 5-300$ Hz. PIJ and SIJ are pulsed and synthetic impinging jets, respectively.

Time-averaged flow velocities for a PIJ and an SIJ at the orifice exit section ($x = 0$): $U_0 = \frac{1}{t_C}\frac{1}{A}\int\limits_0^{t_C}\int\limits_A U(t)dtdA$; for an SIJ within an „injection“ half-period: $U_0 = \frac{1}{t_C/2}\frac{1}{A}\int\limits_0^{t_C/2}\int\limits_A U(t)dtdA$, where $L_0 = U_0/f$ is the stroke length [5]), t_C is the complete cycle time („injection“–„suction“), and A is the orifice area. The diaphragm surface for an SIJ is a solid plane wall that oscillates periodically in the vertical direction (Fig. 1, *a*). As was demonstrated in our recent study [8] on modeling of a synthetic impinging jet, the substitution of an oscillating diaphragm with a solid surface does not induce significant changes in the results of calculations. The measurement data from [10] and the results of recent CFD modeling [11] for a submerged circular synthetic jet were used for comparison. The difference between the numerical results from [11] and the calculations performed in [8] does not exceed 10%. The calculations from [11] and [8] agree qualitatively with the measurement data [9]. The logical conclusion is that the influence of these two input conditions in the given analytical model is insignificant. However, it is possible that the method of characterization of motion of an oscillating diaphragm has a more significant impact in a different geometry [12].

A velocity profile uniform over the cross section was used for pulsed and stationary impinging jets at the orifice exit. In the case of a synthetic impinging jet, a uniform velocity profile was set at the oscillating surface. All calculations were performed with equal mass flow rates of the coolant for unsteady and stationary impinging jets.

The variation of the ratio of averaged friction on the wall in unsteady impinging jets $\bar{\tau}_W$ to friction $\bar{\tau}_{W,st}$ in a stationary flow with the Strouhal number is presented in Fig. 2, *a*.

The averaged friction on the wall was determined using the following relations for a PIJ

$$\bar{\tau}_W = \sum_{n=1}^{NTC\geqslant 10}\left(\frac{1}{NTC\cdot R}\int_0^{t_C}\int_0^{R}\tau_W(r,t)drdt\right)$$

and an SIJ

$$\bar{\tau}_W = \sum_{n=1}^{NTC\geqslant 10}\left(\frac{1}{NTC\cdot R}\int_0^{t_C/2}\int_0^{R}\tau_W(r,t)drdt\right).$$

Here, $R = 5d$ is the integration domain and NTC is the time of averaging (no less than ten periods). Extent R of the averaging domain was chosen for the fact that the change in distributions of friction on the wall for unsteady and stationary impinging jets is the greatest at this distance from the jet axis.

Stationary equations of the model were used to calculate the flow and heat transfer for a stationary jet. The friction on the wall for unsteady impinging jets is higher (by as much as 25%) than the corresponding value for a stationary impinging jet. It is most likely that the friction on the wall reaches a fixed level as the Strouhal number (dimensionless pulse frequency) increases (Fig. 2, a). This is attributable to the fact that pulses follow each other with short time intervals. Dynamic and thermal boundary layers, which fail to renew by the time the next pulse „arrives“, form on the barrier surface.

The results of numerical examination of the effect of pulse frequency on the time-averaged parameter of heat transfer enhancement at the stagnation point are presented in Fig. 2, b, where $\mathrm{Nu}_{0,st}$ is the Nusselt number for a stationary impinging jet. With boundary condition $T_W = \mathrm{const}$, the Nusselt number is given by

$$\mathrm{Nu} = \frac{-(\partial T/\partial x)_W d}{T_W - T_m},$$

where T_m is the mass-average temperature in section $x/d = 2$ and $r = 0$. The time of averaging was no shorter than ten periods. Depending on the Strouhal number, enhancement or impairment of the heat transfer in synthetic and pulsed impinging jets (relative to a stationary flow) were observed. At $\mathrm{St} = 0.08$ ($f = 150\,\mathrm{Hz}$), the averaged heat transfer at the stagnation point for an SIJ and a PIJ reaches its maximum intensity: 20% for a synthetic impinging jet and 10% for a pulsed one. in unsteady jets, the heat transfer is typically inhibited (relative to a stationary flow) at higher Strouhal numbers ($\mathrm{St} > 0.1$); the magnitude of this inhibition reaches 30% for an SIJ and 20% for a PIJ.

Funding

Numerical calculations for a synthetic impinging jet were supported financially by the Russian Foundation for Basic Research (grant No. 20-58-26003 Chekhiya_a), and the data for pulsed impinging single-phase jets were obtained under the state assignment of the Kutateladze Institute of Thermophysics of the Siberian Branch of the Russian Academy of Sciences (121031800217-8).

Conflict of interest

The authors declare that they have no conflict of interest.

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Translated by D.Safin