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Technical Note

Heat transfer characteristics of precessing jets impinging on a flat plate: Further investigations

J.W. Zhou [a,b,*], H. Herwig [a]

[a] Institute for Thermo-Fluid Dynamics, TU Hamburg-Harburg, D-21071 Hamburg, Germany
[b] Department of Thermal Engineering, China Jiliang University, Hangzhou 310018, PR China

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1. Introduction

In a recent study [1] published in this journal heat transfer under precessing impinging jets has been investigated using nozzles of two different sizes. It turned out that their heat transfer coefficients are smaller than those for the corresponding steadily impinging inline jets. As an interesting trend, however, they approach those of the inline jets when the nozzle diameter is decreased as shown in Fig. 1. Since in the previous study only two differently sized nozzles have been tested, we now wanted to find out whether this trend prevails, i.e., whether inline jet coefficients are reached and finally exceeded when the size of the nozzles is further decreased.

In Section 2, we describe the precessing jet nozzle and how we decrease it in size. A short description of the test facility and the heat transfer plate follows in Section 3. For more details Ref. [1].

2. The precessing jet nozzle

A precessing jet nozzle shown in Fig. 2 consists of a cylindrical chamber with a small axisymmetric inlet at one end. The flow separates at the abrupt inlet expansion and reattaches non-symmetrically at the wall of the chamber induced by an ambient air entrainment. Fluid instability causes the reattaching flow from the inlet to precess around the inside wall of the chamber and thus to produce a precessing exit jet.

Although the emerging flow is highly 3-dimensional and unsteady, a nearly fixed precessing frequency can be obtained as long as suitable parameters of the nozzle are adopted. Fig. 3 shows the nozzles of our study which are easy to build and have the property of a stable, uninterrupted precessing flow. They simply consist of an additional envelope tube added to the inner nozzle exit. In our previous study [1], we found the ratio $D_2/D_1 = 10$ and $h/D_1 = 20$ as the best choice for the parameter values. With a 90°-phase at the inner nozzle exit as shown in Fig. 3 the stability of the flow precession can be improved. In the present study, three different sized nozzles are used with diameters $D_1 = 3.5$ mm, 1.75 mm and 0.8 mm, respectively. The first two (Fig. 3a) are exactly similar to those of the previous study [1], the third one of diameter $D_1 = 0.8$ mm is slightly different.

Since one small nozzle alone would only give a very small flow rate, we arranged four nozzles as shown in Fig. 3b. They are made of two plate-inserts (1) and (2) at the end of the envelope tube of the biggest nozzle. The corresponding four inline jets emerge when only plate (1) is fixed at the end of the envelope tube. The flow rate through the four smallest nozzles roughly corresponds to the flow rate through the middle sized nozzle.

With this arrangement the three nozzles of diameter $D_1 = 3.5$ mm, 1.75 mm and 0.8 mm are geometrically similar. The far field conditions downstream of the nozzles are not the same, however, since the four jets out of the smallest nozzle may interact with each other. In this situation one can either assume that the effect of such an interaction is negligible or modify the main question underlying our study. Instead of asking whether reducing D_1 can eventually lead to a heat transfer enhancement by unsteadiness one may ask whether this happens when at the same time a single nozzle is replaced by an array of nozzles. From a

* Corresponding author. Address: Institute for Thermo-Fluid Dynamics, TU Hamburg-Harburg, D-21071 Hamburg, Germany.
E-mail address: h.herwig@tu-harburg.de (J.W. Zhou).

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Nomenclature

A	(m^2) surface	V	(V) electrical voltage
D	(m) diameter	w	(m/s) velocity
f	(s^{-1}) frequency	v	(m^2/s) kinematic viscosity
h	(m) nozzle height		
H	(m) nozzle-to-plate distance	*Indices*	
I	(A) electrical current	C	conduction
k	(W/mK) thermal conductivity	IJN	inline jet nozzle
Nu	(–) Nusselt number	PJN	precessing jet nozzle
q	(W/m^2) heat flux density	SF	single field
Re	(–) Reynolds number	W	wall
Sr	(–) Strouhal number	∞	ambient
T	(K) temperature		

practical point of view this is the relevant question since nozzle arrays have to be used anyway when D_1 is reduced considerably. Otherwise flow rates for cooling a surface would be much too small.

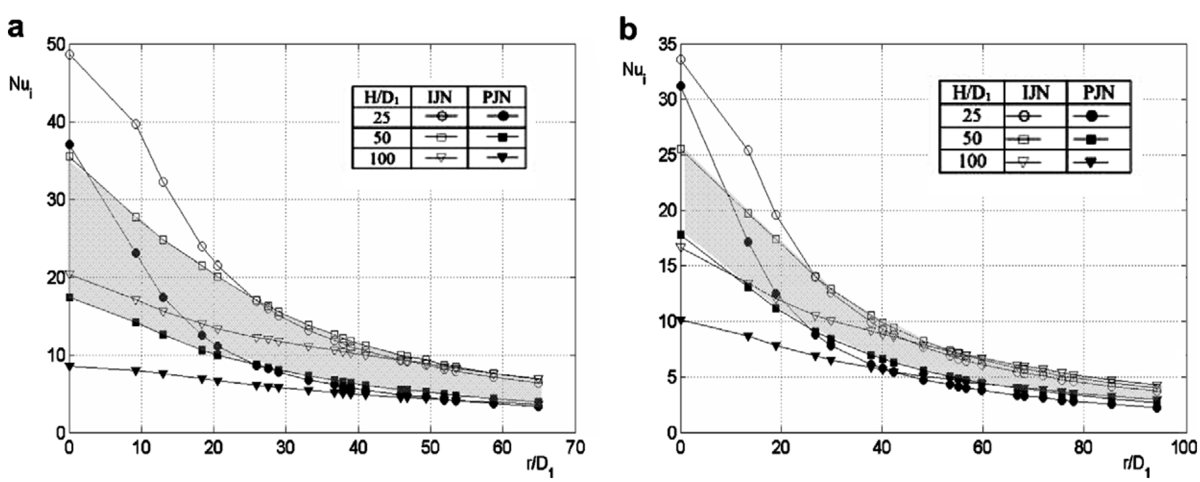

Fig. 1. Difference in Nu numbers for precessing and inline jets for two different nozzle sizes (here shown for $H/D_1 = 50$) (a) $D_1 = 5.0$ mm (figure 10 in [1]) (b) $D_1 = 3.5$ mm (figure 11 in [1]).

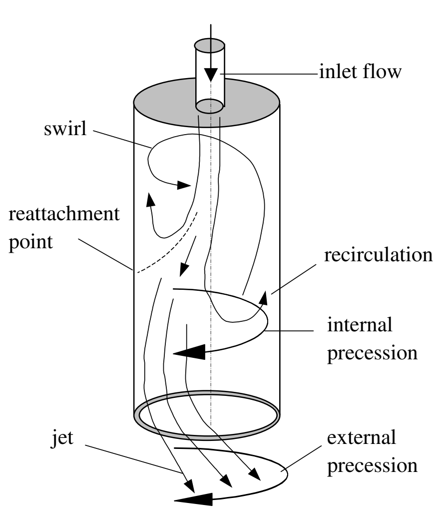

Fig. 2. Flow in a precessing jet nozzle according to [2].

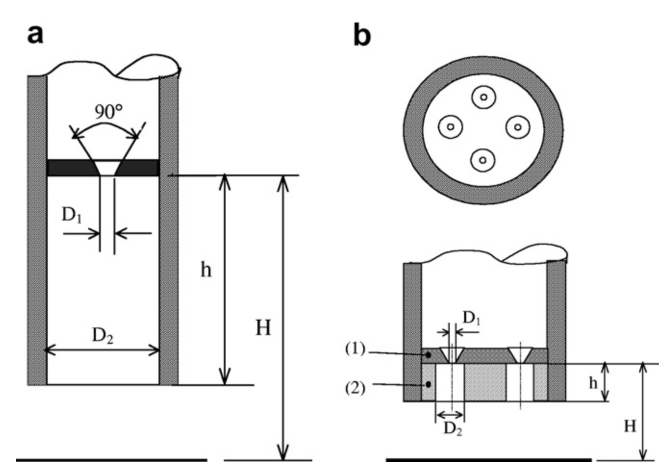

Fig. 3. Geometry of the precessing jet nozzles; $D_2/D_1 = 10$, $h/D_1 = 20$ (a) $D_1 = 3.5$ mm and $D_1 = 1.75$ mm and (b) $D_1 = 0.8$ mm but four nozzles at once.

3. The test facility

Fig. 4 is a sketch of our experimental set-up with the five main parts being the air supply, flow control, nozzle, heat transfer surface and the data acquisition system. The air flow comes from a high pressure net with a pressure reduction valve and a precision pressure control facility providing a constant airflow of adjustable rate. The temperature of the jet is close enough to that of the ambient air to neglect the temperature influence of air entrainment. A hot wire anemometry probe near the exit of the nozzle is used to detect the precessing frequency.

The heat transfer plate is an electrical circuit board with a special circuit design on both surfaces. The wall heat flux density and wall temperature can be measured simultaneously. The top surface which faces the impinging jet is covered with a densely meandering strip conductor between two current contacts. If there is uniform dissipation due to the electrical resistance of the conductor we thus get a local wall heat flux density which is almost constant. The central part of the plate is subdivided into 25 single fields of size $76 \times 76\,\mathrm{mm}^2$ so that the whole heat transfer surface covers an area of $380 \times 380\,\mathrm{mm}^2$. Around the central part there is an additional edge heating in order to reduce the heat losses towards the edges of the heat transfer plate.

For each of the 25 single fields the voltage can be measured between contacts that are led through the plate and continued to the edge of it on the rear part (bottom surface) of the board by appropriate strip conductors.

The board is heated with a constant electrical current I. We can obtain the average wall heat flux density $q_{\mathrm{SF,W}}$ for each single field (SF) from its voltage V_{SF} by subtracting the radiation and conduction heat losses from the total heat flux

$$q_{\mathrm{SF,W}} = \frac{I \cdot V_{\mathrm{SF}}}{A_{\mathrm{SF}}} - \varepsilon\sigma(T^4_{\mathrm{SF,W}} - T^4_{\infty}) - q_{\mathrm{SF,C}}. \quad (1)$$

The single field temperature $T_{\mathrm{SF,W}}$ can be determined as soon as the relationship between the electrical resistance and the temperature is known through a calibration process.

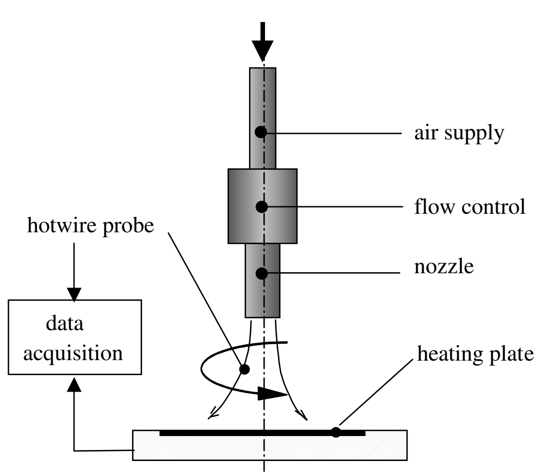

Fig. 4. Main parts of the experimental set-up.

With $q_{\mathrm{SF,W}}$ and $T_{\mathrm{SF,W}}$ from the measurements the quasi-local heat transfer coefficients Nu_{SF} can be determined that individually hold for each single field of area A_{SF} as

$$Nu_{\mathrm{SF}} = \frac{q_{\mathrm{SF,W}} D_1}{(T_{\mathrm{SF,W}} - T_{\infty})k}, \quad (2)$$

where $T_{\mathrm{SF,W}}$ is the average temperature of the single field surface and k is the thermal conductivity of the air. Here, the nozzle diameter D_1 is taken as the characteristic length in the Nusselt number. The Nusselt number with respect to more than one single field of the heating plate is the appropriate average of that of the single fields then under consideration.

4. Results and discussion

The Nusselt numbers for precessing jet impingment on the heat transfer plate are compared to those of steady inline jet nozzle (IJN) impingement which is used as reference case. The reference nozzle has the same orifice diameter D_1 as the precessing jet nozzle. Therefore, the only difference is the additional envelope tube of diameter D_2 which transfers the inline jet to a precessing jet. The Reynolds number for the inline reference nozzle is defined as $Re_{\mathrm{IJN}} = w_{\mathrm{IJN}} D_{\mathrm{IJN}}/\nu$. Here, w_{IJN} is the average velocity in the orifice of diameter $D_1 = D_{\mathrm{IJN}}$ and ν is the viscosity of air.

The nozzle is exactly positioned above the center of the heating plate with distance H/D_1 as an additional parameter. The Nusselt numbers for the central single field and for the central nine fields of the heating plate are determined in this study. Comparisons are always based on the same Reynolds number for the IJN and PJN cases. This condition is crucial though different nozzles and their IJN/PJN heat transfer relation is not strongly Reynolds number dependent, as will be shown next.

Comparing the results of the largest two nozzles, Fig. 5a and b, there is a clear trend that the difference in heat transfer between the inline and the precessing case becomes smaller. At the Reynolds number $Re = 2.5 \times 10^4$ and for the central single field, for example, we have $Nu_{\mathrm{PJN}}/Nu_{\mathrm{IJN}} = 0.49$ for the $D_1 = 3.5$ mm case, but $Nu_{\mathrm{PJN}}/Nu_{\mathrm{IJN}} = 0.81$ for the $D_1 = 1.75$ mm case. This is exactly what was found in [1] already, see Fig. 1. For a much smaller Reynolds number $Re = 10^4$ these ratios are only slightly different: 0.55 instead of 0.49 for the $D_1 = 3.5$ mm nozzle and 0.85 instead of 0.81 for the $D_1 = 1.75$ case.

The expectation and thus the hypothesis was, that the $Nu_{\mathrm{PJN}}/Nu_{\mathrm{IJN}}$-trend might prevail for still smaller nozzles so that there could be a ratio above 1 for very small nozzles. However, Fig. 5c shows that this is not the case, since again $Nu_{\mathrm{PJN}}/Nu_{\mathrm{IJN}}$ is well below 1. Comparing the three cases in Fig. 5 there is obviously no uniform trend with respect to the $Nu_{\mathrm{PJN}}/Nu_{\mathrm{IJN}}$-ratio in contrast to what we expected from our previous results.

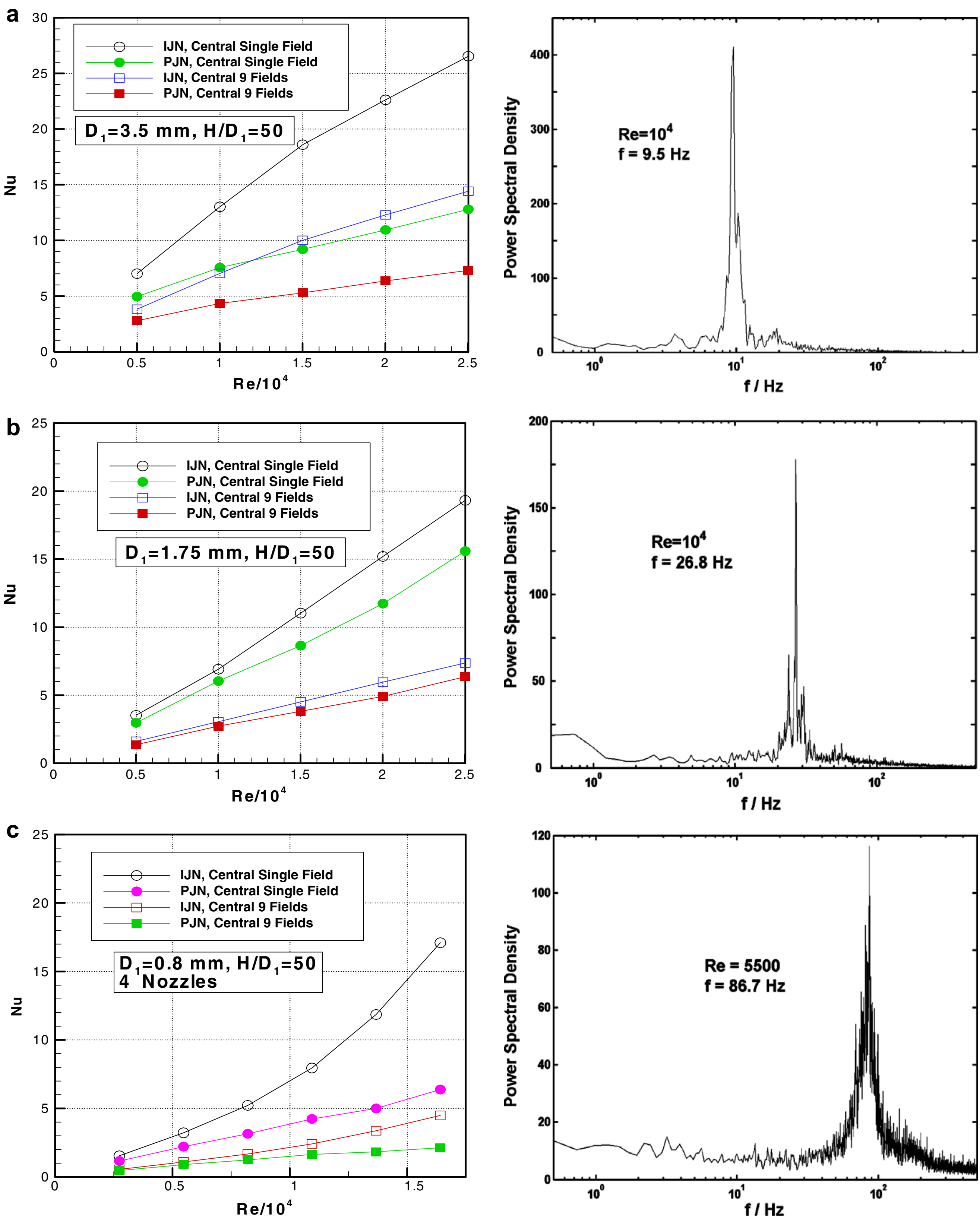

Fig. 5. Heat transfer results for the precessing jet impinging on a heating plate compared to the corresponding inline jet results (a) $D_1 = 3.5$ mm, $H/D_1 = 50$, precessing frequency $f = 9.5$ Hz at $Re = 10{,}000$; (b) $D_1 = 1.75$ mm, $H/D_1 = 50$, precessing frequency $f = 26.8$ Hz at $Re = 10{,}000$ and (c) $D_1 = 0.8$ mm, $H/D_1 = 50$, precessing frequency $f = 86.7$ Hz at $Re = 5\ 500$.

In Fig. 5, the power spectral density from which the precessing frequency can be immediately determined is shown for different Reynolds number ($Re = 10{,}000$ for $D_1 = 3.5$ and 1.75; $Re = 5500$ for $D_1 = 0.8$), since the smallest nozzle at $Re = 10^4$ had the same problem as the bigger nozzles at 5500: the precessing motion was not really stable but interrupted by non-precessing modes. Since, however, the $Nu_{\mathrm{PJN}}/Nu_{\mathrm{IJN}}$ ratio is only slightly Reynolds number dependent comparing nozzles with respect to their unsteadiness behaviour at different Reynolds numbers is not a problem.

With respect to the question whether the four-nozzle arrangement leads to a very different flow field compared to the single nozzle case it is interesting to determine the Strouhal number $Sr = fD_1/w$ which formally can be rewritten as $Sr = fD_1^2/\nu Re$. With the results of Fig. 5, we get $Sr = 7.6 \times 10^{-4}$ for the biggest nozzle, $Sr = 5.4 \times 10^{-4}$ for the middle sized one and $Sr = 6.6 \times 10^{-4}$ for the smallest nozzle. Since Sr for the smallest nozzle lies between the values for the two bigger ones, the physics of the four-nozzle arrangement probably is not fundamentally different from the single nozzle case.

With all our results in mind we finally conclude: unfortunately, for very small precessing jet nozzles the same holds as for the bigger ones. There is no increase in heat transfer compared to that of the corresponding inline jet nozzle. This again shows what was found in many studies concerning heat transfer under impinging jets: it is hard to beat the performance of the simple inline jet.

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