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Influence of a pulsation on heat transfer and flow structure in submerged impinging jets

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Abstract

An experimental investigation on pulsating impinging jets has been performed. The effect of the pulsation on the flow structure and heat transfer have been investigated. Frequency and amplitude were varied separately and the effect of each parameter was examined for different Reynolds numbers and nozzle-to-plate distances.

The jet was found to become broader and the core jet length smaller with the pulsation. The reason for this behavior is that pulsation enhanced entrainment of air into the jet, which results in a change of mean velocity of the jet. Nevertheless, the behavior at lower frequencies (up to 140 Hz) is still quasisteady. This means that the amplitude of the pulsation behaves similar to the mean velocity of the jet, that the shapes of the velocity profiles are comparable to steady jets and that the jet behavior is independent of frequency.

At moderate frequencies heat transfer is only affected by the pulsation when nozzle-to-plate distance and amplitude are large enough. At small nozzle-to-plate distances enhanced entrainment has no influence and no difference between steady and pulsating jets can be recognized. At large nozzle-to-plate distances entrainment increases and jet velocity reduces. This yields a reduction of heat transfer in the stagnation point of up to 50%.

But besides of this effect of enhanced entrainment a theoretical limit could be determined, above which the jet is not anymore quasisteady. Above $Sr = 0.2$ heat transfer is affected by the pulsation also at small nozzle-to-plate distances. At this frequency boundary layer is also affected by the pulsation. This yields increased heat transfer coefficients at the stagnation point. For larger nozzle-to-plate spacings this effect is superposed by the reduction of heat transfer due to increased entrainment, resulting in a strong decrease of heat transfer coefficient.

Keywords: Jet impingement; Heat transfer; Intermittency; Pulsation; Experiment; Flow structure

1. Introduction

The mechanisms of convective heat transfer in steady single phase flows has been widely examined, but only few knowledge exists about how periodic intermittency affects heat transfer. In many technical applications intermittent flow occurs due to moving parts, like in pumps or turbines or by vibrations or flow oscillations. It is still not clear, which mechanisms take place and how heat transfer is influenced by these phenomena. While for pipe flow only a small influence has been observed [8], for free shear flows no information is available. In order to approach this problem an experimental investigation on the influence of a pulsation on heat transfer and flow structure in impinging jets has been performed. Heat transfer measurements have been carried out by means of thermography and flow measurements have been performed with a laser-doppler-velocimeter. The investigation on the influence of frequency and amplitude on heat transfer at different Reynolds numbers and nozzle-to-plate-spacing has

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Nomenclature

Latin symbols
- A amplitude (m/s)
- D nozzle diameter (m)
- f frequency (Hz)
- H nozzle-to-plate spacing (m)
- Nu Nusselt number ($=\frac{\alpha D}{\lambda}$)
- K number of classes (–)
- Pu pulsation intensity (–)
- r radial distance from stagnation point (m)
- Re Reynolds number ($=\frac{uD}{\nu}$)
- Sr Strouhal number ($\frac{fD}{u}$)
- u velocity component in axial direction (m/s)
- z axial distance from nozzle (m)

Greek symbols
- α heat transfer coefficient (based on $T_{\mathrm{plate}} - T_{\mathrm{nozzle}}$) (W/(m^2 K))
- λ thermal conductivity (W/(m K))
- ν kinematic viscosity (m^2/s)
- Φ phase shift (°)

Subscripts
- N normalized on mean axial jet exit velocity
- k value averaged over particular phase angle range
- eff effective (based on RMS, not on maximum deviation)

Superscripts
- $'$ turbulent velocity fluctuation
- $-$ mean velocity

resulted in a guideline, how far pulsating jets can be treated like steady jets and which changes occur above this limit.

2. Background

Submerged impinging jets have been examined widely in the second half of the last century. Numerous works on this topic can be found in the literature, which cover the main influence factors on heat transfer and flow structure. Overviews can be found in [1–4]. Newer investigations on steady jets, performed by the authors of the present contribution, can be found in [5]. Already since the forties of the last century the influence of a pulsation on convective heat transfer has been examined. Most of the works deal with a tube flow with a superposed pulsation. A comprehensive illustration can be found in [6,7]. When water as medium is used, cavitation can occur in cases with flow reversal. This causes a drastic increase in heat transfer, but in systems with no flow reversal or in gaseous jets influence of pulsation on heat transfer can be neglected. Fallen [8] has found no influence of pulsation on heat transfer in laminar flow. In turbulent flow he has observed slight increase or decrease, depending on frequency. Fallen has assumed that pulsation influences turbulence intensity and flow in the inlet region, which results in slight changes in heat transfer.

Mladin [9–12] has investigated, how heat transfer in impinging jets is influenced by a pulsation. An analytical model, which describes the response of the boundary layer on changes in the mean flow, predicts a decrease of heat transfer by the pulsation of up to 17% over a large range of frequencies. Only in cases of low amplitude and high frequency a slight increase of 1% in heat transfer has been observed. The results have been validated with experimental data at Reynolds numbers of up to 14,000 and frequencies of up to 70 Hz.

The influence of pulsation on axisymmetrically impinging jets has been examined by Vejrazka [13] at $Re = 10{,}000$ with an amplitude of 1% of the mean flow. He has performed numerous investigations on the temperature distribution in the jet, vortex generation and flow structure, but he has not found any influence of pulsation on heat transfer. Camci and Herr [14] have performed measurements with self-oscillating jet at large distances from the nozzle. The jet became broader, compared to the steady jet. At Reynolds numbers of 7500–14,000 and very large nozzle-to-plate distances ($H/D = 24$–60) strong increases in heat transfer has been observed.

From these investigations it can be concluded that there is still the need for an investigation on the main influence parameters and mechanisms of pulsating flow, especially with large shear layers.

3. Experimental setup

The experimental setup is sketched in Fig. 1. Details about the setup and the procedure are described in detail in [5,15,16]. The experiments were performed with a single round nozzle with a diameter of 25 mm at the exit, where the jet is contracted from a diameter of 50 mm, resulting in a mean velocity of 49 m/s for $Re = 78{,}000$. Nozzle and plate are surrounded by a cylindrical chamber with a diameter of $33 \cdot D_{\mathrm{Nozzle}}$ to ensure, that the experiment is affected by external influences. By initial CFD calculations, it could be shown, that at this size, the flow in observed area is not influenced by the surrounding. The temperature in the far field of the jet was recorded. It was determined to be about

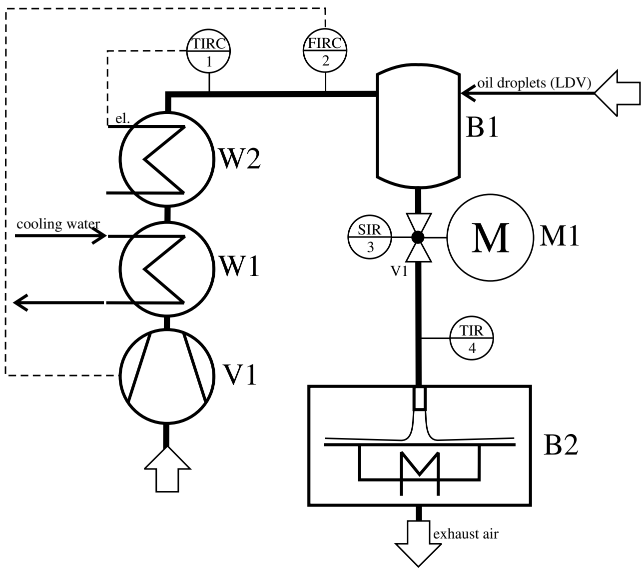

Fig. 1. Experimental setup.

1 K higher than the jet temperature, resulting in an entrainment factor of less than 0.1. Striegl and Diller [17] shown, that an entrainment factor of 0.8 lead in their experiments to a reduction in heat transfer of 20%. As the entrainment factor in the present experiments is roughly one dimension smaller, it is concluded that the effects of that slightly higher temperature can be neglected for the present experiments.

The determination of the heat flux density has been realized by heating a glass plate on its bottom face by steam to ensure constant temperature, which was measured with resistance thermometers. The top face of the glass plate is cooled by the jet. The temperature distribution on the top of the plate is recorded with an infrared camera. The camera was calibrated before each run by taking an image of an isothermal plate with known temperature. Both plates were painted with the same black paint to avoid effects of different emission factors. With the temperature of the bottom face of the glass plate and the temperature distribution of the top face of the glass plate the temperature gradients within the glass plate and the heat flux densities from the glass plate to the jet could be determined:

$$\dot{q}(r) = \frac{\lambda_{\text{plate}}}{s_{\text{plate}}}\left(T_{\text{plate,top}}(r) - T_{\text{plate,bottom}}\right) \tag{1}$$

A sinosoidaly alternating flow was superposed on the mean flow. The amplitude and frequency of the pulsation were adjusted separately from the mean flow parameters. This was realized with a modified version of a pulsation facility developed by Büchner [18], which is based on a rotating cage periodically opening and closing an orifice in a surrounding cylinder. A detailed description can be found in [16]. With this facility frequencies of 1–750 Hz have been investigated.

For measuring the local jet velocity at selected points a laser-doppler-velocimeter has been used. For these mea-

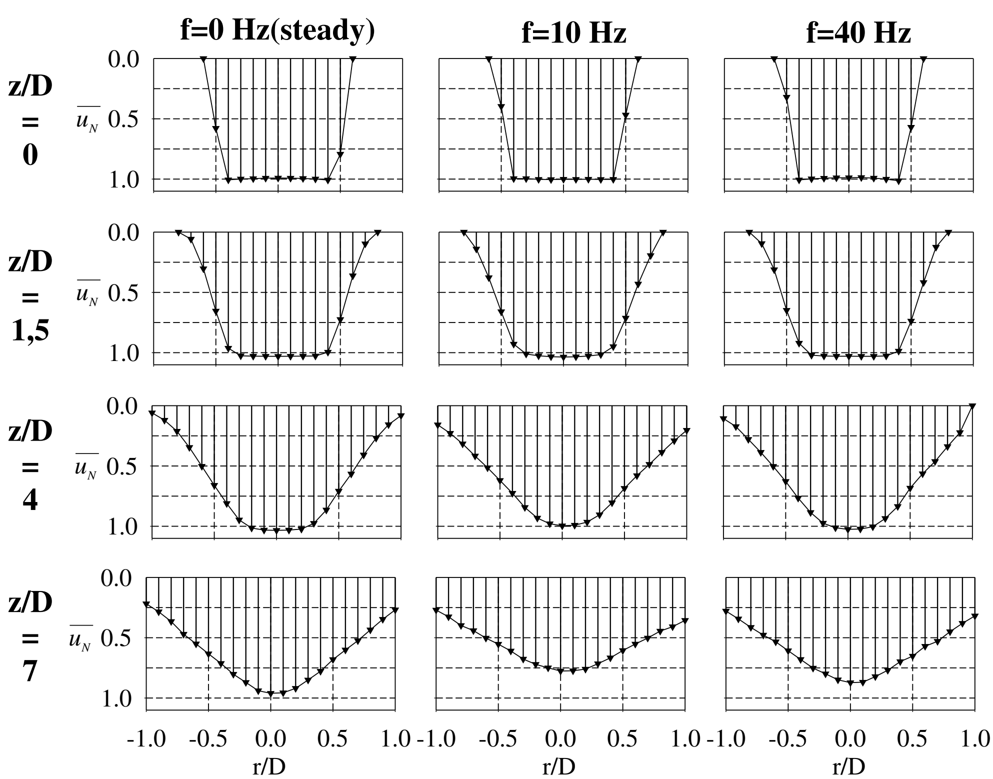

Fig. 2. Normalized velocity profiles in steady and pulsating free jets at different axial distances from the nozzle ($Re = 78{,}000$).

surements upstream of the flow oil droplets were inserted as tracer particles.

A method was developed to split the flow signal into mean flow, periodical (pulsating) parts and stochastically fluctuating (turbulent) parts. In this method, described in detail in [5,16], the pulsation period is divided into 30 classes. Together with each velocity signal the time since the beginning of the last pulsation period is recorded. According to this time the signal is referred to one of the classes. The mean value of each class and therefore a mean velocity versus the phase angle of the pulsation can be determined. When the mean values of all classes are averaged, a mean velocity is determined, the standard deviation gives the effective pulsation amplitude. The standard deviation in each class gives the turbulence intensity at each phase angle and its average value the mean turbulent fluctuations at this location. The mean velocity at any location in the jet is normalized with the mean velocity at the nozzle exit and labelled as

$$\overline{u_{\mathrm{N}}}. \tag{2}$$

The turbulent fluctuation is also normalized with the mean velocity at the nozzle exit and labelled as

$$\overline{u'_{\mathrm{N}}}. \tag{3}$$

A detailed description of the procedure can be found in [5].

Another interesting information is, how the amplitude of the pulsation changes in the flow. The amplitude is characterized by an effective amplitude, determined as root mean square of the phase-averaged mean velocity and its mean over time:

$$A_{\mathrm{eff}} = \sqrt{\frac{1}{K-1}\sum_{k=1}^{K}[\bar{u}_k - \bar{u}]^2}. \tag{4}$$

when this is normalized with the local mean velocity, one obtains an effective amplitude

$$Pu_{\mathrm{eff}}(r,z) = \frac{A_{\mathrm{eff}}(r,z)}{\bar{u}(r,z)}. \tag{5}$$

Another interesting parameter for analyzing turbulent pulsating flow is the phase shift of the pulsation at a location in the flow, compared to the pulsation at the nozzle exit. This is done by normalizing the velocity curves and shifting the curve manually until it is identically with the curve at the exit of the nozzle.

4. Flow structure

4.1. Evolution of mean flow parameters

Fig. 2 shows the distribution of mean axial velocity in a free jet at frequencies of $f = 10$ Hz and $f = 40$ Hz at a constant pulsation intensity of $Pu_{\mathrm{eff}} = 30\%$ and $Re = 78{,}000$. In the next figures, turbulence intensity (Fig. 3), pulsation intensity (Fig. 4) and phase shift (Fig. 5) along the jet are illustrated in comparison with the steady jet values. The turbulence in the core jet is only slightly increased (Fig. 3), but entrainment of air from the environment into

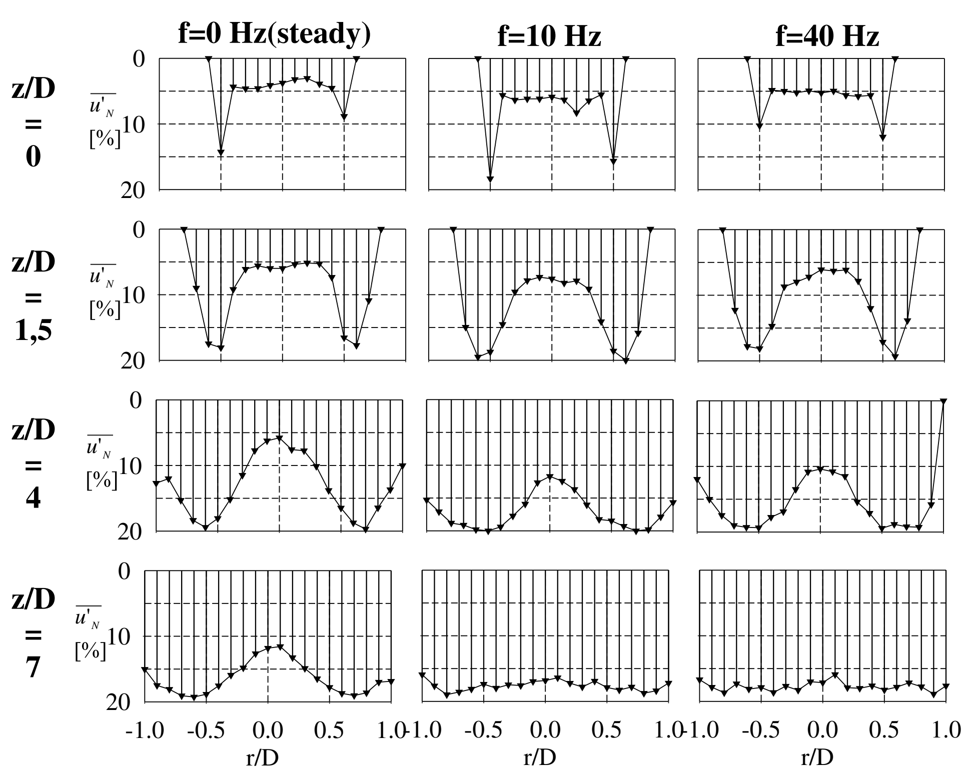

Fig. 3. Normalized turbulence profiles in steady and pulsating free jets at different axial distances from the nozzle ($Re = 78{,}000$).

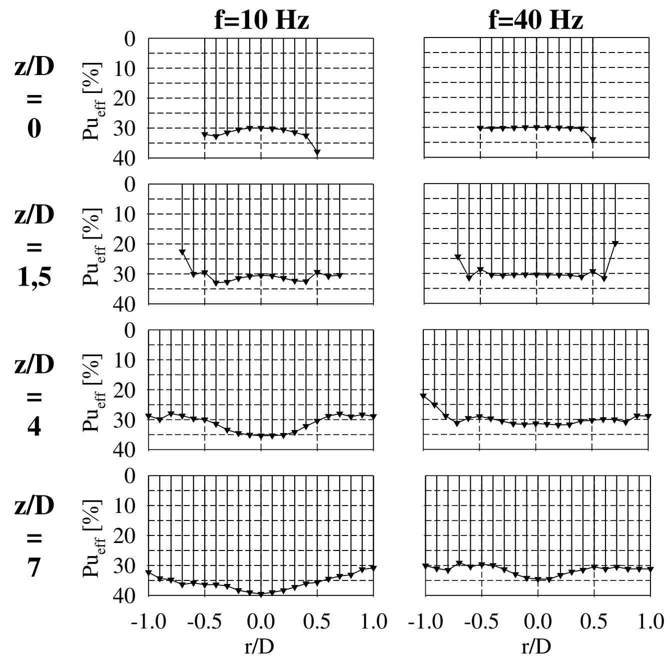

Fig. 4. Evolution of effective normalized amplitude in pulsating free jets at different axial distances from the nozzle ($Re = 78{,}000$).

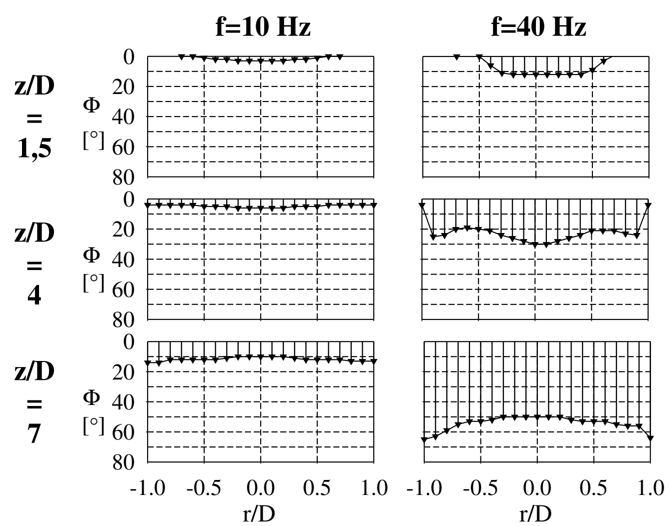

Fig. 5. Evolution of phase shift of the pulsation at different axial distances, compared to the exit of the nozzle ($Re = 78{,}000$).

the jet is enhanced by the pulsation. The jet becomes broader and the core jet becomes shorter. This can be seen in comparison of the velocity profiles, where the gauss-curve-like velocity profile is achieved at smaller nozzle-to-plate distances than in the steady jet. Also the turbulence intensity becomes more homogeneous in the pulsating jets than in the steady jet. Pulsation intensity (Fig. 4) remains nearly constant in the jet. This means that the amplitude of the pulsation reacts like the mean velocity and that its decay follows the same rules as the decay of the mean velocity. When the phase shift of the pulsation is regarded, it can be recognized that on the one hand side the phase transition increases with increasing axial distance from the nozzle and with increasing frequency, and that on the other side a slightly increased phase shift occurs in the boundary regions of the jet. When a signal speed of the pulsation is calculated as

$$u_s = z \cdot \frac{360^\circ}{\Phi} f \tag{6}$$

it can be seen that it has the same order of magnitude as nozzle exit velocity and is far away from sound speed, as it would be expected for self-similar spreading of the pulsation. This is a sign for the existence of vortex rolls in pulsating free and impinging jets, also for small frequencies. It is assumed that a better agreement is achieved, when the signal speed is compared to a mean velocity along the steam line. But for this investigation knowledge of the whole flow field would be necessary, which could not be performed in the present work with the point measurement technique.

This phenomenon shows that the mechanisms in a pulsating jet are completely different from pulsations in channel flow. It shows that the flow pulses mainly perpendicularly to the mean flow direction and not along the mean flow direction as it is done in channel flows. This enhances mixing between jet and environment drastically and affects mean flow and turbulence intensity of the jet, as shown before. Therefore it is clear that pulsation can affect heat transfer for those nozzle-to-plate distances, where the shear surface of the jet is large enough to enable mixing down to the centerline.

4.2. Influence of frequency and influence of impingement plate

In the following part dependency of pulsating jet heat transfer on pulsation frequency was investigated. At three

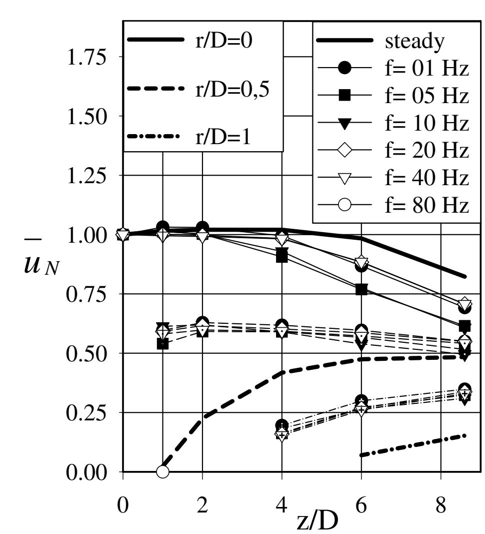

Fig. 6. Influence of pulsation frequency on axial mean velocity along three vertical lines in a free jet ($Re = 78{,}000$).

radial distances ($r/D = 0$ (centerline), $r/D = 0.5$ (below nozzle boundary) and at $r/D = 1$), the jet was examined in vertical directions. Fig. 6 shows the evolution of the axial mean velocity versus the axial distance from the stagnation point for a Reynolds number of $Re = 78{,}000$ and a pulsation intensity of $Pu_{\text{eff}} = 30\%$. A slight increase of axial velocity by deviations of the nozzle shape from a quarter-circular shape, the decrease of core jet length and the increased jet diameter are clearly visible. An interesting aspect is, that these effects are nearly independent of the pulsation frequency, as long as pulsation intensity is kept constant. The curves for one radial distance come together at two lines. Only the profiles of the two neighboring frequencies of ($f = 5$ Hz and $f = 10$ Hz) differ from the other measurements. This is in agreement with the measurements of the profiles, where frequencies of $f = 10$ Hz and $f = 40$ Hz were examined (Fig. 2). In Fig. 7 it is shown that this phenomenon can be transferred to impinging jets. For all variables, there is nearly no influence of the plate visible at a distance of 1 nozzle diameter above the plate.

4.3. Extension to higher frequencies

In Fig. 8 the flow structure at $Re = 78{,}000$, $Pu_{\text{eff}} = 30\%$ and a frequency of $f = 500$ Hz is illustrated. $\overline{u_N}$, $\overline{u'_N}$ and Pu_{eff} are given for axial distances of $z/D = 4$ and $z/D = 7{,}5$. The flow structure has significantly changed, compared to a steady jet and a jet pulsating at low frequencies (see Fig. 2). The profile of mean axial velocity no more exhibits a Gaussian-like shape. Turbulence intensity is

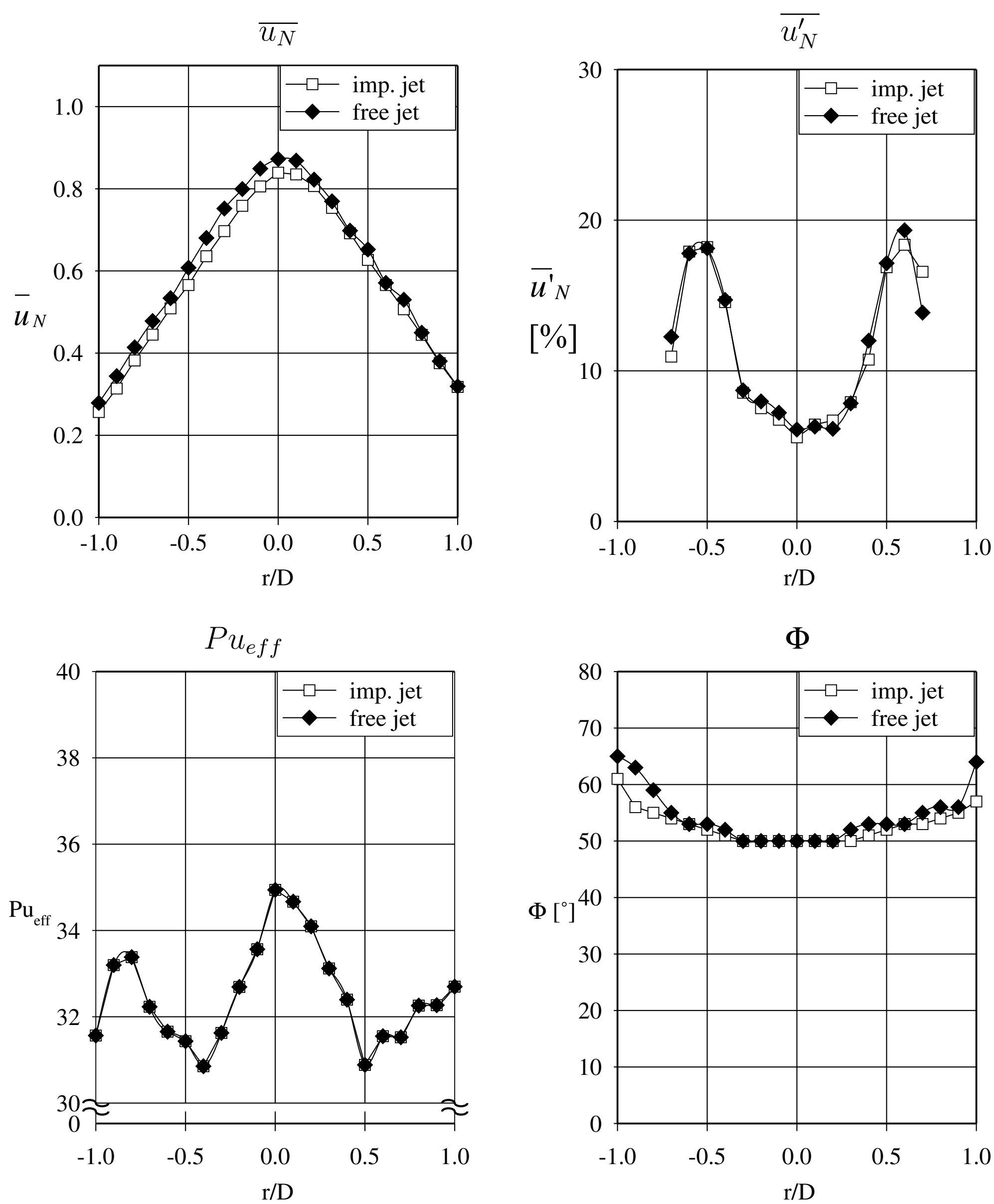

Fig. 7. Influence of an impingement plate on flow characteristics of a pulsating flow at a distance of 1 nozzle diameter above the plate ($Re = 78{,}000$, $f = 40$ Hz, $Pu_{\text{eff}} = 30\%$).

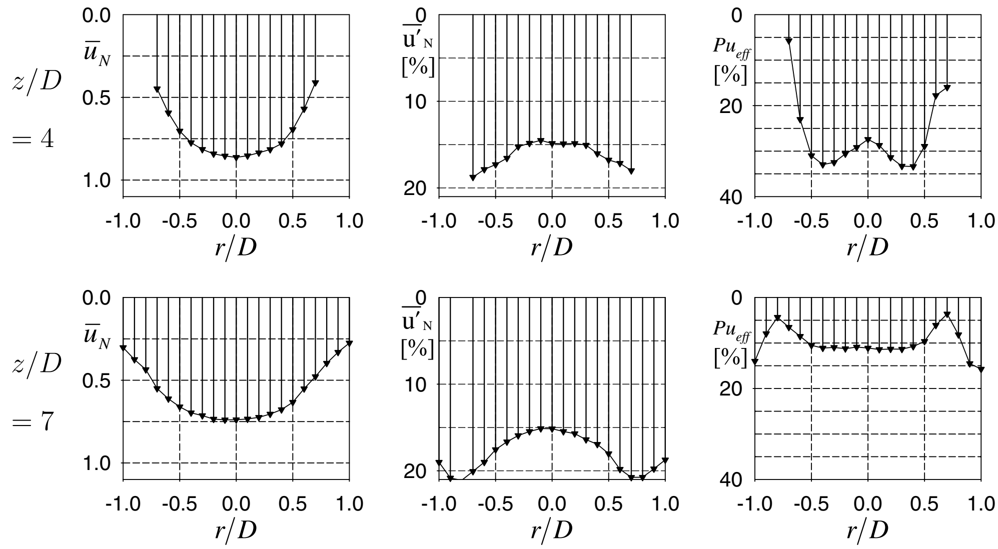

Fig. 8. Flow characteristics in a free jet pulsating at high frequency ($Re = 78{,}000$, $Pu_{eff} = 30\%$, $f = 500$ Hz).

increased, compared to the steady and low-frequency pulsating jet. The amplitude is dampened in the jet, while it remained constant in the low-frequency-pulsating jet.

It can be concluded that the pulsation at $f = 500$ Hz cannot be treated as quasisteady state behavior. Therefore it is assumed that a limit exists, where the pulsation can be treated as quasisteady.

5. Heat transfer

It was examined, how heat transfer in impinging jets is affected by a pulsation. Reynolds number (Re), nozzle-to-plate distance (H/D) and radial distance from the stagnation point (r/D) and the frequency of the pulsation (or the Strouhal number respectively) and the pulsation amplitude (Pu_{eff}) have been varied. The examinations were performed for Reynolds numbers of $Re = 14{,}000$, $Re = 34{,}000$ and $Re = 78{,}000$, for nozzle-to-plate distances of $H/D = 2$, $H/D = 5$ and $H/D = 8.5$ at effective normalized amplitudes of $Pu_{eff} = 3.5\%$, $Pu_{eff} = 15\%$ and $Pu_{eff} = 30\%$. In the first step the frequency was varied from $f = 2$–140 Hz. In a second step the examination was extended to frequencies up to 750 Hz. In the following subsections the influences of the main parameters are discussed. For clear identification of the effect of the pulsation the corresponding steady jet was measured for each series.

5.1. Influence of frequency

It became clear that for the frequencies $f = 2$–140 Hz and for large nozzle-to-plate spacings (H/D), which were examined in the first step, pulsation reduces heat transfer coefficient. Fig. 9 shows a comparison of local Nusselt number in a steady jet with local Nusselt numbers in a pulsating jet at a large nozzle-to-plate distance ($H/D = 8.5$). At a Reynolds number of $Re = 34{,}000$ and a pulsation

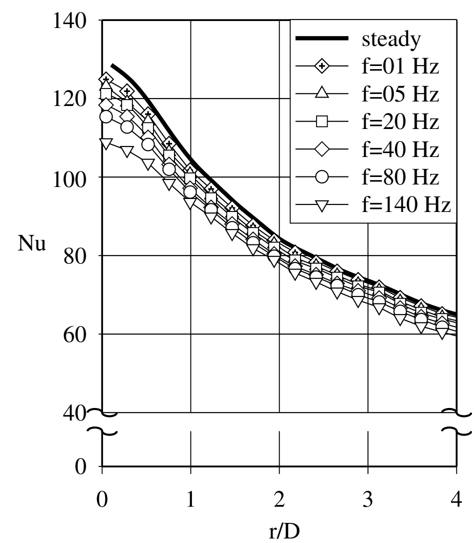

Fig. 9. Influence of pulsation frequency on heat transfer coefficient ($Re = 34{,}000$, $H/D = 8.5$, $Pu_{eff} = 15\%$).

intensity of $Pu_{eff} = 15\%$ frequency was varied. A reduction of heat transfer coefficient of up to 20%, compared to the steady state is visible. The same effect can be seen at $Re = 14{,}000$ (Fig. 10).

5.2. Influence of pulsation intensity

At small pulsation amplitudes no or only very small influence is visible (Fig. 11), whereas significant differences from the steady state are visible for medium (Fig. 9) and large pulsation intensities (Fig. 12).

With increasing pulsation intensity the amplitude and therefore pressure fluctuation increases also in radial direction. Mixing between the jet and the environment is enhanced, which causes a reduction of axial mean velocity

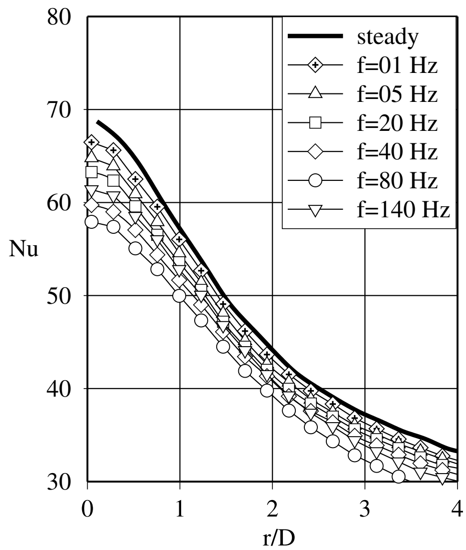

Fig. 10. Influence of pulsation frequency on heat transfer coefficient ($Re = 14{,}000$, $H/D = 8.5$, $Pu_{\mathrm{eff}} = 15\%$).

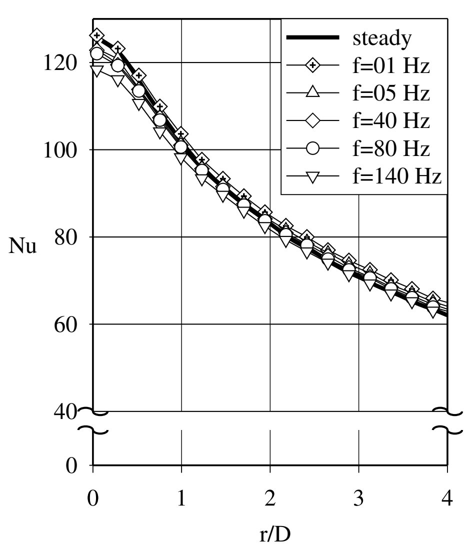

Fig. 11. Influence of pulsation on heat transfer coefficient ($Re = 34{,}000$, $H/D = 8.5$, $Pu_{\mathrm{eff}} = 3.5\%$).

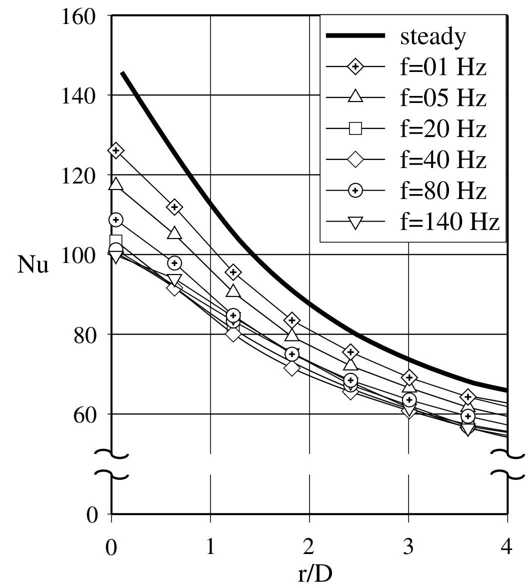

Fig. 12. Influence of pulsation on heat transfer coefficient ($Re = 34{,}000$, $H/D = 8.5$, $Pu_{\mathrm{eff}} = 30\%$).

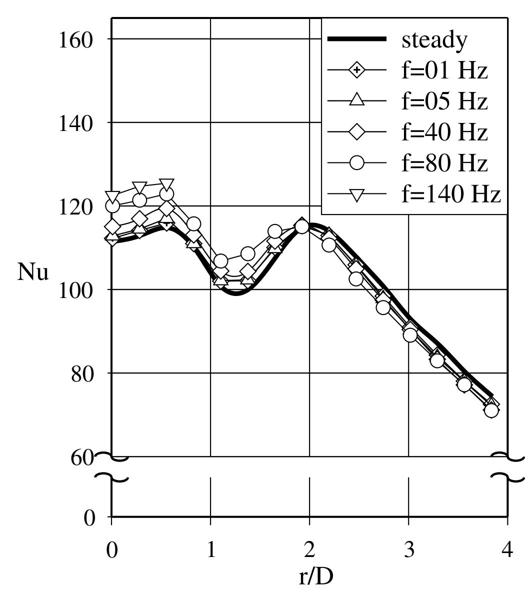

Fig. 13. Influence of pulsation on heat transfer coefficient ($Re = 34{,}000$, $H/D = 2$, $Pu_{\mathrm{eff}} = 30\%$).

of the inner jet and reduces temperature difference between the jet and the impingement plate.

5.3. Influence of nozzle-to-plate distance

At $Re = 34{,}000$ and $Pu_{\mathrm{eff}} = 30\%$ the influence of a pulsation is negligible for small nozzle-to-plate distances ($H/D = 2$, Fig. 13). In the stagnation point even a small enhancement of heat transfer was recognized. With increasing nozzle-to-plate distance the reduction of the Nusselt number becomes larger, compared to the steady jet. At a nozzle-to-plate distance of $H/D = 8.5$ (Fig. 12) the influence of a pulsation becomes larger. Here a reduction of up to 30% was observed.

A reason for this phenomenon is, that at small nozzle-to-plate distances enhanced entrainment of air from the environment and the decrease of jet velocity still has no influence on the core region of the jet, which mainly affects near wall flow. With increasing distance between nozzle and plate the area between jet and environment and therefore the effect of entrainment increased. At $H/D = 8.5$ (Fig. 9) this becomes clear. The entrained air has reached the jet centerline. Heat transfer decreases.

5.4. Influence of Reynolds number

With the experiments of this section it was examined, how a pulsation effects heat transfer at different Reynolds numbers. It became clear, that heat transfer affects heat transfer at different Reynolds numbers similarly. This is concluded from the experiments already presented in Fig. 10 for $Re = 14{,}000$ and in Fig. 9 for $Re = 34{,}000$ and a further investigation for $Re = 78{,}000$ (Fig. 14). In all cases the reduction of heat transfer is clearly visible. Also the slight increase of heat transfer in the stagnation point at small nozzle-to-plate distances occurs at the different Reynolds numbers. Fig. 15 shows the influence of a strong pulsation ($Pu_{\mathrm{eff}} = 30\%$) on heat transfer at a small nozzle-to-plate distance ($H/D = 2$) at $Re = 78{,}000$. The slight increase near the stagnation point and the independency of pulsation in the wall jet region, as recognized in other experiments can be seen here.

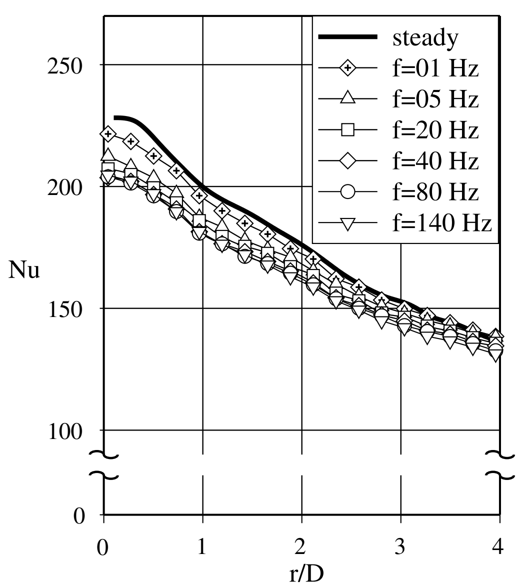

Fig. 14. Influence of pulsation on heat transfer coefficient ($Re = 78{,}000$, $H/D = 8.5$, $Pu_{\mathrm{eff}} = 15\%$).

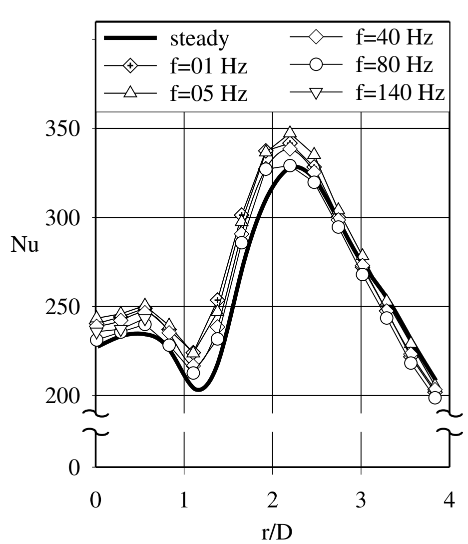

Fig. 15. Influence of pulsation on heat transfer coefficient ($Re = 78{,}000$, $H/D = 2$, $Pu_{\mathrm{eff}} = 30\%$).

5.5. Extension of the range of parameters: high frequencies

It was the aim of this part of the investigation, to test whether the results from the experiments in the previous sections can be transferred to higher frequencies. A theoretical assessment of a possible limit for the quasisteady approach has been performed: Heat transfer can only be affected by a pulsation, when the frequency of the pulsation has the same order of magnitude as the vortex frequency of the turbulence. If it is too small, it cannot cause any further enhancement, compared to the turbulence. The frequency of the turbulent vortices in the jet, which cause the mixing effects is assessed by the following approach: Vortices, created at the boundary of the nozzle, are transported convectively along the jet. To enable high mixing effects, the period of a vortex must be smaller than the residence time from the exit of the nozzle to the tip of the core jet. The same frequencies, causing the decay of the core jet, affect heat transfer from the plate to the jet. Therefore the frequency of the pulsation must be in the range of this frequency to affect boundary layer thickness and heat transfer. The length of the core jet is given by 4–6 nozzle diameters, depending on the author. For a mean value of 5 nozzle diameters the residence time can be calculated as

$$t_{\mathrm{res}} = \frac{5D}{u_{\mathrm{N}}}. \tag{7}$$

With a critical frequency of $f_{\mathrm{crit}} = 1/t_{\mathrm{res}}$ a critical Strouhal number can be determined:

$$Sr_{\mathrm{crit}} = \frac{f_{\mathrm{crit}} D}{u_{\mathrm{N}}} = \frac{u_{\mathrm{N}} D}{5D \cdot u_{\mathrm{N}}} = 0.2 \tag{8}$$

can be determined. Below this number the quasisteady behavior is expected, while above this value an influence of pulsation on boundary layer thickness and therefore on heat transfer is assumed. For the system in this examination the critical Strouhal number is reached at 400 Hz for $Re = 78{,}000$ and at $f = 175$ Hz for $Re = 34{,}000$. For testing this assumption experiments at frequencies of $f = 100$ Hz to $f = 750$ Hz have been performed. Due to the high damping of the high frequency pulsation already between pulsation generator and nozzle exit, it was not possible, to obtain high pulsation intensities at high frequencies.

Fig. 16 shows a variation of pulsation frequency at a pulsation intensity of up to $f = 275$ Hz at a pulsation intensity of $Pu_{\mathrm{eff}} = 30\%$ at $Re = 34{,}000$ and $H/D = 2$. In opposite to low-frequency pulsation the shape of the curves is affected by the pulsation. At a frequency of $f = 250$ Hz stagnation point heat transfer is significantly increased. The shape of the curves looks similar to the curves when varying nozzle-to-plate distance. It can be assumed that the pulsation at high frequencies reacts like an increase in turbulence intensity, while low-frequency pulsation only effects mixing between jet and environment. The frequency, where the enhancement of heat transfer starts, is in the order of magnitude of the theoretically determined critical

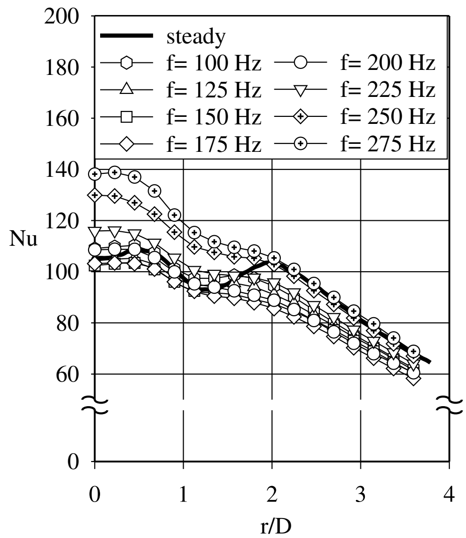

Fig. 16. Influence of pulsation on heat transfer coefficient ($Re = 34{,}000$, $H/D = 2$, $Pu_{\mathrm{eff}} = 30\%$).

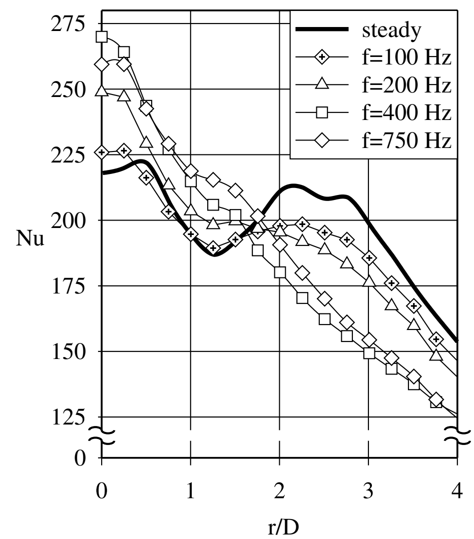

Fig. 18. Influence of high frequency pulsation on heat transfer coefficient ($Re = 78{,}000$, $H/D = 5$, $Pu_{\mathrm{eff}} = 15\%$).

frequency of $f_{\mathrm{crit}} = 175$ Hz. An analog behavior can be seen in Fig. 17 for $Re = 78{,}000$ at a pulsation intensity of $Pu_{\mathrm{eff}} = 3.5\%$ and a nozzle-to-plate distance of $H/D = 5$. Enhancement of heat transfer takes place at a frequency of $f = 400$ Hz. The theoretical limit was determined for this case at $f_{\mathrm{crit}} = 400$ Hz. Also at a pulsation intensity of $Pu_{\mathrm{eff}} = 15\%$ an increase in heat transfer could be observed in this range (Fig. 18).

When the nozzle-to-plate distance is further increased these effects are superposed by the enhanced mixing of air from the environment, which cause a reduction in heat transfer coefficients. At $Re = 34{,}000$, $Pu_{\mathrm{eff}} = 15\%$ and $H/D = 8.5$ a significant reduction of the Nusselt number was observed (Fig. 19).

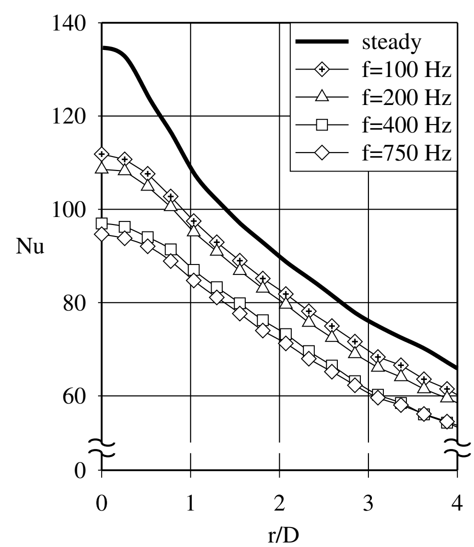

Fig. 19. Influence of high frequency pulsation on heat transfer coefficient ($Re = 34{,}000$, $H/D = 8.5$, $Pu_{\mathrm{eff}} = 15\%$).

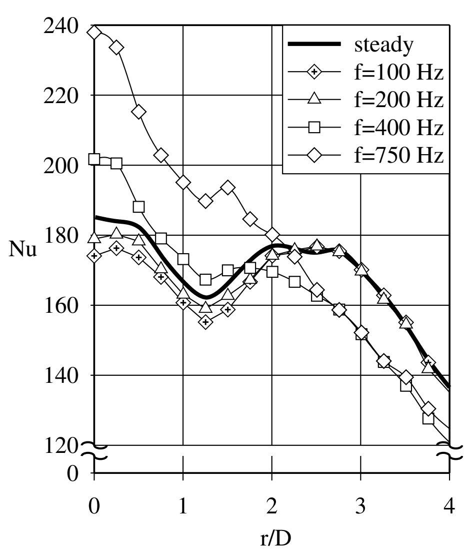

Fig. 17. Influence of high frequency pulsation on heat transfer coefficient ($Re = 78{,}000$, $H/D = 5$, $Pu_{\mathrm{eff}} = 3.5\%$).

6. Conclusions

Convective heat transfer can be influenced by periodic fluctuations in the mean flow. The main mechanism, which was identified is, that the pulsation enhances mixing between the jet and the environment and yields to a reduction in the jet velocity. This becomes important for large nozzle-to-plate-distances, where the free shear layers are large. At small nozzle-to-plate distances nearly no influence on heat transfer was detected, when the frequency is low enough. For large frequencies heat transfer can be enhanced by the pulsation for small nozzle-to-plate distances, when the pulsation frequency is in the order of magnitude of the turbulence. The threshold frequency for this transition has been determined as $Sr = 0.2$.

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