THERMOHYDRAULIC EFFICIENCY ASSESSMENT FOR PULSATING FLUID FLOW IN CHANNELS

Sandra Sidenko, Egils Dzelzitis, Natalia Sidenko
Riga Technical University, Latvia
sandysid@inbox.lv, egils.dzelzitis@rtu.lv, nsidenko@inbox.lv

Abstract. Superimposed flow pulsations promote heat transfer intensification. Usage of such flows allows to increase the thermal characteristics aimed at the energy saving. However, the question of the thermohydraulic efficiency of this method remains open, since there are little data of changing in hydraulic resistance in the case of non-isothermal pulsating flow in the channels. To solve this problem, CAD/CFD software SolidWorks/ FlowSimulation was used. Smooth and profiled channels were investigated. A model of periodically pulsating fluid flow in channels with sinusoidal velocity pulses was considered. Pulsating frequency range was set from 0 to 30 Hz. The results showed that a periodic change in the Reynolds number leads to a change in the value of hydraulic resistance over time. At a flow pulsation frequency of $0.5 \leq f \leq 5$ Hz and $2.8 \leq \lg Re \leq 3.5$ for a smooth channel there are characteristic zones of a laminar ($Re \approx 2000$) and transient flow regime (Re = 2000-4000), and a turbulent zone at $3.5 \leq \lg Re \leq 4.7$. With increasing of the flow pulsations frequency $f > 5$ Hz, there are no zones of laminar and transient regimes, and the flow can be characterized as a stabilized turbulent flow. In contrast to the pulsating flow in a smooth channel, in a profiled channel a stabilized turbulent flow appears already at a pulsation frequency $f = 2$ Hz ($Sh = 0.09$). The relative coefficient of hydraulic resistance for non-isothermal fluid movement in a profiled channel is 10-15% higher than in a smooth one. Increasing in the thermohydraulic efficiency, both in smooth and profiled channels, can reach up to 20 - 40%.

Keywords: thermohydraulic efficiency, heat transfer, hydraulic resistance, flow pulsation, profiled channel.

Introduction

At present, the hydraulic resistance and heat transfer in smooth circular channels at a stationary developed turbulent flow have been studied in detail.

Obtained experimental results in hydraulic losses for isothermal fluid flow are well described by well-known dependencies, such as Poisel's law, Blasius's law of resistance and Nikuradze's equation for various ranges of the Reynolds criterion [1; 2].

Also known are modern studies [3-10] of hydrodynamics and heat transfer in profiled channels at a steady flow of the media, where it is shown that an artificially created profile in the channels, which can be made in the form of ribs, concavities, swirlers, etc., makes it possible to significantly intensify of the heat transfer.

In study [11], generalized dependences of the resistance coefficients were obtained, both for isothermal and non-isothermal fluid movement in channels, and which can also be applicable for droplet and elastic fluids.

However, in practice, unsteady flows are most often encountered, which have their own characteristics. The complexity of studying such flows is due to the changing in the parameters of the fluid flow over time. And this requires additional research using various methods and approaches. These flows include turbulent flow with superimposed periodic pulsations of the fluid flow, and which can occur in various power equipment, where pulsations of the fluid flow can occur as a result of the design features of power equipment, or be created artificially. Such non-stationary processes can significantly affect the hydrodynamics and heat transfer, and be accompanied by both an increase and a decrease in the intensity of heat transfer, and as a consequence, effect on the thermal-hydraulic efficiency of the installation [10-13].

Therefore, the question of the thermohydraulic efficiency of such unsteady flows remains open, since there are less data on the change in hydraulic resistance in the case of non-isothermal pulsating flow in the channels.

The purpose of this work is to numerically investigate the effect of a pulsating fluid flow on the thermohydraulic efficiency in smooth and profiled channels.

Materials and methods

Two variants of the channels, smooth and profiled, were investigated. The inlet diameters D = 0.018 m, and the length of the channels L = 1.6m, are the same in both cases. Concavities located in the corridor order were chosen as the channel profile. The step of the concavities S/h = 10, the angle of the concavities between the axes φ = 120º, [8-10].

Channel 3D models were built by using of the SolidWorks software. The numerical solution was carried out by using of the Flow Simulation CFD software. The mathematical modelling of the medium motion and heat transfer used the non-stationary Navier-Stokes equations, the energy equation (the first law of thermodynamics), and the equation of state are used [14].

This paper considers the phenomenon of heat transfer during forced motion of a fluid in the channel. It is assumed that the forced movement is created by external influence.

The following were accepted as the boundary conditions: at the channel inlet – velosity V_0 = 0.4 m·s^{-1} at the fluid temperature T_{fluid} = 20°C. Fluid – has physical properties of water. At outlet - pressure. The channel walls have physical properties of aluminum and are heated up to the temperature T_{wall} = 105°C.

By the nature of the fluid motion, laminar, turbulent and transient flow regimes can be considered, characterized by the Reynolds number Re.

For the case of periodic pulsations of the fluid flow in the channel, the Reynolds number Re_t depends on time and is described by the equation (1):

$$Re_t = 1 + A\sin(2\cdot\pi\cdot f\cdot t)\cdot\frac{V_0\cdot D\cdot\rho}{\mu} \quad (1)$$

where t – time, (s);
A – amplitude of the pulsations, A = 1;
f – frequency of the working fluid flow's pulsations, Hz;
ρ – density, kg·m^{-3};
μ – dynamic viscosity , Pa·s.

The time-averaged flow velocity V_0 is the same for all cases and V_0 = 0.4 m·s^{-1}. The considered flow frequency is $0 < f < 30$ Hz.

The channel's flow is considered as a turbulent. The boundary layer of the flow near the solid wall of the channel is like the laminar and turbulent flow layers, as well as the transition from laminar to turbulent. And it is modeled with using of the modified universal wall functions.

The FlowSimulation software uses the finite volume method with an adaptive rectangular mesh [14]. Note that, in accordance with the calculation method, any stationary problem is initially solved as non-stationary. The solution is considered found after its establishment in time.

As a result of the numerical solution, the following values of physical parameters were determined: velocity, temperature, density, viscosity, Prandl number, Reynolds number, specific heat at a constant pressure, thermal conductivity coefficient of a liquid and other parameters necessary to determine the average heat transfer coefficient, according to equation (2).

$$Nu_t = 0{,}021\cdot\left((1 + A\sin(2\cdot\pi\cdot f\cdot t)\frac{V_0\cdot D\cdot\rho_1}{\mu_1}\right)^{0.8}\cdot\left(\frac{\mu_1\cdot C_{p_1}}{\lambda_1}\right)^{0.43}\cdot\left(\frac{\mu_1\cdot Cp_1\cdot\lambda_2}{\lambda_1\cdot\mu_2\cdot Cp_2}\right)^{0.25} \quad (2)$$

where $\mu_{1,2}$ – viscosity, Pa·s;
ρ_1 – density, kg·m^{-3};
$Cp_{1,2}$ – specific heat capacity;
$\lambda_{1,2}$ – thermal conductivity, W·(m·K)$^{-1}$.

Parameters of the liquid are the results of numerical calculations at the given coordinate points during steady motion. Index 1 – values in coordinate points along the channel diameter, 2 – at a point located in the near-wall region of the channel.

The distinctive feature, when calculating the Nusselt criterion Nu for a profiled channel, is that the correction ε (2) is introduced into equation (3). It takes into account the increasing in the heat transfer coefficient due to artificial roughness.

$$\varepsilon = 1.04 \cdot Pr_{water}^{0.04} \cdot \exp\left[0.85 \cdot f \frac{(s/h)_{opt}}{s/h}\right] \tag{3}$$

At

$$\left(\frac{s}{h}\right) \leq \left(\frac{s}{h}\right)_{opt} \quad f\left(\frac{s}{h}\right) = \left(\frac{s/h}{(s/h)_{opt}}\right)$$

where s – distance between the axes of the concavities;
h – inner radius of the concave, $(s/h)_{opt} = 13 \pm 1$.

These parameters were determined at points on the selected transverse planes of the channel model and within the computational domain of the fluid by setting the corresponding coordinates *X*, *Y*, *Z*. The required planes were installed at a certain distance from the inlet section when the flow in the channel can be considered stabilized. The defining points were located on the horizontal axis along the channel diameter with a step of 0.5 mm.

In order to extend the obtained numerical experimental results to all similar processes, they are processed according to the second similarity theorem.

To compare the thermal efficiency for smooth and profiled channels on the basis of the obtained experimental data, for the flow pulsation frequency $0 < f < 30$ Hz used relation (4):

$$f(Re) = \frac{Nu_{t(1,2)}}{Nu_{1,2}} \tag{4}$$

where Nu_t – corresponds to the values in the pulsating flow;
Nu – Nusselt criterion for stationary mode;
index 1 – smooth channel; 2– profiled.

The hydraulic resistance ξ in the channels for the laminar fluid flow was determined according to the Poiseuille law, for the turbulent mode of motion according to the Blasius law [2].

As a criterion for evaluating the thermohydraulic efficiency in the channels, the coefficient *E* (5) was considered, which characterizes the intensification of heat transfer at the same losses in the channels with superimposed flow pulsations and at a steady flow regime. When $E > 1$ the thermohydraulic efficiency is considered more beneficial for the intensification of heat transfer.

$$E = \frac{Nu_{t(2)} / Nu_{(1)}}{\left(\xi_{t(2)} / \xi_{(1)}\right)^{1/3}} \tag{5}$$

where ξ_t – hydraulic resistance in the pulsating flow;
ξ – hydraulic resistance for stationary mode.

Results and discussion

The numerical solution results showed that the maximum increase in the average heat transfer coefficient $Nu_{1t\ (ever)}$ in a smooth channel is achieved at $Sh = 0.441$ and is 37% compared with the stationary regime.

In the case of a profiled channel, the maximum $Nu_{2t\ (ever)}$ is reached at the Strouhal number $Sh = 0.0225$ and is 25% compared with the stationary regime.

$Nu_{2t\ (ever)}$ for a profiled channel is 2.5 times more than in a smooth one for the same Strouhal numbers Sh = 0.0225.

For the study of the hydraulic resistance ξ in a smooth channel with non-isothermal fluid motion stationary turbulent flow regimes were considered for the initial fluid flow rates $0.2 \leq V_0 \leq 1$ m·s^{-1}, with the corresponding Reynolds numbers $3500 \leq Re_0 \leq 25000$. The results of the obtained numerical solutions were compared with the known Blasius dependence [14] for turbulent non-isothermal flow in smooth channels Fig. 1.

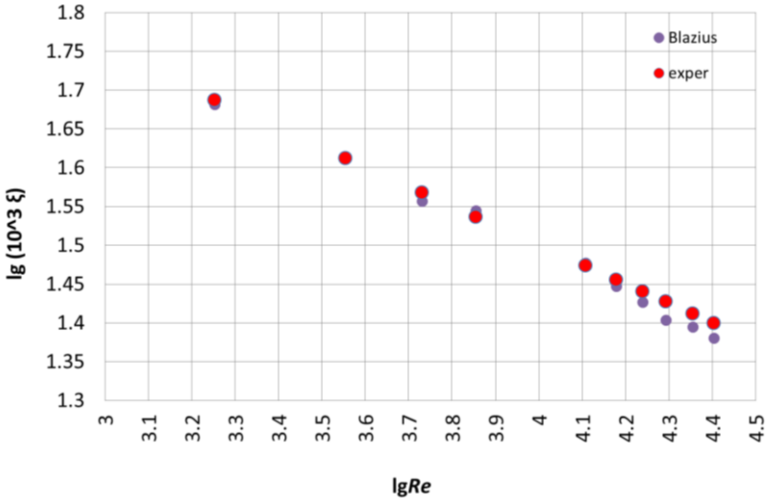

Fig. 1. **$\xi = f$ (*Re*, ΔT) - change in hydraulic resistance during non-isothermal fluid movement in smooth channel:** ξ_{exper} – results of numerical solution, $\xi_{Blazius}$ – Blasius dependence

The deviation of the results of the numerical solution for hydraulic resistance from the known Blasius dependence did not exceed 5-7%, which is a satisfactory convergence of the results.

The following numerical studies for the case of periodic pulsations of the fluid flow in smooth and profiled channels, where the Reynolds number Re_t depends on time and is described by equation (1), showed that a periodic change in the Reynolds number, respectively, leads to a change in the value of hydraulic resistance in time, Fig. 2.

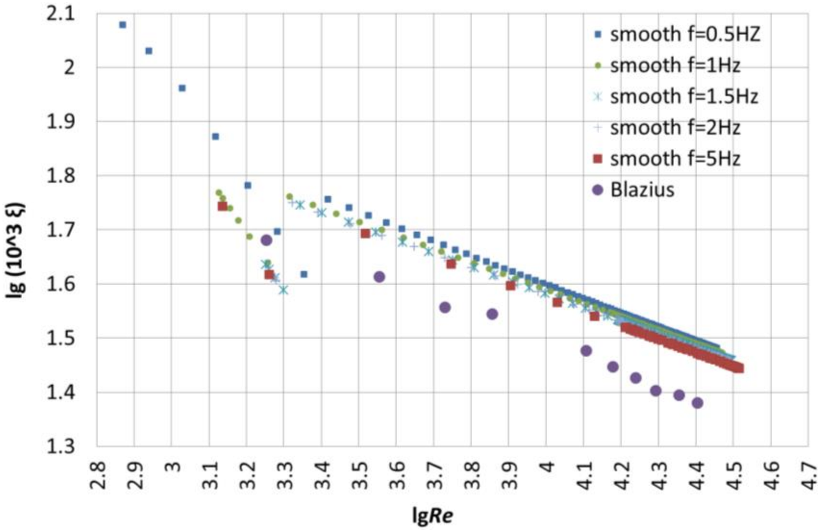

Fig. 2. **Change in hydraulic resistance with time for non-isothermal fluid movement in smooth channel, pulsation frequency $0.5 \leq f \leq 5$ Hz, Struhal $0.0225 \leq Sh \leq 0.225$**

In Fig.2 it can be seen that at a flow pulsation frequency of $0.5 \leq f \leq 5$ Hz at $2.8 \leq \lg Re \leq 3.5$ for a smooth channel, there are characteristic zones of laminar (Re≈2000) and transient flow regimes (*Re* = 2000-4000), and an area of turbulent regime at $3.5 \leq \lg Re \leq 4.7$. With increasing in the frequency of flow pulsations in the channel $f > 5$ Hz, there are no areas of laminar and transient regimes and all the calculated points fall on one straight line, that is, a stabilized turbulent regime occurs in the channel.

A similar picture is observed with a pulsating fluid flow in a profiled channel. Fig. 3. shows the change in hydraulic resistance in time for non-isothermal fluid movement in a profiled channel at a pulsation frequency of $0.5 \leq f \leq 10$ Hz, Struhal $0.0225 \leq Sh \leq 0.441$.

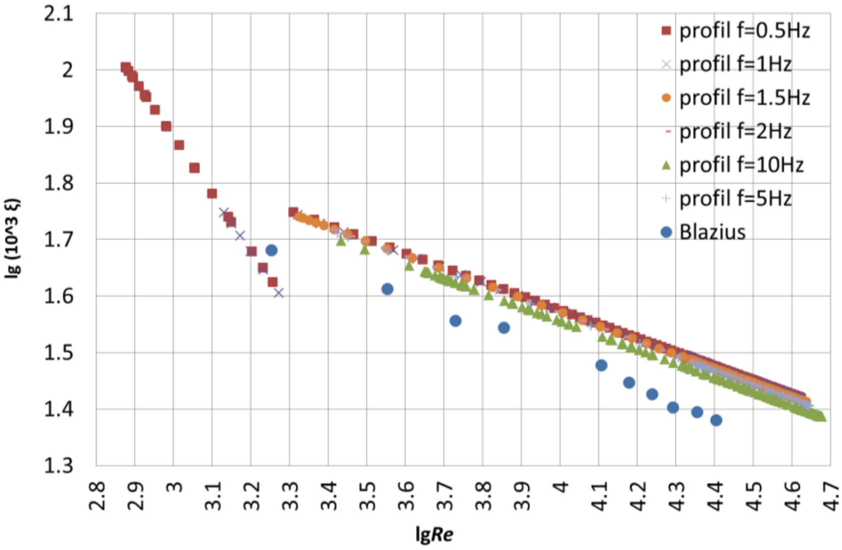

Fig. 3. **Change in hydraulic resistance over time at non-isothermal fluid movement in profiled channel, at $0.5 \leq f \leq 10$ Hz, $0.0225 \leq Sh \leq 0.441$**

In contrast to the pulsating flow in a smooth channel, in a profiled channel a stabilized turbulent flow arises already at a pulsation frequency $f = 2$ Hz ($Sh = 0.09$).

In Fig. 4. the thermohydraulic efficiency E is shown as a function of the relative Reynolds number for the frequency of pulsations of the fluid flow $0 \leq f \leq 20$ Hz, $0.0225 \leq Sh \leq 0.9$. Point 1 – corresponds to the average value of the thermohydraulic efficiency of the profiled channel to the smooth one at $f = 0$. Point 2 – average value at $0 \leq f \leq 2$ Hz, Point 3 – average value at $f = 20$ Hz.

In Fig.4 it can be seen that the thermohydraulic efficiency E at a given frequencies of flow pulsations does not have an unambiguous value, and varies depending on the relative Reynolds number, limiting itself to its minimum and maximum values. But at the same time, intensification effect can be noted, which can be more than 25%, which depends on many factors, such as the channel geometry, fluid flow pulsation frequency, flow vortex structure, as well as the values of heat transfer and hydraulic resistance, and many others.

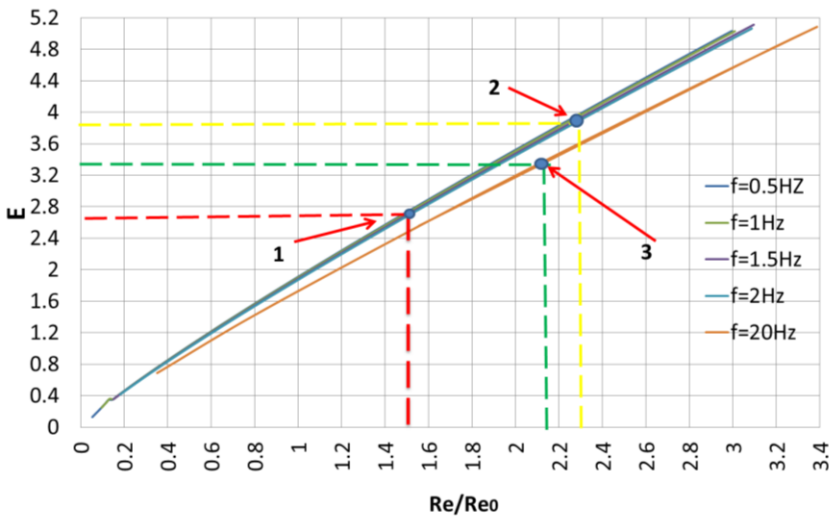

Fig. 4. **Thermohydraulic efficiency E depending on the relative Reynolds number, $0 \leq f \leq 20$ Hz, $0.0225 \leq Sh \leq 0.9$**

Average values of thermohydraulic efficiency E for flow pulsation frequency $0 \leq f \leq 20$ Hz ($0.0225 \leq Sh \leq 0.9$) are shown in Fig. 5. Here, line 1 is the average values of the thermohydraulic efficiency E of a smooth channel at given frequencies of pulsations to a smooth channel at a steady flow regime. Line 2 – average values for the profiled channel at the given frequencies of pulsations to the profiled channel at a stationary flow regime. Line 3 – average values for the profiled channel at the given pulsation frequencies to the smooth channel at a steady flow regime.

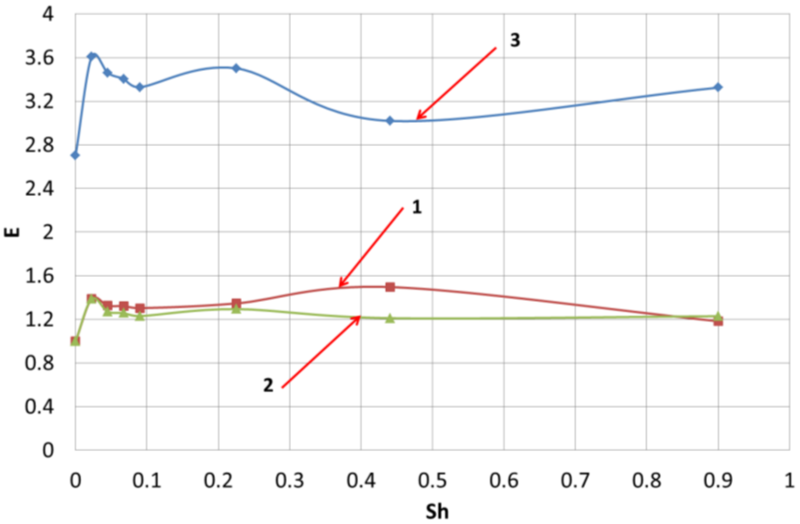

Fig.5. **Average values of thermohydraulic efficiency *E* for flow pulsation frequency $0 \leq f \leq 20$ Hz ($0.0225 \leq Sh \leq 0.9$)**

In Fig. 5. it can be seen that the increase in thermohydraulic efficiency, both in smooth and profiled channels, can have a range from ≈ 20-40%. And with respect to the steady-state regime of flow in smooth channels in a profiled channel at given frequencies it can reach ≈ 360%.

Conclusions

1. The numerical study results have shown that one of the effective ways to intensify heat transfer in the channels can be using of the artificially created forced oscillations of the fluid flow in the channels.
2. Experimental studies have shown that with forced flow pulsations the best indicators of the average heat transfer coefficient in a smooth channel are achieved in the range of flow pulsation frequencies $0 \leq f \leq 10$ Hz ($0.0225 \leq Sh \leq 0.441$). And compared to a steady flow of a liquid in a smooth channel, the average heat transfer coefficient can be 20-40% higher.
3. The smallest relative coefficient of hydraulic resistance for non-isothermal fluid movement in a smooth channel is achieved at a fluid flow pulsation frequency $f = 5$ Hz ($Sh = 0.225$), which is approximately 10-15% lower than in a stationary flow.
4. According to numerical calculations, the highest average value of thermohydraulic efficiency in a smooth channel with non-isothermal fluid motion is achieved at a flow pulsation frequency $f = 10$ Hz ($Sh = 0.441$), which is approximately 50% higher than in a steady flow.
5. Carried out experimental studies in a profiled channel showed that with forced pulsations of the flow, the best indicators of the average heat transfer coefficient are achieved at a pulsation frequency of $0 \leq f \leq 30$ Hz ($0.0225 \leq Sh \leq 1.8$). Here, the average heat transfer coefficient is approximately twice higher in comparing with a smooth channel at the same pulsation frequency. The relative coefficient of hydraulic resistance for non-isothermal fluid movement in a profiled channel is 10-15% higher than in a smooth fluid flow under the same conditions. But it is numerically lower than with a steady flow by 7%.
6. Increasing in thermohydraulic efficiency, both in smooth and profiled channels, can have a range from ≈20 - 40%. And with respect to the steady-state flow in smooth channels in a profiled channel at frequencies of $0.5 \leq f \leq 30$ Hz, it can reach ≈ 360%. As well as in a smooth channel, in practice, it is optimal to use a low frequency range of flow oscillations $2 \leq f \leq 5$ Hz ($0.09 \leq Sh \leq 0.225$).

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