Numerical study of the effects of pressure gradient on the heat transfer in the KMS heat exchanger channel

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Abstract. The work is devoted to studying the influence of the pressure gradient on the heat transfer intensity in the channel of the KMS air heater. The aim of the work is to develop methods for improving the heat engineering characteristics of a heat exchanger using a pressure gradient. Numerical simulation is implemented based on the RANS approach using the Code Saturne software package using the k-w-sst turbulence model. The simulation conditions were set in accordance with the manufacturer's instructions for the KMS-2 type air heat exchanger. In the work, the heat transfer coefficient was studied in the presence of a longitudinal pressure gradient in the heat exchanger channel from the gas path. As a result of the simulation, data were obtained on the effect of the pressure gradient on the heat transfer intensity for different working fluids in the channel of an advanced heat exchanger.

1. Introduction

Heat exchangers are widely applied in microclimate maintenance systems and production plants. Some examples include ventilation plants employed for warm-air heating in public and manufacturing buildings. Moreover, heat exchangers can be used for heating (cooling) technical gases (carbon dioxide, helium, nitrogen, and others). They are divided into recuperative (direct-flow or counter-flow) and regenerative. However, these units have performance limitations that concern gas flow as the thermal physical properties of the gas (air in microclimate maintenance systems) are significantly worse than the characteristics of the heating or cooling heat carrier in these units. Furthermore, the increase of gas flow (air) velocity in the heater aimed to enhance the heat transfer intensity is limited by the noise-level control requirements for operation of the units of microclimate support systems [1, 2]. The existing velocity limits reduce the possibility of increasing the heat transfer coefficient from the air flow. This in turn leads to an unchanged heat transfer coefficient that is indicative of the efficiency of the heat exchanger.

The enhancement of heat transfer intensity is possible due to the application of active and passive methods of enhancement. The active method implies that the heat transfer is influenced by external factors, such as electric field, pulsation, blowing, sound field, etc. The passive method includes the use of intensifiers that act by changing the topography of the heat transfer surface (fins, dimples, pins, and pressure gradient). It is known that the superposing of flow pulsations [3] in the separation area or other influences contribute to the enhancement of heat transfer intensity. As the main method of heat transfer enhancement in the designed heater, it is suggested to use a variable section of channels in which a longitudinal pressure gradient is realized. The interest in this method of increasing heat
transfer intensity can be explained by the fact that the use of this method of enhancement does not complicate the development of new heater constructions.

It should be noted that the pressure gradient sign has a significant impact on heat transfer. Therefore, when designing heat exchangers in ventilation systems, it is necessary to choose the method of performance estimation that will give the most correct results. It is important to note rather high errors of calculation of heat transfer characteristics determined on the basis of dimensionless equations (about 30%). Thus, while developing new samples of heat transfer equipment, using only engineering calculations is not enough to assess the characteristics of heat transfer processes. Therefore, when developing new models of heat exchangers-heaters it is recommended to use the numerical method of research. This research method does not require significant material costs and makes it possible to get the real picture of the object under study. The influence of pressure gradient on heat transfer intensity was analysed by numerical simulation of the heater channel from the air flow side using the RANS approach based on free software [4, 5] provided that the channels of this heat exchanger (KMS-2) are divergent or convergent.

In spite of the fact that at present there are some works [6-9] devoted to the research of influence of pressure gradient on heat transfer, some lack of data for operating conditions of the KMS-2 heat exchanger was revealed. Thus, in work [6] the Reynolds number range is $Re=5000-14000$, the acceleration factor is $K=1.4\times10^{-6}-2\times10^{-5}$, i.e. the characteristics of KMS-2 heat exchanger operation will correspond to a very narrow range of data, especially since the acceleration factor may not correspond to the parameters of the proposed KMS-2 heat exchanger modernisation.

In work [7] a numerical study of heat transfer in a diffuser was conducted with an opening angle $\alpha=3^\circ$, relative channel length $L_d=10-100$ calibres, $Pr=0.026-80$ (working masses: air $Pr=0.71$, helium-xenon mixture $Pr=0.24$, water $Pr=5.40$, transformer oil $Pr=55.7$, mercury $Pr=0.024$). Nevertheless, in work [7], the Reynolds number range $10^{4}-10^{5}$ does not correspond to the operating conditions of the KMS-2 heat exchanger ($Re=30000-6000$). In addition, although the numerical study in work [7] was conducted using the RANS approach, another model of turbulence was used which may give insufficiently accurate results in the selected range $Re=3000-6000$.

Work [8] is devoted to the numerical study of heat transfer in a flat channel with a converging channel and relative length of the converging section $Lk/h_1=13.33$, $Lk/h_2=18.18-100$, acceleration parameter $K=(3-13)\times10^{-6}$ and $Re=10500$. In the proposed study, the geometrical characteristics of the examined converging and diverging channels have other parameters.

2. Numerical study
The design scheme of a heat exchanger with gradient channels that we suggested for microclimate maintenance systems for air heating (cooling) is shown in figure 1 [9].

In order to check the adequacy of the obtained results of the numerical study, they were compared with the results of experimental studies of the influence of turbulence and pressure gradient on the characteristics of heat transfer in the boundary layer given in [10]. The analysis carried out in comparison with the results of numerical modelling [11] and experimental data [10] showed that deviations in the flow rate and pressure parameter do not exceed 10%. It should be noted that there is also a qualitative coincidence of results on the distribution of the heat exchange along the length of the plate [11].
**Figure 1.** Design scheme of the suggested heat exchanger with a longitudinal pressure gradient: $L$ – the length; $S_o$ – the fin spacing; $S_1$ – the longitudinal tube spacing; $S_2$ – the transverse tube spacing; $n_p$ – the number of tubes in a row; $n_c$ – the number of circulation loops; $n_{tc}$ – the number of tubes in the circulation loop.

During numerical research, a heat exchanger of type "KMS" of the manufacturer JSC "TST" was chosen [12] as a basic model of a heat exchanger. In this unit, the heating medium moves through metal finned tubes while the heated air is directed between the plates with dimensions of $136 \times 117$ mm located in increments of 5 mm [12].

A modified heater channel formed between two adjacent plates with a longitudinal pressure gradient through which the heated air passes was chosen as the object of the research. It is assumed that the heat exchanger channel from the air flow side is diverging or converging. The geometry of the calculation area is shown in figure 2. To create a pressure gradient, each of the plates deviates from the longitudinal axis by 3°. The geometry of the calculation area was developed on the basis of cloud software Onshape [13] used for free subscription.

**Figure 2.** Geometry of the calculation area: $a$ – the expanding channel; $b$ – the converging channel.
The working grid was created with the help of Tetrahedron Negten (3D) algorithm, Negten 1D-2D in Salome software package [5] distributed under a free licence and providing access to open source code. The grid is constructed with near-wall viscosity layers and consists of 795911 elements: three-dimensional (3D) elements - 732709 (tetrahedrons - 403645, prisms - 329064); two-dimensional (2D) objects - 62322; nodes - 880.

The carried out analysis of the grid convergence has shown that the reduction of the number of computational cells to fewer than 700 thousand in this problem leads to the appearance of unreasonable fluctuations of flow characteristics. Cell sizes are specified in the range from 0.0001 to 0.01 m. The characteristics of the near-wall viscosity layers in the grid are determined on the basis of the Courant number by the value of $y^+=1$, the value of the initial velocity and equivalent channel diameter; the total thickness of the $\delta$ layers is equal to $2.7 \times 10^{-4}$ m, the number of layers $nsl=6$, the layer expansion coefficient is given equal to 1.5.

The following modelling conditions were set on the calculation area (figure 2): the working mass is air whose density depends on temperature. The air temperature at the inlet is assumed to be equal to the temperature of five cold days for Samara (Russia) (-30 °C), the air velocity at the inlet of the heater model range "KMS-2" takes values of $v_1=3.36$ m/s, $v_2=4.68$ m/s, $v_3=6.7$ m/s, the Reynolds numbers on the minimal section are equal to 3000, 4177, 6000, respectively. The turbulence level is set to $Tu=8.4\%$. On the channel side walls which are heated plates, a heating rate is set. It is determined according to the operating conditions of the KMS heater at a water velocity of $v_w=0.25$ m/s. The temperature difference in the heat exchanger $\Delta t_{cp}$ is measured according to the formula:

$$\Delta t_{cp}=\frac{\Delta g_2-\Delta t_m}{\ln(\Delta t_{m2}/\Delta t_{m1})} = 86.8 \, ^\circ\text{C}. \quad (1)$$

The temperature of the heating agent at the inlet and outlet of the unit is $t_{w1}=95^\circ\text{C}$, $t_{w2}=70^\circ\text{C}$, respectively; the temperature of the air at the inlet $T_{m1}=-30^\circ\text{C}$ and $T_{m2}=20^\circ\text{C}$ at the outlet of the heater. It follows from the performed calculations that the specific heat flow is set equal to $q_1=1832$ W/m², $q_2=2053$ W/m², $q_3=370$ W/m² at the corresponding air inlet velocities. At the outlet of the channel, a free-flow discharge condition is set (see figure 2).

For the realisation of numerical modelling on the basis of the RANS approach with the application of the $k-w-sst$ turbulence model, the software package Code Saturne [4] was used. The solution was obtained on the basis of Navier-Stokes equations, energy and state presented in [14, 15]. Works [9, 11, 14-16] have shown the rightfulness of using this approach to solving the problem studied in the work.

### 3. Simulation results

As a result of the numerical study, the primary characteristics of the flow are determined: temperature, pressure, velocity, energy of flow pulsation. On the basis of the data obtained, local values of the heat transfer coefficient along the channel length for the KMS-2 heater are calculated:

$$\alpha_{xi} = q_j/(T_{wi} - T_{fi}), \quad (2)$$

where $\alpha_{xi}$ – the local heat transfer coefficient; $T_{wi}$ – the wall temperature; $T_{fi}$ – the flow temperature.

In order to further verify the adequacy of the results, figure 3 shows the results of comparison of the data of the numerical study and the calculation of the local heat transfer coefficient according to Zukauskas formula [17] taking into account the high turbulence and acceleration of the flow:

$$Nu_x = Nu_p \cdot \left(\frac{v_x}{v_0}\right)^b, \quad (3)$$
Figure 3. Local values of heat transfer coefficient along the length of the converging channel at $v=6.7$ m/s, $dP/dx < 0$: 1 – the calculation by Zukauskas formula [18]; 2 – the calculation by formula (4) from work [19]; 3 – our numerical modelling.

Figure 4. Distribution of local values of the heat transfer coefficient along the length of the channel: 1 – $Re_a=3000$, $dP/dx < 0$; 2 – $Re_a=4177$, $dP/dx < 0$; 3 – $Re_a=6000$, $dP/dx < 0$; 4 – $Re_a=3000$, $dP/dx > 0$; 5 – $Re_a=4177$, $dP/dx > 0$; 6 – $Re_a=6000$, $dP/dx > 0$. 
Figure 5. Distribution of the local heat transfer coefficient along the length of the channel: a – at $dP/dx > 0$ and the velocity at the channel inlet $1 - v = 3.36$ m/s, $2 - v = 4.68$ m/s, $3 - v = 6.7$ m/s; b – at $dP/dx < 0$ and the velocity at the channel inlet $1 - v = 3.36$ m/s; $2 - v = 4.68$ m/s; $3 - v = 6.7$ m/s.

In addition, figure 5 shows the results of the calculation for the converging channel with an opening angle from 2° to 17° by formula (4) presented in [19]:

$$
Nu_x = 0.024 \cdot Re_x^{0.8},
$$

where $Re_x$ – the local value of Reynolds number; $x$ – the distance from the channel inlet. The analysis of figure 3 shows that the deviation of the calculation results by the dimensionless equations and the numerical modelling method is within the margin of error of dimensionless equations. It should be noted that expression (4) will give better results at higher values of $Re_x$.

Figure 4 shows the results of numerical simulation of local values of heat transfer coefficient along the length of the channel at different Reynolds numbers in the presence of a negative and positive longitudinal pressure gradient.

It was found that the maximum value of the heat transfer coefficient was achieved at higher velocities. The growth of heat transfer intensity is mostly due to the value of air flow velocity.

Since the influence of velocity significantly affects the intensity of heat transfer, the results are
summarised according to Stanton’s criterion for positive and negative pressure gradients. Figure 5 shows the dependence of the St number on the Reynolds number along the length of the expanding and narrowing channel.

The results are generalized not only by local values but also by the average values of the heat transfer coefficient. Figures 6, 7 present the average values of the heat transfer coefficient and the Nusselt number for the expanding and narrowing channel (the Reynolds number is determined for the minimal section of the channels).

As a comparison, figure 7 shows the results of the heat transfer estimation by the known dimensionless equations (Mikheeva, Petukhova, Gnielinski) for channel flow [19]. It should be noted that the results of calculation by Petukhov formula [19] whose range of applicability at $Re_d=10000-5\times10^6$ are given considering that it is Petukhov formula’s modification that represents Gnielinski expression for $Re_d=2300-5\times10^6$ [19].

According to the results of the numerical study presented in figure 6, the average values of the heat transfer coefficient at $dP/dx < 0$ are 12-16% lower than at $dP/dx > 0$ depending on the Reynolds number. Figure 7 shows that the heat transfer intensity in a channel with a gradient with $dP/dx < 0$ will be lower than for a flow with $dP/dx=0$ (heat transfer in a channel without pressure gradient is determined by the known dimensionless equations of Mikheev and Gnielinski). For a flow with $dP/dx > 0$, there is an area ($Re_d=3000-5000$) in which the heat transfer intensity is higher than for a flow with $dP/dx = 0$ (see figure 9). The Petukhov formula gives somewhat overestimated results as its area of application is $Re_d=10^4-5\times10^6$ which does not correspond to the operating conditions of the KMS-2 heat exchanger.

Since it is known that the cost of compensation for pressure losses significantly depends on the pressure gradient sign, while developing a modified design of heat exchanger should be performed not just converging or diverging channels, but combined channels. In combined channels, the length and angle of opening (or narrowing) of individual sections will determine the efficiency of heat exchange in the unit. Therefore, this raises the importance of elaborating a simple engineering design methodology of such heat exchangers on the basis of diagrams or the dimensionless equations.

It is known that, in the presence of a pressure gradient, heat transfer is significantly impacted by the nature of the working masses. One more cycle of numerical experiments has been carried out whose results are shown in figure 8. This figure presents data on the distribution of local heat transfer coefficient in the channel (at the inlet $v=10$ m/s) in the presence of positive and negative pressure gradient for gases of different physical nature (working masses: air, helium, carbon dioxide). The research results (see figure 8) are presented in relative coordinates: $St_i/St_{00}$, where $St_{00}=0.03\cdot Re^{-0.2}\cdot Pr^{-0.4}$ – Stanton number for undisturbed flow and $x=x_i/L$, where $x_i$ – the distance from the channel inlet, $L$ – the channel length. Line $St_i/St_{00}=1$ in figure 8 shows relative values of heat transfer for a gradientless flow since in a gradientless flow $St_i=St_{00}$.
Figure 6. Channel heat transfer coefficient: 1 – at $dP/dx < 0$; 2 – at $dP/dx > 0$.

Figure 7. Heat transfer in the channel: 1 – the numerical modelling at $dP/dx < 0$; 2 – the same at $dP/dx > 0$; 3 – the calculation by Petukhov formula [19]; 4 – the calculation by formula (5); 5 – the calculation by Mikheev dimensionless equation for air.
Figure 8. Distribution of the local heat transfer coefficient for different working masses at inlet velocity $v = 10 \text{ m/s}$ with positive pressure gradient: 1, 2, 3 – air, helium, carbon dioxide, respectively, at $dP/dx < 0$; 4, 5, 6 – air, helium, carbon dioxide, respectively, at $dP/dx > 0$.

According to the results presented in figure 8, the heat transfer intensity at $dP/dx > 0$ will be higher than at $dP/dx < 0$. Moreover, the data in figure 8 show that the presence of a pressure gradient has a significant effect on the heat transfer compared to a gradientless flow. A negative pressure gradient leads to some reduction in heat transfer compared to a gradientless flow; the minimum values of heat transfer coefficient are observed at $dP/dx < 0$ for helium used as a working mass. With a positive longitudinal pressure gradient, the heat transfer coefficient can be higher than in a gradientless flow. The heat transfer intensity for air is higher than for carbon dioxide but lower than for helium at $dP/dx > 0$.

4. Conclusion

As a result of numerical modelling of the KMS-2 heat exchanger-heater channel (assuming that the channel is made in the form of a converging or diverging channel) with a pressure gradient, it was found that the heat transfer intensity significantly depends on the gradient sign.

Thus, the average values of heat transfer coefficient at $dP/dx < 0$ are $12 \text{ – } 16\%$ lower than at $dP/dx > 0$ in the range of Reynolds numbers $Re=30000–6000$ and the acceleration coefficient $K=8.04 \times 10^{-6}–1.56 \times 10^{-5}$. The use of various working masses shows that a negative pressure gradient leads to some reduction in heat transfer compared to a gradientless flow; the minimum values of heat transfer coefficient are observed at $dP/dx < 0$ for helium used as a working mass. With a positive longitudinal pressure gradient at $dP/dx > 0$, the heat transfer coefficient can be higher than in a gradientless flow and the heat transfer intensity for air is higher than for carbon dioxide, but lower than for helium.

When developing the design of a heat exchanger with modified channels, it is assumed to perform not just converging or diverging channels, but combined channels, in which the length and angle of opening (or narrowing) of individual sections will determine the efficiency of heat exchange in the unit. Therefore, this raises the importance of elaborating a simple engineering design methodology of such heat exchangers on the basis of diagrams or the dimensionless equations.

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