AN EXPERIMENTAL INVESTIGATION OF LOCAL HEAT-EXCHANGE COEFFICIENTS ON THE WALLS OF A VALVE MECHANISM

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The widespread application in diverse technical equipment of valve mechanisms with a shaft regulator of the flow rate operating with hot gases has made a detailed investigation of the heat and mass exchange on the walls of the inner cavities of the regulators a pressing matter. Complicated three-dimensional vortex flows with an enhanced degree of turbulence which arise in the region beyond the shaft exert a significant effect on the processes of convective heat exchange and erosional removal of the valve walls on the boundary with the gas [1, 2]. The appearance of factors which affect the intensity of vortex flows as well as the heat exchange parameters in connection with the action of vortices will permit optimizing the geometrical characteristics of valve cavities and creating a procedure for the calculation of protective coatings which protect the walls of the regulators from premature failures.

Some results of an experimental investigation of convective heat exchange on the inner surfaces of a valve mechanism with a shaft are presented in this paper. A model apparatus was designed and fabricated for doing the experiments, the layout of which is given in Fig. 1 [1) baffle, 2) valve housing, 3) shaft, and 4) cavity of the valve beyond the shaft]. Local heat-exchange coefficients between the walls and the process gas on the most thermally stressed part of the valve structure were measured with this apparatus: on the upper A and end B walls of the valve cavity beyond the shaft. This cavity is intended to improve the filling of the inner spaces by the process gas, which exerts a favorable effect on the flow rate characteristic of the entire valve.

Cold air with a temperature $T^* = 263°K$ and $k = 1.4$ was used as the process gas. During the experiments the pressure of the retardation of the incoming flow before the baffle (Fig. 1), which regulates the flow velocity at the input to the valve, was kept equal to $p^* = 8.4 \times 10^5$ Pa. The outflow regime from the valve was constantly critical, and the flow at the valve input was subcritical for all values of the baffle height. Variation of the parameters of the medium at the valve input and of the flow parameters in the inner cavity of the valve was achieved by varying the baffle height H and the position of the valve shaft t. Variation of the baffle height within the limits from H = 0 to H = 75 mm results in a stepwise variation of the velocity at the valve chamber input from 30 to 100 m/sec.

The experimental investigation of heat exchange in channels of complex shape presents definite difficulties due to the complexity of measuring thermal fluxes through different sections of the channel. The measurement of local heat-exchange coefficients in such channel constructions with ordinary calorimetry is very difficult in view of the impossibility of taking rigorous account of heat overflows (in three measurements in the general case) on a streamlined surface. When the heat-exchange coefficients are measured by the method of a regular regime, errors are added due to the heat drain into a wall having a variable thickness.

In this paper a special procedure was used for determining the local heat-exchange coefficients α under conditions of a complicated flow which permitted a rather simple and rapid measurement of the size of the thermal fluxes on arbitrarily specified sections of the wall [3]. The essence of the procedure consists of the following. A conducting heat-releasing graphite-based film was deposited on the heat-insulating surface of the model in the form of a rectangle. An electric current is supplied through current leads to the lateral boundaries of the film, and the power liberated on the film is determined from the measured current strength and voltage. The ratio of this power to the known area of the film determined the average specific thermal flux q_{av} from the film.

Recording of the voltage and the current strength was done by Class 0.2 instruments. A model of the structure being investigated is equipped with 74 Chromel—Copel thermocouples

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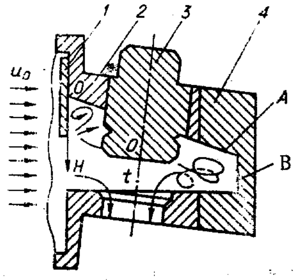

Fig. 1

with an electrode diameter of 0.2 mm for measurement of its surface temperature. The pickups of the thermocouples were electrically insulated from the conducting film by a thin layer of heat-resistant lacquer. In order to reduce the heat leaks through the wall, the valve model was made out of a heat-insulating material — Textolite 30 mm thick with a thermal conductivity coefficient of 0.3 W/m·°C. For the conditions of the experiment the heat losses into the wall are no more than 1.5% of that supplied to the film. Calculations made of the heat drain from the film along the thermocouple electrodes and due to overflows along the length of the conducting film, whose thickness was ~20 μm, introducing in addition ~0.5% of the heat losses for the most thermally stressed tests. The total losses of heat not participating in convective heat exchange do not exceed 2%.

For a steady flow of the process stream the temperature difference for each thermocouple is fixed upon the atainment of steady thermal conditions on the wall with and without heating of the film. This temperature difference at a specified point of the valve surface is caused by the thermal flux from the film and is decisive for the calculation of the local heat-exchange coefficient. Thus, if the film were uniform in resistance, i.e., uniform over its thickness on an entire section, then the local heat-exchange coefficient α_i would be determined from the average specific thermal flux supplied q_{av} and the controlling temperature differential $\Delta T_i = T_i - T_{i_0}$ measured by the i-th thermocouple, where T_i is the temperature at the i-th point with heating of the film, T_{i_0} is the same without heating of the film, and $\alpha_i = q_{av}/\Delta T_i$.

It has turned out in practice to be complicated to achieve constancy of the film thickness and uniformity in connection with the deposition of a conducting film on the wall under investigation. Therefore, it is necessary to perform calibration tests section by section at the thermocouple seal sites for the distribution of the local specific thermal fluxes q_i. The essence of calibration reduced to the fact that calibration coefficients for each of the thermocouples were determined for different thermal loads on the film in the absence of convective heat exchange on the wall. It was also necessary to exclude the effect of free convection in the volume of the valve on the thermocouple readings. It was assumed when performing the calibration that the deviation of the local thermal fluxes q_i from the average over the wall being investigated is proportional to the deviation of the corresponding local temperatures from the weighted mean temperature. The calibration coefficient k_i for the i-th thermocouple is the ratio of the local temperature difference with heating and without heating of the film to the known average specific thermal flux q_{av}. These calibration coefficients permitted determining the local specific thermal flux from the wall section where the i-th thermocouple was embedded as the appropriate fraction of the average q_{av} with respect to $q_i = q_{av}k_i/k_{av}$, $k_i = (T_i - T_{i_0})/q_{av}$, and $k_{av} = (T_{av} - T_{av_0})/q_{av}$ are the calibration coefficients for the section with the i-th thermocouple and for the entire film, respectively; T_{av} and T_{av_0} are the weighted mean surface temperatures of the model over its area with and without heating of the film.

The need for simultaneous recording of all the thermal and pneumatic detectors, reduction of the time to make the measurements, and processing of the information required the application of a system of automated collection and processing of information using a Minsk-22 computer. The system was developed at the Institute of Theoretical and Applied Mechanics of the Academy of Sciences of the USSR (ITPM SO AN SSSR), and includes, in addition to the computer, an Analog system for the collection and conversion of analog information and Type F-240 commutators of analog signals in an assembly with a Type 4014 digital voltmeter, which provide for measurement of the thermal emfs from the thermal detectors in a time ~200 msec.

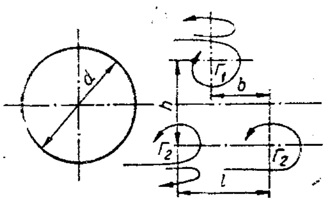

Fig. 2

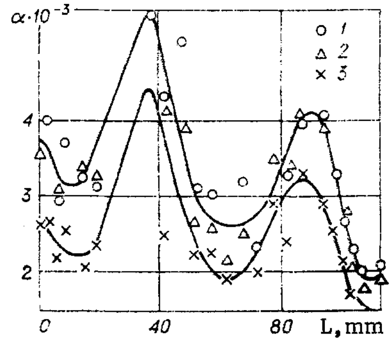

Fig. 3

An algorithm has been developed for processing of the experimental results, and a corresponding program has been coded whose end result is the output of the temperature fields on the wall surface, the local thermal fluxes, and the heat-exchange coefficients.

As investigations of gasdynamics in valve cavities, the results of pneumatic measurements of the pressure fields in the volume and on the walls, as well as the results of carbon oil visualization have shown, the flow pattern has a complicated spatial structure. A recirculation detached zone is formed behind the lowered baffle (see Fig. 1). The protruding valve shaft even in the extreme upper position (t = 0) is a source of formation of two vortex structures with circulations Γ_1 and Γ_2 (Fig. 2), which then flow into the exit aperture of the valve, which is located under the shaft, in the form of vortex ropes. Accumulations from both sides of the wall of a carbon oil emulsion used for visualization of the flow near the wall indicate the presence of such vortex structures interacting with the upper wall of the valve cavity beyond the shaft.

Vortex formations occur near the side walls of the valve which also then flow into the exit aperture of the valve. In order to link the parameters of the flow running into the shaft u_0 with the circulation of the vortices formed beyond it, one can make use of the relationships of von Karman [4, 5] for a vortex trail:

$$u_0 = (\Gamma/2l)\ \mathrm{th}\,(\pi h/b),\quad h/l = 0.28, \tag{1}$$

where h, l, and b are the geometrical parameters of the vortex structures according to [4, 5].

The results of measurements of the local heat-exchange coefficients on the upper and end walls of the valve cavity beyond the shaft are shown in Figs. 3-5.

Typical experimental data on the heat exchange on the upper wall for one of the baffle positions H = 75 mm and three positions of the valve shaft: the highest [1) t = 0] and two intermediate positions [2) t = 10 mm and 3) t = 20 mm] are given in Fig. 3. It is evident that the heat exchange on this wall is characterized by two clearly expressed maxima with an intermediate minimum. Lowering of the valve shaft to the position t = 10 mm, while the total flow rate through the valve decreases by up to 10%, has practically no effect on the size of the heat-exchange coefficients. Further lowering of the shaft (t = 20 mm) leads to an appreciable decrease in the flow rate (up to 50%) through the valve and a corresponding decrease by ~30% in the intensity of the heat exchange.

A stepwise variation of the baffle height within the limits from H = 0 to H = 75 mm leads to a significant intensification of the processes of convective heat exchange on the inner surfaces, which is explained by an appreciable increase in the intensity of the vortex motion as the input area to the valve decreases.

As is evident from the measurement results given in Fig. 4, the maximum of the heat-exchange coefficient on the upper wall increases from α = 1500 W/m²·°C for H = 0 to α = 5300 W/m²·°C for H = 75 mm [1) H = 0, 2) H = 55 mm, 3) H = 65 mm, and 4) H = 75 mm]. An asymmetry is observed in the size of the maxima of the thermal fluxes. The difference in the sizes of the heat-exchange maxima for arbitrary positions of the baffle is more significant on the end surface than for the upper wall for the very same positions of the baffle (Fig. 5; the notation corresponds to Fig. 4). Just as on the upper wall, two maxima of the heat-exchange coefficients are observed on the end wall. In contrast to the upper wall, the position of the maxima on the end wall is fixed at different coordinates L. This fact may indicate non-

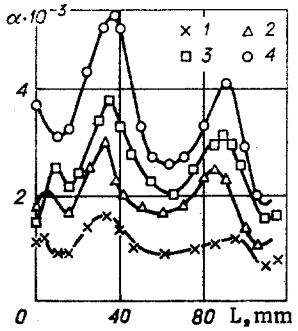

Fig. 4

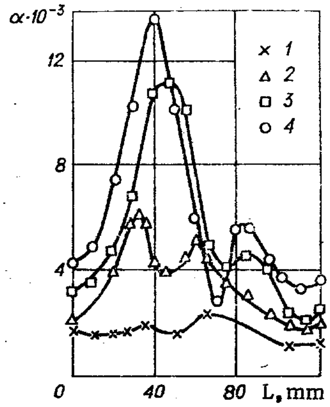

Fig. 5

stationarity of the process upon the descent of the vortices beyond the valve shaft and their interaction with the valve walls.

When the baffle moves from the position H = 0 to H = 75 mm, the maximum values of the heat-exchange coefficients on the end wall increase from 2140 to 13,500 $W/m^2 \cdot {}^\circ C$, i.e., appreciably larger than under the corresponding conditions on the upper wall of the cavity beyond the valve shaft. The heat-exchange process on the end surface is more intense.

The systematic error in the determination of α is ±21%. One can evidently explain the difference in the sizes of the maxima of the thermal fluxes on the end surface by the formation of vortices beyond the shaft in staggered order [2]. The interaction of such a vortex structure by the end (see Fig. 2) with the upper wall of the valve has less effect both on the absolute values of the maxima of the heat-exchange coefficients and on the size of their difference in comparison with the end wall. In the latter case the vortex structure interacts with the wall through its own lateral surface; the staggered position of the vortices decreases the intensity of the interaction of one of them with the wall. In order to generalize the experimental data on heat exchange on both walls under discussion, a model was selected for the growth of the boundary layers from the stagnation points B and Γ of the flow jets on both sides of the x axis (Fig. 6a). For simplicity in perceiving this model we shall consider the heat exchange on the end wall of the cavity beyond the valve shaft. This computational model has been successfully used for very small-scale cell structures which arise in connection with free convection in narrow gaps between planes and cylinders or on the interaction boundary of two media [6, 7]. One can distinguish in Fig. 6a three main zones of variation of the heat-exchange coefficient on the coordinate L (see Fig. 5). Zones I and II are regions of return flows next to the side walls of the valve. The central zone III lies between two stagnation lines of the flow jets B and Γ, which separate the vortex structures and the return flows. The origin of the growth of the boundary layers on both sides of the maxima starts from the points O.

One should note that in connection with generalization of the experimental data for all flow regimes in the valve some experimental quantities are used. Thus, the distance between the maxima of the heat-exchange coefficients was taken as the parameter h. It was assumed that the boundary layers which developed on both investigated walls are turbulent. The Reynolds number Re_x constructed for the experimental conditions in the x coordinate lie in the range of values 10^4–10^6. The heat exchange is determined by the relationship

$$\alpha \sim (\rho u)^{0.8}. \tag{2}$$

for a turbulent boundary layer on a plate.

Using experimental data on the sizes of both maxima of α for one wall and the expression (2), one can determine the relationship between the circulation values of both vortices which arise beyond the shaft:

$$\Gamma_1/\Gamma_2 = (\alpha_1/\alpha_2)^{1.25}. \tag{3}$$

With Eq. (3) taken into account we find from the relationship (1) the tangential flow velocity u_t, which we adopt as the velocity on the outer edge of the boundary layers in the model under discussion (Fig. 6a). The controlling Reynolds number is constructed from this same velocity.

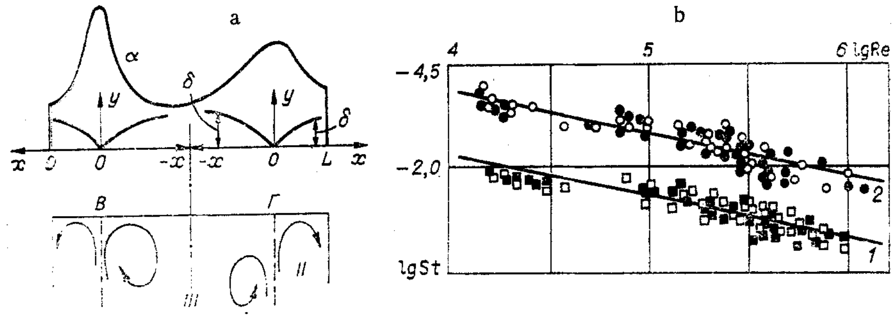

Fig. 6

All the experimental points given in Figs. 4 and 5 are processed with the assumptions expressed taken into account and are presented in Fig. 6b in the form of the dependence $St = f(Re_x)$. One should note that the experimental points for both walls clearly stratify among themselves (curve 1 and the small squares correspond to the upper wall; curve 2 and the small circles correspond to the end wall). The data for zone III on both walls are denoted in Fig. 6b by the open circles; the data for zones I and II are denoted by the filled circles. As has already been pointed out above, a more intense heat exchange process occurs on the end surface. Groups of data for each valve wall are generalized by the equations:

$$St_x = 0.05\,Re_x^{-0.2}Pr^{-0.6} \tag{4}$$

for the upper valve wall (Fig. 6b, curve 1) and

$$St_x = 0.1\,Re_x^{-0.2}Pr^{-0.6} \tag{5}$$

for the end wall (Fig. 6b, curve 2).

It is evident that Eqs. (4) and (5) differ from the heat-exchange law for flow around a smooth impenetrable plate by an incompressible turbulent boundary layer under quasiisothermal conditions [8]

$$St_x = 0.029\,Re_x^{-0.2}Pr^{-0.6} \tag{6}$$

only in the proportionality constant. Dependence (6) has been obtained for heat-exchange conditions in which the incoming flow has a degree of turbulence of the order of 1% or less. In the case of an enhanced degree of turbulence of the flow washing the surface a significant (up to 20-80% and more) intensification of the heat exchange [9, 10] occurs in the Reynolds number range 10^3-10^6. It is well known that vortex structures which arise locally in a two-dimensional flow which flows into an obstacle can intensify the heat exchange at the interaction site of these local vortex structures with a surface by a factor of two in comparison with the average value [1]. In addition, the degree of turbulence of the vortex formations which arise beyond poorly streamlined bodies and upon additional retardation of them in connection with interaction with the surface can significantly increase in comparison with the turbulence of the stream flowing into the body [2, 10]. These investigations offer a basis for assuming that in the case under discussion of flow in a valve there exists an enhanced degree of turbulence in the flows interacting with the valve walls.

An empirical dependence is given in [10] for estimating the effect of an enhanced degree of turbulence on the drag coefficient in a turbulent boundary layer:

$$c_f = c_{f0}[1 + 200(E_t - 0.01)^2], \tag{7}$$

which is valid in the interval $1\% \leqslant E_t \leqslant 8\%$. Here E_t is the enhanced degree of turbulence of the external flow and c_{f_0} is the local drag coefficient for $E_t \leqslant 1\%$. For the conditions of our experiments one can assume [8] $c_{f_0} = 0.059Re_x^{-0.2}$.

Assuming the existence of an analogy for the flow around a plate between drag and heat exchange, we have

$$St = c_f Pr^{-0.6}. \tag{8}$$

For the conditions in which the tests were conducted the heat-exchange law (6) for the upper wall will coincide with the approximate dependence I (Fig. 6b) if the degree of turbulence E_t of the flow on the outer boundary of the boundary layer is ~7%. The effect of an enhanced degree of turbulence on the heat exchange is taken into account by the formula (7) with (8) taken into account. Making similar calculations for the end surface, we obtain the degree of turbulence of the incoming flow to be ~12%, which may be related to additional turbulence of the flow near the wall due to retardation of the flow. The results obtained on the enhanced degree of turbulence can be assumed to be of an estimative nature. Additional experimental investigations of its direct measurement are necessary for the accurate determination of E_t.

In conclusion, one should note the simplicity and effectiveness of the procedure for the determination of the local heat-exchange coefficients in channels of complicated shape, which in combination with an automated system for the collection and processing of the information permits obtaining the heat-exchange parameters of interest to us sufficiently rapidly and reliably. The model of the development of a boundary layer can be used in engineering calculations of the heat exchange coefficients for complex conditions of gas flow with vortex formations.

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