EFFECT OF HEAT TRANSFER ON THE LIMITING CHARACTERISTICS OF MAGNETOFLUID SEALS

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A procedure is developed for calculating the maximum temperature in the working gap of a magnetofluid seal and the limiting rate of rotation of hermetically sealed shafts.

The most promising seals for rotating shafts, especially under conditions of a high vacuum, are at the present time considered to be magnetofluid seals [1]. However, further improvement in their working characteristics is possible only if the harmful consequences of dissipative heating of the fluid magnetic seal are successfully overcome [2]. For high linear velocities of the surface of the hermetically sealed shaft, viscous heat liberation is not compensated by natural processes of heat removal, which can cause the working body to boil and lead to subsequent breakdown of the seal. In vacuum magnetofluid seals, the effect of this factor is aggravated, on the one hand, by a decrease in the boiling point of the magnetic fluid sealing the gap, and on the other, by the significant intensification of vaporization of the magnetofluid. Therefore, the limiting characteristics of high-speed magnetofluid seals are determined in many ways by the thermal processes inside the seal layer.

The problem of heat transfer in the working gap of a magnetofluid seal was formulated in [3], where the stationary two-dimensional system of equations of thermal hydrodynamics are numerically solved for a two-step seal with triangular teeth under isothermal conditions at the boundaries of the magnetic fluid plug. The calculations showed that the maximum heat flux is removed through the tip of the teeth, i.e., the sealing fluid is subjected to the action of a considerably inhomogeneous temperature field.

The one-dimensional stationary problem of heating of a layer of magnetic fluid in a gap between axial cylinders, the inner cylinder in which rotates at constant velocity, was analytically solved in [4]. Comparison of the velocity and temperature profiles beneath a tooth in the seal in [3] and those obtained in [4] for identical boundary conditions show that they coincide to within 7%. Therefore, in engineering calculations, it is completely sufficient to use a one-dimensional approximation.

The considerations stated in [4] concerning the special cooling system for optimizing the characteristics of a magnetofluid seal are developed in [5], where the conjugate problem of heating of a shaft, hermetically sealed by a magnetofluid seal, was solved using the finite-differences method.

The results of [3-5] were obtained assuming that the coefficients of transfer of the working body of the magnetofluid seal are constant and that the thermohydrodynamic regime is stationary. However, the thermophysical characteristics of known magnetic fluids depend appreciably on temperature [6], while any machine reaches the stationary thermal state over a finite time, during which it is necessary to take into account the dependence of the characteristics of the process on time. In this connection, it is important to keep in mind the characteristics indicated in order to obtain more complete information concerning the magnitude of the limiting parameters of the magnetofluid seals.

In real structures of magnetofluid seals, the thickness of the layer of magnetic fluid filling the gap δ is much less than the radius of the sealed shaft, while the width of the base of the teeth focusing the magnetic field is $(2-5)\delta$ [7, 8]. For this reason, the thermohydrodynamic processes in the working gap of magnetic fluid seals must be well-modeled by a system of equations for a flat infinite layer of viscous heat conducting fluid [9]:

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$$\rho \frac{\partial v}{\partial t} = \frac{\partial}{\partial x}\left(\eta \frac{\partial v}{\partial x}\right), \tag{1}$$

$$\rho c \frac{\partial T}{\partial t} = \frac{\partial}{\partial x}\left(\lambda \frac{\partial T}{\partial x}\right) + \eta\left(\frac{\partial v}{\partial x}\right)^2 + \alpha (T_1 - T). \tag{2}$$

The magnetic field is assumed to be given as in [3]. The coordinate x axis is directed across the magnetic fluid layer. The additional term $\alpha(T_1 - T)$ introduced into Eq. (2), in contrast to [9], simulates heat transfer through the end faces of the seal, taking into account indirectly the two-dimensional nature of the real problem. From physical considerations, it can be shown that the quantity α does not exceed the ratio λ/δ^2. T_1 is taken to mean the temperature at the end faces of the free surfaces of the magnetic fluid layer.

System (1)-(2) was analytically and numerically studied for two variants of the boundary conditions:

$$v|_{x=0} = v_0,\ v|_{x=\delta} = 0,\ T|_{x=0} = T|_{x=\delta} = T_0 \tag{3}$$

and

$$v|_{x=0} = v_0,\ v|_{x=\delta} = 0,\ \lambda \left.\frac{\partial T}{\partial x}\right|_{x=0} = q_0,\ T|_{x=\delta} = T_0. \tag{4}$$

The following conditions were chosen as initial conditions

$$v|_{t=0} = 0,\ T|_{t=0} = T_0. \tag{5}$$

If it is assumed that the thermophysical characteristics are constant, then system (1)-(2) separates and in the limiting case of long times, the velocity profile in the gaps is linear [10]:

$$v = v_0 (1 - x/\delta).$$

Then, under isothermal conditions (3) at the boundaries and in the absence of heat transfer through the end faces, the temperature regime in the layer is characterized by the following relations:

for the profile of the temperature field

$$T(x) = T_0 + \frac{\eta (v_0)^2}{2\lambda} (x/\delta)(1 - x/\delta);$$

for the temperature of maximum heating after reaching the steady state

$$T_{\max} = T_0 + \eta (v_0)^2/8\lambda; \tag{6}$$

for the average temperature over the layer

$$\langle T \rangle = \frac{1}{\delta}\int_0^\delta T(x)\,dx = T_0 + \eta (v_0)^2/12\lambda. \tag{7}$$

The following heat flux is removed into the shaft and into the housing:

$$q = \eta (v_0)^2/2\delta. \tag{8}$$

Practical realization of isothermal boundary conditions in the sealing unit is technically quite awkward (in this case, it is necessary to cool both the surface of the teeth and the surface of the shaft). Boundary temperature conditions of form (4) assume only single-sided cooling of the layer and a removal of a thermal heat flux of magnitude q_0 through the secondary boundary due to the thermal conductivity of the shaft. In this situation, after reaching the steady state, the temperature profile has the form

$$T(x) = T_0 + \frac{\eta (v_0)^2}{2\lambda}(1 - x^2/\delta^2) - \frac{q_0 \delta}{\lambda}(1 - x/\delta), \tag{9}$$

the maximum temperature in the layer is

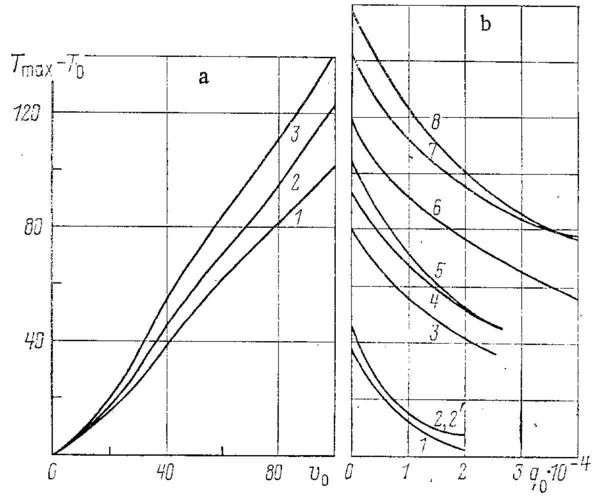

Fig. 1. The maximum temperature of heating of a sealing layer as a function of the linear velocity of the surface of a hermetically sealed shaft under isothermal boundary conditions (a) (1, solution of the transcendental equation (6); 2, a calculation using the analytic expression (6) with $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$; 3, numerical solution), as a function of the heat flux given on the boundary for mixed boundary thermal solutions (b) (1, 3, and 6, solutions of the transcendental equation (10) for v_0 = 20, 40, and 60 m/sec; 2, 4, and 7, a calculation using the analytic expression (10) for v_0 = 20, 40, and 60 m/sec with $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$; 2', 5, and 8, a numerical solution with v_0 = 20, 40, and 60 m/sec). q_0, W/m²; T, °C.

$$T_{\max} = \begin{cases} T_0 + \dfrac{\eta (v_0)^2}{2\lambda} \, [1 - q_0\delta/\eta (v_0)^2]^2, & \text{if} \quad 0 \leqslant q_0\delta/\eta (v_0)^2 \leqslant 1, \\ T_0, & \text{if} \quad q_0\delta/\eta (v_0)^2 > 1, \end{cases} \tag{10}$$

and the average temperature over the layer is

$$\langle T \rangle = \frac{1}{\delta} \int_0^\delta T(x)\,dx = T_0 + \eta (v_0)^2/3\lambda - q_0\delta/2\lambda. \tag{11}$$

Since, in reality, the thermophysical characteristics are some functions of temperature $\eta = \eta(T)$ and $\lambda = \lambda(T)$, in order to use the analytical expressions obtained in specific calculations, it is necessary to know at what characteristic temperature T_* the values of η and λ should be substituted into the equations. Two natural and simplest suggestions are $T_* = T_{max}$ or $T_* = \langle T\rangle$.

Then, in the first case, (6) goes over into a transcendental equation for T_{max} with the parameter v_0 and expression (10) goes over into a transcendental equation for T_{max} with parameters v_0 and q_0. The roots of these equations are the values of T_* sought. The magnitudes of T(x), $\langle T\rangle$, and q can then be found from the values $\eta(T_{max})$ and $\lambda(T_{max})$.

In the second case, expressions (7) and (11) for $\langle T\rangle$ already transform into transcendental equations, while the quantities T(x), T_{max}, and q are calculated from the values of $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$.

Equations (6), (7), (10), and (11) were graphically solved using the following functions [11]:

$$\eta = \eta_0 \exp(A/T + 273), \; \lambda = \lambda_{30}[1 - B(T - 30)], \tag{12}$$

characteristic for magnetic fluids based on silicone carriers, which are used in vacuum magnetofluid seals [12]. Here, $\eta_0 = 8.7 \cdot 10^{-5}$ N·sec/m²; A = 1864.2°; λ_{30} = 0.153 W/m·deg; $B = 11.3 \cdot 10^{-4}$ deg⁻¹.

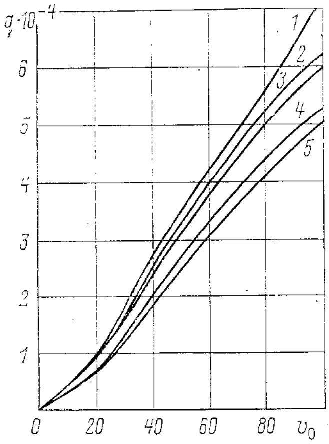

Fig. 2

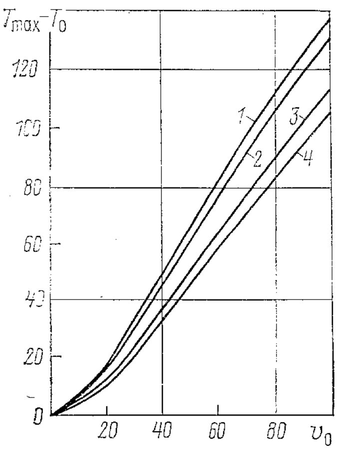

Fig. 3

Fig. 2. Dependence of the magnitude of the heat flux passing through the boundaries of the sealing layer on the velocity of rotation of the surface of the hermetically sealed shaft under isothermal boundary conditions for different values of the coefficient of heat transfer through the end faces: 1) calculation using the analytic expression (8) with $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$ with $\delta = 10^{-3}$ m; 2-5) numerical solutions of the self-consistent problem for $\delta = 10^{-3}$ m with $\alpha = 0$, $1.5\cdot10^5$, $7\cdot10^5$, and 10^6 W/(m$^3\cdot$deg). q, W/m^2; v_0, m/sec.

Fig. 3. Effect of heat transfer through the end faces on the magnitude of the maximum heating of the sealing fluid: 1) $\alpha = 0$; 2) $1.5\cdot10^5$; 3) $7\cdot10^5$; 4) 10^6 W/(m$^3\cdot$deg).

The solutions of the transcendental equations (6) and (10) are presented in curve 1 in Fig. 1a and curves 1, 3, and 6 in Fig. 1b has a function of the parameters. Curve 2 in Fig. 1a and curves 2, 4, and 7 in Fig. 1b were obtained as a result of calculations using relations (6) and (10) with the values $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$, where the average temperature were taken as the roots of the transcendental equations (7) and (11). As v_0 increases, the divergence between the curves, corresponding to the same boundary conditions, but constructed for different T_*, increases.

A numerical solution permits taking into account systematically the dependence of the thermophysical characteristics on temperature, and it also permits estimating the time for the system to reach the steady thermal state. Assuming that it corresponds best to reality, it is possible to establish which of two assumptions is most useful for approximate calculations. In this work, we used the implicit conservative finite-difference scheme [13]:

$$v_{i-1}^{n+1}\,\frac{\eta_{i-1/2}^{n+1}}{h^2}+v_i^{n+1}\left(-\frac{\eta_{i-1/2}^{n+1}+\eta_{i+1/2}^{n+1}}{h^2}-\frac{\rho_i^{n+1}}{\tau}\right)+v_{i+1}^{n+1}\,\frac{\eta_{i+1/2}^{n+1}}{h^2}=-\frac{\rho_i^{n+1}}{\tau}\,v_i^n,$$

$$T_{i-1}^{n+1}\,\frac{\lambda_{i-1/2}^{n+1}}{h^2}+T_i^{n+1}\left(-\frac{\lambda_{i-1/2}^{n+1}+\lambda_{i+1/2}^{n+1}}{h^2}-\frac{\rho_i^{n+1}\,c_i^{n+1}}{\tau}\right)+$$

$$+\,T_{i-1}^{n+1}\,\frac{\lambda_{i+1/2}^{n-1}}{h^2}=-\frac{\rho_i^{n+1}\,c_i^{n+1}}{\tau}\,T_i^n-\eta_i^{n+1}\left(\frac{v_{i+1}^{n+1}-v_{i-1}^{n+1}}{2h}\right)^2-\alpha\,(T_1-T_i^{n+1}).$$

The calculation was carried out using the method of intervals on a uniform grid with parameters $h/\delta = 1/20$ and $\tau = 1/20$ sec. Three iterations were carried out in each time layer. The dependences in [11] were used for the density and heat capacity: $\rho = \rho_{30}[1 - \beta(T - 30)]$, $c = c_{30}[1 + \gamma(T - 30)]$, where $\rho_{30} = 950.6$ kg/m^3; $\beta = 8.8\cdot10^{-4}$ deg^{-1}; $c_{30} = 1.499\cdot10^3$ J/kg$\cdot$deg; $\gamma = 0.193\cdot10^{-2}$ deg^{-1}.

Analysis of the results obtained showed that with a sudden startup in the rotation of the shaft, the time for heating the sealing layer to maximum temperature was 0.1-0.3 sec, after which the steady-state thermal regime is observed. As the thickness of the layer decreases, this time decreases proportionally to δ^{-2}. On the strength of the smallness of the heating time, in this work we present only the steady-state dependences, although the investigation was carried out in the entire volume. The stationary solution of problem (1)-(3) is presented in curve 3 in Fig. 1a, i.e., the analytic solution of (6) is closest to the numerical solution for $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$. The stationary solution of system (1)-(2) with boundary conditions (4) is presented in Fig. 1b (curves 2, 5, and 8). These results also indicate the fact that the higher accuracy of calculations is attained with the use of analytic expression (10) with $T_* = \langle T\rangle$.

Since the disagreement between the numerical and analytic results in the velocity range examined does not exceed 13% for approximate calculations, expression (6) and (10) can be used for the maximum temperature in the working gap of the magnetofluid seal, if the transcendental equations (7) and (11) for $\langle T\rangle$ taking into account the dependences (12) are solved beforehand.

In order to calculate the limiting rotational velocity of the shaft with the magnetofluid seal from the permissible thermal loads, it is only necessary to invert the functional dependences (6) and (10) for v_0, or to find the corresponding values of v_0 from the graphs in Figs. 1a and b. Under isothermal boundary conditions, a convenient approximation can be given:

$$v_0 = [18.2\,(T_{max} - T_0)\exp(T_{max}/100)]^{0.5},$$

obtained from curve 3 in Fig. 1a.

Since the quantity T_{max} is bounded from above by the maximum allowable temperature of the magnetofluid, in order to increase the velocity of rotation of a hermetically sealed shaft, it may become necessary to maintain, using a cooling system, the refrigerant cooling the sealing layer at a negative temperature, the magnitude of which in its turn, is limited below by the freezing point of the magnetic fluid. Undoubtedly, such a cooling system, involving the use of cryogenic technology, introduces considerable complications into the structure of the sealing unit. However, it should be noted that when hermetically sealing gaseous helium flows, as done, e.g., in turbogenerators with superconducting excitation winding, the magnetic fluid seal can be cooled by the hermetically sealed medium itself [14].

The dependence of the magnitude of the heat flux q passing through the boundary of the sealing layer on the linear velocity of rotation of the surface of the hermetically sealed shaft v_0 is shown in Fig. 2. Curve 2 was constructed using the results of the numerical solution of the system (1)-(2) with isothermal conditions (3) and $\alpha = 0$; curve 1 was constructed with the help of the analytic expression (8) and the values $T_* = \langle T\rangle$, obtained by solving the transcendental equation (7) for $\langle T\rangle$. The gap δ, as before, is assumed to be equal to 10^{-3} m. The maximum disagreement between the numerical and analytic results is 13%, i.e., the use of expression (8), corrected by the transcendental equation (7) for $\langle T\rangle$, is justified in practice here as well. Curve 2 in Fig. 2 is well approximated by the equation

$$q = v_0^2 \exp(-15.7 - 0.0001\, v_0^2).$$

A numerical experiment over a wide range of variation of the quantity α showed that heat transfer from the end faces of the surfaces of the fluid plug does not have a significant effect on the temperature regime in the layer (curves 2-5 in Fig. 2 and curves 1-4 in Fig. 3). Figure 4 shows the effect of the temperature dependences of the thermophysical characteristics of the packing on the hydrodynamics and heat transfer in the layer. Curves 2, 4, and 6 in Figs. 4b were constructed based on the numerical solution of the system (1)-(2) with boundary conditions (4); curves 1, 3, and 5 represent the analytic expression (9), corrected with the help of the solutions of the transcendental equation (11) for $\langle T\rangle$ with corresponding values of q_0. Here, the previously arrived at conclusion concerning the possibility of taking into account approximately the temperature dependence of the thermophysical characteristics in the final analytic expressions by transforming the equations for calculating $\langle T\rangle$ into transcendental equations, which simplifies the procedure of the thermal calculation, is confirmed here as well.

The numerical and analytical procedures constructed for calculating the maximum temperatures in the working gap of a magnetofluid seal permit calculating the limiting velocity of rotation of a hermetically sealed shaft and to determine, from the given characteristics of the fluid magnetic seal, the possible range of operation of the seal. In engineering calculations, it is permissible to use the analytic procedure. It is useful to use the numerical algorithm when high accuracy is necessary for the calculation and for analyzing the transient period. The calculations carried out show that for a seal based on silicone and a

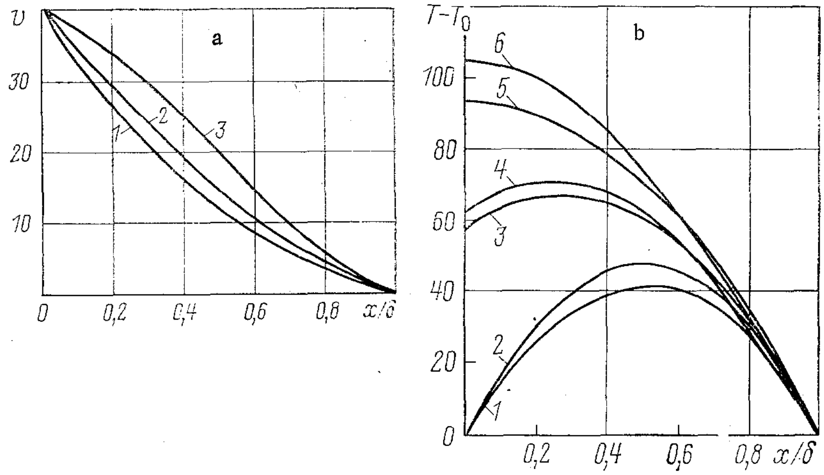

Fig. 4. Velocity and temperature fields in the working gap of a magnetofluid seal taking into account the temperature dependence of the thermophysical characteristics under mixed boundary thermal conditions: a) velocity profile: 1) $q_0 = 0$; 2) 10^4; 3) $2.5 \cdot 10^4$ W/m^2; b) temperature profile: 1, 3, and 5 are a calculation using the analytic expression (9) with $\eta(\langle T\rangle)$ and $\lambda(\langle T\rangle)$ for $q_0 = 2.5 \cdot 10^4$ W/m^2 and 10^4 W/m^2 and 0 W/m^2; 2, 4, and 6) are the numerical solution with $q_0 = 2.5 \cdot 10^4$ W/m^2, 10^4 W/m^2, and 0. δ, m and T, °C.

cooling system that maintains a temperature of $T_0 = 10$°C at the boundary of the layer, it is impossible to stabilize the temperature in the fluid at a level of 120°C, if the linear rate of rotation of the shaft does not exceed $v_0 = 80$ m/sec. We note that in order to take into account the thermal resistance of the section between the refrigerant and the boundary of the sealing layer, it is necessary to solve the conjugate problem.

NOTATION

T_{max}, maximum temperature of heating of the sealing fluid, °C; δ, thickness of the sealing layer, m; v_0, linear velocity of rotation of the surface of the hermetically sealed shaft, m/sec; ρ, density, kg/m^3; η, viscosity, N•sec/m^2; c, specific heat capacity at constant pressure, J/(kg•deg); λ, coefficient of thermal conductivity, W/(m•deg); α, transfer coefficient, W/(m^3•deg); q, heat flux, W/m^2.

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THERMODIFFUSION OF ^{32}S AND ^{34}S IN SULFUR HEXAFLUORIDE

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The article describes the experimental investigation of the separation of sulfur isotopes in a thermodiffusion column.

To obtain sulfur hexafluoride with an isotope composition different from the natural one, it becomes necessary to investigate whether thermodiffusion could be used for this purpose.

We know of only one work that presents the results concerning sulfur isotope enrichment in the gaseous phase [1]. The working medium used was sulfur dioxide. The separation efficiency was low because of the small reduced thermodiffusion constant which amounted to no more than 0.0143.

The glass column used by the authors of [1] was 3.5 m high, with 11.6-mm diameter of the working cylinder; it was heated by an electric heater of platinum wire with d_2 = 0.4 mm. The wire was centered by washers welded to it every 30 cm. It was heated to 100°K. The optimum pressure was 34.6 kPa at which the maximum degree of separation within 220 h was q = 1.065 between the isotopes ^{32}S and ^{34}S.

This result did not provide grounds for assuming that the use of sulfur hexafluoride would make it possible to attain a noticeable shift in the isotope composition. On the other hand, a calculation by the formulas of the kinetic-molecular theory on the assumption of the Lennard-Jones interaction potential being applicable to sulfur hexafluoride yielded the value $\alpha_0 = +0.254$ for $T^* = 2$ and $+0.431$ for $T^* = 3$, where the reduced temperature is $T^* = kT/\varepsilon$, and ε/k = 200.9°K according to data taken from [2]. On the basis of these calculations we undertook the experimental verification of the possibility of separating the sulfur isotopes in SF_6 in a thermodiffusion column (Fig. 1).

The column consisted of two concentric cylinders, one of which was the tubular electric heater (TEH), and the second a brass tube. The working height of the column was 2.4 m, the working gap 2.25 mm; its accuracy was ensured by centering bosses of which four in each section were welded 300 mm apart to the TEH. The brass tube was cooled by circulating water. The ends of the TEH were sealed by stuffing boxes which ensured air-tight sealing in case of thermal expansion. The column was firmly mounted on a bracket, and its vertical position in two mutually perpendicular directions was checked by plumbs. The column was provided with nozzles for sampling and evacuation.

In choosing the temperature regime of column operation, we took into account the chemical properties of sulfur hexafluoride which becomes chemically active at elevated temperatures (>500°C) [3]. The experiments were therefore carried out at a temperature of the hot surface of 427°K and a mean temperature $\overline{T}$ = 357°K. The temperature was measured on the basis of the change of pressure in the thermodiffusion column. If, with the heater switched off and water with T_1 = 287°K circulating through the cooler, the pressure P_0 established itself, and after

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