HEAT AND MASS TRANSFER DURING SUBLIMATION – CONDENSATION IN A CYLINDRICAL ANNULAR GAP

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The sublimation – condensation process is studied at a pressure below the pressure at the triple point in a narrow annular channel in order to assess the possibility of maintaining a constant temperature of an object in a vacuum under conditions of heating at a single side.

An extremely high effective thermal conductivity can be achieved by using heat-transfer equipment (heat pipes) in which the heat transfer occurs as a result of a double phase transition of the heat carrier, with this carrier being continuously returned to the liquid phase through a wick in the evaporation zone [1]. The efficiency of the device can be raised at a pressure of the vapor (sublimate) below the pressure at the triple point, if there is wick-free transport of the solid condensate to the sublimation zone. An obvious way to achieve this situation is to use a heat-transfer ring rotating around its symmetry axis. This ring would be a slotted channel, filled with the sublimating heat carrier, between two thin, coaxial, cylindrical or conical shells.

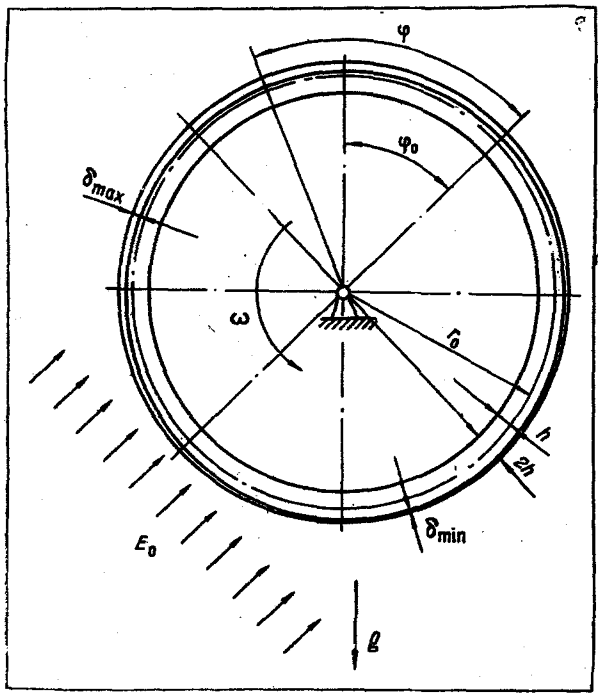

Fig. 1. Distribution of the solid condensate in a cylindrical annular gap.

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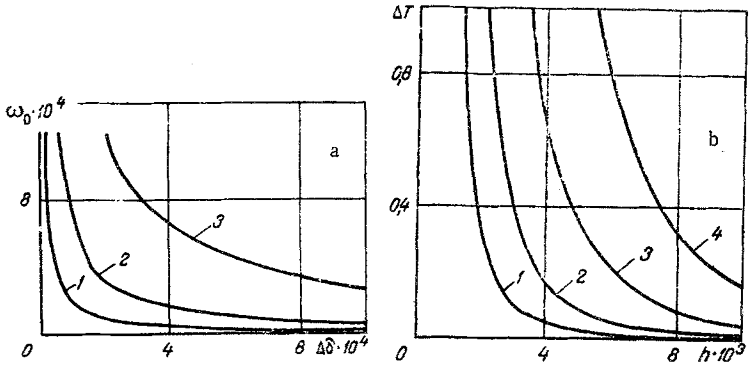

Fig. 2. Calculated curves for the thermal balancing of a cylindrical ring with $r_0 = 2$ m for E = 350, 1400, 5600, and 22,400 W/m^2 (curves 1-4, respectively). a) Effects of the rotation velocity of the ring on the distribution of solid condensate; b) dependence of the accuracy of the thermal balancing on the width of the cylindrical annular gap, 2h. Here $\omega_0 \cdot 10^4$ is expressed in sec^{-1}, $\Delta\delta \cdot 10^4$ is expressed in m, ΔT is expressed in °K, and $h \cdot 10^3$ is expressed in m.

Such arrangements can be used to effectively level the temperature field in appropriate thin shells at low angular velocities ω of an object subjected to heating from a single side in vacuum.

The case of most practical interest is that of cylindrical and conical heat-transfer rings, since in spherical shells the solid condensate eventually accumulates near the rotation axis.

In the present paper we restrict the analysis to the case of a temperature-balancing ring (Fig. 1) consisting of two coaxial cylindrical shells. The narrow gap between these shells ($2h \ll r_0$) is originally purified of noncondensing gases and filled with a certain amount of the working medium (heat carrier), whose triple-point temperature is above the temperatures maintained in the thermostat. To avoid end effects, we assume that both bottoms are ideal thermal insulators. The object is in a vacuum and subjected to heating from a single side by a radiative heat flux in the direction perpendicular to the rotation axis. Furthermore, the outer shell radiates energy into free space in accordance with the Stefan – Boltzmann law. Accordingly, the resultant heat flux across the outer shell is

$$q = \begin{cases} -\varepsilon\sigma T_w^4 \ \text{for} \ \varphi < \dfrac{\pi}{2} \ \text{and} \ \varphi > \dfrac{3\pi}{2}, \\ -A_s E\cos\varphi - \varepsilon\sigma T_w^4 \ \text{for} \ \dfrac{\pi}{2} < \varphi < \dfrac{3\pi}{2}. \end{cases} \tag{1}$$

If radiative heat transfer occurs between the elements of the inner shell and the inner side, the specific heat flux across the corresponding shell is

$$q_1 = \varepsilon_1 q^0 - \varepsilon_1 \sigma T_{w_1}^4. \tag{2}$$

We assume that the length of the cylinder, l, is much larger than its radius, r_0. Accordingly, in the approximation of infinite cylindrical shells it is simple to show that the specific heat flux q^0 to a surface which radiates diffusely (according to the cosine law) is independent of the angular coordinate ($dq^0/d\varphi = 0$; the end effect is neglected):

$$q^0 = \frac{\sigma}{2\pi}\int_0^{2\pi} T_{w_1}^4\, d\varphi. \tag{3}$$

The inner surface of the outer wall of the annular channel is coated with a layer of solid condensate, whose profile is given below. The rate of phase transitions (sublimation – condensation) is given by

$$J_m = \frac{q + q_1}{L} - \frac{\omega\,(c'\rho'\delta' + c_w\rho_w\delta_w)}{L}\,\frac{dT_m}{d\varphi}. \tag{4}$$

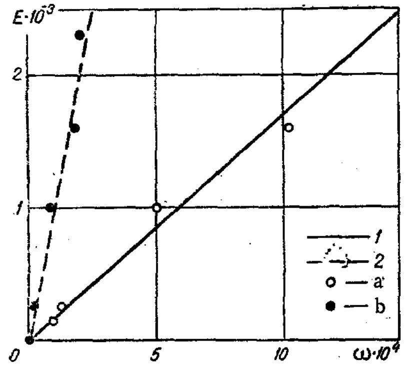

Fig. 3. Rotation velocity of the ring as a function of the heat flux density. a, b) Experimental data; 1, 2) calculated data; a, 1) $\varphi_0 = 45°$; b, 2) $\varphi_0 = 83°$. Here $E \cdot 10^{-3}$ is in W/m^2 and $\omega \cdot 10^{-4}$ is expressed in sec^{-1}.

We neglect the heat flow along the shells themselves. The gap is assumed to be relatively narrow: $2h/r_0 \ll 1$. Then the properties of the sublimate flowing in the gap can be characterized by the potential Ψ, introduced in [2] or [3], if we wish to take into account the effect on this flow of the thermal inhomogeneities across the slotted channel, and if we wish to take into account the temperature dependence of the coefficients μ and λ. In the present case it is not very important to take these factors into account, since by its very nature the rotating ring is a thermostat, which should hold the thermal inhomogeneities along the surface to a low level, so that there will be small temperature drops across the gap also. Since the external heat supply is assumed independent of the axial coordinate, the gasdynamic parameters of the sublimate flowing in the gap also have this property.

Accordingly, following [2], we can write

$$\frac{d^2\Psi}{d\varphi^2} = -\frac{3\mu r_0^2 J_m}{2h^3}; \quad \Psi = \int_{p_*}^{p} \Phi(p)\,dp,$$

$$\Phi(p) = \frac{p}{RT} + \frac{2-\theta}{\theta}\frac{3.78\mu}{h\sqrt{RT}} - \frac{9}{4}\frac{\mu^2}{h^2}\frac{d\ln F(p)}{dp}, \tag{5}$$

where

$$F(p) = \frac{RT_*/L}{1-(RT_*/L)\ln(p/p_*)}.$$

In Eq. (5) we have neglected the additional sublimate velocity due to rotation of the thermostat, since an angular velocity $\omega \ll V/r_0$ [see (18)] is sufficient for effective operation of this thermostat.

The temperatures T_W and T_{W_1} are related to the temperature T(p) of the phase-transition surface (we neglect the phase resistance [2]; i.e., we assume that this latter temperature is related by the Clausius – Clapeyron equation to the pressure p of the sublimate moving over the surface) by the following equations:

$$T_w - T(p) = qR_w; \quad T_{w_1} - T(p) = q_1 R_{w_1},$$

$$R_w = \frac{\delta_w}{\lambda_w} + \frac{\delta'}{\lambda'}; \quad R_{w_1} = \frac{\delta_{w_1}}{\lambda_{w_1}} + \frac{1}{\alpha_l + \lambda/2h}; \quad T(p) = T_* F(p). \tag{6}$$

We assume $\delta' \ll 2h$. The thermal resistances need not be evaluated highly accurately here, since all the temperatures which appear in these equations are approximately the same.

Below we assume that the thermal resistance of the layer of solid condensate, δ'/λ', is negligibly small; i.e., we assume $R_W = \delta_W/\lambda_W$.

By introducing the potential Ψ we can describe the flow in the gap by differential equation (5), whose right side contains a small nonlinearity due to the dependence of $(q + q_1)$ on the sublimate temperature. Accordingly, we should expand T_W^4 and $T_{W_1}^4$ in Taylor series around some average value of the fourth power of the sublimate temperature, T_0^4. For this purpose we introduce $t(p) = T(p) - T_0$; $t(p) \ll T_0$. Then using only the first two terms from (3) and (6), we find

$$q^{(0)} = \sigma\left[T_0^4 + \frac{2T_0^3}{\pi}\int_0^{2\pi} t(p)\,d\varphi\right]. \tag{7}$$

Analogously, from (1), (2), and (6) we find

$$q=\frac{A_sEf(\varphi)-\varepsilon T_0^4\sigma[1+4t(p)/T_0]}{1+4\varepsilon\sigma T_0^3R_w};\tag{8}$$

$$q_1=\frac{\varepsilon_1q^0-\varepsilon_1\sigma T_0^4[1+4t(p)/T_0]}{1+4\varepsilon_1\sigma T_0^3R_{w_1}};\ f(\varphi)=\begin{cases}0 \text{ for } \varphi<\frac{\pi}{2} \text{ and } \varphi>\frac{3\pi}{2};\\ -\cos\varphi \text{ for } \frac{\pi}{2}<\varphi<\frac{3\pi}{2}.\end{cases}$$

Since for the pressure drops Δp in the annular gap of the thermostat we have

$$\frac{\Delta p}{T_0}\frac{dT}{dp}\ll 1,$$

we can write

$$t(p)=\frac{(dT/dp)_0}{(d\Psi/dp)_0}(\Psi-\Psi_0)=\frac{RT_0^2}{Lp_0\Phi(p_0)}(\Psi-\Psi_0).\tag{9}$$

We show below that for a working medium which has a high heat of sublimation (water), and at angular velocities sufficient for effective operation of the thermostat, the second term on the right side of (4) is negligibly small. Then using (7)-(9) we can rewrite Eq. (5) as

$$\frac{d^2\Psi}{d\varphi^2}-N_1\Psi=D-MHA_sEf(\varphi),\tag{10}$$

where

$$H=(1+4\varepsilon\sigma T_0^3R_w)^{-1};\ H_1=(1+4\varepsilon_1\sigma T_0^3R_{w_1})^{-1};$$

$$M=\frac{3\mu r_0^2}{2h^3L};\quad N=\frac{6\sigma RT_0^5\mu r_0^2}{h^3L^2p_0\Phi(p_0)};\ N_1=(\varepsilon H+\varepsilon_1H_1)N;$$

$$D=M\varepsilon H\sigma T_0^4-N\left[\varepsilon_1H_1\int_0^{2\pi}(\Psi-\Psi_0)\,d\varphi-2(\varepsilon H+\varepsilon_1H_1)\Psi_0\right].$$

Since $\omega\ll V/r_0$, we can assume that the flow is symmetric about the $\varphi=0$ axis. Then the potential Ψ must satisfy the boundary conditions

$$\frac{d\Psi}{d\varphi}=0 \text{ for } \varphi=0 \text{ and } \varphi=\pi.\tag{11}$$

A solution of Eq. (10) satisfying the first boundary condition is

$$\Psi=C\operatorname{ch}(\sqrt{N_1}\,\varphi)+\frac{1}{\sqrt{N_1}}\int_0^{\varphi}[D-MHA_sEf(\vartheta)]\operatorname{sh}[\sqrt{N_1}(\varphi-\vartheta)]\,d\vartheta.$$

For $\varphi<\pi/2$

$$\Psi-\Psi_0=\frac{MHA_sE}{(1+N_1)\sqrt{N_1}}\,\frac{\operatorname{ch}(\sqrt{N_1}\,\varphi)-1}{2\operatorname{sh}(\sqrt{N_1}\,\pi/2)},$$

while for $\pi/2<\varphi<3\pi/2$

$$\Psi-\Psi_0=\frac{MHA_sE}{1+N_1}\left\{\frac{\operatorname{ch}[\sqrt{N_1}(\pi-\varphi)]-1}{2\sqrt{N_1}\operatorname{sh}(\sqrt{N_1}\,\pi/2)}-\cos\varphi\right\}.\tag{12}$$

By virtue of (9) the maximum temperature drop of the sublimation surface is

$$T(\pi)-T_0=t[p(\pi)]=\frac{1}{1+N_1}\,\frac{3\mu RT_0^2r_0^2}{2h^3L^2p_0\Phi(p_0)}\,\frac{A_sE}{1+4\varepsilon\sigma T_0^3R_w}.\tag{13}$$

From the condition for steady-state thermal conditions of the ring, $\int_0^{2\pi} q\,d\varphi=0$, and using (9) and (12), we find a transcendental equation for the unknown temperature $T_0=T(0)$:

$$T_0^4+\frac{\varepsilon H}{\varepsilon H+\varepsilon_1 H_1}\,\frac{A_s E}{2\pi\sigma(1+N_1)}\,\frac{2(1+N_1)\,\mathrm{sh}\left(\sqrt{N_1}\,\frac{\pi}{2}\right)-\sqrt{N_1}\,\pi}{\mathrm{sh}\left(\sqrt{N_1}\,\frac{\pi}{2}\right)}=\frac{A_s E}{\pi\varepsilon\sigma}. \tag{14}$$

Neglecting the second term in (4), as in the derivation of (10), we write an equation for the steady-state profile of the layer of solid condensate:

$$\omega\frac{d\delta'}{d\varphi}=-\frac{q+q_1}{L\rho'};$$

we thus have

$$\delta'=\delta_0'+\frac{H}{\omega L\rho'}\left\{\varepsilon\sigma T_0^4\,\varphi-A_s E f_1(\varphi)-\frac{A_s E}{2(1+N_1)}\left[\frac{\varepsilon_1 H_1 f_2(\pi)}{\varepsilon_1 H_1+\varepsilon H}\,\frac{\varphi}{\pi}-f_2(\varphi)\right]\right\}. \tag{15}$$

where

$$f_1(\varphi)=\begin{cases}0 & \text{for } \varphi<\frac{\pi}{2};\\ 1-\sin\varphi & \text{for } \frac{\pi}{2}<\varphi<\pi;\end{cases}$$

$$f_2(\varphi)=\begin{cases}\dfrac{\mathrm{sh}(\sqrt{N_1}\,\varphi)-\sqrt{N_1}\,\varphi}{\mathrm{sh}(\sqrt{N_1}\,\pi/2)} & \text{for } \varphi<\frac{\pi}{2};\\[2ex] \dfrac{2[1+N_1(1-\sin\varphi)]\,\mathrm{sh}\left(\sqrt{N_1}\,\frac{\pi}{2}\right)-\mathrm{sh}[\sqrt{N_1}\,(\pi-\varphi)]-\sqrt{N_1}\,\varphi}{\mathrm{sh}\left(\sqrt{N_1}\,\frac{\pi}{2}\right)} & \text{for } \frac{\pi}{2}<\varphi<\pi.\end{cases}$$

By virtue of (13) we have

$$\frac{N}{1+N_1}=\frac{4\sigma T_0^3}{A_s E}(1+4\sigma\varepsilon T_0^3 R_w)\,\Delta T,\ \text{where}\ \Delta T=T=T(\pi)-T(0).$$

Below we will see that the temperature $T_0=273°K$ is reached at $A_s\approx 0.13$ ($E=1392$ W/m^2, $\varepsilon=0.18$). Accordingly, for water near the triple point we have

$$\frac{N}{1+N_1}=0.0258\Delta T\,(1+4\varepsilon\sigma T_0^3 R_w),$$

where $N_1\le 2N$, since $H\le 1$ and $H_1\le 1$.

If the thermal resistance of the outer shell, R_w, is small in comparison with the effective resistance $(4\varepsilon\sigma T_0^3)^{-1}$ corresponding to radiation into space, we have $N/(1+N_1)\approx 0.0258\Delta T$; i.e., for $\Delta T<1°K$ we have $N\lessdot 0.0258$ ($N_1\lessdot 0.0516$), and for $\Delta T>0.1°K$ we have $N\lessdot 0.00258$ ($N_1\lessdot 0.00516$).

Neglecting quantities on the order of N^2, we find instead of (14) and (15) the following:

$$T_0=\sqrt[4]{\frac{A_s E}{\pi\varepsilon\sigma}}\left[1-\frac{N}{4}\,\varepsilon^2 H\left(1-\frac{\pi^2}{24}\right)\right], \tag{16}$$

$$\delta'=\delta_0'+\frac{HA_s E}{\omega L\rho'}\left\{\frac{\varphi}{\pi}-f_1(\varphi)-N\left[B\,\frac{\varphi}{\pi}-(\varepsilon H+\varepsilon_1 H_1)\,\bar{f}_2(\varphi)\right]\right\}. \tag{17}$$

where

$$B=\varepsilon^2 H\left(1+\frac{\pi^2}{24}\right)+\frac{\varepsilon_1 H_1}{2}\left(1+\frac{\pi^2}{12}\right);$$

$$\bar{f}_2(\varphi)=\begin{cases}\dfrac{1}{3}\,\dfrac{\varphi^3}{\pi} & \text{for } 0<\varphi<\frac{\pi}{2};\\[2ex] \dfrac{1}{\pi}\left[\dfrac{\pi^3}{12}+\pi(1-\sin\varphi)-\dfrac{(\pi-\varphi)^3}{3}\right] & \text{for } \frac{\pi}{2}<\varphi<\pi.\end{cases}$$

Since we have $N \ll 1$ if the temperature equalization is sufficiently effective, we can substitute into these latter expressions the value of N calculated for $T_0 = \sqrt[4]{A_S E / \pi \varepsilon \sigma}$.

The extremal thicknesses of the layer of solid condensate are found from the condition $d\delta'/d\varphi = 0$.

If we neglect quantities on the order of N, we find that in this case the maximum and minimum of δ' correspond to the angles

$$\varphi = \arccos(-1/\pi); \quad \varphi = 2\pi - \arccos(-1/\pi);$$

$$\delta'_{\max} = \delta'_0 + 0.545 \frac{HA_s E}{\omega L \rho'}; \quad \delta'_{\min} = \delta'_0 - 0.545 \frac{HA_s E}{\omega L \rho'}.$$

Accordingly, if the thickness δ' is to be smaller than δ'_1 and larger than δ'_2 everywhere, the thermostat must rotate at an angular velocity

$$\omega \geqslant \omega_0 \simeq 1.09 \frac{HA_s E}{L\rho' (\delta'_2 - \delta'_1)}. \tag{18}$$

As an illustration we consider an annular thermostat (Fig. 1) rotating about an axis perpendicular to the flux of solar radiation ($E = 1392\ W/m^2$), with $A_S = 0.13$, $\varepsilon = 0.18$, $R_W = 0$, and $T_0 \simeq (1 - 0.286N) \cdot 273°K$. If the working medium (heat carrier) is water [$L = 3.06 \cdot 10^6$ J/kg, $R = 461.36\ m^2/(sec^2 \cdot deg\ K)$, and $\mu = 0.81 \cdot 10^{-5}\ N \cdot sec/m^2$], then with $r_0 = 2$ m the gap height 2h must be at least 0.004 m in order to achieve precise thermal equalization, $\Delta T = T(\pi) - T(0) \leq 1°K$ (curve 2 in Fig. 2b); to achieve $\Delta T \leq 0.1°K$ we would need $2h \geq 0.009$ m. As the heat flux density E and the radius r_0 are increased at small values of h, the maximum temperature drop at the sublimation surface increases rapidly.

For this thermostat, whose axis is in a horizontal plane, the earth's gravitation will produce a torque due to the nonuniformity of the distribution of the layer of solid condensate with respect to the angle φ. If this torque is larger than the moment of the static friction in the bearings, the thermostat will begin to rotate. The torque is

$$M_{to} = \rho' g r_0^2 l \int_0^{2\pi} \delta' \sin(\varphi - \varphi_0)\, d\varphi.$$

If the friction moment is $M_{fr} = \text{const}$ (dry friction in the bearings), by substituting (17) into this latter equation and neglecting quantities on the order of N, we find from the condition $M_{to} = M_{fr}$ the angular velocity of the thermostat ($-\pi/2 < \varphi_0 < \pi/2$)

$$\omega = \frac{\pi g r_0^2 l H A_s E \cos\varphi_0}{L M_{tr}}. \tag{19}$$

In the case of a linear function $M_{fr} = K\omega$ (viscous friction in the bearings) we have

$$\omega = \sqrt{\frac{\pi g r_0^2 l H A_s E \cos\varphi_0}{LK}}. \tag{20}$$

The minimum necessary angular velocities calculated from Eq. (18) for $R_W = 0$ ($H = 1$) are shown by curves 1-3 in Fig. 2a. Accordingly, with $E = 1392\ W/m^2$ and $\delta'_2 - \delta'_1 = 0.05$, 0.10, and 1.00 mm, a single revolution of the thermostat should be completed in time intervals no longer than 1.35, 2.70, and 27 h, respectively.

A model thermostat has been developed for an experimental test. It is of light construction, consisting of an outer ring (0.2 m in diameter, $\delta = 0.003$ m, and $l = 0.07$ m) and an inner (Plexiglas) ring, which together form an annular insulated gap of width 0.001 m. This gap communicates with three apertures at the axis by means of six hollow "needles" (thin-walled tubes 0.002 m in diameter, made of stainless steel). The last of these needles is mounted on ruby jewel bearings, and the entire assembled drum is mounted on type VLTK-500 scales in a pressure chamber held at a constant temperature between a cooled nitrogen shield and a flat electric heater, mounted in the pressure chamber at an angle of $\varphi_0 = 83°$ (45°). All the surfaces of the ring held at the constant temperature; the nitrogen shield, and the electric heater which face each other are coated with a thin layer of lamp black. The ring can be easily cranked with an unbalanced 0.001 kg at a lever arm of $r_0 = 0.1$ m. After a careful static balancing of the ring in its annular gap, degassed doubly-distilled water is supplied to the slotted gap through the hollow needles; the gases which have not condensed are evacuated from the pressure chamber, and a directed heat flux of the specified density is produced from a single-sided radiative heat source. After the final distribution of the solid condensate in the gap is produced, with the desired operating conditions of this gap as a thermostat established, we measure the constant rotation velocity of the ring. Figure 3 shows data obtained in these experiments along with those calculated from Eq. (19). Since the discrepancy does not exceed 20%, we can assume that the raw data used for the analytic theory are valid. It should be noted that when the ring begins to operate, and while uncondensed gases are still present in the gap, the condensation in a solid state occurs in the form of isolated crystals and complexes of such crystals. Later, as the residual gases are displaced from the gap by the pure water vapor (the sublimate), a more continuous layer of the solid condensate (ice) forms on the inner side of the Dural ring. The loss of sublimate mass from the gap is continuously measured and is taken into account in the heat- and mass-balance equations.

In conclusion, we evaluate the ratio of the second term on the right side of (4) to the first term, which is a measure of the heat supplied to the phase-transition surface. Using (5), (8), (9), (12), and (18), we find ($\omega \sim \omega_0$):

$$\frac{\omega c' \rho' \delta_0' dT/d\varphi}{q + q_1} \sim \gamma = \frac{c' \delta_0' R T_0^2 \mu r_0^2 H A_s E}{L^3 p_0 \Phi(p_0)(\delta_2' - \delta_1') h^3}. \tag{21}$$

For water with $\delta_2' - \delta_1' = \delta_0'$, $T_0 = 273°K$, $r_0 = 2m$, and $2h = 0.004$; 0.01, and 0.02 m we find for $\gamma = 10^{-3}$; $6.4 \cdot 10^{-5}$; and $8 \cdot 10^{-6}$, respectively.

Accordingly, for practical calculations of heat-transfer rings with a heat carrier with a high heat of sublimation, the assumption we have used here is completely justified.

NOTATION

J_m, sublimation rate; q, q_1, q^0, heat flux densities across the outer shell, the inner shell, and the inner surface of the ring, respectively; E, intensity of the parallel beam of shortwave radiation; A_s, ε, ε_1, absorption coefficient for the shortwave radiation and emissivities of the outer and inner surfaces of the ring; σ, Stefan—Boltzmann constant; T_w, T_{w_1}, temperatures of the outer and inner ring surfaces; φ, angular coordinate; L, latent heat of sublimation; c', ρ', δ', λ', specific heat, density, thickness, and thermal conductivity of the solid condensate; T, p, ρ, μ, λ, temperature, pressure, density, dynamic viscosity coefficient, and thermal conductivity of the sublimate; p_*, T_*, parameters of the triple point; θ, coefficient of diffuse Maxwell reflection; r_0, 2h, l, radius of the mean surface and height and length of the generatrix of the annular cylindrical channel; ω, angular velocity of ring; V, typical sublimate velocity; R, gas constant of sublimate; δ_w, λ_w(δ_{w_1}, λ_{w_1}), thickness and thermal conductivity of the outer (inner) shell; α_r, Ψ, radiative heat-transfer coefficient and stream potential of the low-density gas in the slotted channel; $T_0 = T(0)$; $t(p) = T(p) - T_0$.

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