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TEMPERATURE FIELD OF THE ACTIVE ELEMENT OF A SOLID-STATE LASER WITH A LIQUID COOLING SYSTEM

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Results of a numerical solution of a nonstationary conjugate convective heat-transfer problem are presented. Temperature fields are determined in the coolant stream and in the body being cooled.

The distinguishing feature of solid-state laser operation in the continuous mode is the constant liberation of a part of the energy being pumped into the active element, as heat. Heating of the active element can result in a change in the laser generation characteristics, the appearance of substantial thermal deformations in the material, and failure of the element. In order to prevent overheating of the active element, a part of the heat being liberated there is eliminated through the side surface in the cooling system. In the majority of cases it is an optically transparent, external annular channel through which a liquid or gaseous coolant is pumped.

Existing methods for computing the temperature field in active elements [1, 2] are based on the traditional methods of computing the convective heat transfer in channels by using the Newton—Rikhman (Riemann) relationships in the equations or boundary conditions. However, it has already been shown in [3, 4] that such a formulation of the problem does not take account of the influence of the thermophysical properties and thickness of the wall material, as well as of the presence of internal heat sources, on the heat-transfer coefficient. Moreover, it is known that the index of refraction in the active element and the intensity of the thermal lens, that occurs, depend mainly on the temperature gradient over the radius and the length. The magnitudes of these gradients also determine the thermal stresses, disturbing the anisotropy of the active laser element.

In this connection, a conjugate problem must be solved to determine the temperature fields in the active element and coolant, and to take account of their mutual influence, i.e., a system of differential equations describing the heat transfer in the fluid and the solid must be solved. But it is first necessary to estimate for which cooling parameters is the traditional formulation of the heat-transfer problem possible by using the Newton—Rikhman relationships.

According to Lykov [3], it is customary to take the Bruhn number

$$\mathrm{Br}_z = \frac{\lambda_2}{\lambda_1}\,\frac{r}{z}\,\mathrm{Pr}^m\,\mathrm{Re}_z^n, \quad z \in [0,\ l] \tag{1}$$

as conjugate criterion.

If the Bruhn number is small ($\mathrm{Br}_z \leq \mathrm{Br}_{\min}$), then the convective heat transfer is computed by the traditional method. The quantity $\mathrm{Br}_{\min}$ is determined from estimates of the exact analytic solutions or by experiments. Often, the dimensionless number K, related to the Bruhn number:

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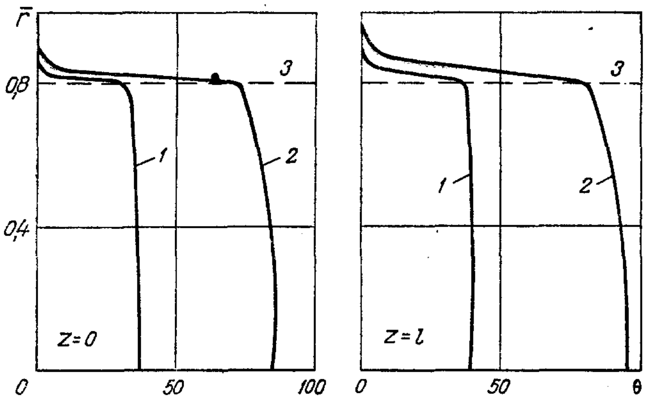

Fig. 1. Temperature distribution over sections of the active element and the cooling system channel: 1) τ = 1 sec; 2) τ = 6 sec; 3) active element surface.

$$K = \mathrm{Br}_z \left[2\mathrm{Pr}^m \left(\frac{r}{l} \right)^{0.5} \right]^{-1}, \tag{2}$$

is used in place of the Bruhn number to esimate the conjugate conditions.

According to [3, 4], if $\mathrm{Br}_{min} \geq 0.1$ but $K \geq 1$, then the convective heat-transfer problem must be solved as a conjugate problem.

Liquid and air cooling systems of existing and prospective solid-state lasers were analyzed in [5]. It was shown that the heat-transfer coefficient is in the 2000–7000-W/m^2 band when cooling the active element by a liquid stream. Values of the heat-transfer coefficient reach 10,000 W/m^2·K and more in cooling systems supported by a large excess head and not associated with a constraint on the fluid discharge.

Let us use the generalized dependences of Petukhov [6] for heat elimination in tubes of annular cross section in the case of heating just the inner wall (the active element):

$$\mathrm{Nu} = \frac{2\alpha (r_2 - r_1)}{\lambda_2} = \varphi_0 (\mathrm{Pr})\, \varphi_1 (\mathrm{Re}) \left(\frac{r_2}{r_1} \right)^{\varphi_2(\mathrm{Pr})}. \tag{3}$$

Assuming the length of the active element given, a transcendental equation relating the number K to the heat-transfer coefficient α can be obtained from (1)-(3) with the form of the function φ taken into account from [6] for known thermophysical properties of the active element and the cooling fluid:

$$\left(f_0 + f_1 \frac{1}{\sqrt[n]{K}} \right) \lg^2 a_0 K - f_2 \frac{1}{\alpha} \sqrt[n]{K} + f_3 \alpha = 0, \tag{4}$$

where

$$f_0 = \frac{56.66 (r_2 - r_1)}{n^2 \lambda_2} - \frac{16.68}{1 + 10\mathrm{Pr}} \frac{1}{n^2};$$

$$f_1 = \frac{47664 (r_2 - r_1)}{n^2 \lambda_2} \left[\frac{\lambda_2}{2\lambda_1} \left(\frac{r_1}{l} \right)^{0.5} \right]^{\frac{1}{n}};$$

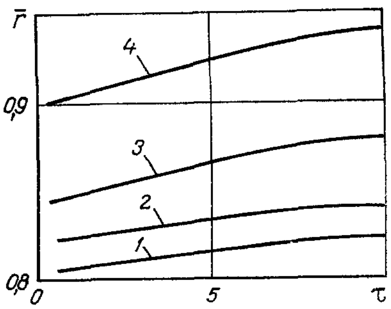

Fig. 2. Change in liquid thermal boundary layer boundary in a channel as a function of the time τ: 1) v = 0.25 m/sec, $W_2 = 0$; 2) v = 0.25 m/sec, $W_2 = 130$ W; 3) 0.1 m/sec and 0; 4) 0.1 and 130 W.

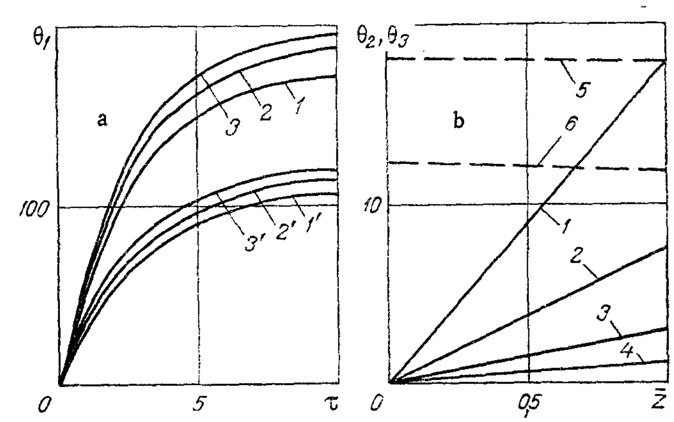

Fig. 3. Change in temperature along the axis and the radius of the active element: a) as a function of time for $z = l$ and $W_2 = 0$ (1 — $v = 1.0$ m/sec, $W_1 = 210$ W; 2 — 0.5 and 210, respectively; 3 — 0.25 and 210; 1' — 1.0 and 130; 2' — 0.5 and 130; 3' — 0.25 and 130) and b) on the dimensionless length for $\tau = 10$ sec and $W_2 = 0$ (1 — θ_2, $v = 0.25$ m/sec, $W_1 = 210$ W; 2 — θ_2, respectively 0.5 m/sec and 210; 3 — θ_2, 1.0 m/sec and 210; 4 — θ_2, 1.5 and 210; 5 — θ_3, 0.5-1.5 and 210; 6) θ_3, 0.5-1.5 and 130).

$$f_2 = \left(\mathrm{Pr} - \frac{0.45}{2.4 + \mathrm{Pr}}\right)\left[\frac{2\lambda_1}{\lambda_2}\left(\frac{l}{r_1}\right)^{0.5}\right]^{\frac{1}{n}}\left(\frac{r_2}{r_1}\right)^{0.16\mathrm{Pr}-0.16};$$

$$f_3 = \frac{71.96\,(\mathrm{Pr}^{2/3} - 1)\,(r_2 - r_1)}{\lambda_2};$$

$$a_0 = \frac{\lambda_1}{2\lambda_2 (7.9433)^n}\left(\frac{l}{r_1}\right)^{0.5}, \quad n = 0.5-0.8.$$

The dependence of the heat-transfer coefficient α on the volume mass-flow rate of the cooling system liquid for a solid-state laser is presented in [5]. By giving values of α, the value of the number K can be obtained from (4) for specific dimensions of the cooling system channel. It turns out that the number is $K > 1$ for the majority of real cooling system parameters of solid-state lasers, and the problem of confective heat transfer of the active element with a coolant must be solved as a conjugate problem.

The nonstationary conjugate problem of convective heat transfer for an annular channel with known profile of the axial liquid flow velocity is described by a system of two partial differential equations:

$$\rho_1 c_1 \frac{\partial T_1}{\partial \tau} = \frac{1}{r}\frac{\partial}{\partial r}\left[r\lambda_1 \frac{\partial T_1}{\partial r}\right] + \lambda_1 \frac{\partial^2 T_1}{\partial z^2} + W_1, \tag{5}$$

$$\rho_2 c_2 \left[\frac{\partial T_2}{\partial \tau} + v_z(r)\frac{\partial T_2}{\partial z}\right] = \frac{1}{r}\frac{\partial}{\partial r}\left[r\lambda_2 \frac{\partial T_2}{\partial r}\right] + W_2. \tag{6}$$

In the case under consideration, the heat fluxes through the active element endface can be neglected and only the side surface need to be taken into account [1, 2, 5]. Let us also neglect the wall thickness of the cooling system annular channel.

Under the assumptions selected for the axisymmetric coordinate system (r, z), the boundary conditions become

$$r = 0, \quad \frac{\partial T_1}{\partial r} = 0;$$

$$r = r_1, \quad \lambda_2 \frac{\partial T_2}{\partial r} = \lambda_1 \frac{\partial T_1}{\partial r}, \quad T_2 = T_1;$$

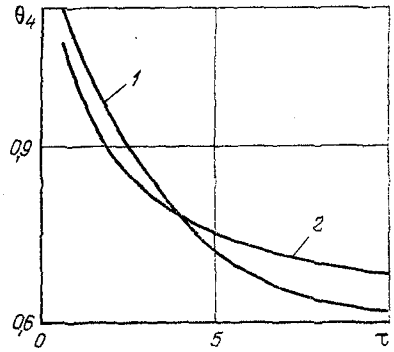

Fig. 4. Influence of heat evolution in a liquid stream on the temperature in the active element ($r = 0$, $z = l$ and $v = 0.5 - 1.5$ m/sec): 1) $W_1 = 210$ W; $W_2 = 210$ W; 2) $W_1 = 130$ W; $W_2 = 130$ W.

$$r = r_2, \quad T_2 = T_{\text{н}} = \text{const},$$
$$z = 0, \quad 0 \leqslant r \leqslant r_1, \quad \frac{\partial T_1}{\partial z} = 0,$$
$$z = l, \quad 0 \leqslant r \leqslant r_1, \quad \frac{\partial T_1}{\partial z} = 0,$$
$$z = 0, \quad r_1 \leqslant r \leqslant r_2, \quad T_2 = T_2^0 = \text{const}.$$

We consider the heat sources given by the following functions

$$W_1 = \begin{cases} 0, & \tau < 0, \\ W_1 = \text{const}, & 0 \leqslant \tau < \infty; \end{cases} \qquad W_2 = \begin{cases} 0, & \tau < 0, \\ W_2 = \text{const}, & 0 \leqslant \tau < \infty. \end{cases}$$

We select as initial conditions

$$\tau = 0, \; T_1(r, z) = T_1^0 = \text{const}, \quad T_2(r, z) = T_2^0 = \text{const}, \quad v_z(r) = f(r).$$

Selection of the coolant flow velocity profile in an annular channel of the active element is difficult in connection with the complex conditions for insertion of the fluid in the channel. Radial, tangential, or some other lateral insertion of the fluid into the active element cooling channel is realized for the majority of modern lasers. Computation of the velocity components of such a flow is an independent problem and is not examined in this paper. A survey of the literature permits the recommendation of the following velocity profile for turbulent flow in a short annular channel, as proposed by Buleev [7], and written as follows for convenient utilization

$$v_z(r) = u\varepsilon\left(\frac{r_1}{r_2}\right)\left[\frac{r - r_1}{r_2 - r_1}\ln\frac{r - r_1}{r_2 - r_1} - \frac{r^2 - r_1^2}{r_2^2}\ln\frac{r}{r_2} - (r - r_1)(r_2 - r)\right]. \tag{7}$$

Here $\varepsilon(r_1/r_2)$ is a correction factor obtained during processing the data [7], which is dependent on the channel geometry, and the fluid motion mode.

For a laminar flow mode, the velocity profile proposed by Petukhov [6]

$$v_z(r) = 2u\,\frac{(r_2^2 - r^2)\ln\dfrac{r_2}{r_1} - (r_2^2 - r_1^2)\ln\dfrac{r}{r_1}}{r_2^2 - r_1^2 + (r_1^2 + r_2^2)\ln\dfrac{r_2}{r_1}} \tag{8}$$

was used to compute $v_z(r)$.

The system of equations (5) and (6) was solved numerically according to an explicit scheme [8] for selected boundary and initial conditions and a velocity profile determined by (7) and (8).

The selection of (7) or (8) for insertion into the computation was performed according to an estimate of the Reynolds criterion in the channel $Re = 2u(r_2 - r_1)/\nu \gtreqless Re_{cr}$.

The following values of the parameters were used for computing the coefficients in Eq. Eqs. (1)-(8): for the active element: $\lambda_1 = 12.6$ W/m·K, $r_1 = 5 \cdot 10^{-3}$ m, $l = 50 \cdot 10^{-3}$ m; for the cooling (coolant—water) channel $\lambda_2 = 0.599$ W/m·K, $u = 0.01$–1.5 m/sec, $(r_2 - r_1) = (2\text{-}5) \cdot 10^{-3}$ m.

An illustration of the temperature distributions over the active element and coolant stream sections, obtained from the solution of system (5) and (6) is presented in Fig. 1. Sections for the insertion of the coolant into the channel ($z = 0$) and for its emergence from the channel ($z = l$) are considered. As expected, the maximum temperature over the active-element section is realized at the end of the channel ($z = l$), where the coolant stream is heated up sufficiently.

The computed values of the thermal boundary layer boundary in the coolant stream are presented in Fig. 2 for the stream exit section from the channel. As is seen from the figure, the degree of stream heating depends on the liquid flow velocity, the presence of a heat source in the stream, and the time of active-element operation. An increase in the heat evolution power in the active element also contributes to stream heating.

To investigate the combined influence of both the stream velocity and the heat evolution in the active element and the stream on the temperature field, more than 30 versions with a different combination of initial parameters were computed. The stream velocity varied between 0.01 and 1.5 m/sec, the heat evolution in the stream between 0 and 210 W, and the heat evolution in the active body was 130 or 210 W.

The change of temperature with time on the axis of the active element at the point ($r = 0$, $z = l$) is shown in Fig. 3a for different stream velocities and heat evolutions in the element. Judging by the results in the graph, the maximal heating on the active element axis depends mainly on the heat evolution level in the solid, and depends weakly on the change in velocity of coolant motion. From the viewpoint of diminishing the cooling system size, the lower stream velocities should be preferred. However, as is shown in Fig. 3b, the longitudinal pressure drop along the active element axis increases abruptly with the diminution in the flow velocity below 0.5-0.4 m/sec, which can result in producing a nonuniform thermal lens.

It should be noted that an increase in the stream velocity above 1.5 m/sec is also illogical since the stream acts as a water screen in this case, reducing the stream residence time on the cooled surface of the active element and the cooling system efficiency as well. In our opinion, the 0.8-1.0 m/sec velocity range of coolant motion is most efficient for up to 300 W of heat evolution in the laser active element.

If the longitudinal temperature drop along the length of the active element θ_2 depends on the magnitude of the stream velocity, then the radial temperature drop along the section θ_3 is practically independent of the velocity in the range 0.4-1.0 m/sec, as follows from curves 5 and 6 in Fig. 3b.

In recent times the cooling fluid performs still another function, it absorbs the nonworking (parasitic) part of the radiation spectrum of the gas-discharge laser-pumping lamp. Light-absorbing admixtures are hence added to the cooling liquid (water in our case). If the change in the thermophysical properties of the coolant is neglected here, then the insertion of the light-absorbing admixtures is equivalent to the insertion of a heat source W_2 in (6), which is distributed uniformly over the whole volume of the liquid. The question of the influence of the heat source on the temperature distribution in the active element arises.

As computations showed, the presence of a heat source in the stream raises the temperature level in the active element, however, for 0.5-1.5 m/sec velocities of coolant motion the nature of the temperature distribution over the section and length of the active element is practically invariant. Time dependences of the dimensionless temperature θ_4 on the active-element axis are presented in Fig. 4 for heat evolution of different intensity present in the stream. For the computations, the heat evolution in the stream was selected commensurate to the heat evolution in the active element.

The numerical computation program developed for the nonstationary conjugate heat-transfer problem permits determination of the coolant and active element temperature fields of a solid state laser with sufficient accuracy for practice. In our opinion, the results obtained are the initial information for an analysis of thermooptical distortion in the head of a solid-state laser and for the selection of a cooling system of minimal size and power consumption, and should contribute to the search for light-absorbing admixtures in the coolant stream to raise the quality of laser radiation.

NOTATION

Dimensional quantities: λ, heat-conduction coefficient; $v_z(t)$, velocity of liquid motion; u, average liquid velocity over the section; $T(r, z, \tau)$, temperature; W, heat evolution in the stream or solid; ρ, density; c, specific heat; r, running radius (coordinate along the vertical); z, longitudinal (axial coordinate); l, length of the channel and active elements; τ, time; ν, kinematic viscosity coefficient. Dimensionless quantities: Pr, Prandtl number; Re, Reynolds number; Re_{cr}, critical Reynolds number; $\bar{r} = (r - r_1)/(r_2 - r_1)$; $\theta = T - T_2^0$; $\theta_1 = T - T_1^0$, active element temperature; $\theta_2 = [T_1(0, \bar{z}) - T_1(0, 0)]$, temperature on active element axis; $\bar{z} = z/l$; $\theta_3 = [T_1(0, l) - T_1(r_1, l)]$, temperature drop over the active element section. Subscripts: 1, active element parameters; 2, fluid parameters.

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