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THERMAL REGIMES OF ANISOTROPIC-THERMOELEMENT THERMAL CONVERTERS

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A computational procedure is developed for the thermal regime of anisotropic-thermoelement thermal converters of cavity and plane types used to investigate heat fluxes and energy parameters of radiation. Heat losses from different elements of the thermal-converter structure are analyzed and the parameters of a battery of anisotropic thermoelements are optimized, providing maximum sensitivity at minimum measurement errors. The results obtained were used in developing a fundamentally new cavity-type filter pyrheliometer and commercially produced automatic absolute radiometers.

To investigate energy parameters of radiation, use is made of cavity- and plane-type heat-flux converters [1, 2], whose metrological characteristics can be improved by optimization of the thermal regime of the thermal converter. Use of fundamentally new sensors, namely, anisotropic thermoelements (ATs), provides the maximum sink of heat from detecting elements and improves the operating characteristics of thermal converters [3, 4]. The greater the heat sink through ATs that are in thermal contact with a dissipative element, the higher the sensitivity of the thermal converter; but the lower the remaining heat losses, the smaller the measurement error. For cavity-type thermal converters, a characteristic of the heat losses is overheating of the cavity, while for plane-type ones, a characteristic is heat losses through the side faces.

In the present work a calculation of the thermal regime of cavity- and plane-type thermal converters in which ATs are used as the sensing element is made; heat losses from different elements of the thermal-converter structure are analyzed; the parameters of the AT battery are optimized to ensure maximum sensitivity at minimum measurement error. The stationary process of the heat exchange of the thermal converter with the surrounding medium at room temperature is considered. The calculation is carried out for heat fluxes with a density q = 1000 W/m^2. Moreover, the maximum temperature difference arising in the thermal converter does not exceed 5 K, and therefore, radiation losses can be neglected, and the calculation of the thermal regime is reduced to the problem of conductive-convective heat transfer and to solving the equation of the heat balance of the system.

I. We investigate a cavity-type thermal converter. As the radiation detector, use was made of a cone-cylindrical element made of 30-μm-thick copper, manufactured by the method of electrodeposition; the inner surface of the element was coated with deep dull KhS-11-07-1M enamel. This geometry of the cavity together with the coating provides nonselectivity in a wide spectral range and an absorption coefficient that is very close to unity. The heat flux investigated was limited by a precision diaphragm with a diameter of 5 mm, and the flux power entering the cavity was equal to Q_0 = 19.625 mW. In order to investigate the thermal regime of the thermal converter, we calculated the cavity overheating and the heat losses from different elements with complication of its structure by stages.

1. The detecting cavity in an infinite air volume (Fig. 1). In this case the heat losses are determined from the heat-balance equation

$$Q_0 = Q_1 + Q_2 , \tag{1}$$

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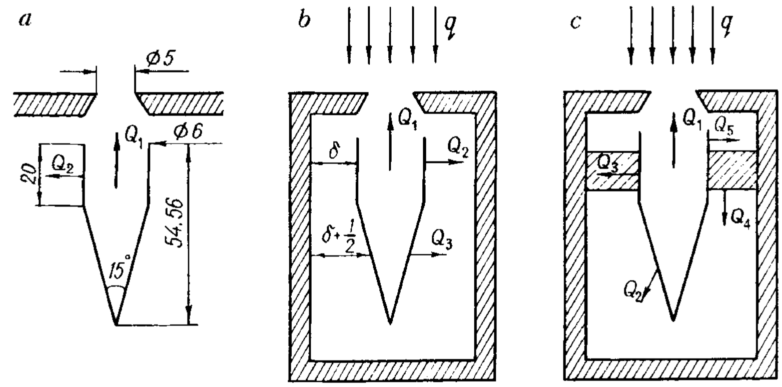

Fig. 1. Detecting cavities with a diaphragm in an infinite air volume (dimensions in mm) (a), inside the body with a diaphragm (b), and with a battery of anisotropic thermoelements in the body with a diaphragm (c).

where Q_1 is the heat transfer from inside the cavity; Q_2 is the heat transfer from the outer surface of the cavity. According to [5], for Q_1 we may write the expression

$$Q_1 = \alpha_{in} F_{in} \Delta T, \tag{2}$$

where α_{in} is the heat-transfer coefficient of the inner surface of the cavity; F_{in} is the area of the inner surface of the cavity; $\Delta T = (T - T_0)$ is the cavity overheating relative to the temperature T_0 of the surrounding medium.

Let us determine the heat-transfer coefficient α_{in} using the Nusselt number, which characterizes the intensity of the heat transfer process:

$$\mathrm{Nu} = \alpha_{in} l / \lambda,$$

where $\lambda = 2.59 \cdot 10^{-2}$ W/(m·K) for air at a temperature of 293 K.

The similarity equation for convective heat transfer processes is in general of the form

$$\mathrm{Nu} = C\,(\mathrm{Gr\,Pr})^n,$$

where the coefficients C and n depend on the argument (GrPr).

The Grashof number is determined by the expression

$$\mathrm{Gr} = g\beta l^3 \Delta T / \nu^2,$$

where g is the free-fall acceleration; l, in the present case, is the cylinder diameter.

Since the Prandtl number for air is equal to 0.703, the argument is

$$(\mathrm{Gr\,Pr}) = 19.204\Delta T < 1000$$

and, according to [5], for heat transfer in a finite volume under these conditions Nu = 1, while the heat transfer coefficient α_{in} = 0.475 W/(m^2·K), and consequently:

$$Q_1 = 0.033 \cdot 10^{-3} \Delta T. \tag{3}$$

To determine the heat transfer from the outer surface of the cavity, we take into consideration the fact that the heat transfer for it occurs in an infinite space at a vertical wall, then (GrPr) = 14,493.921 $\Delta T > 1000$, the Nusselt number will be written as Nu = 0.76 (GrPr)$^{0.25}$, and the heat transfer coefficient α_{out} = $3.955\Delta T^{0.25}$ W/(m^2·K). As a result, the heat transfer from the outer surface of the cavity is equal to

TABLE 1. Overheating of the Detecting Cavity in the Body for Different Gaps between Them

δ, mm	0.250	0.500	0.750	1.000	1.250	1.500
ΔT, K	0.533	0.849	1.102	1.323	1.525	1.715

$$Q_2 = \alpha_{out} F_{out} \Delta T = 2.778 \cdot 10^{-3} \Delta T^{1.25} . \tag{4}$$

From Eq. (1) with account for Eqs. (3) and (4) we determine the cavity overheating:

$$\Delta T = 4.749K .$$

The result otained confirms the assumption about the degree of overheating of the cavity.

2. *The detecting cavity inside the converter body (Fig. 1b).* We denote the distance between the cylindrical part of the cavity and the body by δ. For the conical part of the cavity we use the distance from the body to the middle of the cone ($\delta + 1.5$). In this case the heat-balance equation will be written as

$$Q_0 = Q_1 + Q_2 + Q_3 , \tag{5}$$

where Q_0 and Q_1 have the same values as above; Q_2 and Q_3 are the heat transfer from the cylindrical and conical exterior parts of the cavity, respectively.

Since in actual structures δ does not exceed 2 mm, the heat transfer is determined by the thermal conductivity through the air layer:

$$Q_2 = \lambda \varepsilon_{cyl} F_{cyl} \Delta T / \delta .$$

By analogy with item 1, for the argument (GrPr) we obtain the following value:

$$(\mathrm{Gr\,Pr}) = 711.265\, \Delta T .$$

For the expected values of ΔT this quantity is close to 1000, and therefore the coefficient of convection can be determined as

$$\varepsilon_{cyl} = 0.76\, (\mathrm{Gr\,Pr})^{0.25} ,$$

and for Q_2 we obtain

$$Q_2 = 9.072 \cdot 10^{-6}\, \Delta T^{1.25} / \delta . \tag{6}$$

In a similar way we determine the heat transfer from the conical part of the cavity:

$$Q_3 = \lambda \varepsilon_c F_c\, \Delta T / (\delta + 1.5) .$$

In this case (GrPr) = $3669.974\Delta T$ and $\varepsilon_c = 1.401 \Delta T^{0.25}$, and the heat transfer from the exterior part of the cone is

$$Q_3 = 11.813 \cdot 10^{-6}\, \Delta T^{1.25} / (\delta + 1.5) . \tag{7}$$

Table 1 presents calculated values of the cavity overheating as a function of the air gap, whence it is evident that the size of the latter affects substantially the cavity overheating; this should be taken into account in developing the structures of thermal converters and optimizing the dimensions of ATs that determine δ.

3. *The detecting cavity with a battery of anisotropic thermoelements inside the body (Fig. 1c).* For a battery of anisotropic thermoelements (BAT) we use ATs of rectangular cross section with a length a of 12 mm and a width c of 0.25 mm that are connected in series and are located uniformly along the generatrix of the cylinder over the side surface so that one face of the BAT is in contact with the cylinder, and the other

with the cavity. The thermoelements are glued to the cylinder and the body by means of a thin layer of electrically insulating and heat-conducting adhesive, and because of its small thickness the influence of the thin layer on the thermal fields will not be taken into account. This location of the BAT creates the condition for a heat sink from the cavity to the body. In this case the heat-balance equation has the form

$$Q_0 = Q_1 + Q_2 + Q_3 + Q_4 + Q_5 ,$$

where Q_1 is the heat transfer from inside the cavity (Eq. (3)); Q_2 is the heat transfer from the conical part of the cavity (Eq. (6)); Q_3 is the heat flux through the BAT; Q_4 is the heat transfer from the side faces of the BAT; Q_5 is the heat transfer from the part of cylinder free of the BAT.

The heat flux through the BAT is determined by the expression

$$Q_3 = kF_{\mathrm{BAT}}\,\Delta T/h ,$$

where k is the thermal conductivity of the single crystal in the direction of the heat sink; h is the height of the anisotropic thermoelements, which virtually coincides with the gap between the cavity and the body; F_{BAT} is the total area of contact of the BAT with the cylinder surface.

According to [6], the thermal conductivity coefficient of CdSb single crystals is 1.25 W/(m·K), and $F_{\mathrm{BAT}} = mac$, where m is the number of ATs in the battery. For the heat absorbed by the BAT we obtain the following expression:

$$Q_3 = 3.750 \cdot 10^{-6}\,\Delta T m/h .$$

In calculation of the heat transfer from the side faces of the BAT it is necessary to take into account the possibility of convective heat transfer:

$$Q_4 = \lambda\,\varepsilon_c\,F_s \Delta \check{T}/h .$$

Here $\Delta\check{T}$ is the temperature difference between the side surface of the BAT and the body. Actually, this quantity changes for different parts of the surface within the limits from 0 to ΔT between the cavity and the body. In calculations, we restrict ourselves to the maximum value $\Delta\check{T} = \Delta T$, since preliminary calculations show that under insignificant overheatings this equality is approximately admissible. The area of the side surface of the BAT consists of the side surfaces of the ATs and their end parts, through which commutation of the elements proceeds:

$$F_s = 2mh\,(a + c) = 24.5 \cdot 10^{-6}\,mh .$$

In this case (GrPr) < 1000 and $\varepsilon_c = 1$, and then we obtain

$$Q_4 = 0.6 \cdot 10^{-6}\,m\Delta T .$$

The heat flux from the free part of the cylinder is determined by a formula similar to that for Q_4:

$$Q_5 = \lambda\varepsilon_{\mathrm{fr.c}}\,F_{\mathrm{fr.c}}\Delta T/h .$$

The free-surface area of the cylinder is

$$F_{\mathrm{fr.c}} = (376.8 - 3m) \cdot 10^{-6} .$$

In calculating $\varepsilon_{\mathrm{fr.c}}$, the generatrix of the cylinder is the characteristic dimension, (GrPr) = $0.721 \cdot 10^{-3}$, $\Delta T < 1000$, and $\varepsilon_{\mathrm{fr.c}} = 1$. As a result, we have

$$Q_5 = 2.59\,(376.8 - 3m) \cdot 10^{-5}\Delta T/h .$$

TABLE 2. Magnitude of the Overheating, Heat Fluxes, emf, and Volt-Watt Sensitivity of the Cavity of an AT Thermal Converter

b, mm	m	ΔT, K	Q_1, mW	Q_2, mW	Q_3	mW	Q_4, mW	Q_5, mW	E, mW
0.5	20	0.115	0.004	0.396	17.250	0.001	1.887	6.90	0.352
	40	0.061	0.002	0.361	18.360	0.002	0.814	7.32	0.373
1.0	20	0.227	0.007	0.740	17.025	0.003	1.863	6.81	0.347
	40	0.123	0.004	0.344	18.450	0.003	0.818	7.38	0.376
1.5	20	0.336	0.011	1.007	16.800	0.004	1.838	6.61	0.336
	40	0.183	0.006	0.471	18.300	0.004	0.811	7.32	0.373

Thus, we have determined all the components of the heat-balance equation, expressed in terms of the two variable parameters m and b. In Table 2 we present calculations of ΔT, the emf of the BAT E, and the volt-watt sensitivity S of the thermal converter. In calculation of the emf we used the Thomson formula [7], which presupposes the presence of a constant temperature gradient, and we determined the volt-watt sensitivity from the relation

$$S = E/Q_0 .$$

The application of the Thomson formula is justified, since in our case the ratio a/b is rather large. It should also be noted that in the calculations we used a value of the coefficient of anisotropy of the thermoelectromotive force that is equal to 250 μW/K, although at the present time CdSb single crystals are grown with an anisotropy coefficient of the thermoelectromotive force exceeding this value by a factor of 2–3.

On the basis of the results obtained we can draw the following conclusions:

a) employment of ATs leads to a sharp decrease in the overheating of the detecting cavity (from 4.749 to 0.061 K), which cannot be done by using thermocouples or bolometers. As seen from Table 2, more than 90% of the incident heat flux goes through the BAT;

b) heat losses by inverse radiation of the cavity and from the side surfaces of the ATs are insignificant and do not exceed several hundredths of a percent of the total flux;

c) heat losses from the free surface of the cylinder are significant but they can be decreased by optimizing this surface in conformity with the number of thermoelements;

d) heat losses from the conical part of the system are also significant. However, a thin-wire bifilar winding of electrical substitution is usually glued onto the conical part. Subsequently, to decrease the heat transfer from the conical part, the structure of the detecting cavity changed was somewhat, and this made it possible to improve the thermal-converter parameters means of absorption of a portion of the heat losses from the conical section of the cavity. The calculations performed showed that the distribution of heat losses in this case did not change noticeably.

As seen from Table 2, the calculated quantity S remains practically constant on changing the parameters of the thermal converter and is quite sufficient for practical measurements.

II. We consider a plane-type AT thermal converter. These thermal converters have a somewhat greater error than cavity-type converters; however, they are widely used in developing the radiometers that are commercially produced at the present time. Using the procedure stated above, we calculate the parameters of a plane-type thermal converter.

A schematic diagram of the thermal converter is shown in Fig. 2. A battery consisting of ATs with a length a = 6 mm and a width c = 0.3 mm and located on a heat-removing radiator inside the body with a 3-mm diaphragm turned out to be optimum. The plane sensitive area is coated with a special black enamel so that the incident flux is virtually completely absorbed by the surface. Calculations were carried out for different thicknesses b of the ATs (from 0.5 to 1.5 mm) and a different number of elements m in the BAT (from 6 to 18). In evaluating the thermal-converter parameters, we took no account of the adhesive interlayers, which

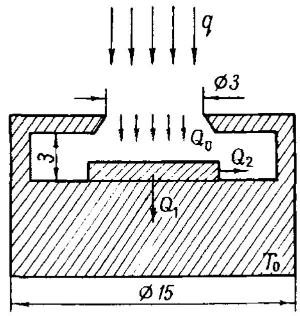

Fig. 2. Converter with a battery of anisotropic thermoelements on a plane sensitive area.

TABLE 3. Magnitude of the Overheating, Heat Fluxes, emf, and Volt-Watt Sensitivity for a Plane-Type AT Thermal Converter

b, mm	m	ΔT, K	Q_1, mW	Q_2, mW	E, mV	S, V/W
0.5	6	0.252	6.804	0.278	2.268	0.32
	8	0.248	6.696	0.365	2.974	0.42
	10	0.245	6.615	0.451	3.675	0.52
	12	0.242	6.534	0.533	4.356	0.62
	14	0.239	6.453	0.613	5.019	0.71
	16	0.236	6.372	0.691	5.664	0.80
	18	0.233	6.291	0.767	6.291	0.89
1.0	6	0.469	6.332	0.728	2.111	0.30
	10	0.283	6.367	0.698	2.123	0.30
	14	0.202	6.395	0.663	2.121	0.30
1.5	6	0.649	5.841	1.222	1.947	0.28
	10	0.394	5.910	1.162	1.970	0.28
	14	0.283	5.943	1.121	1.981	0.28

were much thinner than the BAT, and therefore they did not change substantially the temperature field in the thermal converter.

In this case the heat-balance equation has the form

$$Q_0 = Q_1 + Q_2 ,$$

where Q_0 is the power of the flux incident on the BAT; Q_1 is the flux absorbed by the BAT; Q_2 is the heat transfer through the side faces of the BAT.

In accordance with the conditions of the problem, the power falling on the BAT is Q_0 = 7.065 mW.

The heat flux absorbed by the BAT is determined by the thermal conductivity k of the CdSb single crystal that is used for manufacturing the ATs:

$$Q_1 = kF_{\mathrm{BAT}}\Delta T/b ,$$

where $F_{\mathrm{BAT}} = 1.8\cdot10^{-6}$ m, and consequently:

$$Q_1 = 2.25 \cdot 10^{-6}\Delta Tm/b .$$

In order to calculate the heat losses from the side faces of the BAT, we assume that the heat transfer occurs in an infinite space, since the BAT volume is two orders of magnitude smaller than the body volume, and then

$$Q_2 = \alpha_1 F_s \check{\Delta T},$$

where α_1 is determined from the formula

$$\alpha_1 = \mathrm{Nu}\,\lambda / l.$$

In this case the height of the thermoelements h is the characteristic dimension l. Since $\check{\Delta T}$, in order of magnitude, can only be smaller than or equal to ΔT, hereafter we will assume that $\check{\Delta T} = \Delta T$. Table 3 presents results of calculating the overheating ΔT, the distribution of the heat losses, the emf at the exit of the thermal converter E, and its volt-watt sensitivity S as functions of m and h.

The volt-watt sensitivity increases with m, but the heat losses through the side faces also increase. A BAT with 8–10 elements turns out to be optimum; here a sufficient emf at the exit is ensured with heat losses within 5–6%, which determine the systematic error of the radiometer.

On the basis of the calculations performed it is possible to optimize the structure of an AT thermal converter for a radiometer. A height of the thermoelements of 0.5 mm turned out to be optimum, since in this case the ratio a/b for each thermoelement is equal to 12 and the temperature field is uniform. An increase in the AT height leads to a decrease in the heat flux through the BAT because of the heat losses through the side faces.

We developed a fundamentally new cavity-type filter pyrheliometer based on ATs that was tested under both laboratory and field operating conditions at the high-mountain "Terskol" Observation Center of the Main Geophysical Observatory of the Russian Federation. The optimization of the thermal regime for a plane-type AT thermal converter was used in constructing commercially produced RAT-1P and RAT-2P radiometers of energy illumination.

NOTATION

a, length of the anisotropic thermoelement, mm; b, height of the anisotropic thermoelement, mm; c, width of the anisotropic thermoelement, mm; α, heat transfer coefficient, W/(m^2·K); β, coefficient of volumetric air expansion, K^{-1}; δ, size of the gap between the cylinder and the body, mm; ε, coefficient of convection of the air layer; k, coefficient of thermal conductivity of the single crystal, W/(m·K); λ, coefficient of thermal conductivity of the surrounding medium, W/(m·K); ν, kinematic viscosity of air, m^2/sec; l, characteristic dimension, mm; m, number of anisotropic thermoelements in the battery; n, exponent in the similarity equation; C, numerical coefficient in the similarity equation; q, heat-flux density, W/m^2; Q, power of the radiation flux, W; E, emf of the battery of anisotropic thermoelements, V; S, volt-watt sensitivity of the thermal converter, V/W; F, surface area, mm^2; T, temperature, K; Nu, Nusselt number; G, Grashof number; Pr, Prandtl number. Subscripts: in, inner; out, outer; cyl, cylinder; c, cone; s, side; fr.c, free surface of the cylinder.

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