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TESTS ON A FLOW CRYOSTAT WITH SERIES COOLING

A. B. Gal'chuk, S. V. Lavrenchuk, V. A. Romanenko, and S. I. Khankov

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Measurements and calculations on a flow cryostat with serial cooling have given equivalent thermal schemes that have been tested for adequacy and consequent simple working formulas.

Here we examine a cryostat in which the heat is removed from a series of units by connecting them to a common pipe by thermal bridges: copper-wire bundles. A difference from cryostatting each unit separately [1] is that the system is more compact and economical, but if the units are closely spaced, the heat transfer between them may impose constraints on the independent temperature control. One can select the bundle conductances and the points of attachment to the pipe to match the conductances for the elements, which is a complicated optimization task if one needs to minimize the coolant flow.

Sometimes, it is possible to simplify the thermal model at an early stage, e.g., by classifying the links as strong and weak [2], when simple analytic estimates are possible.

Interest attaches to the more complicated case where the system contains thermal models with lumped and distributed parameters and temperature nonlinearities, while the thermal links do not satisfy the criteria of strong or weak, so one cannot assign them to the model types described in [2]. In the first stage, a full study should be based on numerical calculations tested against measurements on a prototype and the consequent definition of ways of representing the equivalent thermal circuit. The model is thus reduced to a set of algebraic equations, which provides an analytic solution, which simplifies the early design stages.

1. In our cryostat, three units are cooled in sequence (Fig. 1a), and the conclusions can be extended to any number of units, as the formulas show.

Firstly, the working cavity is evacuated to 10^{-3} Pa, so the external heat leaks are determined only by radiative transfer and the conductances in the mounting parts, and secondly, one can restrict the temperature change in the coolant (gaseous helium) to 100-150 K with a mass flow rate of $1\text{-}5\cdot10^{-5}$ kg/sec because dimensionless criteria [3] indicate that the helium flow is laminar at these levels and the heat-transfer coefficient is only slightly dependent on temperature.

We consider only the stationary state and take the units as isothermal, while the temperature distributions in the pipe and flow are one-dimensional along the axis, and the thermal bridges have lumped parameters. The temperature of the body is taken as constant at 295 K.

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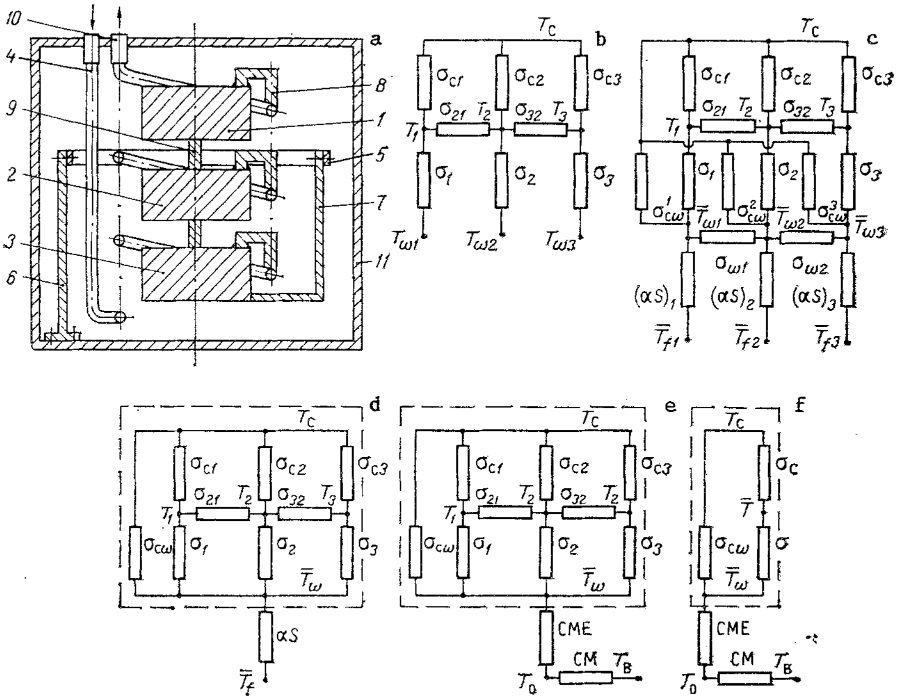

Fig. 1. Thermal model: a) cryostat scheme: 1-3) cooled units; 4) heat exchanger pipe; 5) mounting ring; 6) stand; 7) mounting plates; 8) copper bundles; 9) thermal bridges; 10) cryogenic connections; 11) body; b-f) thermal circuits.

The model has a system of bodies with thermal links between them, and it includes three units, the pipeline, and the coolant, together with the inlet and outlet tubes. The mathematical model corresponding to the thermal one includes:

1. Three heat-balance equations for the units:

$$\sum_{n=1}^{N_i} (\sigma_i)_n (T_i - T_{wn}) + \sum_{j=1}^{J_i} \sigma_{ij} (T_i - T_j) + \sigma_{ci} (T_i - T_c) = 0. \tag{1}$$

2. A differential-equation system for the temperature distributions in the pipeline and tubes; for the pipeline

$$\frac{d}{d\bar{x}} \left(\frac{\lambda F}{L} \frac{dT_w}{d\bar{x}} \right) + \sigma_{cw} (T_c - T_w) - \sigma_{\text{к}} (T_w - T_f) + Q(\bar{x}) = 0;$$

$$\sigma_{cw} = \alpha_r (\bar{x}) S_\sigma; \quad \sigma_c = \alpha_c (\bar{x}) S_i; \; Q(\bar{x}) = \sum_{r=1}^{R} P_r \delta (\bar{x} - \bar{x}_r). \tag{2}$$

The temperature distributions in the tubes, which are made of stainless steel and intended to reduce the conductive heat fluxes from the body to the heat exchanger, may be described by equations analogous to (2) but without the last term.

The boundary conditions at the points where the cryogenic pipes adjoin the heat exchanger are written as conditions for conservation of the heat flux and temperature equality.

3. The differential equation for the temperature distribution in the coolant flow:

$$\frac{dT_f}{d\bar{x}} - \varphi (T_w - T_f) = 0; \; \varphi = \frac{\alpha_c (\bar{x}) S_i}{cM}; \quad T_f|_{\bar{x}=0} = T_0. \tag{3}$$

This model has been implemented with a software suite for solving nonlinear one-dimensional conduction and energy equations, which has been applied to a model represented a graph having any configuration. The arcs in the graph correspond to elements having one-dimensional temperature distributions in the solid and coolant flow, while the nodes are elements with

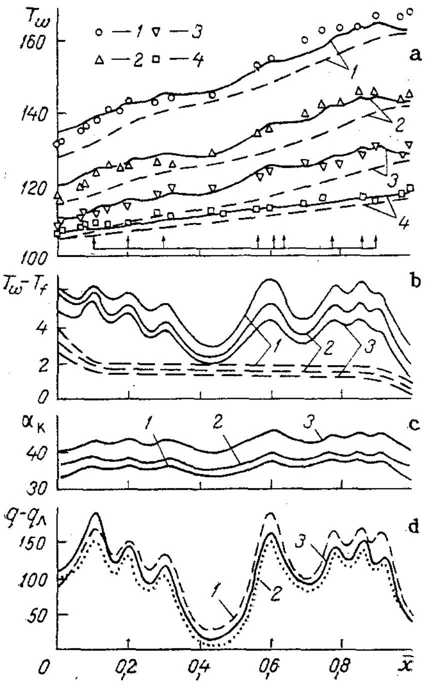

Fig. 2

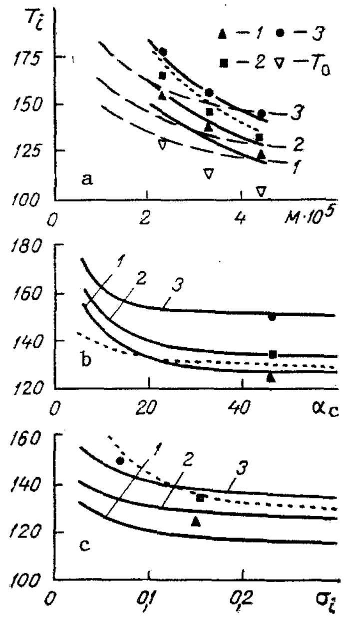

Fig. 3

Fig. 2. Distributions along pipeline: a) temperature, solid line pipe wall, dashed line helium flow; b) temperature difference between pipe wall and helium, solid line for thermal contact with bodies 2, 3, and 1 via wire bundles, dashed line after disconnecting the bundles; c) α_c; d) flux differences (1) $M = 2.2 \cdot 10^{-5}$ kg/sec; 2) $3.2 \cdot 10^{-5}$; 3) $4.2 \cdot 10^{-5}$; 4) $4.2 \cdot 10^{-5}$). T_w, $T_w - T_f$, K; α_c, W/(m²·K); $q - q_r$, W/m².

Fig. 3. Body temperatures as functions of: a) helium flow; b) α_c; c) bundle conductances: 1-3) bodies, solid line from (1)-(3) on the basis of $T_0(M)$, dashed line from (1)-(3) with $T_0 = 105$ K, dotted line calculation on mean cryostatic level for system of three bodies with σ_{21}; $\sigma_{32} = \infty$. T_i, K; σ_i, W/K; α_c, W/(m²·K).

uniform patterns or points of junction, as well as the starts and ends of the arcs [4-6]. The method enables one to calculate one-dimensional temperature patterns in fairly complicated systems, with coordinate and temperature dependence for the coefficients in (1)-(3).

The input data are the temperatures T_0 and T_c, the collant flow rate, and the conductive [7] and radiative [8] conductances defined from standard formulas; the latter can also be derived approximately from the assumed temperature levels in the elements:

$$\sigma_1 = 0{,}150 \text{ W/K}; \ \sigma_2 = 0{,}156; \ \sigma_3 = 0{,}072; \ \sigma_{21} = \sigma_{32} = 0{,}037;$$
$$\sigma_{cw} = 0{,}014; \ \sigma_{c1} = 0{,}005; \ \sigma_{c2} = 0{,}003; \ \sigma_{c3} = 0{,}011 \text{ W/K}.$$

2. Figure 2a shows calculated and measured temperature distributions for the heat exchanger and coolant flow at various rates; the errors in measuring the temperatures with copper-constantan thermocouples did not exceed 1%. The arrows indicate the coordinates where the wire bundles are attached to the pipe (three from each body). Figure 2b gives calculated temperature differences between the wall of the heat exchanger and the helium flow, while the dashed lines give an analogous calculation but without allowance for the heat leak to the exchanger from the bodies. This enables one to estimate the contribution to the heat balance from the parasitic leak by radiation from the body to the pipe.

Figure 2c shows calculations on the heat-transfer coefficient along the pipe. The fluctuations in α_c are governed by ones in $T_w - T_f$, and the maximum change in $T_w - T_f$ (by about a factor three) alters α_c by 10% for a constant helium flow, which corresponds to the relation for laminar flow $\alpha_c \sim (T_w - T_f)^{1/10}$ [7]. For estimates, that small change in α_c along the pipeline can be neglected.

Parts c and d of Fig. 2 have been used in graphs for the specific heat leak to the exchanger pipe without allowance for the leaks from the body, i.e., the useful specific flux $q - q_r$. In the flow-rate ranges used, the M dependence of $q - q_r$ tends to a limit, and the heat flux density averaged over the length is ~80 W/m².

The peaks in Fig. 2b-d correspond to the points where the wire bundles are attached to the heat exchanger. The bundles from the second body are close together on the exchanger, so the peak temperature differences and heat fluxes for them merge because the pipe has high conductivity.

Figure 3a shows measurements and calculations on the body temperatures as functions of helium flow; the helium temperature at the inlet to the exchanger rises as the flow rate decreases (the helium before entering the cryostat is heated from the leak along the pipe), and then the correction for it means that the calculations reflect $T_i(M)$ accurately. The calculations in parts b and c of Fig. 3 show that $T_i(\alpha_c)$ tends to a limit under these conditions, while $T_i(\sigma_i)$ is close to saturation.

Figure 3 shows that one can reduce the body temperatures further by reducing the inlet helium temperature and, to a smaller extent, by raising the helium flow rate, and to a very minor extent by improving the coupling between the bodies and the exchanger.

3. We now describe the heat transfer by means of equivalent thermal schemes (thermal circuits). We assume that each body is coupled to the heat exchanger tube by one bundle, whose thermal conductivity is equal to the sum of the thermal conductivities for the bundles actually linking that body to the pipe, while an average point of attachment is used.

The equivalent schemes show that the main difficulty arises in determining how one describes the heat conditions in the exchanger, since if it is possible to determine the temperatures at the points where the bundles join the pipe, one has a simple thermal scheme as shown in Fig. 1b, and the $T_i(T_{wi})$ relations are

$$T_j = \frac{\sigma_{j2}T_2 + \sigma_{cj}T_c + \sigma_j T_{wj}}{\sigma_{j2} + \sigma_{cj} + \sigma_j}; \quad T_i^e = \frac{\sigma_{ci}T_c + \sigma_i T_{wi}}{\sigma_{ci} + \sigma_i};$$
$$T_2 = \frac{(\sigma_{c2} + \sigma_2) T_2^e + \sigma_1^e T_1^e + \sigma_3^e T_3^e}{\sigma_{c2} + \sigma_2 + \sigma_1^e + \sigma_3^e}, \quad j = 1, 3; \; i = 1, 2, 3, \tag{4}$$
$$\sigma_1^e = \frac{\sigma_{21}(\sigma_{c1} + \sigma_1)}{\sigma_{21} + \sigma_{c1} + \sigma_1}, \quad \sigma_3^e = \frac{\sigma_{32}(\sigma_{c3} + \sigma_3)}{\sigma_{32} + \sigma_{c3} + \sigma_3}.$$

The effective parameters T_i^e and σ_i^e here enable one to extend the algorithm to any number of bodies.

T_{wi} can be determined by closing the thermal circuit on the basis of all the thermal factors; Fig. 1c most fully reflects the latter, but the lower part contains a series of undefindd parameters: $(\alpha S)_1$, $(\alpha S)_2$, $(\alpha S)_3$, σ_{w1}, σ_{w2}, σ_{cw}^1, σ_{cw}^2, σ_{cw}^3, so calculations are difficult. We therefore use less obvious equivalent schemes and specify $T_w(\bar{x})$ and $T_f(\bar{x})$ a priori.

$T_w - T_f$ varies along the pipe, but firstly, there is no tendency to a monotone trend, and secondly, the fluctuations are small by comparison with the temperature level, so one can assume $\Delta T = T_w(\bar{x}) - T_f(\bar{x}) = \bar{T}_w - \bar{T}_f$ = const and perform analogous calculations with the averaged ΔT. One is even more justified in assuming that $\alpha_c(\bar{x})$ = const along the pipe.

This equivalent to assuming a constant heat flux to the coolant along the pipe and the use of a q averaged over the length. The solution to (3) is then

$$T_f(\bar{x}) = T_0 + \frac{Q}{cM}\bar{x}; \quad Q = qS_i = \alpha_c S_i \Delta T.$$

The pipe temperature is then

$$T_w(\bar{x}) = T_0 + \frac{Q}{\alpha_c S_i} + \frac{Q}{cM}\bar{x}.$$

TABLE 1. Thermal-Circuit Calculations

$M \cdot 10^5$, kg/sec	T_0, K	Q_w, W	δ_w	δ_1	δ_2	δ_3
			%			
2,2	128	4,6	1,3	0,9	1,9	2,8
3,2	114	5,1	1,0	0,2	1,6	2,1
4,2	105	5,5	1,0	1,9	2,4	2,7

Figure 1d is used, where Q is calculated from $\bar{T}_w$, with

$$Q = \sigma_\Sigma (T_c - \bar{T}_w) = \alpha_c S_i (\bar{T}_w - \bar{T}_f), \tag{5}$$

in which σ_Σ is derived from Fig. 1d.

The balance equation is

$$\sigma_\Sigma (T_c - \bar{T}_w) = cM (T_e - T_0) \tag{6}$$

and then we solve (5) and (6) with $\bar{T}_f(\bar{x})$ a linear function and $\bar{T}_f = (T_0 + T_e)/2$, to get expressions for $\bar{T}_f$ and Q, which give

$$T_w(\bar{x}) = T_0 + \frac{2(T_c - T_0)}{2(1+\varphi_1)+\varphi}(1+\varphi\bar{x}),\ \varphi_1 = \frac{\alpha_c S_i}{\sigma_\Sigma}. \tag{7}$$

We calculate $T_w(\bar{x}_i)$, in which $\bar{x}_i$ are the bunch attachment coordinates and substitute into (4) to determine T_i. Table 1 gives results as the relative errors in the temperatures by comparison with numerical calculations for three helium flow on the basis of $T_0(M)$.

This method can be applied as follows because (7) provides good accuracy if $\varphi_1 \gg 1$; $\varphi_1 \gg \varphi$ or $\alpha_e S_i \gg \sigma_\Sigma$; $cM \gg \sigma_\Sigma$, i.e., when (7) is described approximately by

$$T_w(\bar{x}) = T_0 + \frac{1}{\varphi_1}(T_c - T_0)(1+\varphi\bar{x}). \tag{8}$$

If the constraints in deriving (8) are not met, (7) may lead to a considerable overestimation for the outlet exchanger temperature up to $T_w(\bar{x} = 1) > T_c$.

Another case where one can use lumped parameters is an exchanger having a nearly uniform temperature (T_w = const), as occurs if the pipe has adequate conductivity in the axial direction. A simplified thermal circuit can be formulated from formulas for the exchanger temperature distribution [7], which give a relation between the characteristic temperatures:

$$\alpha_c S_i (\bar{T}_w - \bar{T}_f) = cME(\bar{T}_w - T_0) = cM(T_e - T_0),\ E = 1 - \exp(-\varphi). \tag{9}$$

We use (9) to represent the circuit as in Fig. 1e. An additional feature is that one can eliminate α_c if $\varphi \gg 1$, which leads to $E \cong 1$ (in our case, φ = 3-4).

As T_w = const is not met, the Fig. 1e scheme gives incorrect results for three bodies, and in particular, the temperature-level relation $T_1 < T_2 < T_3$ is not obeyed. However, if the conductances between the bodies $\sigma_{ij} \gg \max\{\sigma_i, \sigma_{ci}\}$ are high, one can use Fig. 1f, which corresponds to all three bodies acting as one. The formulas then simplify, and it is possible to forecast the mean temperature level as shown in Fig. 3.

One thus has to solve (1)-(3) numerically on the basis of the assumptions in the model to obtain detailed information. The assumptions have been confirmed by experiment.

With the ranges used for the coolant flow rate and temperature, where nonlinearity in the thermophysical characteristics is slight, the model can be represented as a circuit giving simple relations. The approach is applicable subject to $\alpha_c S_i > cM \gg \sigma_\Sigma$, as is evident from a comparison with numerical calculations.

NOTATION

T_c, T_i, T_w, T_f, temperatures of case, body i, tube wall, and flowing coolant in K; T_0 and T_e coolant temperatures at inlet and exit for heat exchanger and pipes in K; T_{wi} mean pipe wall temperature at points of attachment of bundles from body i in K; T_{wn} pipe wall temperature at point of attachment for bundle n in K; $(\sigma_i)_n$ and σ_i thermal conductivities of bundle n and all bundles from body i in W/K; σ_{ij} thermal conductivity between bodies i and j in W/K; σ_{ci}, σ_Σ, σ_{cw} thermal conductivities from case to body i and total and radiative conductivities from case to pipe in W/K; α_c convective heat-transfer coefficient between pipe and coolant in $W/m^2 \cdot K$; α_r radiative heat-transfer coefficient between case and pipe in $W/m^2 \cdot K$; λ pipe material thermal conductivity in W/m·K; c specific heat of helium at constant pressure in J/kg·K; q and q_r correspondingly densities of the total heat flux and radiative flux to the pipe in W/m^2; P_r heat flux along bundle r in W; M coolant mass flow rate in kg/sec; F tube cross section area in m^2; S_i and S_o inside and outside surface areas of pipe in m^2; L pipe length in m; $\bar{x} = x/L$ relative coordinate along pipe axis; $\bar{x}_r$ relative coordinate for bundle r attachment; R total number of bundles; N_i number of bundles cooling body i; J_i number of bodies linked by heat bridges to body i; δ_i relative error in calculating the temperature of body i by comparison with numerical result in %; δ_w mean relative error in heat exchanger temperature calculated numerically by comparison with temperature from (4) taken at ten equally separated points in %; $\delta(\bar{x} - \bar{x}_r)$ Dirac function.

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EXTERNAL HEAT TRANSFER IN POLYDISPERSED FLUIDIZED BEDS AT ELEVATED TEMPERATURES

V. A. Borodulya, Yu. S. Teplitskii, A. P. Sorokin,
V. V. Matsnev, I. I. Markevich, and V. I. Kovenskii

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The authors present results of a theoretical and experimental study of heat transfer in polydispersed fluidized beds of coarse particles at temperatures up to 1273 K.

The technique of fluidization is traditionally widely used in chemical technology, metallurgy, oil processing, etc. Recently its sphere of application has been expanded into a number of new areas, amongst which the foremost is energetics. A prominent place is occupied by high-efficiency processes of combustion and gasification of solid fuel. One important problem arising in introducing fluidized technology into energetics is the need to design heat exchangers, which as a rule meet a constraint in the bed to carry away the heat generated and to maintain the optimal temperature in it.

It is known [1] that in the conditions realized in high-temperature fluidized beds the external heat transfer has a complex conductive—convective—radiative nature. Its intensity is influenced by a number of factors, to compute which is a complex scientific problem.

A. V. Lykov ITMO, Academy of Sciences of the Belorussian SSR, Minsk. I. I. Polnzuov NPO TsKTI, Leningrad. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 56, No. 5, pp. 767-773, May, 1989. Original article submitted December 14, 1987.