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INVESTIGATION OF PLANAR MICROREFRIGERATORS

S. I. Bondarenko, A. A. Svyatukha,
V. N. Fenchenko, and V. M. Bakumenko

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We present the results of calculations and experimental investigation of planar microrefrigerators that employ nitrogen and a nitrogen-hydrocarbon gas mixture as a working body.

Introduction. As is known [1, 2], the practical use of high-temperature superconducting (HTSC) squids, bolometers, and SHF-receivers, whose working temperatures reach 90 K, can be ensured only with the aid of miniature refrigerator-type systems and, moreover, only for HTSC squids of nonmagnetic construction. In this regard, planar microminiature refrigerators (PMR) offer the greatest promise [4]. They can be combined rather simply with planar superconducting and cryoelectronic probes, do not produce vibrations, and are very adaptable to fabrication in batches.

PMRs are not fabricated in the SIS countries, but abroad some types of PMRs have been proposed, in particular, by MMR Technologies Inc., Mountain View, U. S. A. [5].

Construction of the PMR. The PMR developed at the Special Design-Technological Bureau of the Physicotechnical Institute for Low Temperatures of the Academy of Sciences of Ukraine [3] is based on the Joule-Thomson effect. It operates in the Hampson cycle regime [4]. The essence of this cycle is that a compressed gas passes through a heat exchanger into a throttle and an evaporator, where it expands and cools down due to the Joule-Thompson effect. The cold gas returns to the heat exchanger and cools the entering compressed gas.

The PMR consists of three glass plates, each 0.3 mm thick, 6 mm wide, and 48 mm long, all sintered into a single block. The middle plate separates the forward and return gas flows. This plate has a transition hole. The other two plates have holes for the emergence and entrance of the gas and etched channels of the heat exchanger. Moreover, in the plate with forward flow channels (of high pressure) the channel of the throttle is etched. To simulate the object to be cooled, pyroceramic sheets are placed at the ends of the plates, whose presence simultaneously increases the strength of the structure (see Fig. 1).

A high-pressure gas enters through a union from a capillary tube into the forward flow channels of the heat exchanger, passes through the throttle channel, and enters the evaporator. Through the hole in the separating plate the evaporator is connected to the return flow channels of the heat exchanger. The gas, which has a low pressure and temperature after the throttle and evaporator, enters the return flow channels of the heat exchanger and, while passing through them, cools the forward flow. In the present experiments the gas was vented to the atmosphere.

The erosion technology used abroad to produce PMR microchannels is not very adaptable to commercial use and produces channels with an increased coefficient of friction even in the case of a laminar flow. This leads to a shift in the critical Reynolds number to the $Re \cong 900$ region. Therefore, to produce the microchannels we used a chemical etching method, which simplifies the technology of microchannel production and at the same time improves the thermohydraulic characteristics.

After the PMR has been assembled, it is placed in a clamp and sintered in a tube muffle furnace; then it is annealed with cooling at a rate of 1°C min to remove stresses in the glass. Note that this fabrication technology also differs from that adopted abroad where gluing-together is employed. Moreover, when an MPR operates on gas mixtures, a high pressure is not required (4–5 MPa is sufficient), and at such a pressure an MPR produced by the indicated technology operates rather stably.

B. I. Verkin Physicotechnical Institute for Low Temperatures, Academy of Sciences of Ukraine, Khar'kov
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Fig. 1. Overall view of assembled PMR.

Calculation. When we conducted thermal design calculations of a PMR, we did not take into account the longitudinal heat conductivity in the heat exchanger. The heat flux removed from the forward flow is $q_p = m(I_{1p} - I_{2p})$, and the heat flux supplied to the return flow is $q_0 = m(I_{20} - I_{10})$, where I_{1p}, I_{2p}, I_{20}, I_{10} are the enthalpies of the forward and return flows at the inlet and outlet of the heat exhanger; m is the mass flow rate of the gas. The heat flux transferred to the heat exchanger is $Q = kFdT_{\log}$, where F is the heat exchange area; $dT_{\log}$ is the logarithmic mean temperature head; the heat transfer coefficient k is determined from the formula $k = I/(I/a_p + I/a_0 + \delta/\lambda_p)$ and a_p, a_0 are the mean heat-transfer coefficients for the forward and return flows, δ is the thickness, and λ_p is the heat-conductivity coefficient of the partition between the flows. The main complexity in the calculations was the need to calculate the values of the gas mixture enthalpies at different temperatures and pressures. A polynomial approximation was used for temperature and a linear one for pressure.

When the longitudinal heat conductivity in the heat exchanger was taken into account in a verifying calculation of PMRs, we managed to reveal their essential specific feature, i.e., the sensitivity of their characteristics to the material of the structure [6]. Thus, for the heat exchangers of ordinary throttling-regenerative systems one requirement is imposed on the material of the heat exchanger, namely, its thermal conductivity should not be very small. For the heat exchangers of a PMR with small dimensions and low flow rates, a material with an optimum thermal conductivity is needed. The use of a material with a very small or very high thermal conductivity leads to a substantial decrease in the thermal efficiency of the heat exchanger [6].

In contrast to heat exchangers of ordinary throttle-regenerative systems, the heat exchanger of a PMR is characterized by small gas flow rates and, consequently, by a laminar regime of its flow in channels. We note that the laminar region of flow in the microchannels of the heat exchanger depends on the roughness of the walls. The technology we apply makes it possible to obtain channels with a reduced hydraulic resistance. Therefore, in the calculations we used the classical dependences of the friction coefficient on the Reynolds number and assumed that the region of laminar flow extended up to the critical Reynolds number Re $\cong$ 200. Since the regime of flow in the PMR heat exchanger channels is laminar, the mean value of the Nusselt number is independent of the Reynolds number and is equal to Nu $\cong$ 3.66. In this case, the heat transfer coefficient $\alpha = \mathrm{Nu}\,\lambda/D$, where λ is the thermal conductivity of the flow, $D = 4S/P$ is the equivalent hydraulic diameter, S is the area of the flow cross section, and P is the perimeter.

Just as in the thermal calculation, the hydraulic design calculation of the heat exchanger in a first approximation was made using the arithmetic mean values of the temperatures of the flows; moreover, since in the forward flow channels the gas density is high and, therefore, the pressure drop is small, the latter was ignored in design calculations. In verifying hydraulic calculations we took into account the change in temperature along the flows and, as a consequence of this, the change in the density and viscosity of the gas flow. This is especially important for determining the pressure drop in the return flow channels, since precisely this drop substantially affects the temperature level of the PMR.

The determination of the PMR parameters, the thermal and hydraulic calculations must be performed simultaneously. To solve the system of equations that describes thermal and hydraulic processes in a PMR, we

Fig. 2. Test rig with a PMR specimen installed.

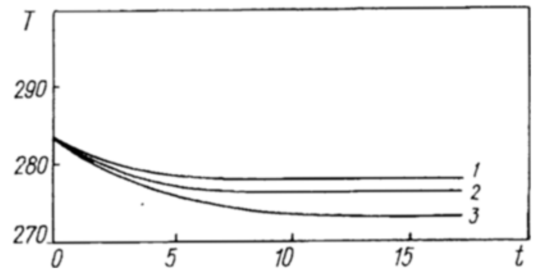

Fig. 3. Temperature of cooled object vs testing time with pure nitrogen at a pressure of 4 MPa: 1) without heat insulation and without vacuum in the chamber; 2) with heat insulation, but without vacuum in the chamber; 3) with heat insulation in the evacuated chamber. T, K; t, min

used the method of successive approximations, namely, in the process of calculations we determined more precisely the values of temperatures before and after the throttle and the value of the gas flow temperature at the exit of the heat exchanger. The algorithm was reduced to the following sequence of steps:

- Assignment of preliminary values for the temperatures of the forward and return gas flows at the inlet and outlet.
- Assignment of preliminary values for pressure differences in the forward and return gas flows.
- Refinement of the temperature value at the outlet of the PMR heat exchanger.
- Calculation of the temperature value at the exit of the PMR throttle at each step of "refinement of the temperature value at the outlet of the PMR heat exchanger."
- Calculation of the temperature value at the inlet of the PMR throttle at each step of "refinement of temperature at the exit of the PMR throttle."
- Finally, calculation of pressure drops in the forward and return flows at each step of "refinement of the value of temperature at the inlet of the PMR throttle."

To select the optimum relationship of the geometric parameters of the PMR, we varied the length, cross-section, and number of channels of both the forward and return flows in the heat exchanger. The temperature level of the PMR was assumed to be the criterion of the quality, i.e., it was assumed that the optimum PMR should ensure the lowest temperature of the cooled object at a given value of refrigeration output.

Tests. The PMR was tested on a specially fabricated rig (see Fig. 2). First, we purged and tested the PMR with pure nitrogen at a reduced pressure. The results of these tests are presented in Fig. 3. Naturally, in this case we could not obtain low temperatures, but, as can be seen from the results presented, an appreciable refrigerating effect is distinctly expressed even without high-quality insulation. This makes it possible to design a PMR for weakly cooled electronic integrated circuits. In the future, a PMR could be mounted directly on an integrated-circuit chip. In our opinion, this opens a promising new direction for the use of PMRs filled with gases at reduced pressures.

In experiments with a gas mixture we used an AUS-1 nitrogen-hydrocarbon mixture (40% nitrogen, 19% methane, 16% ethane, 25% propane) at a pressure of $\cong$ 5 MPa. Calculations carried out for a PMR test specimen

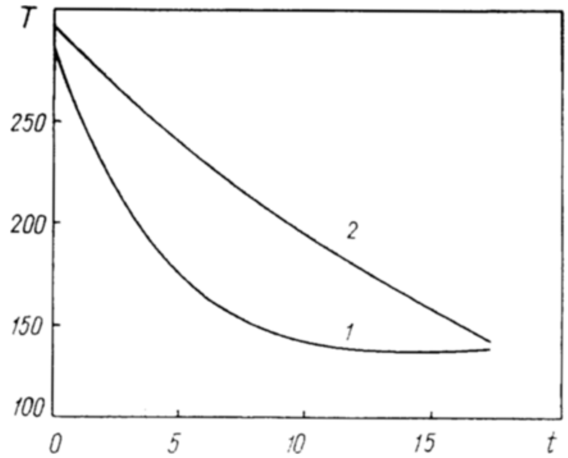

Fig. 4. Temperature of cooled object vs testing time with AUS-1 nitrogen-hydrocarbon mixture at a pressure of 4 MPa: 1)"nonoptimized" version of PMR; 2) "optimized" version of PMR.

whose heat exchanger had 40 channels with cross-sections of 50×40 μm for both forward and return flows and a length of 40 mm showed that at an inlet mixture temperature of 290 K and an inlet pressure of 4.8 MPa we could obtain a minimum temperature of $\cong$ 95 K at a mixture flow rate of 2.9 mg/sec and $\cong$ 125 K at 2.2 mg/sec. The heat load was assumed to be equal to 40 mW. In the experiments, the actual temperature of the object cooled was from 100 to 160 K. Thus, the main points of the procedure used to calculate the PMR parameters were confirmed experimentally.

The results of testing of two PMRs with a gas mixture are presented in Fig. 4. The tested MPRs differed in the parameters of the channels. In the first version the cross-section of the heat exchanger was not optimized, while in the second version such an optimization was performed. It is seen that while the first PMR attained the regime, in the second PMR the temperature drop continued (unfortunately, the experiment was terminated, as one of the plates of the PMR was broken). The difference in the rate of temperature drop is due to the different flow rates of the gas mixture.

Conclusions. A PMR with a nitrogen-hydrocarbon mixture fabricated by our technology is viable and promises temperatures in the "nitrogen" range. The calculated characteristics of the PMR are close to those obtained experimentally, proving the validity of the main postulates of the calculation procedure applied.

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