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THERMOELECTRIC RECUPERATIVE-TYPE HEAT EXCHANGE APPARATUS WITH THERMAL BRIDGES

This paper dwells upon the design of a thermoelectric heat-exchange apparatus with thermal bridges. The results of its mathematical simulation and full-scale test of the prototype are presented. Calculated and experimental plots of temperature variation at different points of the apparatus are given. The results obtained allow for a conclusion on the efficient practical application of heat-exchange apparatus.

Key words: heat-exchange apparatus, thermopile, heat carrier, temperature field, mathematical model, prototype, experiment.

Introduction

At the present stage of science and technology development the problems of studying special facilities for provision of intensive heat transfer from the sources with high thermal loads, creation of basically new high-performance cooling and temperature stabilization systems meeting special requirements, design of heat exchangers with improved characteristics become increasingly vital and relevant [1]. This is due to world market saturation with novel technical facilities offering great functional opportunities and high-speed performance but characterized by increased thermal overloads and overheating problems, which has an adverse effect on their operating reliability.

One of promising lines when creating cooling and temperature stabilization systems is the use of semiconductor thermoelectric converters providing for construction of economical, small-size coolers and temperature stabilizers with broad functional capabilities of maintaining given thermal mode [2, 3]. The theory and possibilities of practical application of such devices are adequately described in the works by A. F. Ioffe, L. S. Stilbance, A. E. Kolenko, A. I. Burshtein, M. A. Kaganov, M. P. Privin, L. I. Anatychuk, A. L. Vainer, E. K. Iordanishvili, I. V. Zorin, et al. In these works, parameters of devices working in different modes are calculated, the energy efficiency of their application is determined. The main emphasis here is made on the investigation of thermophysical processes with constant temperature on the junctions of thermoelectric power converters.

However, there exist many application areas of thermoelectric devices with temperature variation of heat carriers along the thermopile surfaces absorbing and releasing heat. They include various types of heat exchangers: coolers and heaters of liquid flows, air coolers, air conditioners, etc., i.e. all devices where circulation of heat carriers occurs along thermopile junctions. Despite considerable progress in thermoelectric technology, papers on such devices are lacking, their theoretical foundations are not developed to the full extent, no efficient operating modes are indicated, no areas of reasonable application are determined, etc.

Not without interest is investigation of thermoelectric heat exchangers of special structure with improved energy characteristics, their optimization, determination of the basic parameters and rational application areas. The above interest is due to insufficient research in this field, alongside with the urgent need for development of highly efficient heat exchangers with improved properties. Said factors determine the relevance of the present research.

The purpose of the paper is to study a thermoelectric recuperative-type heat exchange apparatus with thermal bridges with improved energy, weight-size and reliability parameters.

Mathematical model of heat exchange apparatus

Design circuit of the apparatus is given in Fig. 1. Here, in the elementary section dx of thermopile length, heat transfer is provided both through thermoelements and high thermally conductive thermal bridges with heat exchange intensity determined by the Biot numbers $Bi_{1,2}$ and $Bi'_{1,2}$ for thermopile and thermal bridge, respectively. The concept of filling factor ξ, is introduced which in this case is characterized by the ratio of thermopile area in the elementary section to the elementary section area. Then, in the elementary section of length dx the area of thermoelement junctions occupies the surface ξ Ldx, and the surface area of thermal bridge is $(1 - \xi)$ Ldx, where L is thermopile width.

Coefficients of heat transfer, thermal conductivity and thickness, respectively, for thermopile and thermal bridge will be denoted through $\alpha_{1,2}$, $\alpha'_{1,2}$, $\lambda_{1,2}$, $\lambda'_{1,2}$, d, d'. Let us assume that the temperatures of cooled and heated heat carriers at thermopile inlet are related as $T_1 > T_2$. The rest of assumptions are generally accepted for straight-flow thermoelectric heat pumps: the flows are absolutely mixed in the direction perpendicular to motion direction; the properties of heat carriers and materials are temperature-independent; heat transfer through unaccounted structural members is absent.

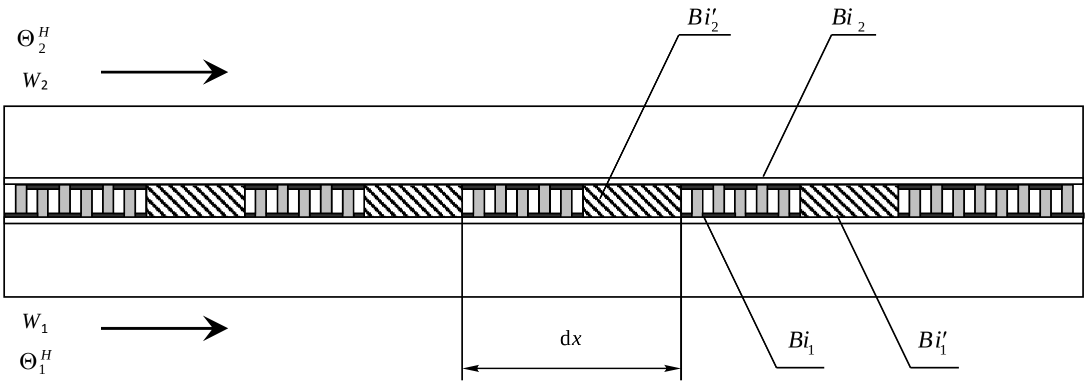

Fig. 1. Design circuit of thermoelectric heat exchange apparatus with thermal bridges.

Heat balance equations according to heat carrier flows for the above circuit under parallel flow conditions are given below:

$$W_1 \frac{dT_1}{dx} = \alpha_1 \xi L (T_{1TEB} - T_1) + \alpha_1^{'} (1-\xi) L (T_{1м} - T_1),$$

$$W_2 \frac{dT_2}{dx} = \alpha_2 \xi L (T_{2TEB} - T_2) + \alpha_2^{'} (1-\xi) L (T_{2м} - T_2),$$

where $T_{1TEB,\ 2TEB}$ are temperatures of thermoelement junctions, $T_{1b,2b}$ are surface temperatures of thermal bridges, $T_{1,\,2}$ are temperatures of cooled and heated heat carriers.

Heat balance equations on the surfaces exposed to heat carrier flows for TE junctions are of the form:

$$\alpha_1(T_1 - T_{1TEB}) = \bar{e}jT_{1TEB} - \frac{1}{2}j^2\rho d - \frac{\lambda}{d}(T_{2TEB} - T_{1TEB}),$$

$$\alpha_2(T_{2TEB} - T_2) = \bar{e}jT_{2TEB} + \frac{1}{2}j^2\rho d - \frac{\lambda}{d}(T_{2TEB} - T_{1TEB}),$$

where $\bar{e}$ is the Seebeck coefficient, j is electric current density;
for thermal bridges:

$$\alpha'_1(T_1 - T_{1м}) = K'(T_1 - T_2),$$

$$\alpha'_2(T_{2м} - T_2) = K'(T_1 - T_2),$$

where $K' = \left(\frac{1}{\alpha'_1} + \frac{1}{\alpha'_2} + \frac{d'}{\lambda'}\right)^{-1}$ is coefficient of heat transfer through thermal bridge.

On excluding the surface temperatures $T_{1TEB,\ 2TEB}$ and $T_{1b,2b}$ and reducing the equation to dimensionless form, the expressions for temperature variation of heat carrier flows along the heat exchange surface of heat exchange apparatus take on the form:

$$\frac{\mathrm{d}\Theta_1}{\mathrm{d}x} = b\xi\left\{\left[m\beta\nu^2 - (1+\nu)\right]\Theta_1 + \Theta_2 + \frac{\nu^2}{2}\left[1 + (2-\nu)m\beta\right]\right\} + b'(1-\xi)(\Theta_2 - \Theta_1), \tag{1}$$

$$\frac{\mathrm{d}\Theta_2}{\mathrm{d}x} = \eta b\xi\left\{\left[\beta\nu^2 - (1-\nu)\right]\Theta_1 + \Theta_2 + \frac{\nu^2}{2}\left[1 + (2+\nu)\beta\right]\right\} + \eta b'(1-o)(\Theta_1 - \Theta_2), \tag{2}$$

where
$$b' = \frac{K'}{W_1}S;\quad \Theta_1 = \frac{\bar{e}^2}{\rho\lambda}T_1;\quad \Theta_2 = \frac{\bar{e}^2}{\rho\lambda}T_2;\quad \nu = \frac{\bar{e}d}{\lambda}j;$$

$$b = \frac{K}{W_1}S;\ K = \left[\frac{1}{\alpha_1} + \frac{1}{\alpha_2} + \frac{d}{\lambda} + \nu\left(\frac{1}{\alpha_1} - \frac{1}{\alpha_2} - \frac{\nu\lambda}{\alpha_1\alpha_2 d}\right)\right]^{-1}.$$

The boundary conditions are written for the case of parallel flow as

$$\Theta_1\big|_{x=0} = \Theta_1^H;\Theta_2\big|_{x=0} = \Theta_2^H. \tag{3}$$

Solution of the resulting system of differential equations (1)-(2) with the boundary conditions (3) for parallel flow condition is written as follows:

$$\Theta_1 = C_1\left(\Psi_1 - b_2\right)\frac{e^{\Psi_1 X}}{b_1} + C_2\left(\Psi_2 - b_2\right) - -\frac{P_1}{\Psi_1 b_1}\left(\Psi_1 - b_2\right) - \frac{P_2}{\Psi_2 b_1}\left(\Psi_2 - b_2\right),$$

$$\Theta_2 = C_1 e^{\Psi_1 X} + C_2 e^{\Psi_2 X} - \frac{P_1}{\Psi_1 b_1} - \frac{P_2}{\Psi_2 b_1},$$

where
$$C_1 = \left[\Theta_1^H b_1 - \left(\Psi_1 - b_2\right)\Theta_2^H + \left(\Psi_2 - b_2\right)V - b_1\delta\right]\big/\left(\Psi_1 - \Psi_2\right),$$

$$C_2 = \left[\Theta_1^H b_1 - \left(\Psi_1 - b_2\right)\Theta_2^H + \left(\Psi_1 - b_2\right)V - b_1\delta\right]\big/\left(\Psi_2 - \Psi_1\right),$$

$$P_1 = \left(a_3 b_1 - b_3\Psi_2 + b_3 b_2\right)\big/\left(\Psi_1 - \Psi_2\right),$$

$$P_2 = \left(a_3 b_1 - b_3\Psi_1 + b_3 b_2\right)\big/\left(\Psi_2 - \Psi_1\right),$$

$$\delta = \frac{P_1}{\Psi_1 b_1}\left(\Psi_1 - b_2\right) - \frac{P_2}{\Psi_2 b_1}\left(\Psi_2 - b_2\right),\ V = -\frac{P_1}{\Psi_1} - \frac{P_1}{\Psi_2},$$

$$\Psi_{1;2} = \frac{b_2 + a_1 \pm \sqrt{\left(b_2 - a_1\right)^2 + 4a_2 b_1}}{2},$$

$$a_1 = b\xi\left[\nu^2 m\beta - \left(1+\nu\right)\right] - b'\left(1-\xi\right),\ a_2 = b\xi + b'\left(1-\xi\right),$$

$$a_3 = b\xi\frac{\nu^2}{2}\left[1 + \left(2-\nu\right)m\beta\right],\ \mathrm{b}_1 = \eta b\xi + \eta b'\left(1-\xi\right),$$

$$b_2 = \eta b \xi \left[\nu^2 \beta - (1-\nu) \right] - b' \eta (1-\xi),\ b_3 = \eta b \xi \frac{\nu^2}{2} \left[1 + (2+\nu) \beta \right] .$$

In the case of counter flow, the sign of the left side of the second equation of the initial system for the description of temperature fields of heat carriers along the heat exchanger should be changed for the opposite. It is the same as if in all the expressions for determination of Θ_1 and Θ_2 in the case of parallel flow instead of b_1, b_2, b_3 it is necessary to take $-b_1$, $-b_2$, $-b_3$.

Moreover, constants C_1 and C_2 must be found from the boundary conditions:

$$\Theta_1 \big|_{x=0} = \Theta_1^H, \Theta_2 \big|_{x=1} = \Theta_2^H .$$

In conformity with the foregoing, for determination of C_1 and C_2 in the case of a counter flow the following relations are used:

$$\begin{cases} \Theta_1^H = C_1 (\Psi_1 - b_2) \dfrac{1}{b_1} + C_2 (\Psi_2 - b_2) \dfrac{1}{b_1} + \delta, \\ \Theta_2^H = C_1 e^{\Psi_1} + C_2 e^{\Psi_2} + V. \end{cases} \tag{12}$$

From the system we obtain:

$$C_1 = \frac{-b_1 \left(\Theta_1^H - \delta \right) e^{\Psi_2} + \left(\Theta_2^H - V \right) (\Psi_2 - b_2)}{(\Psi_2 - b_2) e^{\Psi_2} - (\Psi_1 - b_2) e^{\Psi_2}}, \tag{13}$$

$$C_2 = \frac{-b_1 \left(\Theta_1^H - \delta \right) e^{\Psi_1} + \left(V - \Theta_2^H \right) (\Psi_1 - b_2)}{(\Psi_2 - b_2) e^{\Psi_1} - (\Psi_1 - b_2) e^{\Psi_2}} . \tag{14}$$

In these expressions, one should also replace b_1, b_2, b_3 by $-b_1$, $-b_2$, $-b_3$. And heat carrier temperatures in this case are found by the same formulae as for parallel flow.

The distinctive feature of counter flow mode is that in this case the radical expression in Ψ_1 and Ψ_2 may prove to be a negative value, i.e.

$$(b_2 - a_1)^2 + 4 a_2 b_1 < 0 .$$

If such is the case, then Ψ_1 and Ψ_2 can be represented in a complex form:

$$\Psi_1 = \phi + j\Psi,\ \ \Psi_2 = \phi - j\Psi,$$

where
$$\phi = (b_2 + a_1)/2,\ \ \Psi = \sqrt{\left| (b_2 - a_1)^2 + 4 a_2 b_1 \right|} .$$

In this case, changes in the temperatures Θ_1 and Θ_2 along the heat exchanger will have the nature of harmonic oscillations.

In the case when Ψ_1 and Ψ_2 are real in counter flow mode, full temperature variations in the flows of cooled and heated heat carriers when passing through heat exchanger are equal to:

$$\Delta\Theta_1 = \Theta_1 \big|_{x=0} - \Theta_1 \big|_{x=1} = \frac{C_1}{b_1} (\Psi_1 - b_2) \left(1 - e^{\Psi_1} \right) + \frac{C_2}{b_1} (\Psi_2 - b_2) \left(1 - e^{\Psi_2} \right),$$

$$\Delta\Theta_2 = \Theta_2 \big|_{x=0} - \Theta_2 \big|_{x=1} = C_1 \left(1 - e^{\Psi_1} \right) + C_2 \left(1 - e^{\Psi_2} \right).$$

The results of computational experiment are given in Figs. 2 – 4.

As it follows from Fig. 2 showing plots of temperature variation at heat exchanger outlet versus its length, for the best use of thermoelectric heat exchange apparatus one should proceed from the requirements to it. For instance, to obtain a deeper cooling with the same current value, it is necessary to use a longer thermopile and a lower filling factor. In so doing, saving of thermoelements as compared to the case when they cover the entire surface (ξ = 1) is tangible enough. The plot suggests that at ξ = 1 maximum thermopile length in heat exchange apparatus for the above conditions is 1.1 m, and liquid is cooled from 318 K to 312 K. The surface area of thermopile in this case is $S_1 = L \cdot \xi = 1.1\ L$ m^2. When $\xi = 0.2$, the length of thermopile in the heat-exchange apparatus whereby maximum possible cooling of liquid is attained is 3.6 m. In this case heat carrier is cooled from 318 K to 308 K, with the surface area of thermoelement $S_2 = 0.38$ m^2.

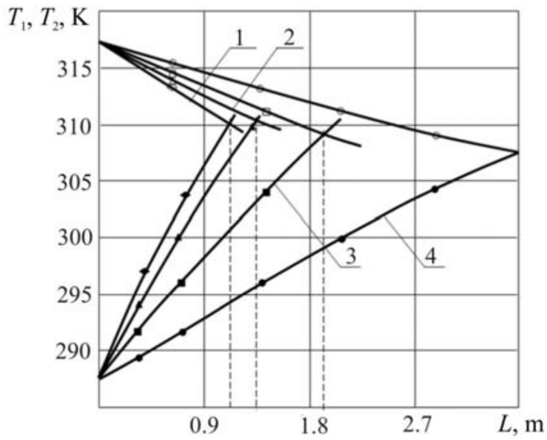

Fig. 2. Temperature variation of liquids at heat exchanger outlet versus length at different filling factors (1 – ξ = 1; 2 – ξ = 0,8; 3 – ξ = 0,5; 4 – ξ = 0,2; I = 10 A).

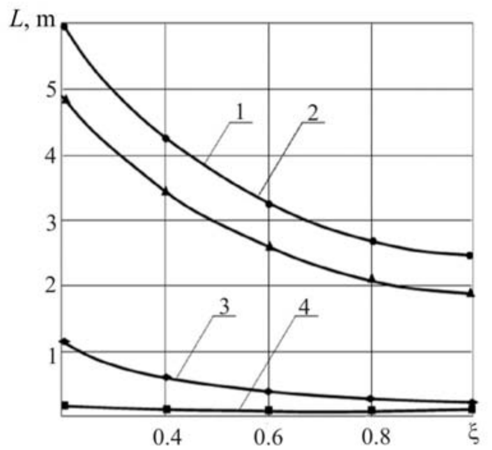

Fig. 3. Ultimate thermopile lengths in the intensification mode versus filling factor at different temperatures (1 – T^H = 327 K; 2 – T^H = 318K; 3 – T^H = 297 K; 4 – T^H = 293 K; T^H_2 = 287K; I = 5 A).

If we now compare these two cases, cooling of outlet liquid in the second case is deeper than in the first one. Moreover, surface area in the second case is smaller than the area in the first case, i.e. $S_2 < S_1$, which means saving of thermoelement material and consumable electric energy.

Fig. 3 represents the plots of change in ultimate thermopile lengths versus filling factor, i.e. the lengths whereby liquid temperatures at heat exchanger outlet are equal. As it follows from the represented data, the larger temperature difference of heat carriers at heat exchanger inlet, the larger thermopile length is necessary to hold the intensification mode. The plots are monotonously decreasing versus filling factor. The larger temperature difference of heat carriers at the inlet, the more drastically functions $L = L(\xi)$ decay at constant supply current $I = 5$ A.

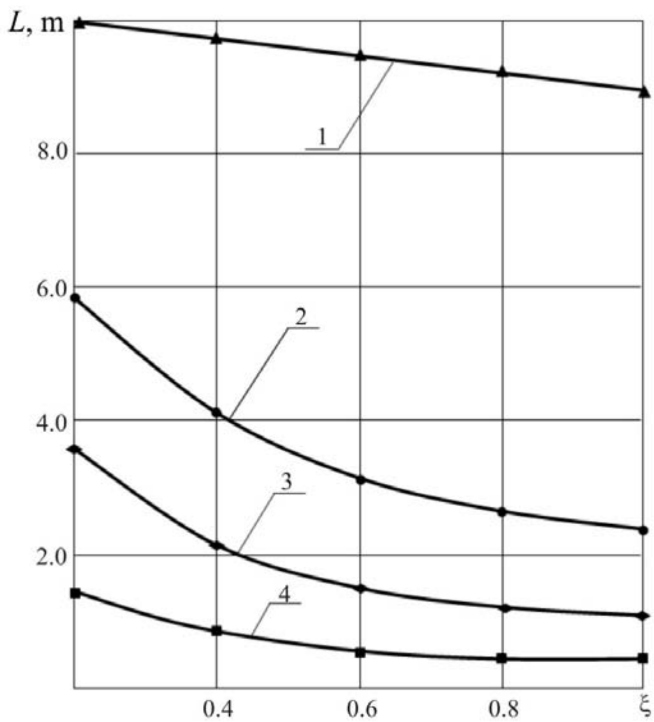

Fig. 4. Ultimate thermopile lengths in the intensification mode versus filling factor at different supply currents(1 – I = 1 A; 2 – I = 5 A; 3 – I = 10 A; 4 – I = 20 A; T^H_1 = 318 K; T^H_2 = 287 K).

Fig. 4 represents the plots of ultimate thermopile lengths in the intensification mode versus filling factor at various supply currents and constant temperature difference of heat carriers at heat exchanger inlet. Functions $L = L(\xi)$ are also monotonously decaying. In so doing, the larger supply current value, the smaller thermopile length is required to hold the intensification mode.

Analysis of calculation results shows that current value whereby the work of heat exchanger in the intensification mode is still possible essentially depends on temperature difference of heat carriers at heat exchanger inlet. The higher this difference, the larger is current magnitude in the intensification mode with otherwise equal parameters.

The results of calculation of coefficient of performance $\varepsilon(\nu)$, coefficient of conversion $\mu(\nu)$ and heat carrier temperatures at heat exchanger outlet 1 m and 0.5 m, T_{1inl} = 45°C, T_{2inl} = 14°C are given, where T_{1inl}, T_{2inl} are inlet temperatures of cooled and heated heat carriers. According to these data, the larger filling factor, the smaller current variation area for the intensification mode. A deeper cooling can be achieved in the intensification mode with smaller currents, but with larger values of filling factor ξ. However, with lower ξ values such cooling may not be, since considerable current increase leads to further heating of cooled heat carrier. As regards heated heat carrier, with the same current value a more intensive heating occurs at low values of factor ξ. Comparison of coefficients of performance or conversion coefficients shows that under otherwise equal conditions these coefficients are higher in the lower length heat exchanger. If we consider one and the same heat exchanger with different ξ values, these coefficients are higher for lower ξ values.

The results of calculations of $\varepsilon(\nu)$, $\mu(\nu)$ and heat carrier temperatures at heat exchanger outlet with thermopile length 0.24 m, T_{1inl} = 30°C, T_{2inl} = 21°C are given. In this case the ranges of current values whereby the intensification mode is observed for different filling factors are close to each other. These ranges decay more slowly than in the case of higher temperature difference of heat carriers at heat exchanger inlet. And the difference in temperatures at heat exchanger outlet for ultimate current values in the intensification mode is not more than one degree.

Experimental investigations of heat exchange apparatus.

To perform full-scale investigations of heat exchange apparatus, a test bench was developed the basic diagram of which is shown in Fig. 5.

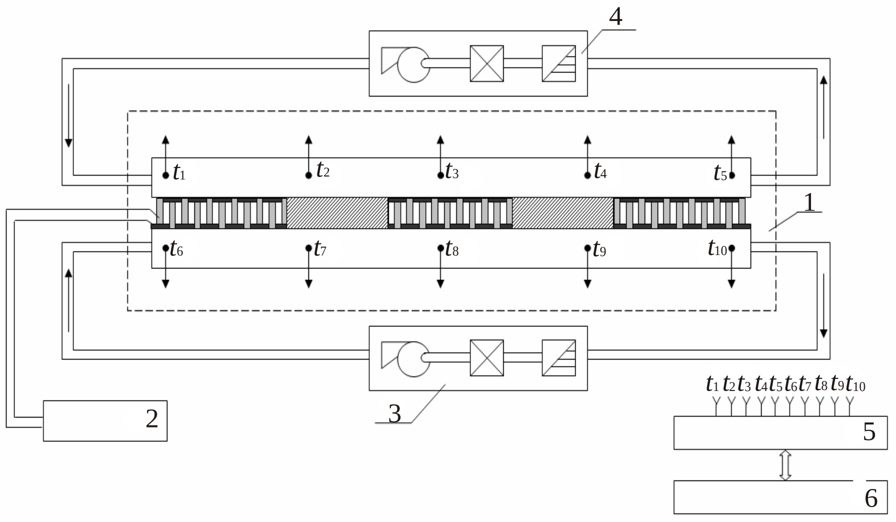

Fig. 5. Basic diagram of test bench.

Heat exchange apparatus 1 is connected to circulation loops of cold and heat carriers in which capacity water is used, as well as to adjustable direct current source 2. Ultra thermostats 3 and 4 maintain given temperatures of cold and heat carriers at heat exchange apparatus outlets to an accuracy of 0.1°C and provide for their circulation. Flow rates in the cold and heat carrier loops are adjusted by valves located on the respective ultra thermostats.

Recording of current temperature values is done by measuring system IRTM 2402/M3 5 which is connected to PC 6 and provides for possible connection of up to 24 temperature sensors.

The object for experimental studies was a thermoelectric heat exchange apparatus of flow-through type (Fig. 6), consisting of two steel pipes with internal diameter 7 mm and length 250 mm. The outer surface of the pipes was polished as a tetrahedron 20 × 20 mm. As thermopiles, unified thermoelectric modules (TEM) of BPTM and TEM types, developed in the laboratory of semiconductor thermoelectric instruments and devices of Dagestan State Technical University, were used.

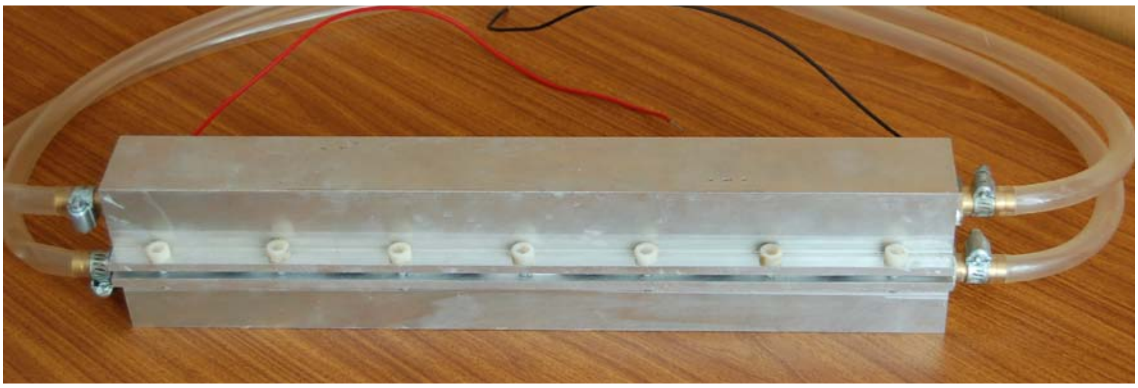

Fig. 6. External view of a thermoelectric heat exchange apparatus with thermal bridges.

Electrically connected in series, TEM and thermal bridges are fixed between the two pipes through thermally conductive paste KPT.

To exclude heat exchange with the environment, the entire structure is placed into a foam package. Mounted on the lateral faces along the length of the pipes are copper-constantan thermocouples $t_1 - t_{10}$ the reference junctions of which are thermally stabilized at 0°C in a Dewar flask.

Experimental curves of temperatures distribution along the length were obtained by measurement of temperatures at different points of heat exchange apparatus:

– for different factors of filling with semiconductor modules,

– for different currents flowing through thermoelement,

– for different materials of thermal bridges.

All the experiments were carried out at given constant temperatures and flow rates of cold and heat carriers at structure inlets. For this purpose, only semiconductor TEM in the amount of 10 were placed at first in the heat exchange apparatus (100% filling with TEM, filling factor $\xi = 1$).

Ultra thermostats were used to maintain the assigned flow rates and temperatures of cold and heat carriers at the inlets. On stabilizing the mode of ultra thermostats (in 15 – 20 minutes), a program of cyclic polling of thermocouples was run on the measuring system with subsequent output of temperature values to PC. At the same time, the heat exchange apparatus was connected to direct current source by the measuring instrument of which the necessary value of supply current was established.

Filling factor was varied as follows. After device assembly, part of TEM were disconnected, the remaining modules were moved apart and uniformly distributed along the length of the pipe, between the TEM thermal bridges were installed made of suitable material (copper, aluminum, steel), the total area of which was equal to the total area of modules removed. For this purpose, thermal bridges of different sizes were manufactured. Thus, for instance, to obtain 80% of filling ($\xi = 0.8$), two modules of 10 were removed, to be replaced by two thermal bridges, each of which is equal to 1/4 of module area.

Based on the test bench, a series of experiments was carried out giving an idea of practical applicability of the elaborated mathematical model of thermoelectric heat exchange apparatus with thermal bridges.

The basic task when performing the experimental investigations of the prototype of heat exchange apparatus was to determine the dependence of temperature at the above control points on thermoelement supply current, filling factor and thermal bridge material.

Fig. 7 presents the results of experimental investigations (points) of heat exchange apparatus under steady-state operating modes with different TEM filling factors and, for comparison, the results of theoretical calculations (solid line). The calculations employed passport data on thermophysical properties of semiconductor substance and characteristics of TEM used in the structure, the geometric parameters of legs, and the values of contact electric and thermal resistances. The heat capacities of the pipes in the calculations were disregarded. Comparison with the results shows an agreement between the calculated and experimental data. Maximum quantitative discrepancies between the theoretical calculations and the experimental results not exceeding 12% were observed in the initial section of the pipelines, which can be considered satisfactory. From the figure it follows that as filling factor increases, temperature distribution curves change more drastically, i.e. heat exchange apparatus operates more efficiently.

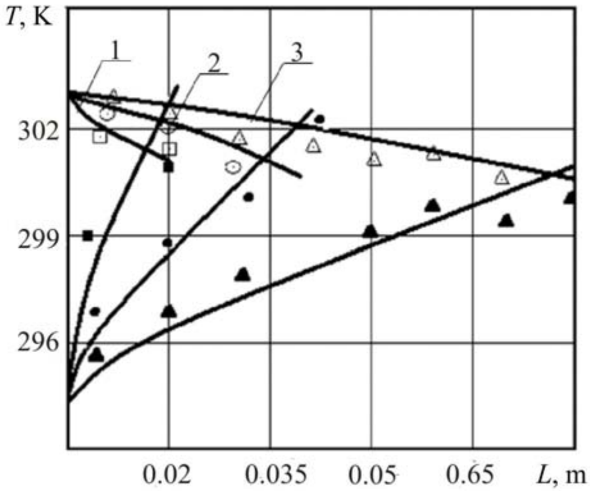

Fig. 7. Experimental and theoretical dependences of heat carrier temperatures at heat exchanger outlet on length at different filling factors (1 – ξ = 1.0; 2 – ξ = 0.5; 3 – ξ = 0.2; TEM supply current 1.8 A).

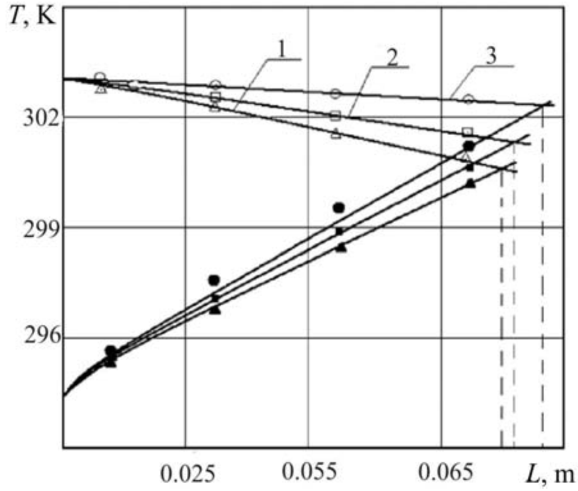

Fig. 8. Experimental and theoretical dependences of heat carrier temperatures at heat exchanger outlet on length for thermal bridges of different materials (1 – copper, 2 – aluminum, 3 – steel; TEM supply current 1.8 A)

Fig. 8 shows the curves of temperature distribution along the length of heat exchange apparatus with filling coefficient $\xi = 0.2$ for thermal bridges made of different materials: copper, aluminum and steel. From the represented data it follows that the operating efficiency of heat exchange apparatus does not essentially depend on thermal bridge material, as confirmed by the results of numerical experiment. However, the use of copper thermal bridges is more efficient, which is due to a higher thermal conductivity of this material.

Fig. 9 shows a dependence of thermopile length with device operation in the intensification mode on supply current at filling factor $\xi = 0.5$ for thermal bridges made of copper. As is obvious, increase in supply current reduces the area of heat exchanger where thermopile works in the mode of heat exchange intensification. For instance, with current increase by 0.5 A for given conditions the above length is reduced by 1.8 cm.

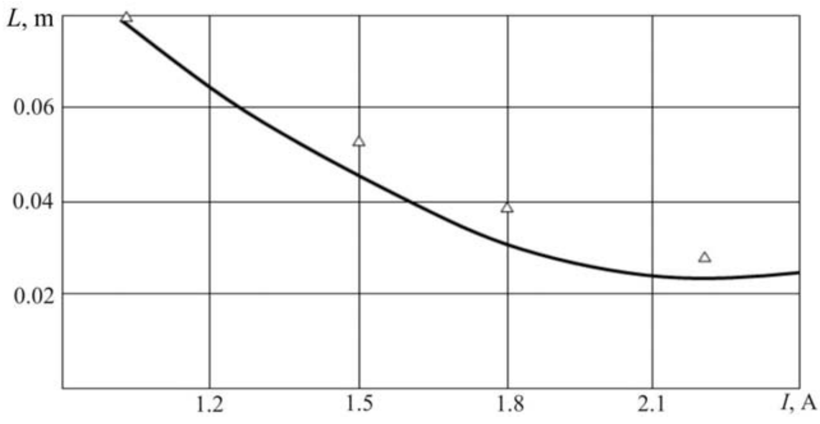

Fig. 9. Dependence of thermopile ultimate length in heat exchange apparatus on supply current (ξ = 0.5, thermal bridge material – copper).

Fig. 10 shows experimental time dependences of temperature variation at a point 1.5 cm distant from heat exchanger inlet. The represented data suggest the onset of stabilization in about 27 minutes after thermopile switching.

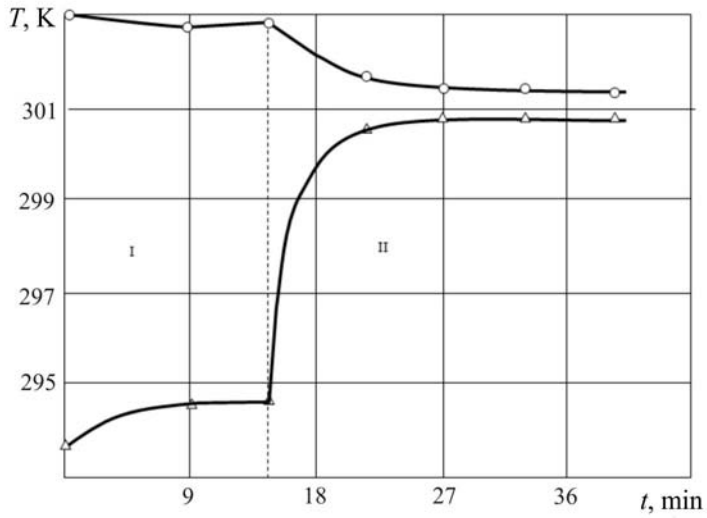

Fig. 10. Dependence of heat carrier temperatures on time without (area I) and with switched thermopile (area II) (ξ = 0.8, L = 0.015 m; I = 1.8 A, thermal bridge material – copper).

On the whole, the results of experimental investigations prove the validity of the elaborated mathematical model.

Conclusions

1. On the basis of investigations performed, the following conclusions can be made:
2. Based on the results of the survey of methods for heat exchange intensification, as well as the structures of heat exchangers it is shown that for the intensification of heat exchange between two heat carrier flows it is advisable to use thermoelectric power converters.
3. A new concept of creating thermoelectric recuperative-type heat exchangers consisting in combination of heat transfer through highly thermally conductive material (thermal bridge) heat exchange intensification with the use of thermopiles.
4. A mathematical model is developed for the calculation of thermal mode of thermoelectric recuperative-type heat exchangers combined with thermal bridges of various configurations.
5. The basic characteristics of thermoelectric heat exchanger with thermal bridges have been calculated, including temperature variation of heat carriers depending on filling factor, thermopile supply current, as well as the length of heat exchange apparatus.
6. The adequacy of the elaborated mathematical model has been proved experimentally; comparison of the experimental and calculated data has shown that their discrepancy has not exceeded the permissible values.

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