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OPTIMIZATION OF THERMAL CONNECTIONS IN LIQUID-LIQUID THERMOELECTRIC HEAT PUMPS FOR WATER PURIFICATION DEVICES OF SPACE APPLICATION

The paper presents the results of computer simulation of liquid-liquid thermoelectric heat pump. Multi-parameter computer optimization was used to determine parameters and arrangement of thermoelectric modules and heat exchangers to achieve the highest efficiency.

Key words: thermoelectric heat pump, computer simulation, liquid-liquid system.

Introduction

General characterization of the problem. The use of thermoelectric heat pumps in air-conditioning systems is related to their unique properties [1 – 5]: environmental friendliness (such devices have no toxic coolants); reliability (mechanical stability, long operational life); independence of orientation in space (possibility of work in the absence of gravitation) [6, 7].

An example of efficient application of thermoelectric heat pumps is provided by systems of water recovery from liquid biowaste on board of manned spacecrafts (urine, atmosphere humidity condensate, sanitary-hygiene water). Efficiency tests of such equipment on NASA stand have shown that in the most important parameters, namely specific power consumption, dimensions, weight and quality of distillate produced the system of water purification with thermoelectric heat pump outperforms known analogs of space application [4, 5].

However, new, more stringent requirements are imposed on such devices due to the possibility of their new applications (piloted missions for exploration of Mars and other planets). This is mainly related to the reduction of their weight, dimensions, as well as energy consumption for the operation of thermoelectric heat pump. The problem of further quality improvement of such devices is very complicated, since the attained values of their efficiency are close to boundary.

One of the methods for solving this problem is optimization of thermoelectric heat pump not as a whole, but of each thermoelectric module and heat exchanger in particular. This approach involves creating such optimal operating conditions for each thermoelectric converter that will assure the best efficiency values of the entire device.

This multi-factor problem was solved with the use of modern methods of computer object-oriented programming.

The purpose of this paper is efficiency increase of thermoelectric heat pump via multi-parameter computer optimization of the arrangement of thermoelectric modules and heat exchangers of the heat pump.

Physical model of thermoelectric heat pump

A physical model of thermoelectric heat pump is presented in Figs.1 – 3. It comprises heat exchangers 1 assuring passage of heat flux Q_h through the hot side of thermoelectric modules, thermoelectric modules 3 as such, heat exchangers 2 assuring passage of heat flux Q_c through the cold side of thermoelectric modules and a system of hydraulically bound channels 4 providing for circulation of liquid in the thermoelectric heat pump.

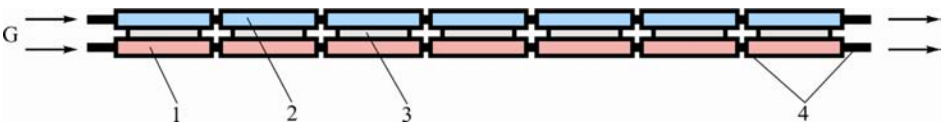

Fig. 1. The simplest physical model of thermoelectric heat pump.

In the simplest case this model represents series-connected hot 1 and cold 2 heat exchangers with thermoelectric modules 3 arranged between them (Fig. 1). However, practical implementation of such design is not always rational. This is due to considerable dimensions of such device.

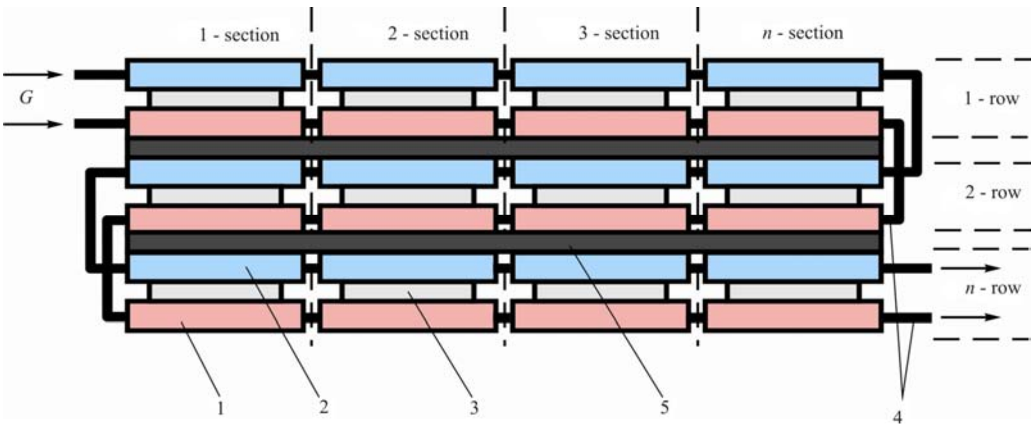

Fig. 2. Physical model of thermoelectric heat pump with thermal insulation.

In practice it is more convenient to connect heat exchangers 1 and 2 with thermoelectric modules 3 in rows with a different number of sections having thermal insulation 5 between them.

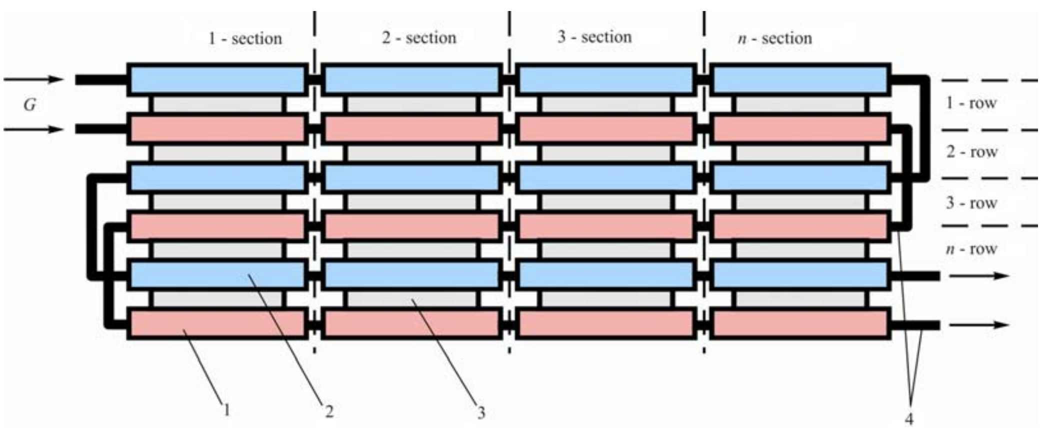

Fig. 3. Physical model of thermoelectric heat pump.

However, to reduce the weight and dimensions of such device, one can simplify the design proposed in Fig. 2. In this case a row of heat exchangers will assure the operating conditions of two rows of thermoelectric modules (Fig. 3). Owing to this, the number of heat exchangers, hence, the weight and dimensions of such device will be reduced considerably.

Mathematical and computer description of the model

To describe heat and electric current fluxes, let us use the laws of conservation of energy

$$div\vec{E} = 0 \tag{1}$$

and electric charge

$$div\vec{j} = 0, \tag{2}$$

where

$$\vec{E} = \vec{q} + U\vec{j}, \tag{3}$$

$$\vec{q} = \kappa\nabla T + \alpha T\vec{j}, \tag{4}$$

$$\vec{j} = -\sigma\nabla U - \sigma\alpha\nabla T. \tag{5}$$

Here $\vec{E}$ is the energy flux density, $\vec{q}$ is the thermal flux density, $\vec{j}$ is the electric current density, U is the electric potential, T is a temperature, α, σ, κ are the Seebeck coefficient, electric conductivity and thermal conductivity.

With regard to (3) – (5) one can obtain

$$\vec{E} = -(\kappa + \alpha^2\sigma T + \alpha U\sigma)\nabla T - (\alpha\sigma T + U\sigma)\nabla U. \tag{6}$$

Then the laws of conservation (1), (2) will acquire the form:

$$-\nabla\left[(\kappa + \alpha^2\sigma T + \alpha U\sigma)\nabla T\right] - \nabla\left[(\alpha\sigma T + U\sigma)\nabla U\right] = 0, \tag{7}$$

$$-\nabla(\sigma\alpha\nabla T) - \nabla(\sigma\nabla U) = 0. \tag{8}$$

Nonlinear differential equations of second order in partial derivatives (7) and (8) determine the distribution of temperature T and potential U in thermoelements.

An equation describing the process of heat transport in the walls of heat exchangers in the steady-state case is written as follows:

$$\nabla(\text{-}k_1 \cdot \nabla T_1) = Q_1, \tag{9}$$

where k_1 is the thermal conductivity of heat exchanger walls, ∇T_1 is a temperature gradient, Q_1 is a heat flux.

The processes of heat-and-mass transfer of heat carriers in heat exchanger channels in the steady-state case are described by equations [8]

$$-\Delta p - f_D \frac{\rho}{2d_h} v|\vec{v}| + \vec{F} = 0, \tag{10}$$

$$\nabla(A\rho\vec{v}) = 0, \tag{11}$$

$$\rho \mathrm{AC_p}\vec{v} \cdot \nabla T_2 = \nabla \cdot Ak_2\nabla T_2 + f_D \frac{\rho A}{d_h}|\vec{v}|^3 + Q_2 + Q_{wall}, \tag{12}$$

where p is a pressure, ρ is a heat carrier density, A is a cross-section of the tube, $\vec{F}$ is the sum of all forces, C_p is the heat carrier heat capacity, T_2 is a temperature, $\vec{v}$ is a velocity vector, k_2 is the heat carrier thermal conductivity, f_D is the Darcy coefficient, $d = \frac{4A}{Z}$ is an effective diameter, Z is the perimeter of tube wall, Q_2 is the heat which is released due to viscous friction [W/m] on the unit length of heat exchanger, Q_{wall} is the heat flux coming from the heat carrier to the tube walls [W/m]

$$Q_{wall} = h \cdot Z \cdot \left(T_1 - T_2\right), \tag{13}$$

where h is the heat exchange coefficient which is found from equation

$$h = \frac{Nu \cdot k_2}{d}. \tag{14}$$

Here Nu is the Nusselt number found from equation:

$$Nu = \frac{\left(\frac{f_d}{8}\right)\left(Re - 1000\right)Pr}{1 + 12.7\left(\frac{f_d}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)}, \tag{15}$$

where $Pr = \frac{C_p \mu}{k_2}$ is the Prandtl number, μ is the dynamic viscosity, $Re = \frac{\rho v d}{\mu}$ is the Reynolds number, $3000 < Re < 6 \cdot 10^6$, $0.5 < Pr < 2000$.

The Darcy coefficient f_D is found with the use of the Churchill equation for the entire spectrum of the Reynolds number and all the values of e/d (e is roughness of wall surface)

$$f_D = 8\left[\frac{8}{Re}^{12} + \left(A + B\right)^{-1.5}\right]^{1/12}, \tag{16}$$

where $A = \left[-2.457 \cdot \ln\left(\left(\frac{7}{Re}\right)^{0.9} + 0.27\left(e/d\right)\right)\right]^{16}$, $B = \left(\frac{37530}{Re}\right)^{16}$.

Solving Eqs.(7) – (12), we obtain the distributions of temperatures, electric potential (for thermoelements), velocities and pressure (for heat carrier).

The above differential equations with the respective boundary conditions were solved using Comsol Multiphysics package of applied programs.

Computer simulation results

Below are given the results of optimization of the arrangement of thermoelectric modules and heat exchangers for real thermal and temperature operating conditions of heat pump for water purification device of space application. The results of computer investigations of heat exchanger design, as well as electric power supply to thermoelectric modules are very important and will be presented in detail in the next work.

The initial data:

electric power supply to thermoelectric modules – 300 W;

the number of thermoelectric modules – 80 pcs;

heat carrier temperature at inlet to hot heat transfer loop – 36 ºC;

heat carrier temperature at inlet to cold heat transfer loop – 31.5 ºC;

hydraulic resistance of each heat-transfer loop – 0.07 atm;

heat carrier flow rate in each loop – 22 ml/s.

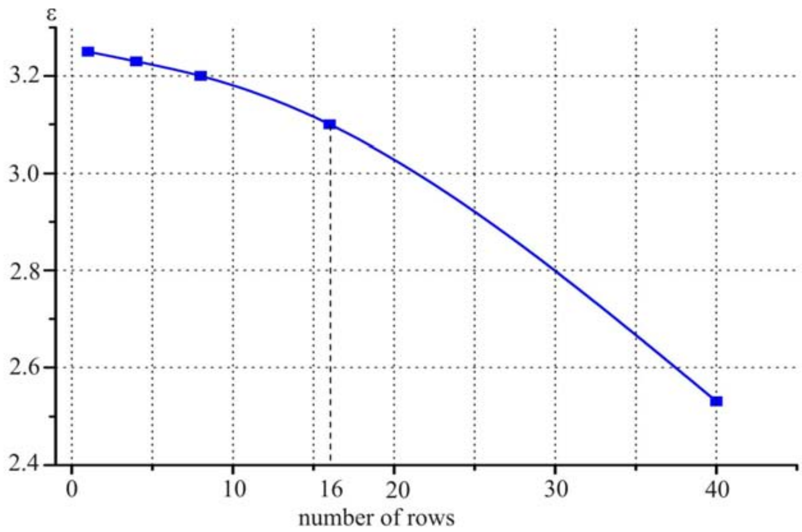

Fig. 4. Dependence of the heating coefficient of thermoelectric heat pump on the number of rows.

In this fashion, the values of the integral heating coefficient and device weight reduction (percentagewise) were calculated for different design variants of thermoelectric heat pump:

1) 1 row of 80 thermoelectric modules and 160 heat exchangers (Fig.1 *a*);

2) 4 rows of 40 thermoelectric modules and 120 heat exchangers (Fig.1 *b*);

3) 8 rows of 10 thermoelectric modules and 90 heat exchangers (Fig.1 *b*);

4) 16 rows of 5 thermoelectric modules and 85 heat exchangers (Fig.1 *b*);

5) 40 rows of 2 thermoelectric modules and 82 heat exchangers (Fig.1 *b*).

Fig. 2 represents a dependence of the heating coefficient of thermoelectric heat pump ε on the number of rows. As was to be expected, the idealized model version (Fig. 1) has the highest efficiency.

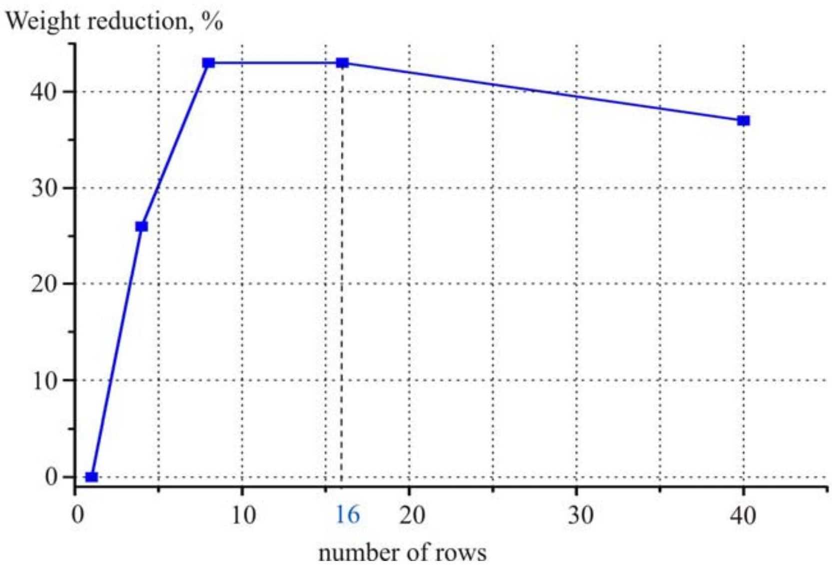

Fig. 5.Dependence of percentage weight reduction of thermoelectric heat pump on the number of rows.

For the selection of the most rational design of thermoelectric heat pump, of great importance are mass-dimensional parameters. Analysis of percentage weight reduction of thermoelectric heat pump depending on the number of rows in it (Fig. 3) suggests the presence of an optimum, since decreasing the number of heat exchangers is attended with increasing the number of connecting elements, which makes the design heavy. Therefore, an optimum has been found which is in the area of 16 rows including 5 thermoelectric modules with 85 heat exchangers.

Comparison of computer simulation results to previous investigations [5] testifies that the efficiency (heating coefficient) of thermoelectric heat pump with the proposed arrangement of thermoelectric modules and heat exchangers is improved by 15 – 20 %.

Conclusions

1. The efficiency of thermoelectric heat pump as a function of its design parameters was calculated.
2. It was established that dependence of percentage weight reduction of thermoelectric heat pump on the number of rows has an optimum in the area of 16 rows.
3. Comparison of the obtained results to previous investigations [5] testifies that the efficiency of thermoelectric heat pump with the proposed arrangement of thermoelectric modules and heat exchangers is improved by 15 – 20 %.

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