
A.V. Prybyla

A.V. Prybyla, *cand. Phys. - math. Sciences*[1,2]
L.I.Anatychuk *acad. National Academy of sciences of Ukraine*[1]


L.I.Anatychuk

[1]Institute of Thermoelectricity
of the NAS and MES of Ukraine,
1, Nauky str., Chernivtsi, 58029, Ukraine;
e-mail: anatych@gmail.com
[2]Yu.Fedkovych Chernivtsi National University,
2, Kotsiubynskyi str., Chernivtsi, 58000, Ukraine

INFLUENCE OF MINIATURIZATION ON THE EFFICIENCY OF A SPACE-PURPOSE THERMOELECTRIC HEAT PUMP

The paper presents the results of calculating the influence of miniaturization on the boundary possibilities of a thermoelectric liquid-liquid heat pump, in particular for its use as a high-efficiency heater for a space-purpose water purification device. Bibl. 10, Fig. 5.
Key words: thermoelectric heat pump, efficiency, distiller.

Introduction

General characterization of the problem. The use of thermoelectric heat pumps (THPs) in air and liquid conditioning systems, special- purpose evaporators is associated with their unique advantages [1 - 7], in particular environmental friendliness (there are no toxic refrigerants in such equipment); reliability (resistance to mechanical impacts, long service life); independence from the orientation in space (the ability to work in the absence of gravity).

An example of the effective use of thermoelectric heat pumps are devices for water regeneration from liquid waste on board manned spacecraft (urine, condensate, sanitary water) [5 - 7]. Tests of their efficiency at the NASA stand showed that in the most important indicators - specific energy consumption, dimensions, weight and quality of the distillate- water purifiers with thermoelectric heat pump outperform known space-purpose analogues [6].

However, such devices are subject to new, higher requirements related to the possibilities of their new applications (manned missions to explore Mars and other planets). This mainly concerns the reduction of their weight and size while maintaining (or even improving) the achieved level of energy efficiency. In [8, 9] the results of calculations of the influence of miniaturization of thermoelectric modules in the heating mode are given. The influence of the height of thermoelectric material legs on the heating coefficient of thermoelectric modules was determined by computer simulation, and the optimal height of the leg of material was found, which provides minimal losses of energy conversion efficiency. However, the complex task of optimizing the thermoelectric heat pump, providing a reduction of its weight and dimensions, has not yet been solved.

The purpose of our work is to study the energy efficiency of a space-purpose thermoelectric heat pump under conditions of reduction of its overall dimensions.

Physical model of THP

Physical model of a thermoelectric heat pump is shown in Figs. 1 - 3. It consists of heat exchangers 1, providing the passage of heat flux *Qh* through the hot side of thermoelectric modules, thermoelectric modules proper 3, heat exchangers 2, providing the passage of heat flux *Qc* through the cold side of thermoelectric modules and a system of hydraulically coupled channels 4, providing circulation of liquid in the thermoelectric heat pump.

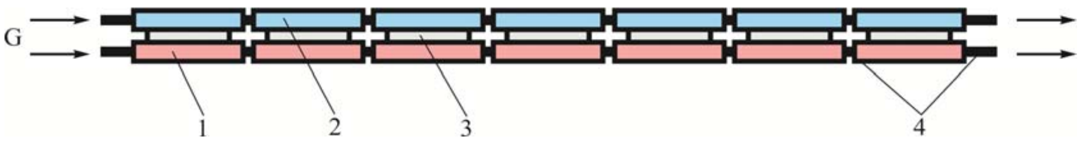

Fig. 1. The simplest physical model of a thermoelectric heat pump.

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

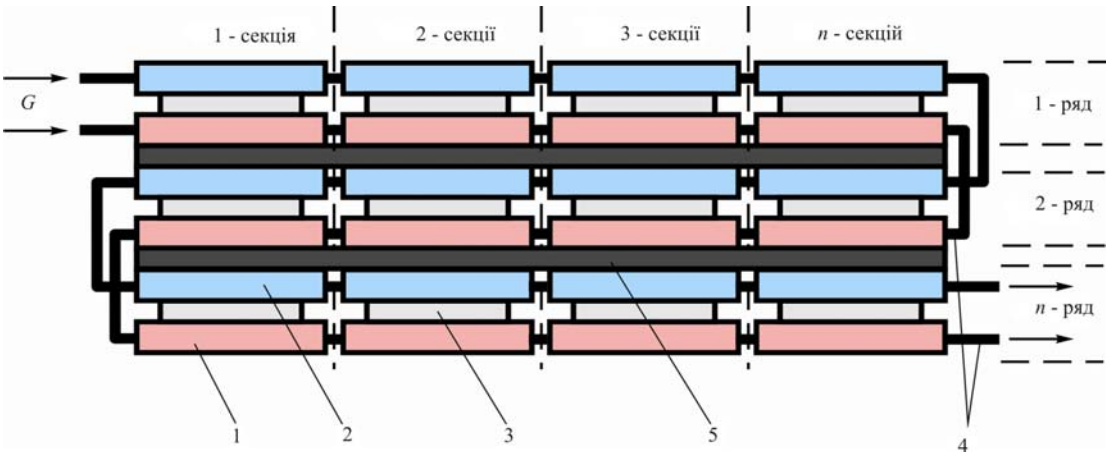

Fig. 2. Physical model of a thermoelectric heat pump with thermal insulation.

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

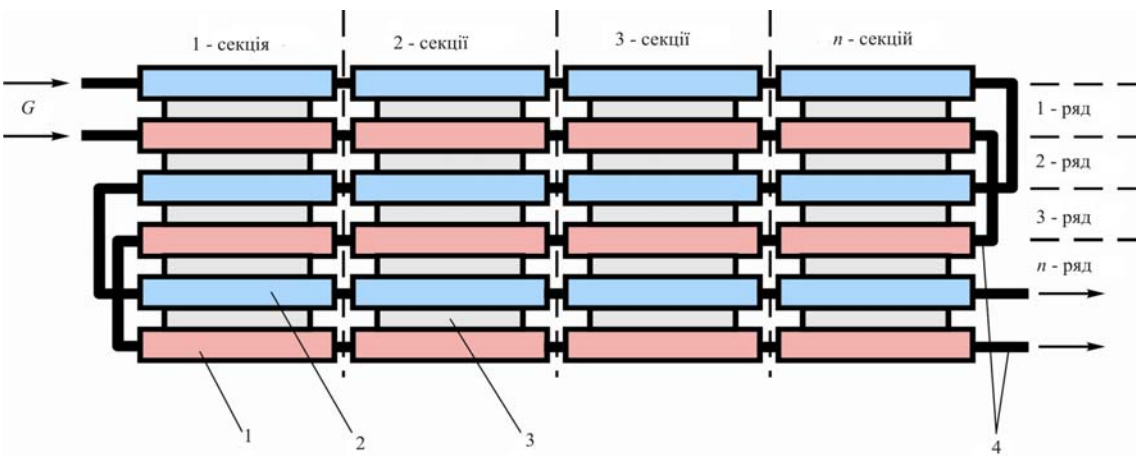

Fig. 3. Physical model of a thermoelectric heat pump.

However, to reduce the weight and dimensions of such equipment, you can simplify the design proposed in Fig. 2. In this case, a number of heat exchangers will provide the operating conditions of two rows of thermoelectric modules (Fig. 3). This makes it possible to significantly reduce the number of heat exchangers, and, consequently, the weight and dimensions of such a device.

Mathematical and computer description of the model

To describe the heat and electricity fluxes, we will use the laws of conservation of energy

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

and electrical 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 energy flux density, $\vec{q}$ is heat flux density, $\vec{j}$ is electrical current density, U is electrical potential, T is temperature, α, σ, κ are the Seebeck coefficient, electrical 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 take on the form:

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

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

The 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 thermal conductivity of heat exchanger walls, ∇T_1 is temperature gradient, Q_1 is 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 [10]

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

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

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

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

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

where h is heat exchange coefficient which is found from equation

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

Here, Nu is the Nusselt number which is found from equation

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

where $Pr = \frac{C_p \mu}{k_2}$ is the Prandtl number, μ is dynamic viscosity, $Re = \frac{\rho v d}{\mu}$ is the Reynolds number, $3000 < Re < 6 \cdot 106$, $0.5 < Pr < 2000$.

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)

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

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

From the solution of equations (7) – (12) we obtain the distribution of temperatures, electrical potential (for thermoelements), velocities and pressure (for heat carrier).

To solve the above differential equations with the respective boundary conditions, the Comsol Multiphysics applied software package was used.

Computer simulation results

Below are the results of calculating the parameters of a thermoelectric pump in accordance with the physical model shown in Fig. 3. The influence of energy consumption W_{pump} on the heat carrier pumping through the heat exchange system on the heating coefficient μ of a thermoelectric heat pump was studied for different heights of thermoelectric power converters (*h* from 0.1 to 1 mm) and different temperature differences of the heat carriers at the inlet to the heat exchange circuit of a thermoelectric heat pump (W_{pump}) (Fig. 4).

Thus, from the analysis of Fig.4 it is seen that the heating coefficient of a thermoelectric heat pump weakly depends on the height of the thermoelectric converter up to the height of the thermoelectric converter leg of 0.5 mm and begins to sharply decrease with its further miniaturization. So, when the height of the thermoelectric converter leg decreases by 2 times (from 1 to 0.5 mm), the heating coefficient decreases by only 5 %, but its subsequent decrease (to a height of 0.25 mm) leads to a decrease in μ already by~22 %, and with a leg height of 0.1 mm μ decreases by~45 %. On the other hand, a twofold decrease in the leg height leads to a decrease in the weight of the heat pump by 25 %, the volume by 28 %, and also its cost by 35 %.

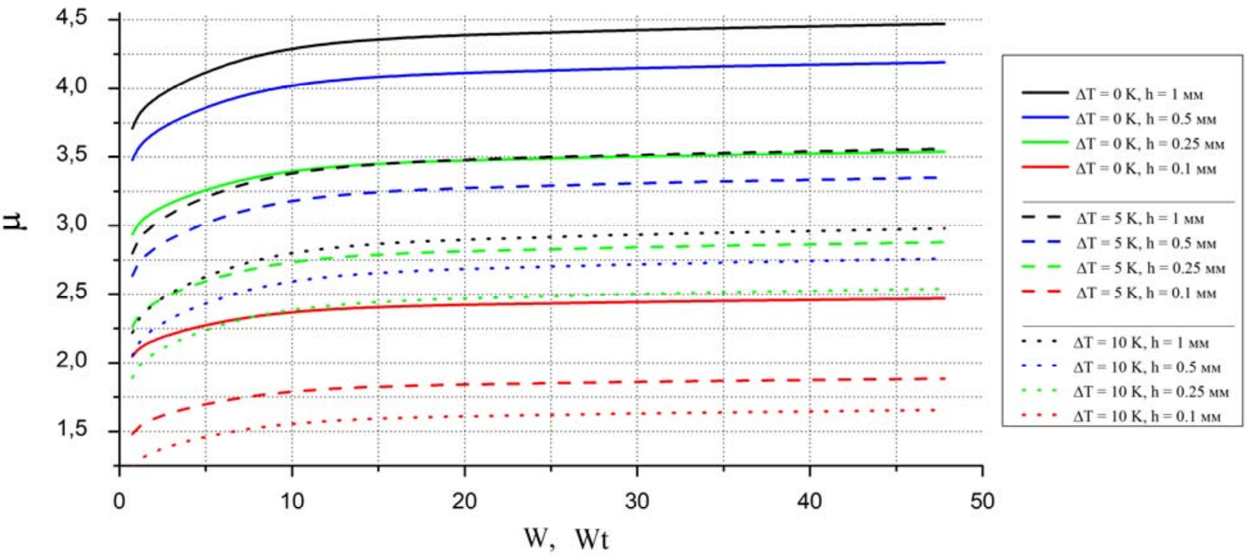

Fig. 4. Dependence of the heating coefficient of a thermoelectric heat pump μ on the supply power of heat exchange system W for different heights of thermoelectric power converters h and different temperature differences of heat carriers at the inlet to heat exchange circuits of a thermoelectric heat pump ΔT.

In addition, we analyzed the losses in the efficiency of the heat pump caused by the need for additional power supply to the heat exchange system (Fig. 5). Analysis of Fig. 5 shows that with an increase in the supply power of the liquid pump, which ensures the circulation of the heat carrier in the heat exchange system, the heating coefficient of the thermoelectric heat pump first increases, which is due to a decrease in the loss of the temperature difference in the heat exchange system due to an increase in the circulation rate of the heat carrier. Taking into account in the expression for the heating coefficient of a thermoelectric heat pump (17) the energy consumption for heat carrier pumping (18) leads to the fact that μ, reaching a maximum, starts gradually decreasing, because energy consumption for heat carrier pumping begin to reach the level of energy consumption for the functioning of thermoelectric modules.

$$\mu = \frac{Q_г}{W_{тн}}, \tag{17}$$

$$\mu_{втр} = \frac{Q_г}{W_{тм} + W_{нас}}, \tag{18}$$

where Q_h is the heat output of the heat pump, W_{TM} is the supply power of thermoelectric modules, W_{pump} is the supply power of the liquid pumps of the heat exchange system.

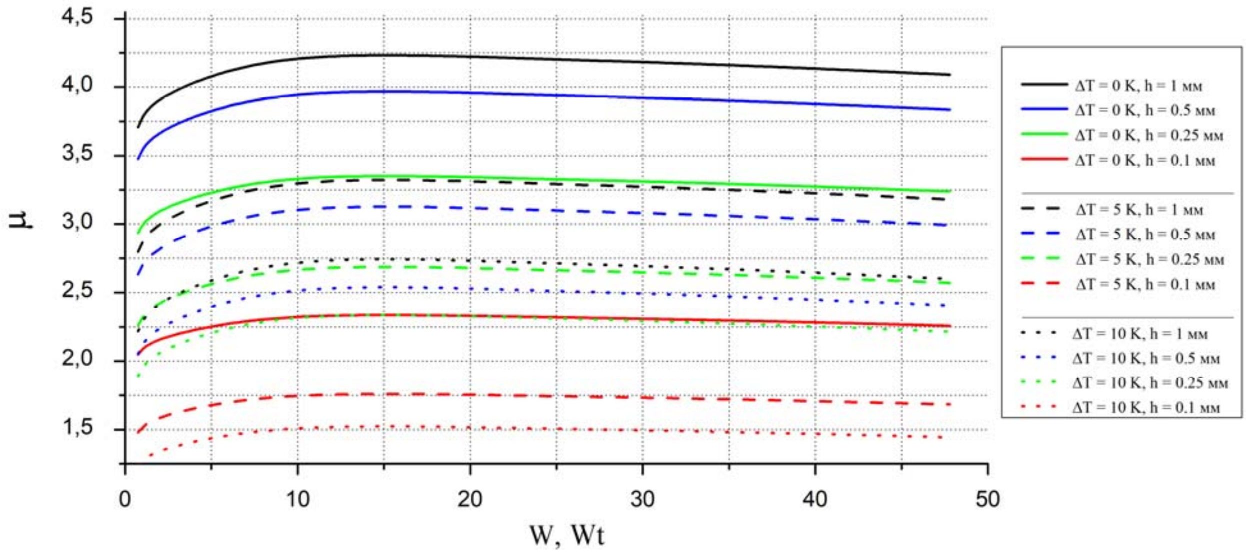

Fig. 5. Dependence of the heating coefficient of a thermoelectric heat pump μ (with regard to energy consumption for heat carrier pumping) on the supply power of heat exchange system W for different heights of thermoelectric power converters h and different temperature differences of heat carriers at the inlet to heat exchange circuits of a thermoelectric heat pump ΔT.

Conclusions

1. The influence of energy consumption for heat carrier pumping through heat exchange system on the heating coefficient μ of a thermoelectric heat pump has been established for different heights of thermoelectric power converters (*h* from 0.1 to 1 mm) and different temperature differences at the inlet to heat exchange circuits of a thermoelectric heat pump (ΔT from 0 to 10 K) (Fig.4).
2. It has been determined that the heating coefficient of a thermoelectric heat pump weakly depends on the height of the thermoelectric converter up to the height of the thermoelectric converter leg of 0.5 mm and begins to sharply decrease with its further miniaturization. So, when the height of the thermal converter leg decreases by 2 times (from 1 to 0.5 mm), the heating coefficient decreases by only 5%, but its subsequent decrease (to a height of 0.25 mm) leads to a decrease in μ already by~ 22%, and with a leg height of 0.1 mm μ decreases by~45%. On the other hand, a twofold decrease in the leg height leads to a decrease in the weight of the heat pump by 25%, the volume by 28%, and also its cost by 35%.
3. It has been established that taking into account in the expression for the heating coefficient of a thermoelectric heat pump of energy consumption for heat carrier pumping leads to the fact that μ, on reaching a maximum, starts gradually decreasing, because energy consumption for heat carrier pumping

begins to reach the level of energy consumption for the functioning of thermoelectric modules.

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Анатичук Л.І., *акад. НАН Украини*[1,2]
Прибила А.В., *канд. физ.-мат. наук*[1,2]

[1]Інститут термоелектрики НАН і МОН України,
вул. Науки, 1, Чернівці, 58029, Україна;
e–mail: anatych@gmail.com
[2]Чернівецький національний університет
ім. Юрія Федьковича, вул. Коцюбинського 2,
Чернівці, 58012, Україна

ВПЛИВ МІНІАТЮРИЗАЦІЇ НА ЕФЕКТИВНІСТЬ ТЕРМОЕЛЕКТРИЧНОГО ТЕПЛОВОГО НАСОСА КОСМІЧНОГО ПРИЗНАЧЕННЯ

У роботі наводяться результати розрахунків впливу мініатюризації на граничні можливості термоелектричного теплового насоса рідина-рідина, зокрема для його використання у якості високоефективного нагрівника для приладу очистки води космічного призначення. Бібл. 10, рис. 5.
Ключові слова: термоелектричний тепловий насос, ефективність, дистилятор.

Анатычук Л.И., *акад. НАН Украины*[1,2]
Прибыла А.В., *канд. физ.-мат. наук*[1,2]

[1]Институт термоэлектричества НАН и МОН Украины,
ул. Науки, 1, Черновцы, 58029, Украина, *e–mail: anatych@gmail.com;*
[2]Черновицкий национальный университет
им. Юрия Федьковича, ул. Коцюбинского, 2,
Черновцы, 58012, Украина

ВЛИЯНИЕ МИНИАТЮРИЗАЦИИ НА ЭФФЕКТИВНОСТЬ ТЕРМОЭЛЕКТРИЧЕСКИХ ТЕПЛОВЫХ НАСОСОВ КОСМИЧЕСКОГО НАЗНАЧЕНИЯ

В *работе приводятся результаты расчетов влияния миниатюризации на предельные возможности термоэлектрического теплового насоса жидкость-жидкость, в частности для его использования в качестве высокоэффективного отопителя для прибора очистки воды космического назначения. Библ. 10, рис. 5.*
Ключевые слова: *термоэлектрический тепловой насос, эффективность, дистиллятор.*

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