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COMPUTER DESIGN OF THERMOELECTRIC OTEC

Method for computer design of thermoelectric OTEC (Ocean thermal energy conversion) plants is developed which allows optimization of electric power plant in many parameters and take into account the losses in OTEC operation. The efficiency of the method is demonstrated by an example of a concrete physical model of OTEC. Optimal parameters of TEG are defined, whereby its maximum generated power is achieved with regard to losses for in-house needs. Economic parameters of a 100 kW thermoelectric OTEC are calculated: relative capital investments are 25 \$/W, the cost of electric energy is 0.15 – 0.3 \$/kW·h, which confirms the economic viability of such projects.

Key words: thermoelectric generator, Ocean thermal energy, Ocean power station

Introduction

General characterization of the problem. The relevance of creating efficient renewable energy sources is generally known. Among them, increasing interest is aroused by the sources using low-grade heat due to its huge amount, for instance industrial waste heat, the exhaust heat from thermal electric power plants, etc. However, the major of these low-grade heat sources is global ocean where the difference in temperatures between its surface and the depth of 1-2 km reaches 20-25°. Creation of ocean thermal into electric energy converters (OTEC) can have a significant impact on solving the problem of power supply to mankind. In fact, these are solar energy converters, but OTEC can work day and night, offering a number of principal advantages. Such attractive opportunities stimulate development of electric power pants working on this heat.

For realization of this opportunity, developments of OTEC with the use of heat engines are underway [1]. Their efficiency can reach 3%. This relatively low efficiency cannot become the basic obstacle on the way to using OTEC, since the advisability of their creation primarily depends on its economic factors, rather than efficiency. In [2,3] it was established that OTEC based on heat engines can be competitive to solar electric power plants at powers higher than 10 MW. At lower powers, relative capital investments increase. Therefore, low-power OTEC are economically unviable. Creation of high-power OTEC is also problematic because of considerable investments with insufficiently defined risks.

Thermoelectric OTEC are free from this disadvantage, since they are almost insensitive to scale factor, which accounts for intense interest in them.

In 1980-1982, in Japan, company Kawasaki Heavy Industries was the first to manufacture a thermoelectric generator (TEG) for OTEC [4]. It experimentally confirmed the possibility of electric energy production at low temperature differences. The generator employed conventional cooling modules. No optimization results for modules and TEG were reported.

Ref. [5] describes the results of theoretical and experimental investigations of TEG design for low temperature differences, including ocean temperature difference. Optimization was performed with a view to achieve maximum efficiency. Calculations were made without regard to loses on pumping of cold water from the bottom of the ocean and hot water from the ocean surface to TEG.

Ref. [1, 6, 7] give the results of optimization of thermoelectric OTEC for the achievement of maximum power. Calculations of relative capital investments for 400 MW OTEC are made. It is established that they are close to capital investments in OTEC with a heat engine. The latter is a convincing proof of the advisability of using thermoelectric generators in OTEC at low powers. In so doing, the losses of electric energy on pumping of cold water from the bottom of the ocean are taken into account, and the losses due to warm water pumping from the ocean surface are disregarded.

A 100 kW thermoelectric OTEC is calculated in [8]. The cost of electric energy 0.15 \$/kW·h is obtained, which proves the viability of using low-power thermoelectric OTEC.

However, designing of such OTEC is rather cumbersome due to the necessity of optimization of many structural components. The above results were obtained by direct mathematical calculations that become increasingly productive with increasing number of variables for OTEC optimization. Therefore, for further progress in this area it is important to pass to computer simulation of thermoelectric OTEC. Below are given the main approaches of this simulation and the results of computer design for a concrete physical model of OTEC.

1. Physical model of thermoelectric OTEC

A general model of thermoelectric OTEC of power W_{OTEC} is given in Fig. 1, *a*. It consists of a thermoelectric generator (TEG) 1 and heat pipes for the hot 2, mixed 3 and cold water 4, with their lengths L_1, L_2 and L_3, and lowering heights H_1, H_2 and H_3, respectively. The outer surface of the cold water pipeline is thermally insulated. Water in the pipes is pumped by two pumps: 5 – of power W_{hp} for the hot water, 6 – of power W_{cp} for the cold water. The model of TEG is shown in Fig. 1, *b*. It comprises rows of thermoelectric modules 7, arranged between the hot 8 and cold 9 heat exchangers in the form of pipes of rectangular shape from the outside. Water flowing in adjacent heat exchangers has equal temperature. The hot and cold water is fed to heat exchangers through inlet collectors 10 and 11 and flows out of heat exchangers through collectors 12 and 13.

The model of TEG can be conveniently described by identical block-sections 14. They are composed of one module located between the hot and cold heat exchangers. Each TEG row comprises *N* block-sections. The TEG consists of *n* rows.

Fig. 1, *c* shows a model of TEG block-section. Here, a thermoelectric module consists of n- and p-type legs 15 of height *h*, length *a* and width *b*, connecting plates 16 and electrically isolating ceramic plates 17. The internal shape of heat exchangers is determined by effective diameter *d*. The module generates power Wm_i which is equal to the difference between thermal power that came to the module from the hot heat carrier Qh_i and thermal power given to the cold heat carrier Qc_i.

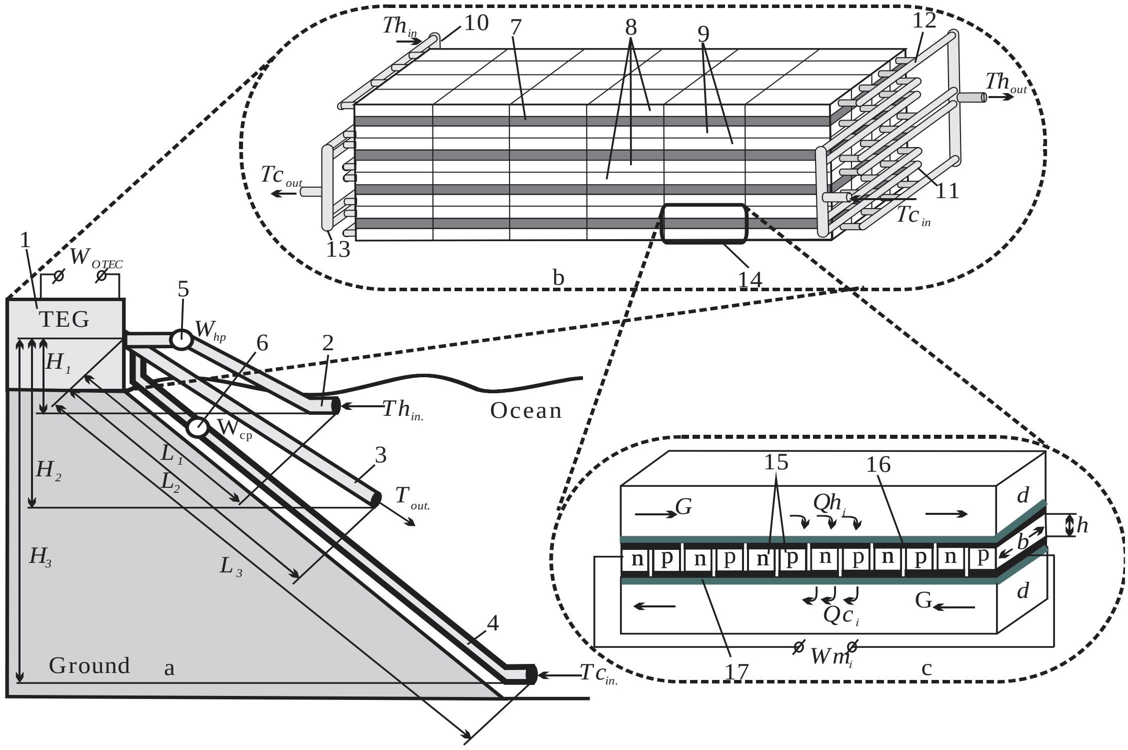


Fig. 1. Physical model of thermoelectric OTEC: A – thermoelectric OTEC; B - TEG; C – block-section of TEG.

The power of TEG is affected by the following parameters:

- pipeline lengths L_1, L_2 and L_3;
- pipeline lowering heights H_1, H_2 *and* H_3;
- pipeline diameter *D;*
- inlet hot water temperature Th_{in};
- inlet cold water temperature Tc_{in};
- number of TEG rows *n*;
- number of block-sections in a row *N*;
- block-section length *l*;
- local resistance coefficients ξ at pipe inlets and outlets and on collectors;
- pumps efficiency η;
- heat carrier flow rate in one row *G;*
- heat exchanger channel diameter *d*;
- thickness of heat exchanger wall *x*;
- height of thermoelectric module legs *h*;
- width of thermoelectric module legs *b*;
- length of thermoelectric module legs *a*;
- distance between the legs l_1;
- connecting plates thickness h_1;
- ceramic plate thickness h_2.

Parameters to be optimized in this model include *G*, *d*, *h*, *b*, *N*. All other parameters during optimization are assigned as input data. Having optimal parameter values, one can determine the number of TEG rows *n* which are necessary for construction of electric power plant of assigned power.

2. Mathematical description of the model

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

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

and electrical charge

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

where

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

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

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

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

Taking into consideration (3) - (5), one can get

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

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

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

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

These nonlinear second-order differential equations in partial derivatives (7) and (8) are determined by the distributions of temperature T and potential U in thermoelements.

Equation describing the process of heat transfer in heat exchanger walls in the steady-case case is written as follows

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

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

The processes of heat and mass exchange of heat carriers in heat exchanger channels in the steady-case are written by equations [9]

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

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

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

where p is pressure, ρ is heat carrier density, A is cross-section of the pipe, $\vec{F}$ 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 pipe wall perimeter, Q_2 is heat released due to viscous friction [W/m] (from heat exchanger unit length), Q_{wall} is heat flux coming from heat carrier to pipe walls [W/m].

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

where h is heat exchange coefficient found from equation

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

The Nusselt number is found with the use of the Gnielinski equation (3000 $<Re< 6\cdot10^6$, 0.5 $<$Pr$<$ 2000)

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

where the Prandtl number $Pr = \frac{C_p\mu}{k_2}$, μ is dynamic viscosity, $Re = \frac{\rho v d}{\mu}$ is the Reynolds number.

To determine the Darcy coefficient f_D, we use the Churchill equation for the entire spectrum of the Reynolds numbers and all values e/d (e is wall surface roughness)

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

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

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

The objective function during optimization of the block-section is the net power (module power with regard to losses for pumping of heat carriers in heat exchangers) which is calculated by the formula:

$$W_i = Wm_i - Wn_{hot} - Wn_{cold}, \tag{17}$$

where

$$Wm_i = \left(\frac{\Delta U}{R+r}\right)^2 R, \tag{18}$$

$$Wn_{hot} = \frac{G\cdot\Delta p_h}{\eta}, \tag{19}$$

$$Wn_{cold} = \frac{G\cdot\Delta p_c}{\eta}. \tag{20}$$

Here Wn_{hot}, Wn_{cold} are powers spent on pumping of the hot and cold heat carriers, respectively, ΔU is potential difference between extreme interconnects of block-sections, R is electrical load resistance, r is thermoelement resistance, Δp_h, Δp_c is hydraulic resistance of the hot and cold heat exchangers (pressure drop), η is pump efficiency.

Module efficiency can be calculated by the formula

$$\eta_m = \frac{Wm_i}{Qe_i}. \tag{21}$$

The power of all modules in a row and the net power of one TEG row are determined by the formulae

$$Wm = \sum_{i=1}^{N} Wm_i, \tag{22}$$

$$W = \sum_{i=1}^{N} W_i \,. \tag{23}$$

After optimization of block-section geometrical dimensions, determination of power and efficiency of optimal block-section, one can find the number of TEG rows *n* which is necessary for building of electric power plant of power W_{OTEC}. For this purpose, we use the equation

$$W_{OTEC} = n \cdot N \cdot Wm_i - W_{hp}(n) - W_{cp}(n)\,. \tag{24}$$

The efficiency of electric power plant is found by the formula

$$\eta_{OTEC} = \frac{W_{OTEC}}{n \cdot N \cdot Qh_i}\,. \tag{25}$$

The power consumed by the pumps can be determined by the formulae

$$W_{hp}(n) = W_h(n) + \frac{1}{2} W_{exit}(n) + n \cdot N \cdot Wn_{hot}\,, \tag{26}$$

$$W_{cp}(n) = W_c(n) + \frac{1}{2} W_{exit}(n) + n \cdot N \cdot Wn_{cold}\,. \tag{27}$$

Power that must be spent on pumping of water by pipelines in the general case is calculated by the formula [10]:

$$W_{c,h,exit}(n) = \frac{1}{\eta}\left[(\rho_1 - \rho_2) G_v(n) g H + \left(\lambda \frac{L}{D} + \xi\right) \cdot \frac{\rho G_v{}^3(n)}{2S^2}\right], \tag{28}$$

where G_v is water consumption in all TEG rows (for mixed water it is twice as large), ρ is the average water density, ρ_1, ρ_2 are the densities of cold and hot water, respectively, *g* is free fall acceleration, H is the required height of water lift, $\lambda = \frac{0.316}{\sqrt[4]{\mathrm{Re}}}$ is pipe friction number, *D* is pipe diameter, *L* is pipe length, ξ is local resistance coefficient (takes into account local resistances at pipe inlets and outlets and on collectors 10 - 13), *S* is pipe cross-sectional area, $\mathrm{Re} = \frac{4\rho G_v}{\mu \pi D}$ is the Reynolds number.

3. Computer model of TEG block-section

Design of thermoelectric power plant is composed of two phases:

1) optimization of one row of TEG block-sections;
2) calculation of thermoelectric power plant.

For solving of the first phase of the problem it is reasonable to use finite-element method that can be realized in Comsol Multiphysics program environment. This method allows solving differential equations (6)-(12) for different values of TEG parameters (*G*, *d*, *h*, *b*, *N*). Of these parameters one can select such whereby the function of TEG input power (17) will reach maximum.

Let us consider construction of a computer model of TEG block-section by example of a TEG with heat exchangers of round shape from inside. This shape of heat exchangers is due to the fact that their hydraulic resistance is 15% lower than that of rectangular heat exchangers. This is determined by means of computer simulation in Comsol of two pipes of identical length and cross section. The results are given in Fig. 2.

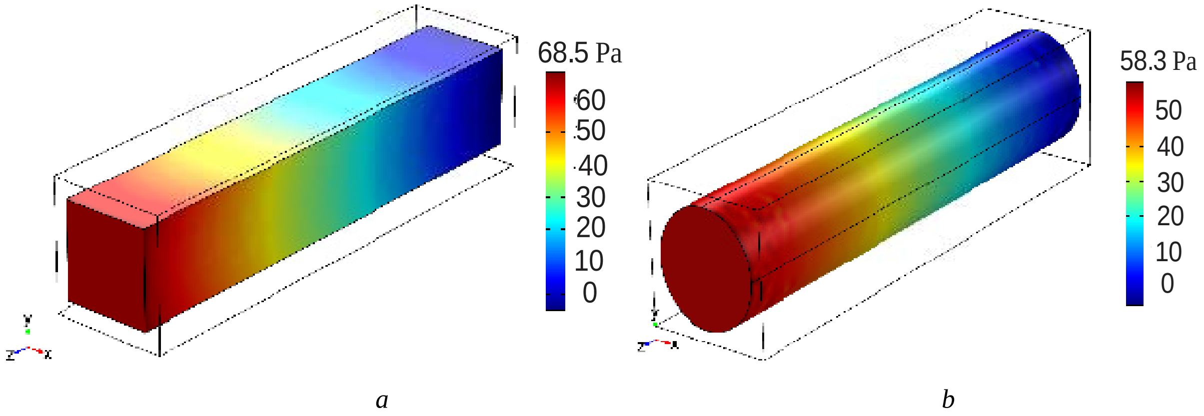


Fig. 2. Pressure difference in a pipe of rectangular A and round B shape from inside.

Fig. 3 shows finite element geometry and mesh built in Comsol for simulation of thermoelectric block-section. The input data are as follows:

- block-section length l = 60 mm;
- pump efficiency η = 80 %;
- heat exchanger wall thickness x = 2 mm;
- length of thermoelectric module legs a = 4 mm;
- distance between the legs l_1 = 1 mm;
- thickness of connecting plates h_1 = 0.5 mm;
- thickness of ceramic plates h_2 = 0.5 mm.

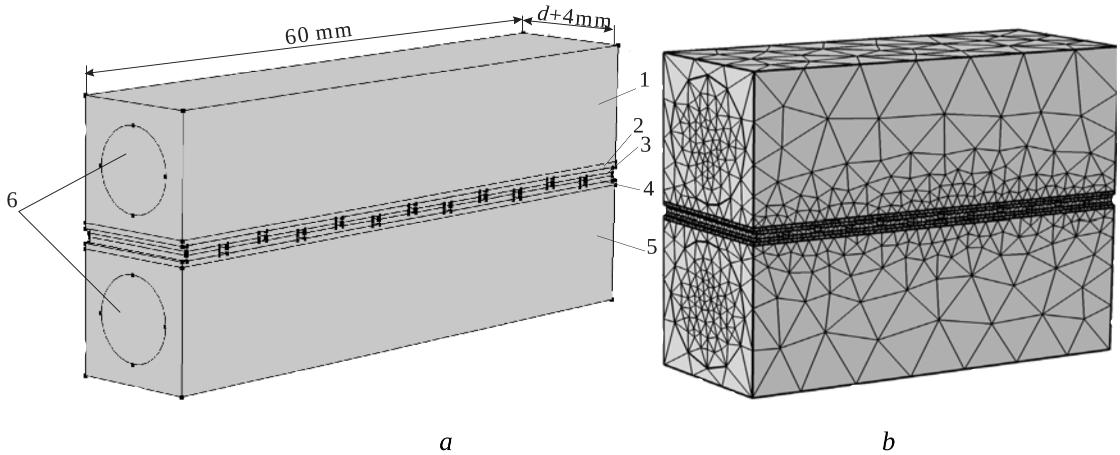


Fig.3. Geometry A and mesh B built in Comsol for TEG block-section:
1 – hot heat exchanger; 2 – ceramic plate; 3 –thermoelectric material; 4 – connecting plate; 5 – cold heat exchanger; 6 – channels of diameter d with a heat carrier.

Material of heat exchangers is aluminum. Material of connections is copper. The model takes into account contact and connecting resistance of thermoelements. Contact resistance is equal to 10^{-5} Ohm·cm^2. To take into account the temperature dependence of α, σ, κ of *n*-type and *p*-type thermoelectric material *Bi-Te*, they were assigned as polynomials obtained from the experimental data. The net power was calculated in the mode of matched load ($R = r$).

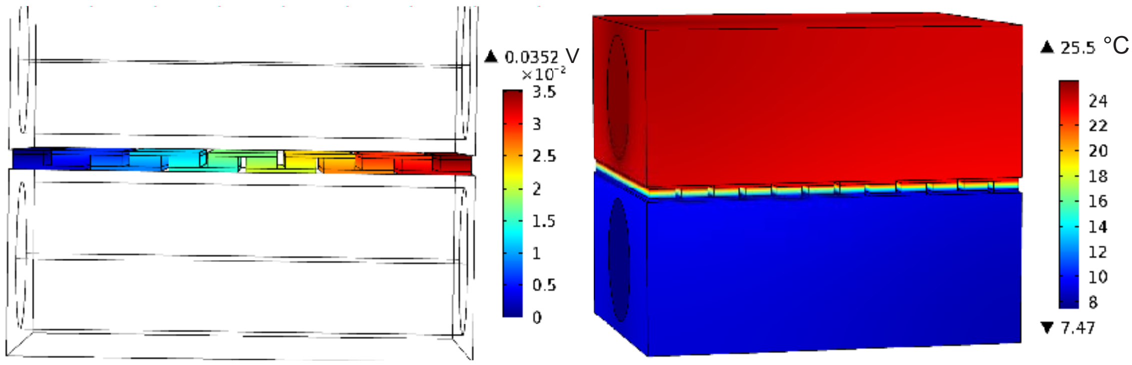
Simulation parameters variation range:

- channel diameter d = 2..140 mm;
- leg height h = 0.04..10 mm;
- leg width b = 12…200 mm;

- heat carriers flow rate G = 1..2000 ml/s;
- the number of block-sections N = 1..400 pcs.Net simulation conditions:
- Hot water inlet temperature Th = 29 ˚C.
- Cold water inlet temperature Tc = 4 ˚C.
- Heat carrier flow rate at channel inlets G.
- Pressure at channel outlets p = 0 Pa.
- Electrical potential on the first interconnect U = 0 V.
- The rest of the boundaries are imposed with thermal and electrical insulation conditions.

4. Computer simulation results

Fig. 4 shows typical distributions of temperature and electrical potential in the block-section.



a) *b)*

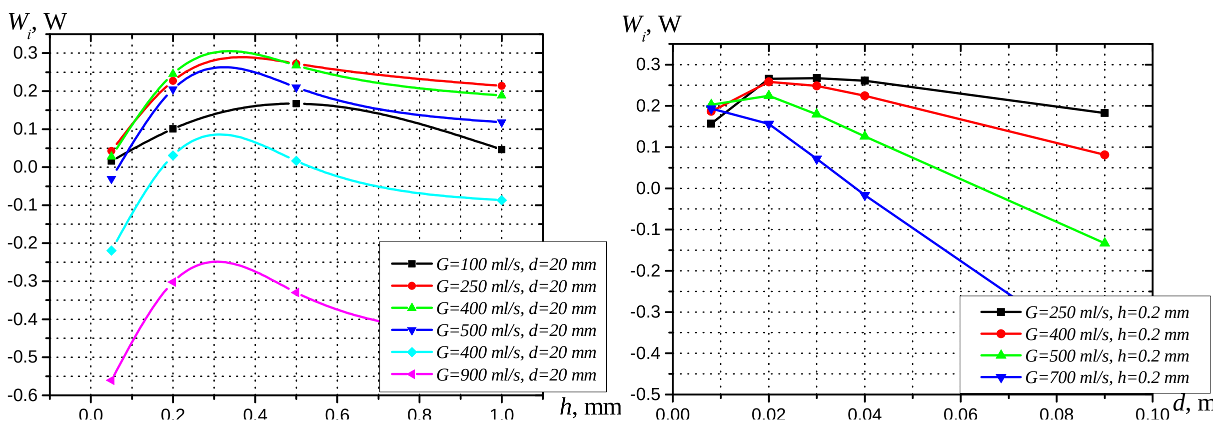
Fig.4. Typical distribution of temperatures a) and electrical potential b) in the block-section.

Figs. 5, 6 show a series of dependences of net power (17) on legs height h for optimal d, heat carrier flow rate G for optimal h and channel diameter d for optimal h (b = 12 mm).

Fig. 7 presents the results of optimization for the number of block-sections in one row.

The following optima are seen from the plots:

- leg height h_{opt}^{*} = 0.35 mm;
- heat carrier flow rate G_{opt} = 300 ml/s;
- channel diameter d_{opt} = 20 mm;
- number of block-sections in one row N = 175 pcs.



a) *b)*

Fig.5. a) Dependence of net power (17) on legs height h for optimal d, b) Dependence of net power on heat carrier flow rate G for optimal h.

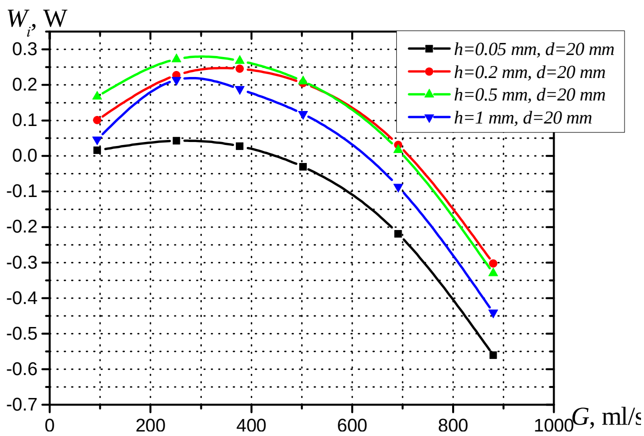


Fig. 6. Dependence of net power on channel diameter d for optimal h.

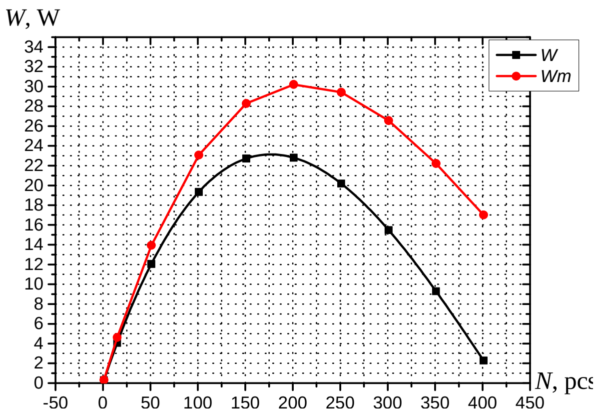


Fig. 7. Dependence of all modules power (22) and net power (23) on the number of block-sections in one row.

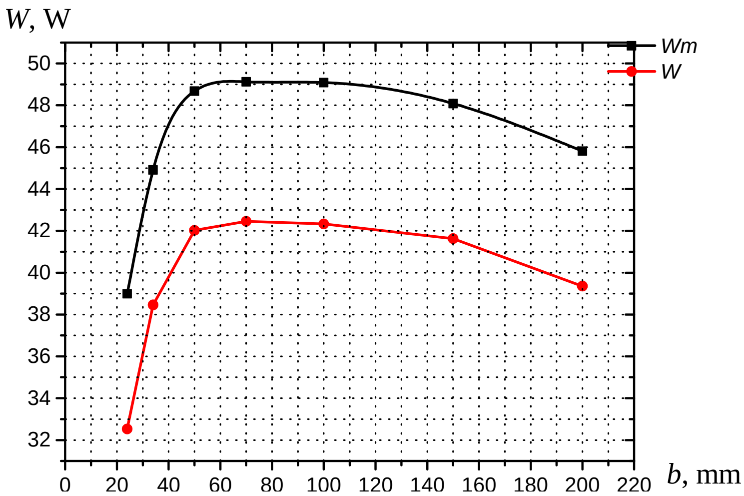


Fig.8. Dependence of all modules power (22) and initial power (23) on leg width.

These results were obtained for a thermoelement with leg width b = 12 mm. In order to increase power due to increased area of contact between heat exchanger and module and reduced effect of contact resistance, one should increase the width and height of leg proportionally in such a way that thermoelectric material completely filled the space between heat exchangers. With further increase of leg width, one should accordingly increase the width of heat exchanger. Simulation showed that optimum is achieved with leg width b_{opt} = 50 mm and the respective height h_{opt} = 1.5 mm (Fig.8).

As a result of calculation of a model with final optimal parameters h_{opt} = 1.5 mm, b_{opt} = 50 mm, G_{opt}=300 ml/s, d_{opt} = 20 mm, N = 175 pcs, the following energy characteristics of one row of block-sections were obtained:

- net power W_{max} = 42 W;
- modules power Wm_{max} = 49 W;
- voltage U_N = 2.8 V;
- current I = 17.5 A;
- generator efficiency η_{TEG} = 0.76 %;
- modules efficiency η_m = 0.88 %.

5. Calculation of thermoelectric power plant

The second phase of electric power plant design consists of calculation of power losses on pumping of water pipelines and calculation of the number of TEG rows that are necessary to achieve the required power of electric power plant W_{OTEC}. For this purpose, one should solve Eq. (24) which can be easily done in Mathcad program.

Consider, as an example, an electric power plant of power W_{OTEC} = 100 kW. One row of modules with regard to power losses on heat carrier pumping generates 42 W of electric energy with water flow rate 2×300 ml/s. It is necessary to use n rows of TEG. Each pump must provide for flow rate $Gv = n \cdot 300 \cdot 10^{-6}$ m^3/s.

Input data for calculations:

- pipeline lengths L_1 = 2000 m, L_2 = 200 m, L_3 = 20 m;
- pipeline lowering heights H_1 = 1000 m, H_2 = 100 m, H_3 = 10 m;
- local resistance coefficients at pipe inlets $\xi_{in} = 0.5$, pipe outlets $\xi_{out} = 1$, on the bends $\xi_b = 0.7$, on inlet collectors $\xi_{\text{c.in}} = 1.2$, on outlet collectors $\xi_{\text{c.out}} = 0.8$;
- pipeline diameter D = 2…10 m.

Fig. 9 shows the results of calculation of the number of rows and power losses on water pumping in the pipelines for electric power plant of power W_{OTEC} = 100 kW. From the plot it is apparent that for diameters smaller than D = 3 m, power losses on water pumping and the number TEG rows increase considerably. The use of pipelines with large diameters is complicated due to increase of mechanical loads on the pipe and its cost.

Making calculations for D = 3 m we obtain the necessary number of module rows n = 3313; powers that should be spent pumping W_c = 36.31 kW, W_h = 1 kW and W_{exit} = 1.94 kW; power that generated by thermoelectric modules– 162.3 kW; power spent on pumping of heat carriers in TEG channels – 23.1 kW; electric power plant efficiency – 0.54%.

It is readily calculated that with the cost of thermoelectric module 3$ (under mass production conditions), the cost of heat exchanger material 2.5 $/kg, the cost of pipelines 60 $/m, pipes 0.45 $/W, inverter 0.1 $/W and other expenses 10%, the relative capital investments to such plant will be up to 25 $/W. For comparison, the same relative capitals investments are made in heat engines of power 6 MW [2]. The cost of electric energy in so doing is 0.15 – 0.3 $/kW·h. Thus, thermoelectric OTEC offer advantage over heat engines in the range of low powers, since they are less sensitive to scale factor.

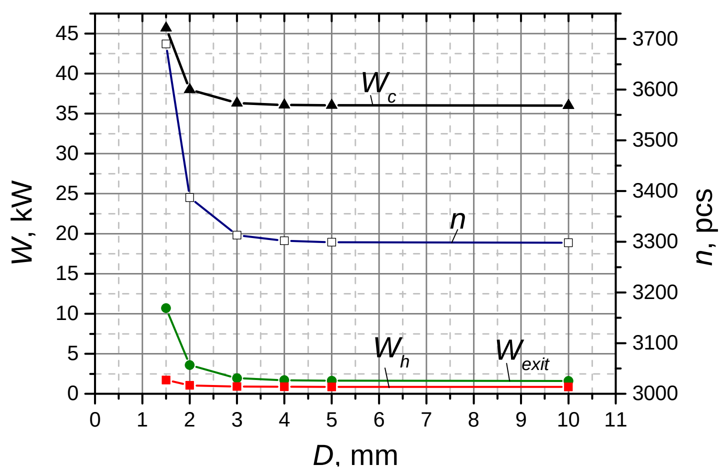


Fig. 9. Dependence of the number of TEG rows and power losses on water pumping on pipeline diameter.

Moreover, the specific cost of solar power plants with all their structural components varies from 3$/W to 6 $/W [11]. Considering that capacity use factor within 24 hours in solar electric power plants is 20% [12], and in OTEC – 100%, one can speak about the competitive ability of these technologies.

Conclusions

1. Design method for thermoelectric OTEC has been developed.
2. Optimal parameters have been determined for a row of TEG block-sections, whereby maximum generated power W_{max} = 42 W is achieved with regard to losses for in-house needs: leg height h_{opt} = 1.5 mm, leg width b_{opt} = 50 mm, heat carrier flow rate G_{opt} = 300 ml/s, heat exchanger channel diameter d_{opt} = 20 mm, number of block-sections in one TEG row N = 175 pcs.
3. A 100 kW electric power plant has been calculated, the relative capital investments in the plant have been determined as 25 $/W, and the cost of electric energy as 0.15 – 0.3 $/kW·hour.

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