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Research Article

Development Of Thermoelectric Refrigerator For Transport Facilities

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Abstract

Objective: To conduct a computer investigation of thermoelectric cooler (TEC) to be used on transport facilities.

Methods: The method of modeling a thermoelectric refrigerator for transport facilities is described. The development of calculation algorithms and optimization of thermoelectric coolers on the basis of theory of optimal control are represented. The work proposes a new approach to modeling a thermoelectric chamber with aluminum inserts in a non-stationary cooling mode using the quasi-stationary method.

Results: The quasi-stationary approach described in the article allows for simulation of the temperature in the chamber. The main modeling results indicate the possibility of ensuring the required temperature regime for cooling the chamber objects with typically powered by direct current sources (12V-18V) with chamber volumes ranging from 40 to 100L.

Conclusion: The above modeling and optimization methods provide a theoretical basis for the design and application of thermoelectric refrigerators in transport facilities.

Keywords: thermoelectric refrigerator, theory of optimal control, cooling efficiency

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1 INTRODUCTION

Thermoelectric refrigeration technology, leveraging the Peltier effect, is emerging as a promising alternative to traditional mechanical refrigeration systems in transport applications. Its potential lies in its compact design, environmental friendliness, and flexibility, making it particularly suitable for small, insulated vehicles and specialized transport needs.

Thermoelectric transport refrigerators (TTERs) are typically powered by direct current (DC) sources, such as 12V-18V vehicle mains or batteries, and employ two-level temperature control for efficient operation. Early implementations were custom-built, often adapting household thermoelectric refrigerators for transport use, with chamber volumes ranging from 40 to 100L. These systems are particularly valuable in settings where external

power is unavailable, such as yachts, railway cars, tourist buses, mobile homes, and food trucks. Reducing the power requirements of TTERs is crucial, as continuous operation from a battery can quickly lead to battery depletion, making energy efficiency a central design focus[1].

Thermoelectric cooling is especially well-suited for short-distance, low-capacity transport-such as insulated vans used for urban home delivery. Traditional mechanical systems are still preferred for long-haul, high-capacity refrigeration due to the higher cooling power required. In small insulated vans, thermoelectric systems can effectively manage multiple temperature zones, keeping cargo temperature fluctuations within 1.5°C during typical 2.5h delivery cycles. This is a significant improvement over non-refrigerated vans, where temperature fluctuations can exceed 5.8°C. However, the cooling performance of current thermoelectric modules is limited by material properties and a low coefficient of merit, restricting their use to applications with modest temperature differentials and cooling loads[2].

Electric and hybrid transport refrigeration units are gaining traction for both short- and long-haul applications, offering significant emissions reductions, operational savings, and quieter operation. However, they require substantial infrastructure investment and are currently more expensive upfront than traditional systems. Thermoelectric refrigeration, while not suitable for high-power needs, is uniquely positioned for small, urban delivery vehicles and specialized applications where its advantages can be fully realized[3-5].

Thermoelectric refrigeration offers a clean, reliable, and maintenance-friendly solution for temperature-sensitive transport, particularly in urban and short-haul contexts. While not yet a replacement for mechanical systems in all scenarios, ongoing advancements in materials and system integration are likely to expand its role in the transport refrigeration landscape, especially as sustainability and operational efficiency become increasingly important[3-5].

One of the components of thermoelectric cooling systems is radiators, which are used to remove heat from the chamber and into the surrounding environment. The design features of radiators are described in detail in the works[6-12]. The main integral characteristic of the radiator is the thermal resistance, which is used in this work to model different thermoelements[13] fore the thermoelectric cooling system.

The purpose of this work is computer investigation of thermoelectric cooler (TEC) to be used on transport facilities.

2 COOLER CALCULATION MODEL

Figure 1 presents a thermal model for the calculation of such cooler.

As a source of cold in cooling chamber 1 use is made of thermoelectric battery 2 consisting of thermoelectric modules electrically connected in series or in parallel. The cooling chamber is thermally isolated with a layer of insulation material 3. Heat transfer from the heat-absorbing junctions of the battery thermoelements to the chamber is realized by means of radiator 4 due to forced air convection in the chamber provided by the fan 5. The heat from the heat-releasing thermoelectric battery junctions is removed to the environment by means of the radiator 6 cooled by the fan 7. Thermal load on the thermoelectric battery Q_0 is a sum of active heat releases in chamber Q_{act} and the quantity of heat Q_1 leaking through the chamber insulation. To reduce heat flow Q_2 that creates additional thermal load on the thermoelectric battery, the space between the free surface of cold radiator 4 and hot radiator 6 is filled with insulation material 8, and aluminum insert 9 is placed between the cold surface of thermoelectric battery and radiator base 4. Thermal contact between the cold radiator base and aluminum insert surfaces, thermoelectric battery cold surface and aluminum insert, the battery hot surface and radiator base is realized by means of contact heat conducting lubricant layer 10.

It should be born in mind that actually the model can consist of an arbitrary of thermoelectric batteries with heat exchangers. The thermobattery being designed can refer not to the whole cooling object , but only to its part. For example, in case of two modules with heat-exchangers the chamber volume is associated with heat inleak and active heat releases twice less.

2.1 Cooler Calculation Algorithm

In accordance with the cooler model in the development of calculation algorithm the following factors are to be taken into account:

Convection heat exchange of the surface of cold radiator 4 with the air in the chamber which leads to the loss in temperature drop - $\Delta T1=T1-T2$;

Loss in temperature drop on the cold radiator 4 - $\Delta T2=T2-T3$;

Loss in temperature drop on the heat conducting lubricant layer between the base of the radiator 4 and the aluminum insert 9 - $\Delta T3=T3-T4$;

Loss in temperature drop on the aluminum insert 9 - $\Delta T4=T4-T5$;

Loss in temperature drop on the heat conducting lubricant layer between the aluminum insert and cold surface of thermoelectric battery 2 - $\Delta T5=T5-T6$;

Loss in temperature drop on the heat conducting lubricant layer between the thermoelectric battery hot surface and the base of hot radiator 6 - $\Delta T6=T7-T8$;

Loss in temperature drop on the hot radiator 6 - $\Delta T7=T8-T9$;

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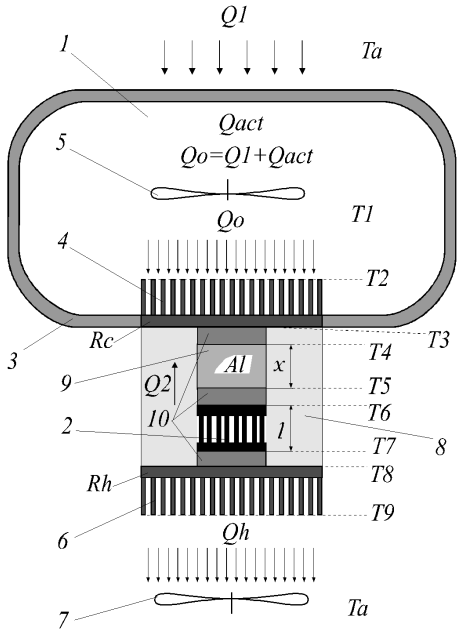


Figure 1. Air Heat Exchange Refrigerator Model. (1) refrigerating chamber; (2) thermoelectric cooler; (3) insulating material; (4) cooling chamber radiator; (5) fan; (6) hot radiator; (7) fan; (8) insulating material; (9) aluminium plate; (10) heat-conducting lubrication layer. The following notation of temperatures is accepted in Fig.1: Ta - is ambient temperature; T1 - is cooling temperature in the chamber; T2 - is temperature of radiator surface in the cooling chamber; T3 - is temperature at the boundary of radiator base in the cooling chamber and lubricant layer; T4 - is temperature at the boundary between the lubricant layer and the aluminum insert; T5 - is temperature at the boundary between the aluminum insert and the lubricant layer; T6 - is temperature of thermoelectric battery cold surface; T7 - is temperature of thermoelectric battery hot surface; T8 - is temperature at the boundary between the lubricant layer and the hot radiator base; T9 - is temperature of hot radiator surface.

Convection heat exchange of the surface of hot radiator 6 with the ambient air that results in the loss in temperature drop - $\Delta T8=T9-T_a$;

Additional thermal load on the thermoelectric battery through the insulation 8 between the free surface of the cold radiator 4 and hot radiator 6 - Q_2.

2.2 System of Heat Balance Equations

As the basis for computer design of a cooler use is made of a system of stationary heat balance equations corresponding to the model selected. This system is written as follows:

(1) Equation of convective heat exchange between the air in the cooling chamber and the surface of the cold radiator 4

$$Q_0 = \alpha_c S_{rc}(T_1 - T_2)(1)$$

where Q_0 is thermal load in the chamber determined as

$$Q_0 = Q_1 + Q_{act}(2)$$

Q_{act} is heat releases in the chamber, Q_1 is the heat leaking to the chamber through the layer of insulation material 3

$$Q_1 = K_1 \frac{S_1 + S_2}{2\,h_1}(T_a - T_1)(3)$$

K_1 is heat conductivity of insulation chamber material; S_1 is the area of external chamber surface; S_2 is the area of internal chamber surface; h_1 is chamber insulation thickness; α_c is coefficient of convective heat exchange; S_{rc} is the area of cold radiator 4 base.

(2) Heat balance equation for the cold radiator in the chamber

$$Q_0 = \frac{1}{R_c}(T_2 - T_3)(4)$$

where R_c is radiator thermal resistance in the chamber.

(3) Balance equation at the boundary between the base of cold radiator 4 and heat conducting lubricant layer

$$Q_0 + Q_2 = \frac{K_3 S_{TEC}}{h_2}(T_3 - T_4)(5)$$

where additional thermal load on the thermoelectric battery through the insulation 8 between the free surface of the cold radiator 4 and hot radiator 6 Q_2 is calculated as

$$Q_2 = K_2 \frac{S_2 - S_{TEC}}{L + h_Al + h_2 + h_3 + h_4}(T_3 - T_4)(6)$$

K_2 is thermal conductivity of the insulation between the free surface of the cold radiator and hot radiator; K_3 is thermal conductivity of the lubricant layer; h_2 is thickness of lubricant layer between the cold radiator and the aluminum insert; h_3 is thickness of lubricant layer between the aluminum insert and thermoelectric battery cold surface; h_4 is thickness of lubricant layer between the thermoelectric battery hot surface and the hot radiator base; h_Al is height of the aluminum insert; S_{TEC} is the area of thermoelectric modules; L is the height of thermoelectric module.

(4) Equation of balance at the boundary between the heat conducting lubricant layer and the aluminum insert

$$Q_0 + Q_2 = \frac{K_Al \cdot S_{TEC}}{\mathrm{h_Al}}(T_4 - T_5)(7)$$

where K_Al is thermal conductivity of the aluminum insert.

(5) Equation of balance at the boundary between the aluminum insert base and heat conducting lubricant layer

$$Q_0 + Q_2 = \frac{K_3 S_{TEC}}{h_3}(T_5 - T_6)(8)$$

Equation of balance at the boundary between the heat conducting lubricant layer and the heat-absorbing surface of thermoelectric modules

$$Q_0 + Q_2 = Q_c(\alpha_{n,p}, \sigma_{n,p}, \kappa_{n,p}, N, S_{TEC}, U, T_6, T_7)(9)$$

where Q_c is refrigerating capacity of thermoelectric battery which is a function of thermoelectric material parameters ($\alpha_{n,p}$, $\sigma_{n,p}$, $\kappa_{n,p}$) battery structural parameters (N,S_{TEC}), supply voltage U, the temperatures of its cold and hot surfaces (T_6,T_7).

(6) Equation of heat balance at the boundary between the heat releasing surface of thermoelectric modules and conducting lubricant layer

$$Q_h = \frac{K_3 S_{TEC}}{h_4}(T_7 - T_8)(10)$$

where Q_h is heating capacity of thermoelectric battery which is determined as

$$Q_h = Q_0 + Q_2 + W \quad (\alpha_{n,p}, \sigma_{n,p}, \kappa_{n,p}, N, S_{TEC}, U, T_6, T_7)(11)$$

W is power consumed by thermoelectric battery which is a function of thermoelectric material parameters ($\alpha_{n,p}$, $\sigma_{n,p}$, $\kappa_{n,p}$), battery structural parameters (N,S_{TEC}), supply voltage U, temperatures of its cold and hot surfaces (T_6,T_7).

(7) Equation of heat balance for the hot radiator

$$Q_h = \frac{1}{R_h}(T_8 - T_9)(12)$$

where R_h is hot radiator thermal resistance.

(8) Equation of convective heat exchange between the environment and the hot radiator surface

$$Q_h = \alpha_h S_{rh}(T_9 - T_a)(13)$$

where α_h is coefficient of convective heat exchange of the hot radiator; S_{rh} is the area of the hot radiator base.

The basic starting data for the cooler calculation include: Ambient temperature T_a, cooling temperature in the chamber T_1, heat releases in the chamber Q_{act}, the area of external chamber surface S_1, the thickness of chamber insulation h_1, thermal conductivity of chamber insulation material K_1.

These starting data completely determine thermal load in the chamber Q_0. In this case a system of heat balance equations (1)-(13) allows to determine unambiguously the unknown temperatures T_2-T_9 and power consumed by thermoelectric battery W which provide the assigned cooling temperature in chamber T_1. In so doing, it is reasonable to require that structural parameters and supply current of thermoelectric battery be such as to provide its maximum coefficient of performance $\varepsilon=Q_0/W$. It will provide chamber cooling to temperature T1 with the minimum expenditure of electric energy. For this purpose the cooler calculation algorithm should be supplemented with the algorithm of calculating optimal parameters of N-stage thermoelectric battery under conditions of maximum coefficient of performance (N is the number of stages).

Calculation of optimal characteristics of thermoelectric cooling battery in the Institute of Thermoelectricity is made on the basis of methods of mathematical optimal control theory presented in detail in Ref[14-18]. We shall cite the basic calculation relationships.

3 CALCULATION OF OPTIMAL THERMOELECTRIC BATTERY CHARACTERISTICS UNDER CONDITIONS OF MAXIMUM COEFFICIENT OF PERFORMANCE

In the development of calculation algorithms and optimization of cascade coolers the basic difficulties arise due to the fact that material properties are essentially temperature dependent. As a result of this, in going to a real thermoelement model a system of initial equations of stationary thermal conductivity in the legs becomes non-linear and can be solved only numerically. Thus, known analytical solutions underlying the so-called simple theory of thermoelectric cooling[14] cease to be satisfied. Nevertheless, practically all existing algorithms of optimization of cascade coolers are based on the use of simple theory balance relationships with the substitution of physical parameters values related to the average cascade temperature[14]. As is shown, this method, giving satisfactory results for one-stage applications, in the case of multi-stage systems gives the unacceptably big errors.

Besides, as a rule, known methods of thermoelectric cooling calculation approximately take into account the irreversible thermal and electric loss in thermoelement related to the availability of contact resistances and external heat inflows, the loss in temperature drop on the inter-cascade ceramics.

The possibilities of computing mathematics today allow to give up all these assumptions. The productive idea consists in replacing approximate balance equations by the differential equations of non-equilibrium thermodynamics with temperature dependent kinetic coefficients and using them as the initial. The differential form of the initial model representation allows to consider cascade cooler in the context of the theory of optimal processes. A model for calculation of N-stage Peltier cooler is presented in Figure 2.

Mathematically, the problem is formulated as follows. The object of control is N-stage cooler whose thermoelements consist of n- and p-type legs, material properties change with coordinate x due to their dependence on temperature.

The initial data for optimization include refrigerating

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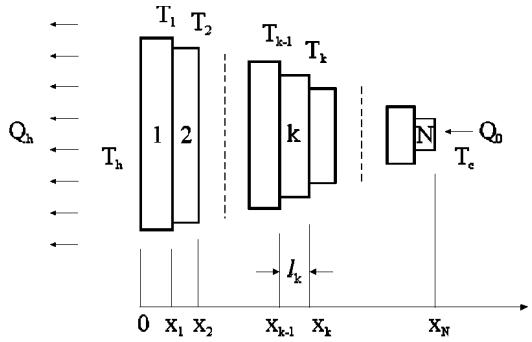


Figure 2. Model for Cascade Cooler Calculation.

capacity Q_0, cooling temperature T_c, temperature of heat-releasing surface T_h. Theoretical or experimental dependencies of Zeebeck coefficient α, electric conductivity σ, thermal conductivity κ on temperature should be also assigned.

Besides, the model takes into account the existence of contact resistances r_0, inter-stage temperature drops δT and heat exchange with the environment at the heat exchange coefficient α_T.

Of greatest practical interest is the problem of attaining the assigned refrigerating capacity Q_0 at the minimum expenditure of electric energy $W=Q_h-Q_0$, where Q_h is heat given to the environment which is equivalent to attaining the minimum of the relationship

$$g = \frac{Q_h}{Q_0} \quad (14)$$

If we introduce the variables

$$g_k = \frac{Q_1^K}{Q_0^K} = \frac{q_1^K}{q_0^K}, k = 1, \ldots N \quad (15)$$

where Q_1^k, Q_0^k are heat flows on the heat-releasing and heat-absorbing faces of the k-th cascade containing n_k thermocouples, and

$$q_1^k = \frac{Q_1^k}{n_k I}, q_0^k = \frac{Q_0^k}{n_k I} \quad (16)$$

are the appropriate specific (related to current I force) heat flows on the thermocouple junctions, and use the obvious conditions of heat balance between the stages $Q_1^{k+1}=Q_0^k$, $k=1,...N-1$, $Q_1^1=-Q_h$,$Q_0^N=-Q_0$, the function Equation (14) can be written in the form

$$g = \prod_{k=1}^{N} \frac{q_1^K}{q_0^K} \quad (17)$$

The distribution of temperature $T(x)$ and specific thermal flows $q(x)$ in the thermoelement legs is determined by the system of $4N$-differential equations of the form

$$\left.\begin{aligned} \frac{dT}{dx} &= -\frac{\alpha i_k}{\kappa} T - \frac{i_k}{\kappa} q \\ \frac{dq}{dx} &= \frac{\alpha^2 i_k}{\kappa} T + \frac{\alpha i_k}{\kappa} q + \frac{i_k}{\sigma} \end{aligned}\right\}_{n,p} \quad k = 1, \ldots N \quad (18)$$

Here Zeebeck coefficient, electric conductivity and thermal conductivity $\alpha_{n,p}=\alpha_{n,p}(T)$; $\sigma_{n,p}=\sigma_{n,p}(T)$; $\kappa_{n,p}=\kappa_{n,p}(T)$ are functions of temperature, $(i_k)_{n,p}$ is $2N$-dimensional vector-parameter of current density.

Boundary conditions for these equations are written as equalities of temperatures between the stages with regard for the inter-stage temperature loss δT and have the form

$$\begin{gathered} T_n(0^+) = T_P(0^+) = T_h, \quad T_n(x_N^-) = T_p(x_N^-) = T_c \\ T_n(x_k^+) = T_p(x_k^+), \quad T_n(x_k^-) = T_p(x_k^-), \\ T_n(x_k^+) = T_n(x_k^-) + \delta T, \end{gathered} \quad (19)$$

and the points x_k, $k=1,...N$ have no fixed location. Besides, the equations of thermal and electric stage mating are to be met, which are written, accordingly, in the form

$$q_1^{k+1} n_{k+1} = q_0^k n_k, K = 1, \quad N; (20)$$

$$(i_n s_n)^k = (i_p s_p)^k = I, \quad k = 1, \quad N, (21)$$

where s_n^k, s_p^k are thermoelement leg cross sections.

Conditions must be found when coefficient of performance attains its maximum under given cooling value. In terms of optimal control theory the problem is reduced to the investigation of functional minimum, equivalent to function Equation (17),

$$J = \sum_{k=1}^{N} 1n\, q_1^k - 1n q_0^k \quad (22)$$

Specific heat flows on the junctions are determined by the expressions

$$\begin{gathered} q_0^k = \sum_{n,p} \left[q(x_k^-) + i_k r_0 + \frac{\alpha T}{i_k} (Th - T(x_k^-)) \right], \\ q_1^k = \sum_{n,p,} \left(q(x_{k-1}^+) - i_k r_0 \right). \end{gathered} \quad (23)$$

They depend on the controlled parameters of current density $i_{n,p}^k$.

The problem is solved by the Pontryagin concept of maximum. According to this concept, for the minimum J the following conditions should be met:

(1) Current densities in the stages are to satisfy the equalities:

$$-\frac{\partial J}{\partial i_{n,p}^k} + \int_{x_{k-1}^+}^{x_k^-} \frac{\partial H^k(\varphi, T, q, i_n^k, i_p^k)}{\partial i_{n,p}^k} dx = 0 \quad (24)$$

where Hamiltonian function H^k has the form

$$H^k \equiv \sum_{n,p} \left(\varphi_1 f_1^k + \varphi_2 f_2^k \right), (25)$$

$(f_1^k, f_2^k)_{n,p}$ are right-hand parts of Equation (18), $\Phi=(\Phi_1,\Phi_2)_{n,p}$ is vector of pulses [1].

(2) Coordinates of the butt points x_k satisfy the equalities

$$H^k(x_k^-) = H^{k+1}(x_k^+), k = 1, \ldots N-1, (26)$$

and allow to find optimal distribution of the inter-stage cooler temperatures.

This problem is solved by a numerical method of successive approximations. The relationships Equations (14-

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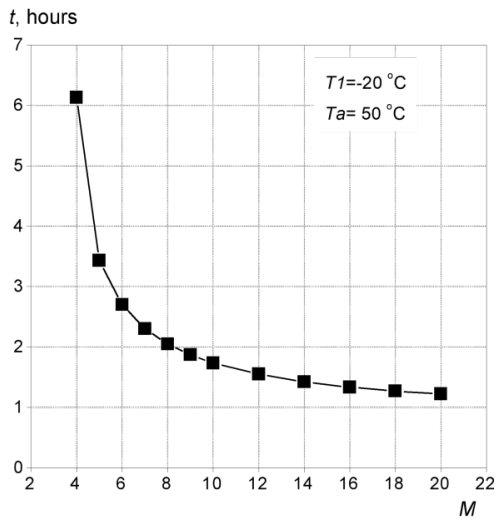


Figure 3. Time of Chamber Cooling *t* in Dependence on Module Quantity *M*. Empty chamber. Cooling temperature T1=-20°C. Ambient temperature Ta=50°C

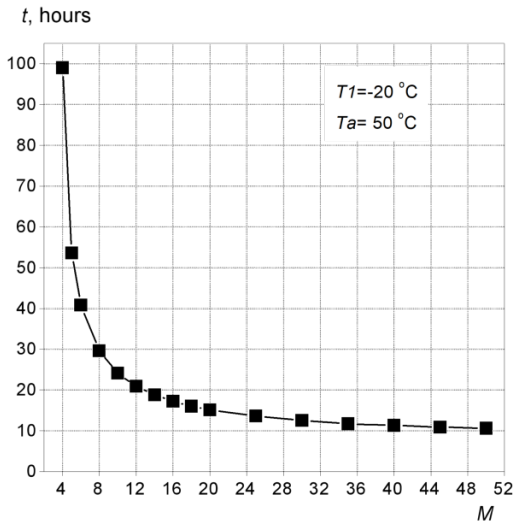


Figure 4. Time of Chamber Cooling *t* in Dependence on Module Quantity *M*. Chamber with 30L of liquid. Cooling temperature T1=-20°C. Ambient temperature Ta=50°C.

26) allow to determine optimal structure and characteristics of *N*-stage Peltier coolers[14]. Theory of optimal control and computer design method can indicate the ways for the optimization of construction and creation of structural elements most suitable for the cooler, as well as their assembly process. The math modeling for TECs of this design are presented in literature[14-22]. Let us consider consecutively the improvements that can be attained due to the introduction of new structural elements of the thermoelectric coolers.

4 RESULTS OF DESIGN

Whit this method was development the program for studies of time dependence of cooling which it takes for a cooler to reach the level of -20°C from the ambient temperature +50°C. The calculations were made for two cases. The results of calculations for the first case are shown in Figure 3. In the first case the cooler is empty. The heat capacity of a radiator inside the cooling chamber and the heat capacity of the chamber body are taken into account. The calculations were made for various number of modules *M* that can be used in a cooler. The results of calculations showed that with four modules, as before, the necessary temperature -20°C is attained approximately within 6 hours. An increase in the number of modules improves this situation, namely, with 5 modules, cooling -20°C is attained in 3.5h. The use of 6 modules allows to expect cooling in 2.7h. The use of 8 modules reduces cooling time to 2.1h. Further increase in the number of modules has already little effect on cooling time. The reason is that the time of cooling is restricted by the transitions from the module to the chamber. They have their own heat capacity, so their cooling requires time. Naturally, a large number of modules can be considered only theoretically, because there is no place for their arrangement and, besides, they will consume tremendous electric power.

Figure 4 shows a similar dependence for the case when a cooler is loaded with 30L of liquid at the temperature of +50°C. This case is the worst possible and difficult for cooling. It is seen that the use of four modules according to the previous calculation allows to cool 30L of liquid in 100h. The use of six modules will reduce this time to 40h. Thus, a loaded cooler will cool 30L of liquid over a sufficiently long period of time. The actual situation will be not so bad, because the cooler will be loaded with liquids at a lower temperature, for instance, +20°C, which reduces cooling time.

On the whole, a conclusion can be made that as a result of using four modules designed before and the electric power used in the cooler, it will take for an empty cooler six hours to be cooled from +50°C to -20°C, and for a loaded cooler approximately four days.

Computer design performed in the previous sections assumes the use of standard heat-exchangers and common ways of creating cooler constructions. However, computer design method can indicate the ways for the optimization of construction and creation of structural elements most suitable for the cooler, as well as their assembly process. The results of this design are presented in Figure 5. Dependence 1 in the figure shows the initial result of the best design.

Our model utilizes mechanical connection of planes between the radiator and the insert and between the module and the insert. Their thermal resistance is equal to 0,01875W/(cm K). In designing of the TECs[20-24] in these transient parts use can be made of soldering. For this purpose in designing a cooler we shall assume these thermal resistances negligibly small, as long as soldering will really make them negligibly small compared, for example, to thermal resistance of the aluminum insert.

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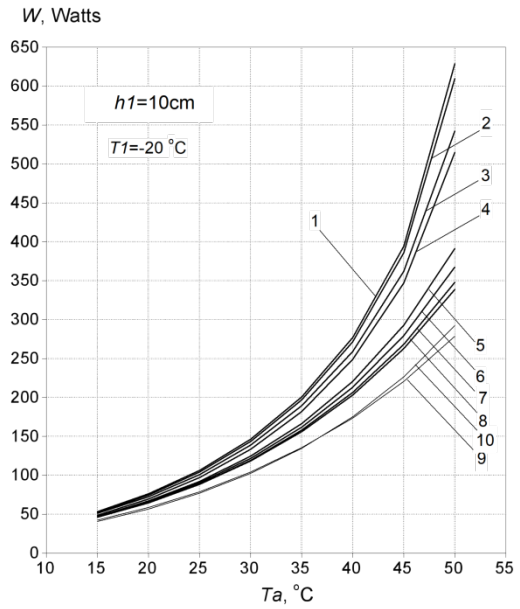


Figure 5. The Results of Thermoelectric Cooler Design for Various Models. Ambient temperature Ta dependences of TEC electric power W for cold temperature T1=-20C. Chamber insulation thickness h1=10 cm. 1 - initial result of the best design; 2 - model with soldering on cold side; 3 - model with soldering on cold and hot side; 4 - model with soldering on cold and hot side and insert of flexible multi-wire conductors; 5 - with special hot radiator and soldering; 6 - with the six modules and radiators; 7 - with the eight modules and radiators; 8 - with the nine modules and radiators; 9 - with special cold radiator; 10 - model of two stage thermoelectric modules.

5 CONCLUSIONS

Computer design performed in the previous sections assumes the use of standard heat-exchangers and common ways of creating cooler constructions. Let us consider consecutively the improvements that can be attained due to the introduction of new structural elements of the cooler.

5.1 Reduction of Thermal Resistance between the Cooler Cold Face and the Cold Radiator

This reduction can be achieved in two ways.

5.1.1 Soldering of Nickel-Coated Contacting Surfaces

The contacting surfaces are covered with nickel and soldered to each other. In this case thermal resistance on the soldered thermal contacts can be ignored. Dependence 2 shown in Fig.5 indicates that this technological practice allows to improve the situation and reduce electric power requirement from 630W to 610W, i.e. the economy in electric power is 20W.

5.1.2 Application of Flexible Intermediate Inserts

A limitation of method is that the construction of cooling device becomes rigid, and impact loads can result in the destruction of the module. Therefore, it is worthwhile to use soldering on one side only, and the cold radiator can be unified with the insert. In this case soldering will become unnecessary. Naturally, it will somewhat aggravate thermal resistance between the module and the insert. To avoid this, one can introduce an intermediate insert made of ductile material, for example, lead 1-2mm thick. Both the module and insert can be soldered to it. Besides, the insert itself can be made of flexible multi-wire conductors with their area equal to the area of the insert. Then the system will be also flexible and at the same time thermally conducting like the solid one. This insert is know-how and will be patented.

5.2 Soldering of the Hot Module Side

Still greater economy in the electric power requirement can be obtained when soldering the hot module side. The results of the module design for the case of soldering the cold and hot module sides are shown in Fig.5, dependence 3. We can see that in this case electric power requirement is further reduced by 95W. The total power requirement will be 515W. It should be borne in mind that the hot module part has the greatest dimension, and soldering of the hot module surface to the heat removing radiator will create mechanical stresses that reduce the reliability of cooler operation and sometimes can even cause module destruction. Therefore, the innovations utilized in the construction should exclude thermal and mechanical stresses.

5.3 Application of Special High-Efficiency Radiators

Standard radiators available are not effective enough, as long as thermal loads on the radiator are sufficiently great. This results in the increase of temperature difference between the environment and the module hot surface. The analysis of standard radiators shows that this temperature drop is about 18°C, causing high values of the cooler electric power. Therefore, a special radiator with improved heat removal efficiency should be specially made for this cooler. This radiator would reduce the value of temperature drop from 18°C to 12°C. The results of designing a cooler with improved special radiators are given in Fig.5, dependence 5. As we see, the improved radiators allow to reduce the electric power requirement to 390W. Special radiator design can be also the object of know-how.

5.4 Increase in the Number of Modules and Radiators

An important opportunity of further economy in electricity consumption is provided by improved heat removal from the radiator. Even on the improved radiator there is loss in temperature drop caused by temperature drop on the aluminum plate of radiator base due to arrangement of module in the center. Besides, power removed by the radiator (about 100W per radiator) proves to be very great which also increases temperature drop between the ambient air and the module hot surface. Hence, it follows that the number of radiators should be increased and the number of modules accordingly. In so doing, the power of heat release of each module will be reduced so that their total cooling capacity could provide cooling -20°C in the chamber.

Computer design was made for the cases of 6, 8 and

9 modules. The results of such design are expressed by dependencies 6,7,8 in Figure 5. Dependence 6 corresponds to six radiators and modules, 7 to 8 radiators and modules, dependence 8 to 9 radiators and modules. It is seen that the use of increased number of modules also reduces power requirement and it becomes less than 350W.

5.5 Improvement of Heat Removal from the Cold Radiator in the Chamber

This improvement also contributes greatly to the reduction of power requirement of the cooler with cooling capacity unchanged. The results of such design when temperature drop between the radiator base and module is reduced from 5 to 2 degrees are shown in Figure 5, dependence 9. This reduction is attained by forced air cooling of cold radiator in the chamber. It should be borne in mind that forced air cooling also leads to a more uniform temperature distribution in the chamber. Thus, power requirement of a cooler with regard for all design optimizations can be reduced more than twice. It will be used in the development of cooler design.

5.6 Optimization of the Number of Cooler Stages

In our previous design we needed a three-stage cooler. Now, when the loss in temperature drop is reduced, it is worthwhile to verify again the number of stages that must be in the cooler. These investigations were performed by means of computer design. The results obtained were as follows. With three stages, electric power requirement will be 280W. It is given by dependence 9 in Figure 5. With two stages, electric power requirement will be 290W. It is given by dependence 10 in Figure 5. It is obvious that for the sake of economy in 10W there is no sense in using more complicated and expensive three-stage coolers. It is better to restrict oneself to a two-stage cooler.

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Conflicts of Interest

No conflict of interest.

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Data Availability

No additional data are available.

Author Contribution

Cherkez RG conceived and designed the physical model, implemented the numerical method for calculating the physical model, performed the calculations and analyzed the calculation results. Vikhor LM conceived and designed the physical model, formulated the correlations for the description of the physical model, constructed the graphs and analyzed the results.

Abbreviation List

TTER, Thermoelectric transport refrigerators
DC, Direct current
TEC, Thermoelectric cooler

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