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EXPERIMENTAL STUDY OF LOW TEMPERATURE REFRIGERATION CHAMBER

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Abstract

An experimental study of a low-temperature chamber included measuring the temperature inside and outside the chamber as it was cooled to a temperature of –150 °C and then heated with the cooling unit turned off. In addition, the temperature of a 14.4 kg steel plate placed inside the cooling chamber was measured as it cooled from ambient to the minimum temperature. It is shown that a regular thermal regime is established very quickly in the process of heating the cooling chamber. From this, we can find the equivalent value of the thermal diffusivity coefficient of the thermal insulation of the chamber. Using the analytical solution of the thermal insulation model with the value of the thermal diffusivity obtained in the experiment with a regular thermal regime of heating, it is possible to calculate the temperature distribution in the thermal insulation of the chamber during its cooling. Having an analytical solution for the temperature in the thermal insulation, we find the temperature gradient on the inner wall of the chamber as a coordinate derivative of the expression for the temperature in the thermal insulation. Given the analytical solution for a temperature gradient in the thermal insulation, it is possible to find the heat flow through the inner wall surface, which is equal to the product of the temperature gradient on the inner wall, inner wall surface and the thermal conductivity of the thermal insulation. However, the thermal conductivity of the polyurethane foam insulation can be assumed to be known with a large margin of error. Therefore, to refine the thermal conductivity coefficient of thermal insulation at low temperatures, we will use the results of the experiment to cool a loaded refrigerator chamber. The steel part inside the refrigeration chamber plays the role of a calorimeter. From this, we find the value of the thermal conductivity of the thermal insulation, which is equal to 0.0407 W/(m·K), and we find the cooling capacity of the refrigerating unit at –150 °C, which is equal to 136 W.

Keywords: Low-temperature chamber; Thermal insulation; Thermal diffusivity coefficient; Regular thermal regime; Thermal conductivity; Temperature gradient; Cooling capacity.

ЕКСПЕРИМЕНТАЛЬНЕ ДОСЛІДЖЕННЯ НИЗЬКОТЕМПЕРАТУРНОЇ ХОЛОДИЛЬНОЇ КАМЕРИ

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Анотація

Експериментальне дослідження низькотемпературної камери включало вимірювання температури всередині та зовні камери, коли її охолоджували до температури –150 °С, а потім відігрівали з вимкненою холодильною машиною. Крім того, вимірювалася температура сталевої пластини вагою 14.4 кг, поміщеної в камеру охолодження. Показано, що регулярний тепловий режим встановлюється дуже швидко в процесі нагрівання холодильної камери. Це дозволяє знайти еквівалентне значення коефіцієнта температуропровідності теплоізоляції камери. Використовуючи аналітичне рішення для нестаціонарної теплопровідності теплоізоляції зі значенням еквівалентного коефіцієнта температуропровідності, отриманим в експерименті з регулярним режимом нагріву, можна розрахувати розподіл температур в теплоізоляції камери під час її охолодження. Маючи аналітичне рішення для розподілу температур в теплоізоляції камери, знаходимо градієнт температури на внутрішній стінці камери як похідну від виразу для температури в теплоізоляції. Маючи аналітичне рішення для градієнта температур на внутрішній стінці камери, можна знайти тепловий потік через внутрішню поверхню стінки, який дорівнює добутку градієнта температури на внутрішній стінці та еквівалентної теплопровідності теплоізоляції. Однак еквівалентну теплопровідність пінополіуретанової ізоляції можна вважати відомою з великою похибкою. Тому для уточнення еквівалентного коефіцієнта теплопровідності теплоізоляції при низьких температурах використано результати експерименту з охолодженням завантаженої холодильної камери. Звідси було знайдено еквівалентне значення теплопровідності теплоізоляції, яке дорівнює 0.0407 Вт/(м·К), і знаходимо холодопродуктивність холодильної установки при –150 °С, яка дорівнює 136 Вт.

Ключові слова: низькотемпературна камера; теплоізоляція; тоефіцієнт теплопровідності; регулярний тепловий режим; теплопровідність; температурний градієнт; потужність охолодження.

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Introduction

Reducing the electrical energy consumption of refrigerators is highly beneficial to global energy conservation. The electric power consumption of refrigerators is mainly related to the heat transfer rate through the insulating walls. In order to predict the heat transfer characteristics of a refrigerator, many researchers [1–10] have studied the heat transfer phenomena in a refrigerator both experimentally and numerically. The amount of heat loss through the insulation of household refrigerators has also been determined experimentally [11–16] by the reverse heat loss method. In this method, steady-state conditions were achieved by adjusting the compartment temperatures to meet the requirements of the tests using electric heaters placed in the fresh and frozen food compartments [11–13].

This paper presents the results of an experimental determination of heat fluxes in a low-temperature chamber cooled by a chiller using a zeotropic mixture of refrigerants.

Low-temperature refrigeration chambers capable of maintaining a temperature of –150 °C for long periods of time can be used to store biological objects, such as donor blood, bone marrow, and other tissues. In addition, such chambers can be used for cold processing of steel or plastic parts to increase their strength and wear resistance.

The purpose of this study is to develop a relatively simple, non-destructive method for determining the cooling capacity of a low-temperature refrigeration machine operating with a zeotropic mixture of refrigerants. In addition, as a result of processing the results of the series of experiments, the caloric and dynamic parameters of the thermal insulation of the refrigeration chamber were determined.

Experimental method

This article discusses the results of an experimental study of a chamber designed for long-term storage of biological objects (see Fig.1).

Fig.1. Foto of the chamber designed for long-term storage of biological objects

The refrigeration unit of this chamber works according to the Joule-Thomson cycle. A zeotropic mixture of natural refrigerants: methane, nitrogen, ethane, propane, and isobutane is used as a working fluid in the refrigeration unit under consideration.

The low-temperature chamber has a rectangular shape and dimensions of 775x460x450 mm.

The thickness of the thermal insulation of the walls of the chamber is 200 mm. The chamber has a cooling unit capable of maintaining the chamber temperature at –150 °C. A schematic diagram of the cryogenic chamber refrigeration unit is shown in Fig. 2.

An experimental study of a low-temperature cooling chamber included measuring the temperature inside and outside the chamber during its cooling to the temperature of –150 °C and during the subsequent heating of the chamber with the cooling unit turned off.

In addition, the temperature of a 14.4 kg steel plate placed inside the cooling chamber was measured as it cooled from ambient temperature to the minimum temperature the cooling unit can provide (approximately –150°C).

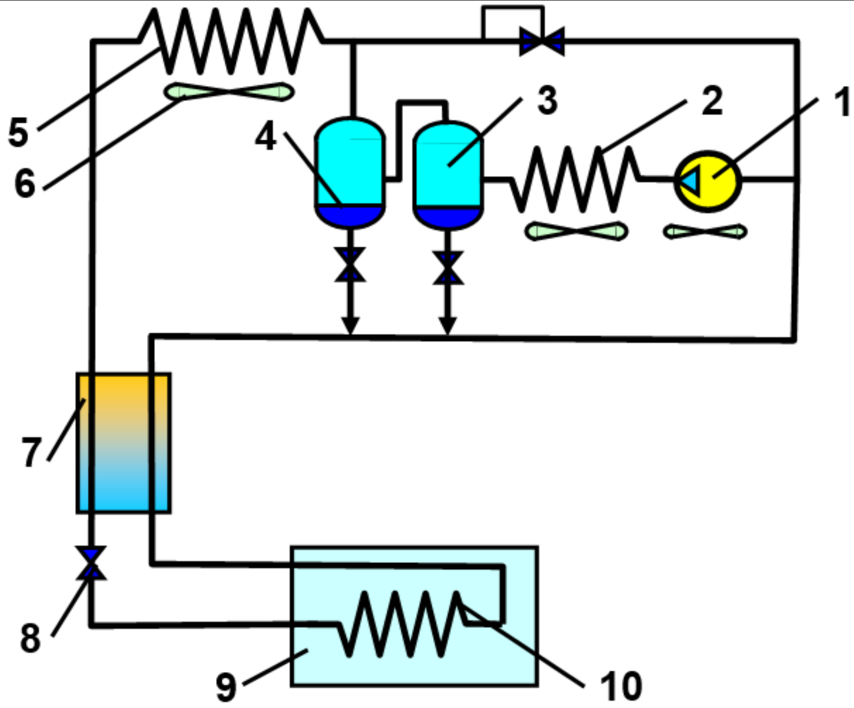

Fig. 2. Schematic diagram of the refrigeration unit of the low-temperature chamber. 1 – lubricated compressor TAG 2513Z; 2 – preliminary condenser; 3 – first oil separator; 4 – second oil separator; 5 – main condenser; 6 – fan; 7 – recuperative heat exchanger; 8 – throttling device (capillary tube); 9 – refrigerating chamber; 10 – evaporator

In order to measure the temperature of the cooling chamber, a calibrated temperature sensor (PT 1000 resistance sensor) was placed in the geometric center of the chamber. The readings of the sensor are automatically recorded in the DLT 10 logger at a frequency of one minute. The sensor has an accuracy of 0.5 °C and a temperature measurement range of –196 to 500 °C (see Fig. 3).

Fig. 3. Calibrated temperature sensor (PT 1000 resistance sensor) to measure the temperature of the cooling chamber

Results and discussion

Let's start the analysis of the obtained experimental data with the results of measuring the temperature inside the chamber during its natural heating from –150 °C to almost room temperature. These data, taking into account the presence of thermal bridges in the form of refrigerant inlet and outlet pipes, fixing elements of the chamber, etc., allow estimating the equivalent value of the thermal diffusivity of the thermal insulation of the chamber. In order to estimate the value of the thermal diffusivity of the thermal insulation of the refrigerating chamber, we will use the regular thermal regime method [17].

It is assumed that the ambient temperature T_E and the coefficient of heat transfer from the outer wall of the chamber α – are constant during the heating process of the refrigeration chamber. Under these conditions, the non-stationary process of the heating of the chamber can be divided into two stages: the initial stage and the stage of the regular mode of heating.

Initially, changes in the temperature field depend mainly on the characteristics of the initial thermal state of the insulation. Gradually, the influence of the initial conditions is lost and the importance of the heating conditions and the physical properties of the insulation become decisive. In this case, a regular thermal regime of chamber heating starts. At the regular thermal regime the law of temperature field change takes simple and universal form: temperature of all points of the chamber insulation increases with time according to exponential law.

The heat transfer coefficient to air during free convection has a value of approximately 5 W/(m²·K), the thermal conductivity of polyurethane foam insulation has a value close to 0.05 W/(m·K), and the thickness of the thermal insulation of the chamber is 0.2 m, so the Biot number, which characterizes the unsteady heating of the chamber, will be equal to Bi~20>>1. This means that the heating of the chamber is determined mainly by the processes inside the thermal insulation, and the temperature of its outer and inner surfaces will be approximately equal to the temperatures outside and inside the chamber, respectively. Therefore, we will consider the temperature of the inner wall of the refrigerating chamber to be approximately the same at all points, and we will make a similar assumption for the outer surface of the chamber.

With this in mind, as a mathematical model of the thermal insulation of the chamber, we choose the model of non-stationary thermal conductivity of an unlimited plate, see Fig. 4. Boundary conditions of the third kind are set on both sides of this plate, and zero boundary conditions of the second kind are set on the axis of symmetry. The differential equation of this form is used to describe the problem of transient heat conduction:

$$\begin{aligned} &\frac{\partial^2 T(x,\tau)}{\partial x^2} = a\frac{\partial T(x,\tau)}{\partial \tau}, \\ &T(x,0) = f(x); \frac{\partial T(0,\tau)}{\partial x} = 0 \\ &-\lambda\frac{\partial T(L,\tau)}{\partial x} + \alpha(T_E - T(L,\tau) = 0; \\ &\lambda\frac{\partial T(L,\tau)}{\partial x} + \alpha(T_E - T(-L,\tau) = 0; \end{aligned} \qquad (1)$$

where $T(x,\tau)$ is the current temperature of the thermal insulation of the refrigerating chamber, [K]; α is heat transfer coefficient from the outer surface of the chamber to air, [W/(m²·K)]; a is coefficient of thermal diffusivity of the thermal insulation of the chamber, taking into account the presence of thermal bridges in the form of pipes for supplying and discharging refrigerant, elements of fastening the chamber, etc., [m²/s]; λ is the value of the thermal conductivity coefficient of thermal insulation, [W/(m·K)].

The solution of the problem is given, for example, in Lienhard's textbook [18], and can be written as:

$$\Theta(x,\tau) = \sum_1^\infty B_k \cos\left(\mu_k \frac{x}{L}\right) \cdot \exp\left(-\mu_k^2 \frac{a}{L^2}\tau\right), \qquad (2)$$

where Θ is the dimensionless temperature $\Theta(x,\tau) = \frac{T_E - T(x,\tau)}{T_E - T_0}$; T_0 is the initial temperature in the chamber; μ_k are the roots of the characteristic equation: $tg(\mu_k) = -\frac{1}{Bi}\mu_k$; B_k is determined from the initial conditions.

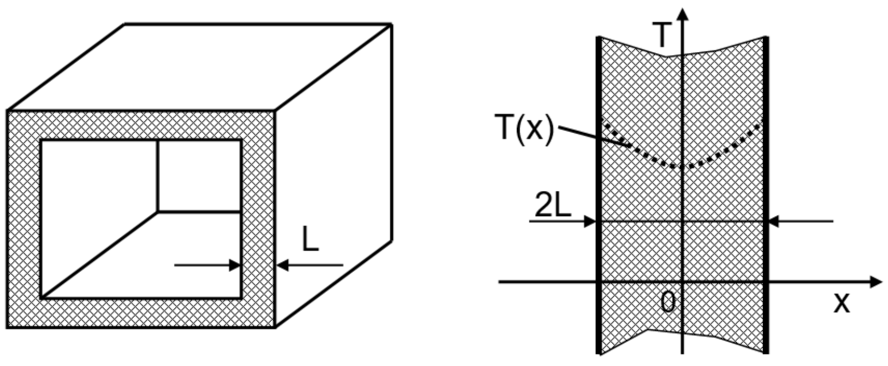

Fig. 4. Design diagram for heating the refrigeration chamber

For a Biot value of 20, the first five values of μ_k are, respectively: 1.4961; 4.4915; 7.4954; 10.5117; 13.5420 [18]. Hence it follows that the leading terms of the series (2) decrease very rapidly. Therefore, the temperature at any point of the thermal insulation of the refrigerating chamber, long before it reaches the ambient temperature, will be determined by the first member of the series (2), i.e. will follow a simple

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exponential law. In other words, a regular thermal regime for heating the chamber will begin.

If we plot the dependence of the temperature inside the chamber on time in a semi-logarithmic coordinate system, then a straight line on such a graph will correspond to the regular thermal regime of heating the refrigerating chamber.

In Fig. 5 shows a graph of the temperature inside the refrigerator compartment versus time in a semi-logarithmic coordinate system. It can be seen from the graph that in the process of heating the refrigerating chamber, a regular thermal regime is established very quickly, which corresponds to the exit of the temperature change curve to a straight line.

The tangent of the slope of this line is equal to the exponent $\mu_1^2 \frac{a}{L^2}$. Taking the value of μ_1 =1.4961, which corresponds to the value of the Biot number equal to 20, we find the equivalent value of the parameter a/L^2 for the refrigerating chamber under consideration:

$$\frac{a}{L^2} = 5{,}157 \cdot 10^{-6}\, 1/\mathrm{s}. \qquad (3)$$

Thus, it is possible to find the value of the parameter determining the rate of heating of the chamber based on the analysis of experimental data on heating of the cooling chamber.

We will now proceed to the analysis of the data on the cooling of the chamber with the cooling unit in operation.

Figure 6 shows a graph of the dependence of the temperature inside the cooling chamber on the cooling time. This dependence is approximated by a fifth-degree polynomial of the time measured in minutes.

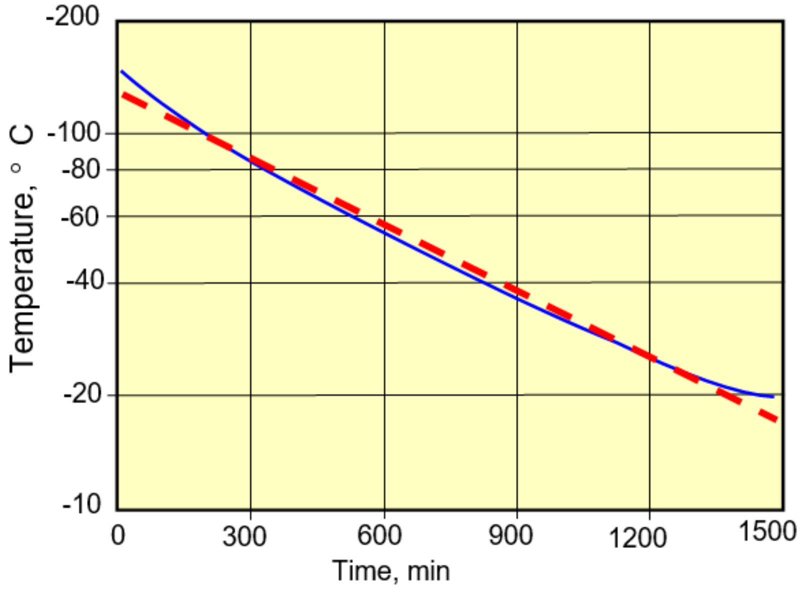

Fig. 5. Change in temperature inside the refrigeration chamber when it is warmed up. Solid blue line – experiment; Red dotted line – regular thermal regime

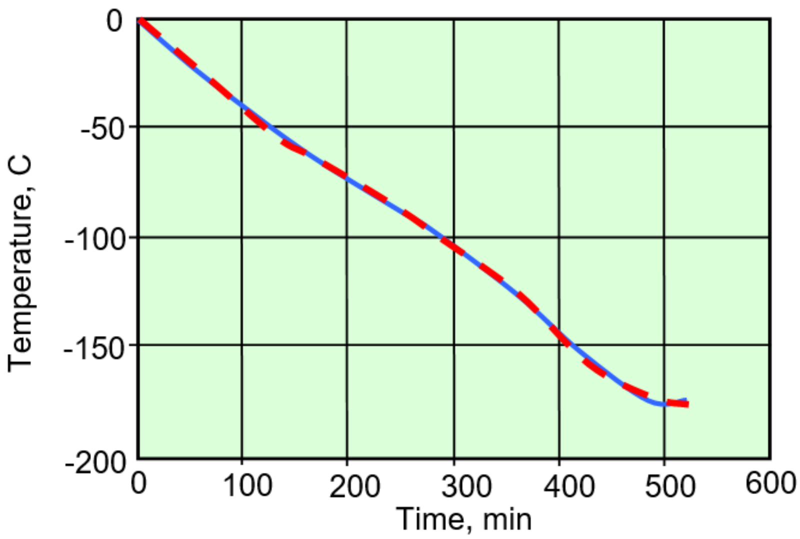

Fig. 6. Approximation of experimental data on the cooling of an empty refrigerator chamber by a fifth-degree polynomial. Solid line – experiment; dotted line - approximation

As can be seen from the above graph, the polynomial of the fifth-degree from time approximates well all the features of the time dependence of the temperature course inside the chamber.

As a mathematical model of the cooling process of the chamber we again choose the transient thermal conductivity of an infinite plate, but as boundary conditions we choose the boundary conditions of the first type on the outer and inner walls of the plate. In addition, the temperature on the outer wall of the plate under consideration is assumed to be constant, and the temperature on the inner wall of the plate (chamber) is assumed to vary according to a certain law. As an initial condition of the model, we assume that the chamber temperature is equal to the ambient temperature T_E. To simplify the task, we assume that the ambient temperature T_E is zero

$$\frac{\partial^2 T(x,\tau)}{\partial x^2} = \frac{1}{a}\frac{\partial T(x,\tau)}{\partial \tau},$$
$$T(L,\tau) = T_E = 0; \qquad (4)$$
$$T(L,\tau) = f(\tau);$$
$$T(x,0) = T_E = 0,$$

where f (τ) is a fifth-degree polynomial approximating the time dependence of the chamber temperature.

After the transition to the operator form in time, the equation of unsteady heat conduction takes the form:

$$\frac{\partial^2 \tilde{T}(x,s)}{\partial x^2} = \frac{1}{a}\left[s \cdot \tilde{T}(x,s) - T_0(x)\right], \qquad (5)$$

where s is the time differentiation operator [19]; $T_0(x)$ is the initial temperature distribution in the thermal insulation.

Taking into account zero initial conditions, we have

$$\frac{\partial^2 \tilde{T}(x,s)}{\partial x^2} = \frac{s}{a} \cdot \tilde{T}(x,s). \qquad (6)$$

We perform a finite integral transformation along the x-axes using the transformation kernel, which has the form [20–22]

$$\bar{T}(k,s) = \int_0^L \tilde{T}(x,s)\sin\left(\pi k \frac{x}{L}\right)dx,$$
$$k = 0,1,2\ldots \qquad (7)$$

where L is the thickness of the thermal insulation of the refrigerating chamber.

The inversion formula for the selected integral transformation along the x-axis is

$$\tilde{T}(x,s) = \sum_{k=0}^{\infty} \frac{\bar{T}(k,s)}{\|\psi_k\|^2}\sin\left(\pi k \frac{x}{L}\right), \qquad (8)$$

where $\|\psi_k\|^2$ is the normalizing factor (the square of the norm of the kernel of the integral transformation)

$$\|\psi_k\|^2 = \int_0^L \sin^2\left(\pi k \frac{x}{L}\right)dx = \frac{L}{2}. \qquad (9)$$

In this way, the integral transformation is introduced, which is called the Fourier sinusoidal transformation. It allows to eliminate the second partial derivative with respect to the coordinate from the differential equation of transient heat conduction.

Integrating the second partial derivative with respect to the x-coordinate twice by parts, we obtain

$$\int_0^L \frac{\partial^2 \tilde{T}(x,s)}{\partial x^2}\sin\left(\pi k \frac{x}{L}\right)dx = \frac{\partial \tilde{T}(x,s)}{\partial x}\sin\left(\pi k \frac{x}{L}\right)\bigg|_0^L - \frac{\pi k}{L}\int_0^L \frac{\partial \tilde{T}(x,s)}{\partial x}\cos\left(\pi k \frac{x}{L}\right)dx =$$
$$= -\frac{\pi k}{L}\tilde{T}(x,s)\cos\left(\pi k \frac{x}{L}\right)\bigg|_0^L - \left(\frac{\pi k}{L}\right)^2 \int_0^L \tilde{T}(x,s)\sin\left(\pi k \frac{x}{L}\right)dx = \qquad (10)$$
$$= -\frac{\pi k}{L}\left((-1)^k T_E - \{f(\tau)\}\right) - \left(\frac{\pi k}{L}\right)^2 \bar{T}(k,s)$$

Taking into account zero boundary conditions on the outer wall of the chamber, we have

$$\int_0^L \frac{\partial^2 \tilde{T}(x,s)}{\partial x^2}\sin\left(\pi k \frac{x}{L}\right)dx = \frac{\pi k}{L}\{f(\tau)\} - \left(\frac{\pi k}{L}\right)^2 \bar{T}(k,s), \qquad (11)$$

where $\{f(\tau)\} = f(s)$ is the time dependence of the temperature of the inner wall in time-operator form or the image of the function $f(\tau)$.

Boundary conditions of the first type were applied to the inner and outer walls of the chamber during the transformations. Therefore, the boundary conditions are already included in the equation for the image of the second partial derivative with respect to the coordinate.

The equation for the thermal conductivity in a layer of thermal insulation, after a sine transformation along the *x*-coordinate, takes the form

$$\frac{\pi k}{L}\{f(\tau)\}-\left(\frac{\pi k}{L}\right)^2\bar{T}(k,s)=\frac{s}{a}\cdot\bar{T}(k,s). \qquad (12)$$

Solve the problem in image space

$$\bar{T}(k,s)=\frac{\frac{\pi k}{L}\{f(\tau)\}}{\left[\frac{s}{a}+\left(\frac{\pi k}{L}\right)^2\right]}. \qquad (13)$$

Using the inversion formula, we find the solution using the time-differentiating operator

$$\tilde{T}(x,s)=\frac{2}{L}\sum_{k=1}^{\infty}\frac{\frac{\pi k}{L}\{f(\tau)\}}{\left[\frac{s}{a}+\left(\frac{\pi k}{L}\right)^2\right]}\sin\left(\pi k\frac{x}{L}\right). \qquad (14)$$

Let's transform the last equation to the form

$$\tilde{T}(x,s)=2\sum_{k=0}^{\infty}\frac{\frac{\pi ka}{L^2}\{f(\tau)\}}{\left[s+\frac{\pi^2k^2a}{L^2}\right]}\sin\left(\pi k\frac{x}{L}\right). \qquad (15)$$

To get to the originals in time, we write the resulting solution as a convolution of two functions

$$\begin{aligned}T(x,\tau)&=2\sum_{k=0}^{\infty}\frac{\pi ka}{L^2}\{f(\tau)\}\cdot\left\{\exp\left(-\frac{\pi^2k^2a}{L^2}\tau\right)\right\}\cdot\sin\left(\pi k\frac{x}{L}\right)=\\&=2\sum_{k=0}^{\infty}\frac{\pi ka}{L^2}\sin\left(\pi k\frac{x}{L}\right)\int_0^{\tau}\exp\left(-\frac{\pi^2k^2a}{L^2}(\tau-t)\right)f(\tau)dt\end{aligned} \qquad (16)$$

If k=0, the resulting expression becomes indeterminate, the uncertainty of the form 0/0. This uncertainty can be resolved by using the rule of L'Hôpital

$$\lim_{k\to0}\left(\frac{\sin(\pi k)}{\pi k}\right)=\lim_{k\to0}\left(\frac{(\sin(\pi k))'}{(\pi k)'}\right)=\lim_{k\to0}\left(\frac{\pi\cos(\pi k)}{\pi}\right)=0. \qquad (17)$$

With this in mind, we finally have

$$T(x,\tau)=2\sum_{k=1}^{\infty}\frac{\pi ka}{L^2}\sin\left(\pi k\frac{x}{L}\right)\int_0^{\tau}\exp\left(-\frac{\pi^2k^2a}{L^2}(\tau-t)\right)f(\tau)dt. \qquad (18)$$

This solution has a fundamental disadvantage, which is typical for all solutions of similar problems obtained in the form of Fourier series. It can be seen from the last formula that all terms of the Fourier series, in the form of which this solution is obtained, satisfy zero boundary conditions. In other words, if we directly calculate the values of this series at x=0 and $x = L$, we will obtain zero values at the boundaries of the thermal insulation. According to our problem, the series should not satisfy such a boundary condition [20]. The reason for this situation is the uneven convergence of the obtained Fourier series at the boundaries of the region. According to Steklov's theorem, absolute and uniform convergence of Fourier series can be obtained only for homogeneous boundary conditions.

For technical calculations, these formal restrictions are not essential, since it is always possible to calculate the temperature value at a point close enough to the boundary surface so that the obtained temperature value can be considered equal to its value at the boundary surface.

By means of equation (18) with the value of the thermal diffusivity obtained in the experiment with a regular regime of heating of an empty chamber, it is possible to calculate the time dependence of the temperature distribution in the thermal insulation of the chamber during its cooling. Figure 7 shows a 3D graph of the temperature change in the thermal insulation layer during the cooling of the chamber.

We find the temperature gradient on the inner wall of the chamber as a derivative with respect to the *x*-coordinate of the expression (18) for the temperature distribution in the thermal insulation

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$$\frac{\partial T(x,\tau)}{\partial x}=-2\sum_{k=1}^{\infty}\frac{(\pi k)^2 a}{L^3}\cos\left(\pi k\frac{x}{L}\right)\int_0^{\tau}\exp\left(-\frac{\pi^2k^2a}{L^2}(\tau-t)\right)f(\tau)dt. \quad (19)$$

Having an analytical solution for the temperature gradient in the thermal insulation on the inner wall of the chamber, it is possible to find the heat flux through the inner wall Q_E, which is equal to the product of the temperature gradient on the inner wall surface and the thermal conductivity of the thermal insulation

$$Q_E(\tau)=\lambda\cdot F\cdot\left.\frac{dT(x,\tau)}{dx}\right|_{x=L}, \quad (20)$$

where λ is the equivalent thermal conductivity of the thermal insulation, taking into account the presence of thermal bridges, [W/(m·K)]; F is the surface of the inner walls of the refrigeration chamber, [m^2].

When cooling an empty refrigerating chamber, all heat inflows into the chamber are removed by the operating refrigeration machine. Therefore, the cooling capacity of the refrigeration unit Q_0 is exactly equal to the heat inflow through the chamber walls

$$Q_E(\tau)=Q_0(\tau). \quad (21)$$

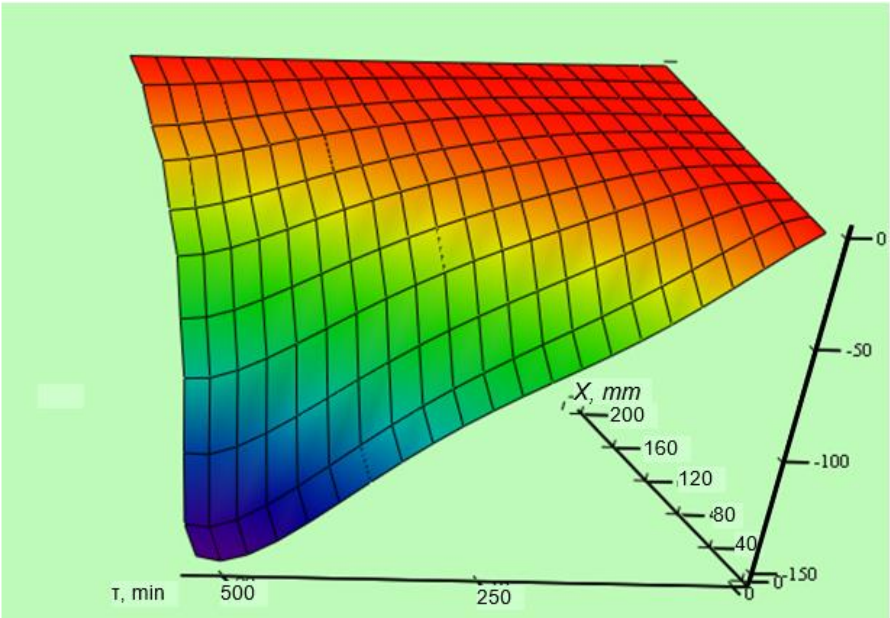

Fig. 7. 3D graph of the temperature profile in the thermal insulation of the cooling chamber during the cooling process

Figure 8 shows the dependence of the refrigeration unit's cooling capacity on the empty chamber cooling time. The equivalent thermal conductivity coefficient of the chamber insulation is assumed to be 0.0407 W/(m·K).

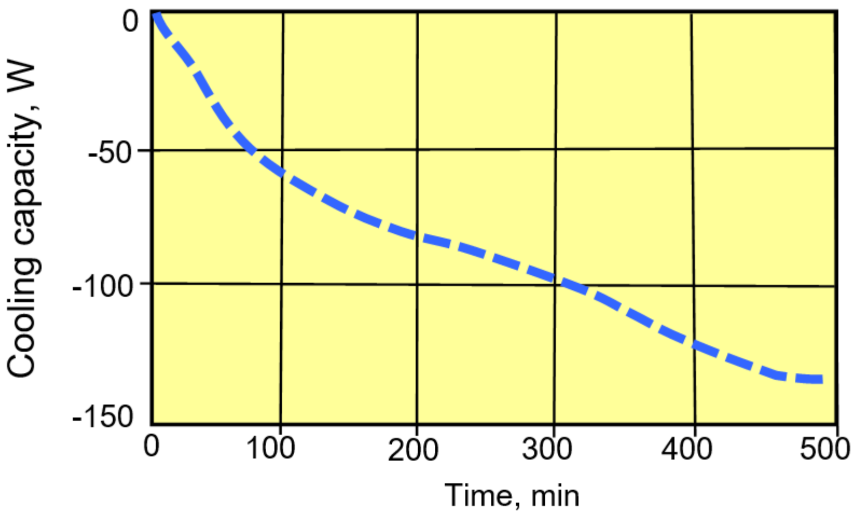

Fig. 8. Changes in the cooling capacity of the refrigeration system during the chamber cooling process without load

The internal dimensions of the refrigerator compartment are well known. However, the equivalent thermal conductivity of the polyurethane foam insulation can be assumed to be known with a large margin of error. This is due to the fact that the thermal conductivity of thermal insulation is highly dependent on the density of the foam and its temperature [23; 24].

Therefore, to refine the equivalent thermal conductivity coefficient of thermal insulation at low temperatures, we will use the results of an experiment to cool a loaded refrigerator chamber. This experiment measured the air temperature in the refrigeration compartment and the temperature of a 14.4 kg steel plate placed in that refrigeration compartment as it cooled from ambient to the temperature –150 °C.

Figure 9 shows the experimental data of air temperature in an empty and loaded refrigerator chamber and an approximation of the air temperature in a loaded chamber by a seventh-degree polynomial.

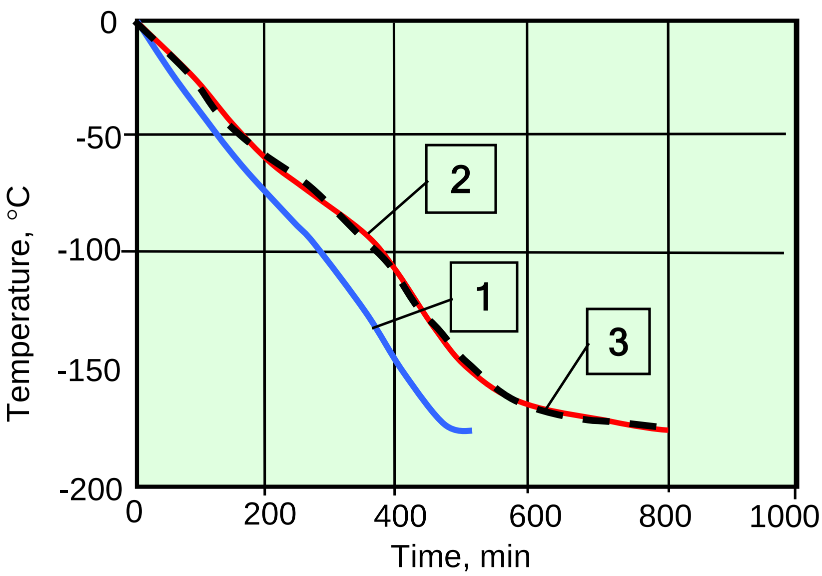

Fig. 9. Approximation of experimental data on the cooling of the empty (1) and loading (2) refrigerator chamber by a seventh-degree polynomial (3).
Solid line – experiment; dotted line - approximation

The amount of heat given off by the metal plate can be calculated by assuming that the temperature of the metal plate is practically the same at all points during the cooling process. This can be done from the measured temperature curve of this metal part, the known weight of the metal part, and the heat capacity of the steel.

In this way, the steel part inside the refrigeration chamber plays the role of a calorimeter, which allows us to measure the cooling capacity of the refrigeration unit. Therefore, we can determine the amount of heat that enters this chamber from the metal parts Q_M:

$$Q_M(\tau) = M \cdot C_m \frac{dT_M(\tau)}{d\tau}, \qquad (22)$$

where τ is time; M is the mass of the parts placed in the refrigeration chamber, [kg]; C_m – heat capacity of the steel, [J/(kg·K)].

In order to determine the temperature gradient on the inner wall of the refrigerator chamber in the process of cooling the loaded chamber, the same mathematical model can be used that was considered in the experiment of cooling the empty chamber.

Therefore, formulas (18) and (19) will also be valid in the case of cooling a loaded refrigeration chamber. However, in this case, the time dependence of the temperature $f(\tau)$ should be given when the loaded refrigeration chamber is cooling.

The heat capacity of steel is highly temperature dependent in the temperature range from 0 to –150 °C. The graph in Figure 10 illustrates this dependence, which has been approximated by a second-degree polynomial based on three known points.

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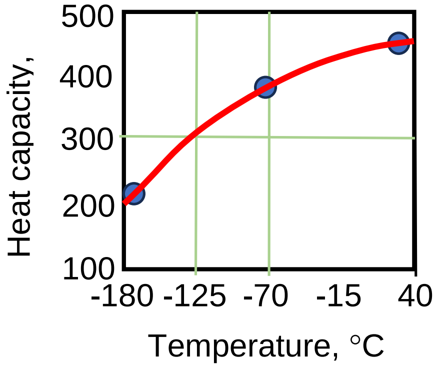

Fig. 10 Dependence of the heat capacity of steel on temperature, which has been approximated by a second-degree polynomial. Circles – experiment; solid line – approximation

If we know the heat capacity of a metal plate as a function of its temperature and its mass, then by multiplying them by the rate of change of the temperature of the solid metal plate, we can obtain the dependence of the heat flux given off by this plate on the cooling time of the loaded chamber.

Figure 11 shows a graph of the heat flux given off by the metal plate during the cooling process of the loaded chamber.

Knowing the heat flux given off by the steel plate, we can write down the heat balance equation for a loaded low-temperature chamber during its cooling. The cooling capacity of the refrigerator at any moment is exactly equal to the sum of the heat flow emitted by the steel plate and the heat flow entering the chamber through its walls

$$Q_0(\tau) = Q_L(\tau) + Q_M(\tau) = \lambda \cdot F \cdot \left.\frac{\partial T_L(x,\tau)}{dx}\right|_{x=L} + M \cdot C_m \cdot \frac{dT_M(\tau)}{d\tau}, \quad (23)$$

where Q_0 is the cooling capacity of the installation, [W]; Q_L is heat input to the refrigerating chamber throughout its wall, [W]; Q_M - heat input from the still plate, [W].

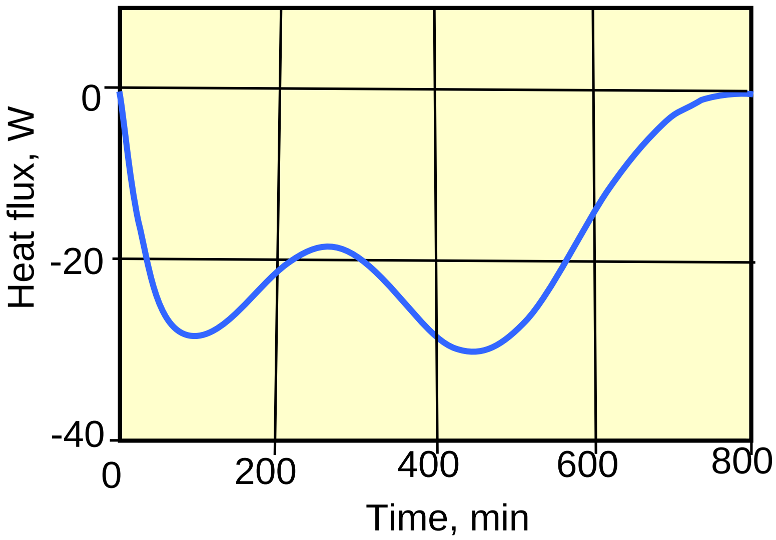

Fig. 11. Heat flux given off by the metal plate during the cooling process of the loaded chamber

At a constant condensing temperature, the cooling capacity of the refrigerator is determined only by the boiling temperature in the system's evaporator. Therefore, at the same temperature in the refrigeration chamber, the cooling capacity of the refrigeration unit will be the same whether it is cooling an empty chamber or a loaded chamber. Hence

$$Q_0(T) = Q_E(T) = Q_L(T) + Q_M(T), \quad (24)$$

where Q_E - heat input to the refrigerating chamber at cooling without load, W.

Substituting the appropriate expressions for the heat flows during cooling of an empty and loaded chamber, we obtain an equation in which there is only one unknown − the equivalent thermal conductivity of the thermal insulation of the chamber.

$$\lambda \cdot F \cdot \left.\frac{\partial T_E(x,\tau)}{dx}\right|_{x=L} = \lambda \cdot F \cdot \left.\frac{\partial T_L(x,\tau)}{dx}\right|_{x=L} + M \cdot C_m \cdot \frac{dT_M(\tau)}{d\tau}, \quad (25)$$

From this we find the equivalent value of the thermal conductivity of the thermal insulation, which is equal to λ(−150°C)=0.0407 W/(m·K), and we find the cooling capacity of the cooling unit at −150°C, which is equal to Q_0 =136 W.

Conclusions

An experimental study of a low-temperature chamber included measuring the temperature inside and outside the chamber as it was cooled to the temperature of −150 °C and then heated with the cooling unit turned off. In addition, the temperature of a 14.4 kg steel plate placed inside the cooling chamber was measured as it cooled from ambient to the minimum chamber temperature. It is shown that a regular thermal regime is established very quickly in the process of heating the refrigerating chamber. From this, was finding the value of the thermal diffusivity coefficient of the thermal insulation of the chamber.

Using the analytical solution of the thermal insulation model, was calculated the temperature distribution in the thermal insulation of the chamber during its cooling. Having an analytical solution for the temperature in the thermal insulation, we find the temperature gradient on the inner wall of the chamber as a coordinate derivative of the expression for the temperature in the thermal insulation. Given the analytical solution for a temperature gradient in the thermal insulation, was fined the heat flow through the inner wall surface, which is equal to the product of the temperature gradient on the inner wall and the thermal conductivity of the thermal insulation.

However, the thermal conductivity of the polyurethane foam insulation can be assumed to be known with a large margin of error. Therefore, to refine the thermal conductivity coefficient of thermal insulation at low temperatures, we will use the results of the experiment to cool a loaded refrigerator chamber. From this, we find the value of the thermal conductivity of the thermal insulation, which is equal to 0.0407 W/(m·K), and we find the cooling capacity of the refrigerating unit at −150°C, which is equal to 136 W.

A fundamentally new method for determining the refrigeration capacity of refrigeration units operating as part of refrigeration chambers is proposed. To do this, it is enough to measure the temperature progression inside the chamber as it cools and warms up. This makes it possible to analyze the condition of the refrigeration unit during the operation of the refrigeration chamber.

In addition, a method of non-destructive testing of the state of thermal insulation of the refrigeration chamber using the measured values of the coefficients of thermal diffusivity and thermal conductivity is proposed.

References

[1] Laguerre, O., Ben Amara, S., Flick, D. (2005). Experimental study of heat transfer by natural convection in a closed cavity: application in a domestic refrigerator. *Journal of Food Engineering*, 70, 523–537.

[2] Tian, S., Shao, S., Liu, B. (2019). Investigation on transient energy consumption of cold storages: Modeling and a case study. *Energy*, 180, 1–9.

[3] Sakallı, Ö., Kerpiççi, H., Kuddusi, L. (2017). A study on optimizing the energy consumption of a cold storage cabinet. *Applied Thermal Engineering*, 112, 424–430.

[4] Hermes, C., Melo, C., Knabben, F., Gonçalves, J. M. (2009). Prediction of the energy consumption of household refrigerators and freezers via steady-state simulation. *Applied Energy*, 86, 1311–1319. doi:10.1016/j.apenergy.2008.10.008

[5] Yilmaz, I.C., Yilmaz, D. (2020). Optimal capacity for sustainable refrigerated storage buildings. *Case Studies in Thermal Engineering*, 22, 100751.

[6] Laguerre, O., Ben Amara, S., Moureh, J., Flick, D. (2007). Numerical simulation of air flow and heat transfer in domestic refrigerators. *Journal of Food Engineering*, 81, 144–156.

[7] Laguerre, O., Flick, D. (2010). Temperature prediction in domestic refrigerators: deterministic and stochastic approaches. *International Journal of Refrigeration*, 33, 41–51.

[8] Yazici, B., Can, Z.s., Calli, B. (2014). Prediction of future disposal of end-of-life refrigerators containing CFC-11. *Waste Management,* 34, 162-166.

[9] Harrington, L., (2017). Quantifying energy savings from replacement of old refrigerators. *Energy Procedia,* 121, 49-56.

[10] Elarem, R., Mellouli, S., Abhilash, E., Jemni, A. (2017). Performance analysis of a household refrigerator integrating a PCM heat exchanger. *Applied Thermal Engineering*, 125, 1320-1333.

[11] Wei-Han Tao, Hyi-Yu Sun. (2001). Simulation and experimental study on the air flow and heat loads of different refrigerator cabinet designs. *Chemical Engineering Communication*, *186*(1), 171–182.

[12] Sim, J.S., Ha, J.S. (2011). Experimental study of heat transfer characteristics for a refrigerator by using reverse heat loss method. *International Communications in Heat and Mass Transfer, 38*(5), 572–576. doi:10.1016/j.icheatmasstransfer.2011.02.007

[13] Paul, A., Baumhoegger, E., Elsner, A., Moczarski, L., Reineke, M. Sonnenrein, G., Hueppe, C., Stamminger, R., Hoelscher, H., Wagner, H., Gries, U., Freiberger, A., Becker, W., Vrabec, J. (2021). Determining the heat flow through the cabinet walls of household refrigerating appliances. *International Journal of Refrigeration*, 121, 235–242. doi:10.1016/j.ijrefrig.2020.10.007
[14] Soltani, Z., Sørensen, K.K., Leth, J., Bendtsen, J.D. (2022). Fault detection and diagnosis in refrigeration systems using machine learning algorithms. *International Journal of Refrigeration*, 144, 34–45.
[15] Thiessen, S., Knabben, F.T., Melop, C. (2014). Experimental evaluation of the heat fluxes through the walls of a domestic refrigerator. *15th Brazilian Congress of Thermal Sciences an Engineering at Belem*, PA, Brazil, 10–13.
[16] Hueppe, C., Geppert, J., Stamminger, R., Wagner, H., Hoelscher, H., Vrabec, J., Paul, A., Elsner, A., Becker, W., Gries, U., Freiberger, A. (2020). Age-related efficiency loss of household refrigeration appliances: Development of an approach to measure the degradation of insulation properties. *Applied Thermal Engineering,* 173, 115113.
[17] Dulnev, G. N. (1960). The theory of thermal regular regime and its application to the determination of thermal characteristics. *International Journal of Heat and Mass Transfer, 1*(2-3), 152–160. doi:10.1016/0017-9310(60)90019-3
[18] Lienhard, J. H. IV, Lienhard, J. H. V. (2020). *A Heat Transfer Textbook* (5th ed.). Mineola, NY: Dover Pub. p. 771.
[19] Mikusiński, J. (1959). *Operational Calculus*. New York: Pergamon Press.
[20] Duhamel, P. (2004). Application of a new finite integral transform method to the wave model of conduction. *International Journal of Heat and Mass Transfer*, *47*(3), 573–588.
[21] Cotta, R. M., Knupp, D. C., Naveira-Cotta, C. P., Sphaier, L. A., Quaresma, J. N. N. (2013). Unified Integral Transforms Algorithm for Solving Multidimensional Nonlinear Convection-Diffusion Problems. *Numerical Heat Transfer, Part A: Applications*, *63*(11), 840–866.
[22] Queiroz, H., Bentes, F. M., Nóbrega, M. J, Martins, F. B. (2020). A generalized integral transformed technique: literature review and comparative results with finite volume method. In book: *Atividades Cientificas e Tecnologicas no Campo da Engenharia Mecanica*. Publisher: Atena Editora. Paraná. Brasil.
[23] Berardi, U., Madzarevic, J. (2020). Microstructural analysis and blowing agent concentration in aged polyurethane and polyisocyanurate foams. *Applied Thermal Engineering,* 164, 114440.
[24] Hermes, C., Melo, C., Knabben. F. (2013). Alternative test method to assess the energy performance of frost-free refrigerating appliances. *Applied Thermal Engineering*, 50, 1029–1034.