NUMERICAL DYNAMIC SIMULATION OF DOMESTIC REFRIGERATING SYSTEMS

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Abstract

The package of software products is developed to provide the proved choice of a perspective design of a refrigerator for realization of design works on the basis of the analysis and comparisons of probable variants of execution of a product using parameters of optimization the energy consumption by a refrigerator and/or heat output in the evaporator and to determine preliminary a possibility of the use inside the structure of refrigerating unit of constructive elements with characteristics being different from standard characteristics of evaporator , condenser, etc.

KEYWORDS
domestic refrigerator calculation thermal characteristics energy consumption numerical dynamic simulation

INTRODUCTION

This work has been initiated on purpose to improve energy efficiency of domestic refrigerating systems [1]. Basis of the software package make the following program components.

1. The program of thermal calculations doorway contour heating tube position area of the freezing chamber.

2. The program of calculation of temperature in chambers , thermal and performance characteristics of refrigerating unit, energy consumption (considering the available data for used compressors) at cycle work of the refrigerating unit of a refrigerator/deep freezer with the bottom freezer (Fig. 1) with the purpose of decrease energy consumption and guarantee of a required temperature in chambers.

CALCULATIONS DOORWAY CONTOUR HEATING TUBE POSITION AREA

Performance of the given calculation serves for data acquisition heat penetration values to the cabinet and, hence, the evaporator thermal loading , required compressors' capacity at different models of refrigerators. The basis of calculated circuit is made with definition of a temperature field in the area of position of the heating tube with use of the numerical alternating direction method for the solution of two-dimensional transient heat conduction problem.

The dynamic thermal mathematical model describes conductive heat transfer through heat insulation of the freezer compartment wall:

$$\frac{\partial T}{\partial \tau} = a_{pf}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right), \qquad (1)$$

where x, y – Cartesian coordinates.

Description of the temperature distribution for the outer metal panel and the inner plastic plate is based on the solution of one-dimensional unsteady heat conduction problem. The system of the equations are solved numerically. Some results of calculations are presented in Fig. 2.

CYCLE WORK OF THE REFRIGERATING UNIT

This component part of the software package represents by itself program of calculation (numerical dynamic simulation) of a 2-chamber 1-compressor refrigerator. Additions, which will

upgrade the program, concern, first of all, more full consideration of compressors' characteristics changes.The dynamic thermal mathematical model describes conductive heat transfer through three-layer walls (metal panel, heat insulation, plastic or pasteboard plate) of the refrigerator:

$$\frac{\partial T_i}{\partial \tau} = a_{pf} \frac{\partial^2 T_i}{\partial x_i^2} \ , \qquad 1 \le i \le N \ , \tag{2}$$

$$-\lambda_{pf} \frac{\partial T_i}{\partial x_i}\Big|_{x=+0} = \alpha_i \left(T_{env} - T_i\Big|_{x=+0}\right) - \rho_i c_i \delta_i \frac{\partial T_i}{\partial \tau}\Big|_{x=-0} \ , \tag{3}$$

$$\lambda_{pf} \frac{\partial T_i}{\partial x_i}\Big|_{x=\delta_i-0} = \alpha_{i,cab}\left(T_{cab} - T_i\Big|_{x=\delta_i-0}\right) - \rho_{i,cab}, c_{i,cab}\delta_{i,cab} \frac{\partial T_i}{\partial \tau}\Big|_{x=\delta+0} \ , \tag{4}$$

$$\lambda_{pb} \frac{\partial T_i}{\partial x_i}\Big|_{x=\delta_i-0} = \frac{1}{\dfrac{1}{\alpha_{evp}} + \dfrac{\delta_{i,cab}}{\lambda_i}}\left(T_{cab} - T_i\Big|_{x=\delta_i-0}\right) -$$

$$-\left(\rho_{i,cab} c_{i,cab} \delta_{i,cab} + \rho_{evp} c_{evp} \delta_{evp}\right) \frac{\partial T_i}{\partial \tau}\Big|_{x=\delta+0} + q_{evp} . \tag{5}$$

Eq. (5) describes thermal state of the inner wall of the storage compartment, where is disposed the foamed in evaporator: $q_{evp} = Q_{cm} / S_{evp.act}$ (compressor on-cycle), $q_{evp} = 0$ (compressor off-cycle).

Compressor performance data is presented according to international standards, which prescribes a 3rd order polynomial equation that shell be used to present refrigerant mass flow, cooling capacity and power consumption . The polynomial equation has the following form :

$$X = A_0 + A_1 t_0 + A_2 t_c + A_3 t_0^2 + A_4 t_0 t_c + A_5 t_c^2 + A_6 t_0^3 + A_7 t_c t_0^2 + A_8 t_0 t_c^2 + A_9 t_c^3, \tag{6}$$

where X - refrigerant mass flow in kg/h (cooling capacity in watt, power consumption in watt);

t_0 - saturation temperature at dew point of suction pressure in °C;

tc - saturation temperature at dew point of discharge pressure in °C;

A_0 to A_9 - polynomial coefficients.

Mass flow rate through the capillary tube is presented according to [2] (non-dimensional model):

$$\pi_1 = B \pi_2^a \pi_3^b \pi_4^c \tag{7}$$

(B is a constant coefficient.),

$$\left.\begin{aligned} \pi_1 &= \frac{G_{ct}}{D\mu_l}, & \pi_2 &= \frac{D^2 \rho_l P_c}{\mu_l^2}, \\ \pi_3 &= \frac{L}{D}, & \pi_4 &= \frac{D^2 \rho_l^2 c_l \Delta t}{\mu_l^2}. \end{aligned}\right\} \tag{8}$$

The mass balance equation for the evaporator has form (in accordance with experimental data [3]):

$$M_{l,evp} = M_{refr} - \int_{o}^{\tau_w} \left(G_{cm} - G_{ct}\right) d\tau \,. \qquad (9)$$

Now it is possible to obtain an active surface area of the evaporator of the storage compartment (an active surface area of the evaporator of the freezer is constant):

$$S_{evp,act} = \left[1 - k_{evp}\left(1 - \frac{M_{l,evp} - M_{fr}}{M_{refr} - M_{fr}}\right)\right] S_{evp,0}\,, \qquad (10)$$

where k_{evp} – an experimental scale coefficient.

Determination of temperature parameters of the condenser is based on the experimental results:

$$Q_c = k_c\, F_c\, (T_{env} + \Delta T_{env} - T_c). \qquad (11)$$

Also we have the basic heat balance equation (without details):

$$\int_{0}^{\tau_w} Q_{cm} d\tau = \int_{0}^{\tau_0} \left(Q_{st} + Q_{fr} + Q_{hp}\right) d\tau, \qquad (12)$$

where τ_w / τ_0 – operation ratio.

The system of the equations are solved numerically.

The program uses as incoming data physical parameters of thermostat, an ambient temperature , walls insulation thickness , etc. As a result of calculation all characteristics received at realization of tests, including the minimal and maximal temperatures in chambers of a refrigerator are determined. The characteristics of a continuous mode of operating of a refrigerator are calculated as well. Programs of calculation of the other developers, the authors have data about, provide only calculated imitation of a cyclic operation of the refrigerating unit , that , in particular , is related with big volume of calculations owing to complexity of mathematical model.

Software package is fulfilled by visual programming environment of Borland C ++, representing an executed exe-module and possessing fast realization. It has visual graphic presentation of the calculation results (Fig 3). Provides sending to the text files the most important calculated parameters . The main application of this software is the determination of energy consumption range decrease due to the improvement of a refrigerator design by maintaining temperature regime according to the standard. Correct product use supposes its calibration by a refrigerator model (prototype).

A calculation example is presented in Fig. 4.

Comparison of calculation results with test data proves its good correspondence to temperature and consumption characteristics of refrigerating unit and refrigerator .

NOMENCLATURE

a – thermal diffusivity, m^2/sec; c – specific heat, J/(kg K); D – inner diameter of capillary tube, m; F – area, m^2; G – mass flow rate, kg/sec; L – length of capillary tube, m;M – mass, kg; N – number of walls; P – pressure, Pa; Q – heat flow rate, W; q – heat flux, W/m^2; S – area, m^2; $S_{evp.0}$ – limit active surface area of evaporator, m^2; T – temperature, K; α – heat transfer coefficient, W/(m^2 K); ΔT_{env} – superheat of air around condenser, K; Δt – subcooling, K; δ – thickness, m; λ – thermal conductivity, W/(m K); μ – viscosity, Pa·sec; ρ – density, kg/m^3; τ – time, sec; τ_0 – total operating cycle time, sec; τ_w – refrigerating operation time, sec.

Subscripts: act – actual; c – condenser; cab – cabinet; cm – compressor; ct – capillary tube; env – environmental; evp – evaporator; fr – freezer; ht – doorway contour heating tube; l – liquid; pf – polyurethane foam; refr – refrigerant; st – storage.

References

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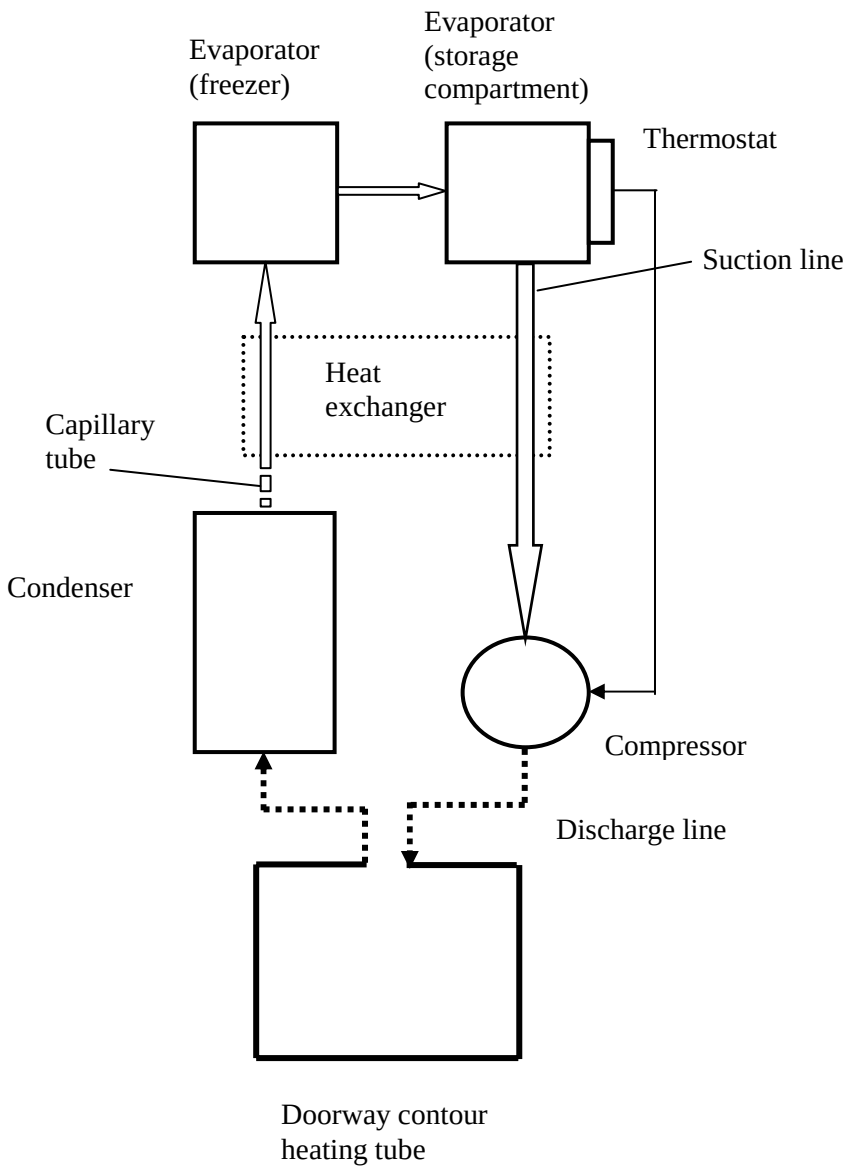

Fig. 1. Refrigerating unit of a 2-chamber 1-compressor refrigerator

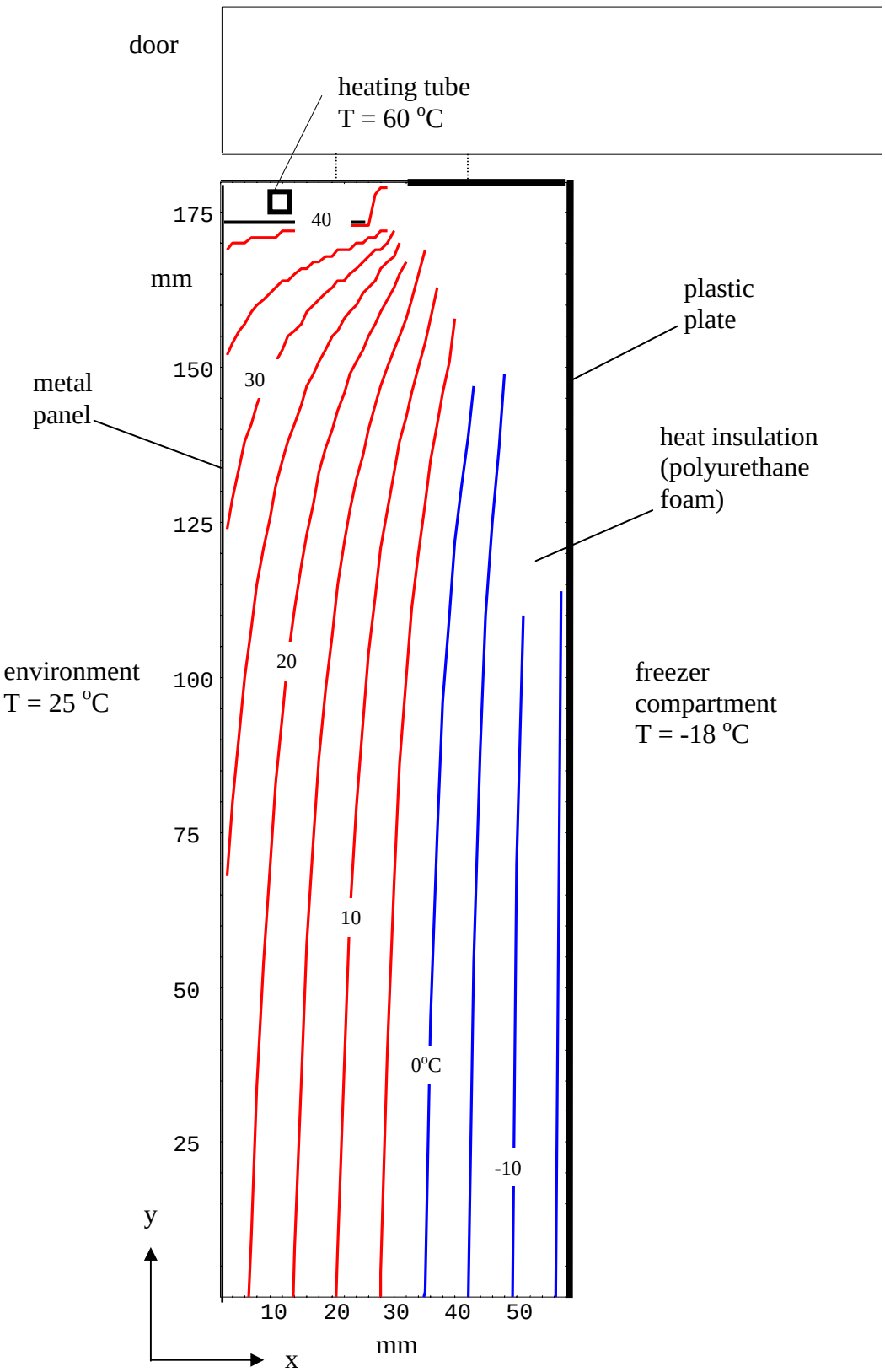

Fig. 2. Change in the temperature
of the freezer wall

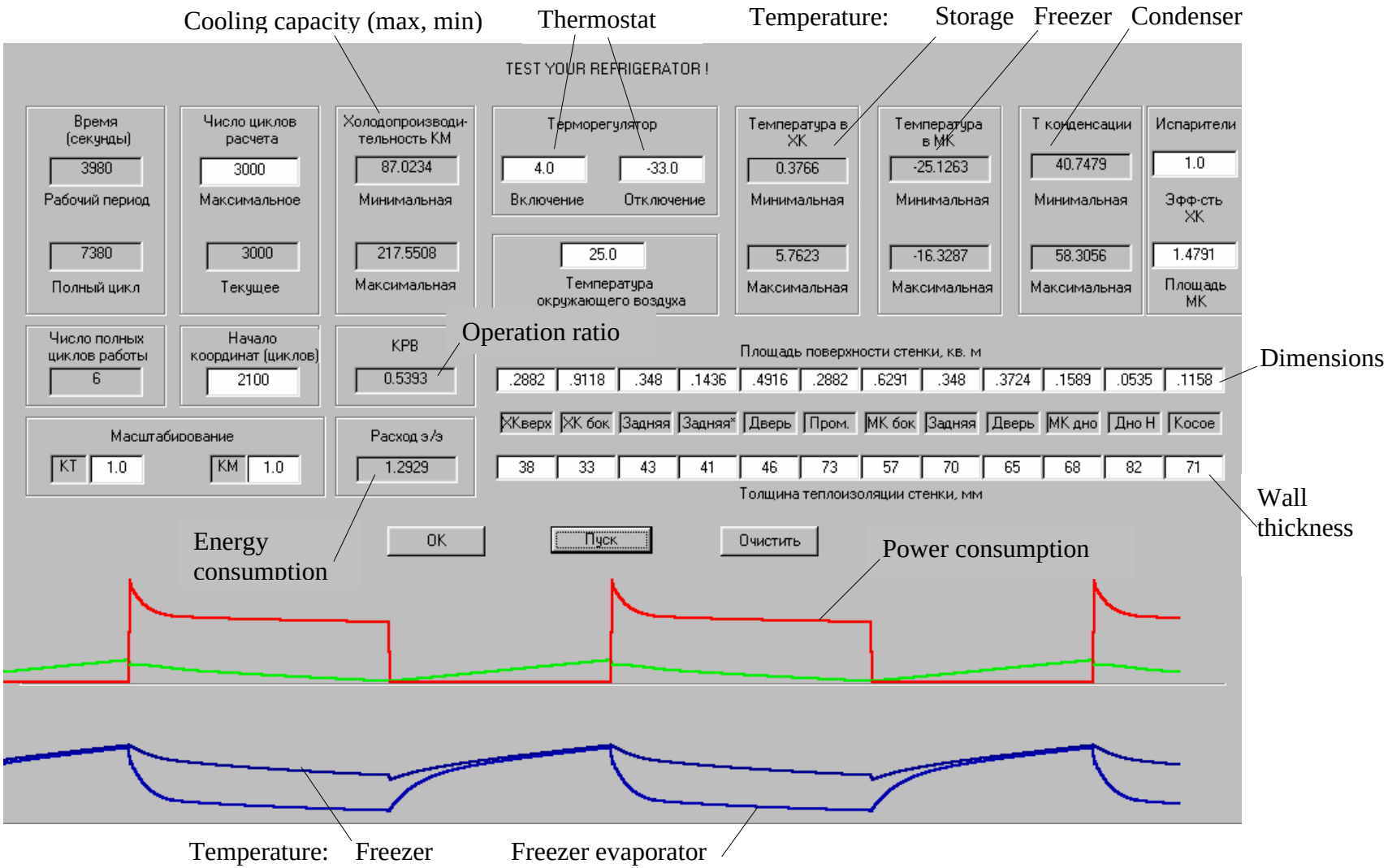

Fig. 3. The main window of the program

Basic technical characteristics of the refrigerator-freezer

Storage compartment gross volume, liters	215
Freezer compartment gross volume, liters	115
Refrigerant	R134a
Compressor cooling capacity (ASHRAE), W	164
Surface area of the storage compartment evaporator, m^2	0.14
Surface area of the freezer compartment evaporator, m^2	1.5
Surface area of the condenser, m^2	0.77

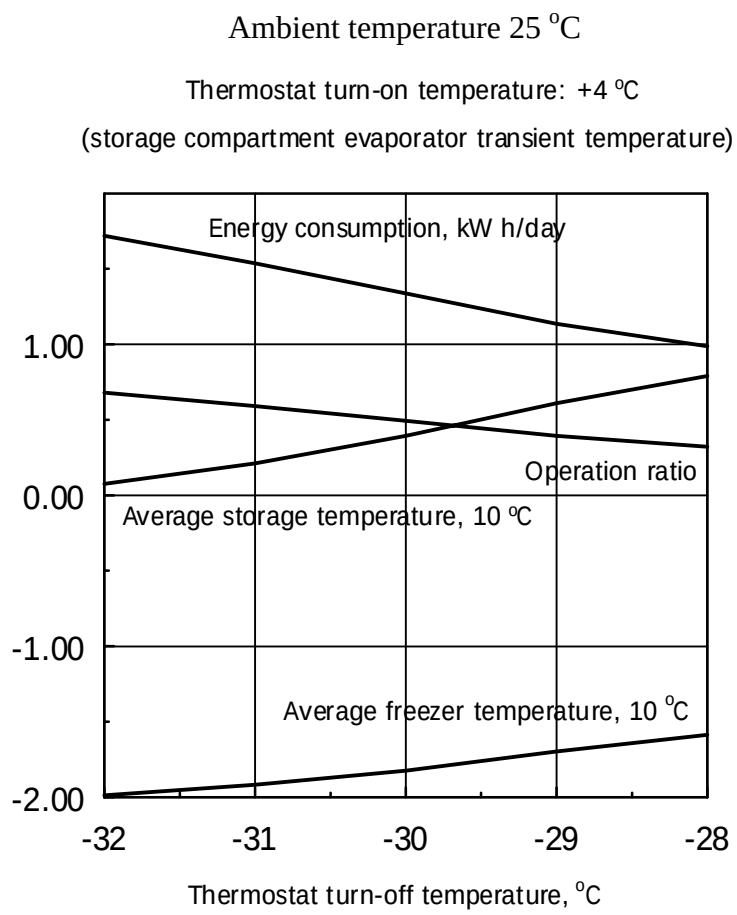

Fig. 4. A calculation example