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MODELS OF REFRIGERATING DEVICES IN THE OPEN MODELING SYSTEM TT-RH

Author: Serhiy Lomovtsev
Advisor: Boris Lomovtsev
Odesa National University of Technology (Ukraine)

***Abstract**. In the TT- RH system, the subject basis is modules that describe elementary processes in various elements of installations, both main and auxiliary, in uniform requirements that ensure the correctness and simplicity of their joint interaction; conditions of joint operation of these modules; universal algorithms for creating arbitrary control programs that depend on external modeling conditions. The structure of the subject area of the TT- RH open system consists of modules of several levels. Thanks to this, it is possible to simulate the scheme of any refrigerating machine in the working range of boiling and condensation temperatures, and the refrigerating (thermal) capacity of the installation itself. You can choose the cooling medium and the cooling coolant and so on. It is possible to make an effective selection of the compression mechanism, and if there is a large range between temperatures, switch to a multi-stage installation. At the same time, it is possible to choose the most effective regular mode of operation, observing the specified limits of power and operating temperatures. All this and much more, for example, the choice of an effective regenerative heat exchanger of its brand and heat exchange surface, as for refrigerating machines, can be successfully and efficiently carried out with the help of the open modeling system TT-RH.*

1. MODELS OF REFRIGERATORS BASED ON THE OPEN MODULAR SYSTEM TT- RH

1.1. Simulation of the operation of refrigeration devices

Today, the means of computer technology for the analysis and synthesis of complex technical objects lead to the need for the development of universal means of forming models of refrigerating units that flexibly implement any possible regulation schemes and programs. The requirements for generating such models are [1...4]:

- both structural and parametric analysis of R M's work;
- display of the physical essence of the formation of the model during the implementation of the production scheme;
- basing the subject area on single universal output modules;
- commonality and availability of means of forming models;
- the stability of the computing process with the possibility of effective accuracy control;
- simplicity and availability of interactive tools.

Taking into account the specified requirements, an open universal multi-level modular system TT- RH was developed for the formation of the installation model, which works according to the combined scheme RH (R - refrigeration - cold; H - heating - heat), which provides a parametric and structural analysis of the operation of various R M schemes on stages preceding the immediate design and construction process (Fig. 1)

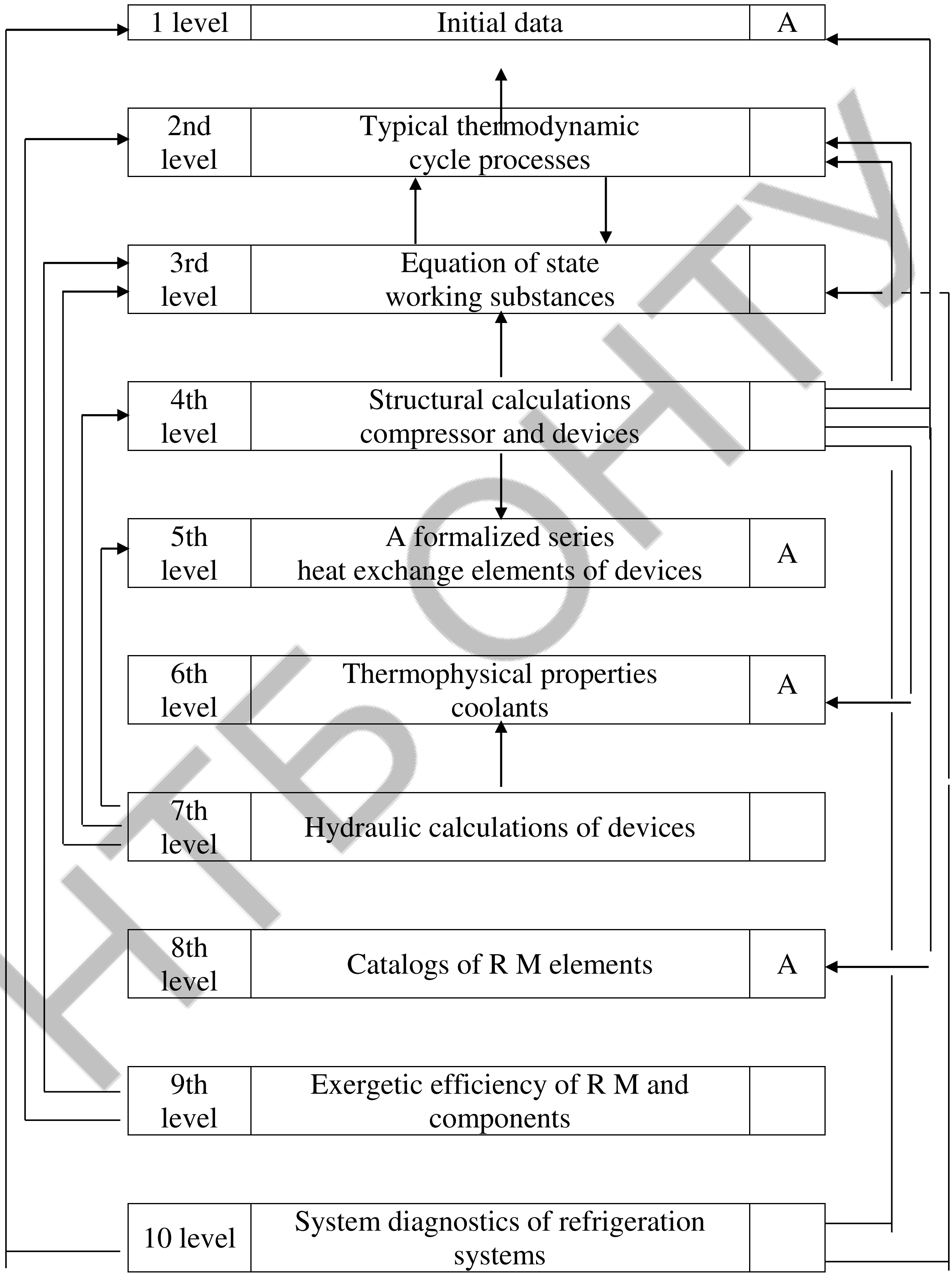

Fig. 1. The structure of the open multi-level modular system TT–RH

The formation of RM modules in the TT-RH system is based on the principle of built-in cycles, which allows for the physical interpretation and universality of the module formation process. In the TT-RH system, the subject basis is modules that describe elementary processes in various elements of installations, both main and auxiliary, in uniform requirements that ensure the correctness and simplicity of their joint interaction; conditions of joint operation of these modules; Universal algorithms for the assignment of arbitrary control programs that depend on external modeling conditions [2...4, 7].

1. 2. The structure of the subject area of the TT-RH open system

The structure of the subject area of the TT-RH open system consists of modules of several levels.

The first level consists of output data modules describing elementary procedures: selection of cooling capacity, condensation temperature depending on the physical properties and aggregate state of the cooling source and its initial temperature; selection of the boiling temperature depending on the physical properties and aggregate state of the cooled medium and its initial temperature, etc. Modules of the first level are autonomous and do not contain references to modules of other levels.

The second level of the module describes typical thermodynamic processes occurring in the selected cycle: compression in the compressor, expansion in the throttle valve, heat exchange in the main and auxiliary heat exchange devices, as well as the main operational characteristics (pressure difference in the cycle, their ratio, etc.). The functioning of the modules of the second level is carried out through calls to the modules of the first and third levels.

The third level of the module is the equations of state of the corresponding types and modifications of the selected working body, which allow unambiguous determination of thermodynamic parameters at the nodal points of the cycle. This level also contains modules that allow you to determine the thermophysical properties of the working body in different aggregate states based on the corresponding equations. The functioning of the modules of the third level is carried out in interaction with the modules of the second level.

With the help of modules of the fourth level, standard design calculations are carried out according to the main indicators and characteristics of devices, main and auxiliary heat exchange devices. The operation of all modules of this level involves addressing them to modules of the first, second and third levels, and some modules of this level additionally interact with modules of the fifth and sixth levels.

The fifth level of the module is a formalized series of elements that are used as elementary heat exchange elements. All their characteristics are standardized. The modules of this level are autonomous and do not depend on the modules of other levels.

Sixth -level modules contain data that determine the thermophysical properties of heat carriers, such as water, soda ash solution, air, etc., which are used in the main heat exchange devices of the HM. They do not depend on other level modules and function autonomously.

Modules of the seventh level. With their help , appropriate hydraulic calculations of the main and auxiliary heat exchangers are carried out. Their functioning is based on interaction with modules of the third, fourth, fifth and sixth levels.

Modules in the eighth level are - catalogs with the main components of RM, formed on the basis of existing standards for them according to the main characteristics and parameters. They can be accessed by modules of the fourth and tenth levels.

Modules of the ninth level consist of modules that implement exergy analysis of thermotransformers in general and for links separately with the determination of their efficiency based on relative losses. These modules interact with modules of the second and third levels.

The corresponding modules of the open diagnostic subsystem are connected to the last , tenth level , which, interacting with the modules of the first, second, and third levels, determine irreversibility in the operating installation, additional costs of energy raw materials for their elimination , and identify effective modes for given values.

Universal principles of synthesis of RM models from modules of these levels are based on the following conditions:

- continuity of the flow, which ensures the preservation of the flow balance;
- power balance, heat balance, conditions imposed by regulatory programs;
- aggregate state of the working body.

This approach allows you to imagine the complex process of model synthesis from typical elementary modules in a simple and accessible way for users who do not have the qualifications of a professional programmer. The RM model of an arbitrary scheme is formed from the output modules of the sequence, which reflects the structure of the selected installation scheme.

The open TT- RH system uses the following standard numerical methods for solving systems of equations; the method of nested cycles, which is divided into the method of bisections and the method of chords [1 , 4, 8]. To develop the TT-RH system , the rapid application development environment was used [6 , 10, 11 , 14].

An example of a work session with the TT–RH system is shown in Fig. 2.

Unfortunately, a fragment with calculations for the working fluid R717–ammonia, which in itself is a very toxic substance, got here. But all this has a relatively negative resonance.

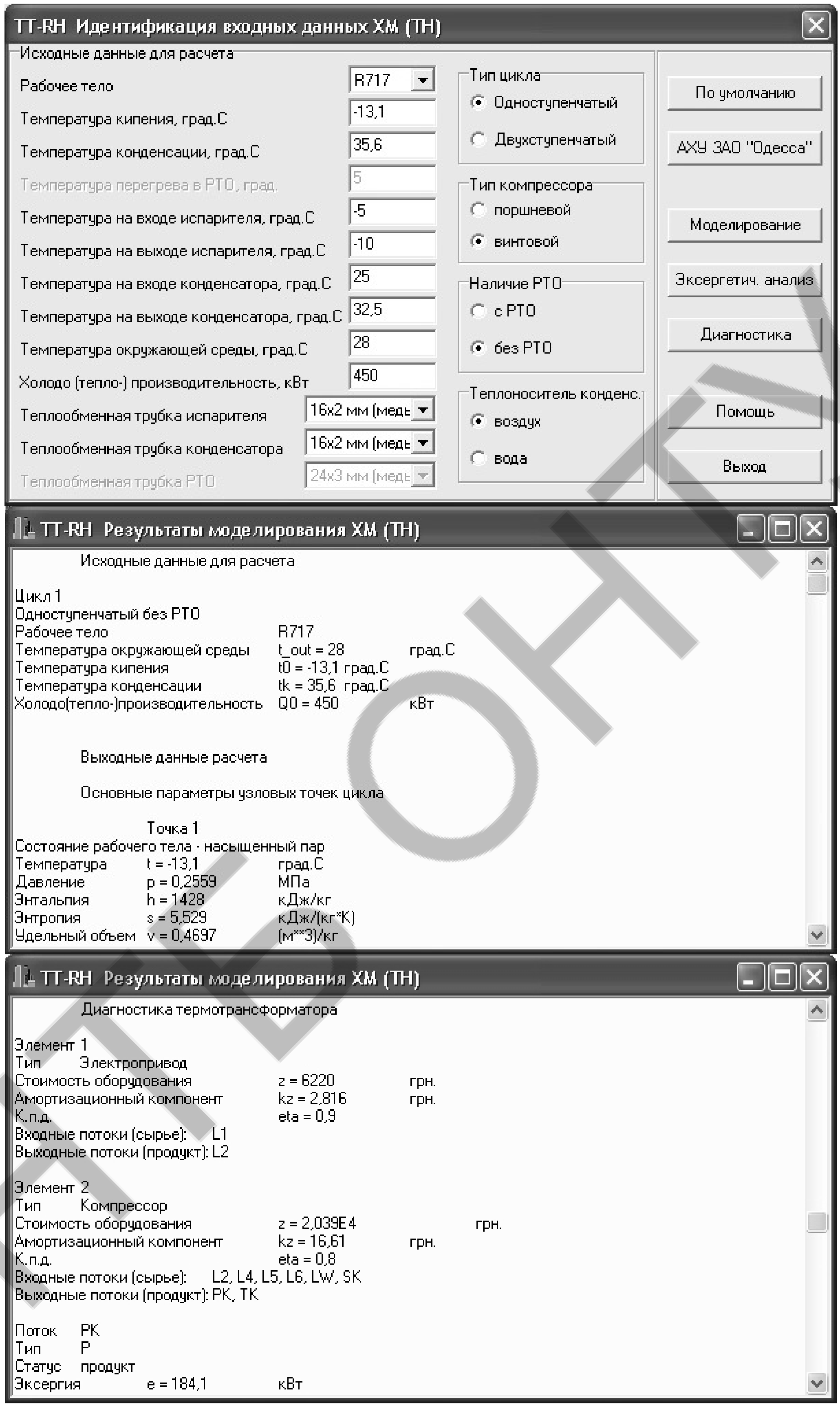

Fig. 2. Working with an open TT–RH system

1.3. Formalization of the scheme of a single-stage vapor compression refrigerating unit

The formalized diagram of a single-stage vapor compression unit in the TT-RH system is shown in Fig. 3. In the analyzed model, two modules: "Input of the working body" and "Output of the working body" are designed to assign flow parameters, respectively at the entrance, i.e. in the place of a virtual break in the closed circuit (the place of connection of the capacitor and the choke is selected) and obtaining the parameters at the output. Thus, by varying the values of the input parameters of the "Input working body" module, it is possible to study the behavior of the installation under different operating conditions for different working bodies. The thermodynamic calculation of the installation according to this model is carried out in accordance with the algorithm that implements the sequential calculation of the component modules with the given values of the cycle parameters.

2. FORMATION OF MODULES OF OPEN SYSTEM TT–RH

2.1. Formation of modules of the first level

modeling of steam compression TT according to the RH scheme of the first level models are selected as autonomous and ultimately determine the choice of thermodynamic cycle, structure and composition of TT elements.

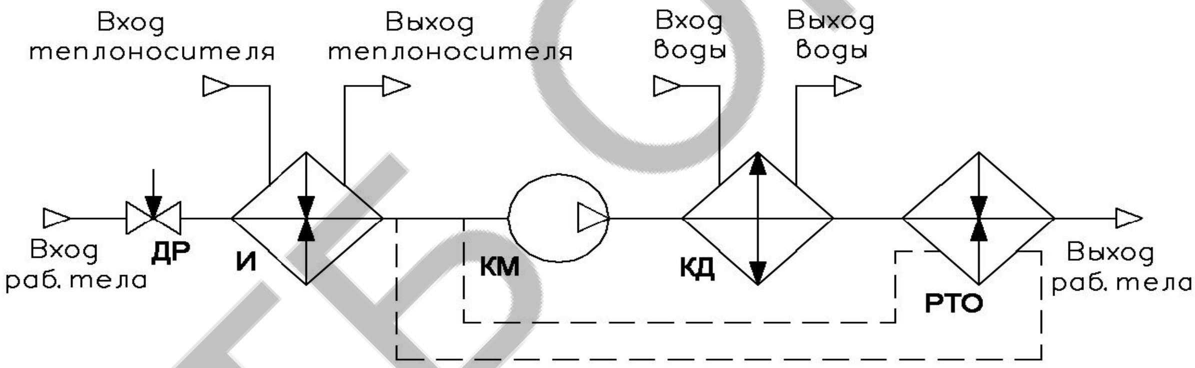

Fig. 3. A formalized scheme of a single-stage vapor compression plant in an open TT-RH system

IN modules of this level are given output data:

cooling capacity Q_o, kW;

boiling point in the cycle $t_O, {}^oC$;

condensation temperature in the cycle $t_K, {}^oC$;

temperature of the cooling coolant, initial $t_{w_1}, {}^oC$;

temperature of the cooling coolant, final $t_{w_2}, {}^oC$;

the initial temperature of the cooling medium $t_{s_1}, {}^oC$;

the temperature of the cooling medium is final $t_{s_2}, {}^oC$.

In the dialogic open system of TT-RH, the modules of the second, fourth and tenth levels address the modules of the first level.

2.2. Formation of modules of the second level

2.2.1. Module "Typical thermodynamic processes in a single-stage vapor compression unit"

The basis of this module is thermodynamic processes during the calculation of the specific thermal characteristics of the cycle.

For this purpose, the output data in the modules of the first level are pre-set: t_o, t_K, $Q_o\,(Q_к)$; in modules of the third level - by the working body and the amount of overheating in the regenerative heat exchanger $\Delta t_{пг}$, *city*

The scheme and cycle of a single-stage steam compression installation are presented in Fig. 4 and 5. The scheme allows the operation of the installation with and without the RTO.

The compressor (KM) compresses the vapors of the working medium to pressure $p_{\ K}$ (process 1–2) and pumps them into the condenser (KD), in which, due to heat exchange with the cooling medium, the aggregate state of the working medium changes

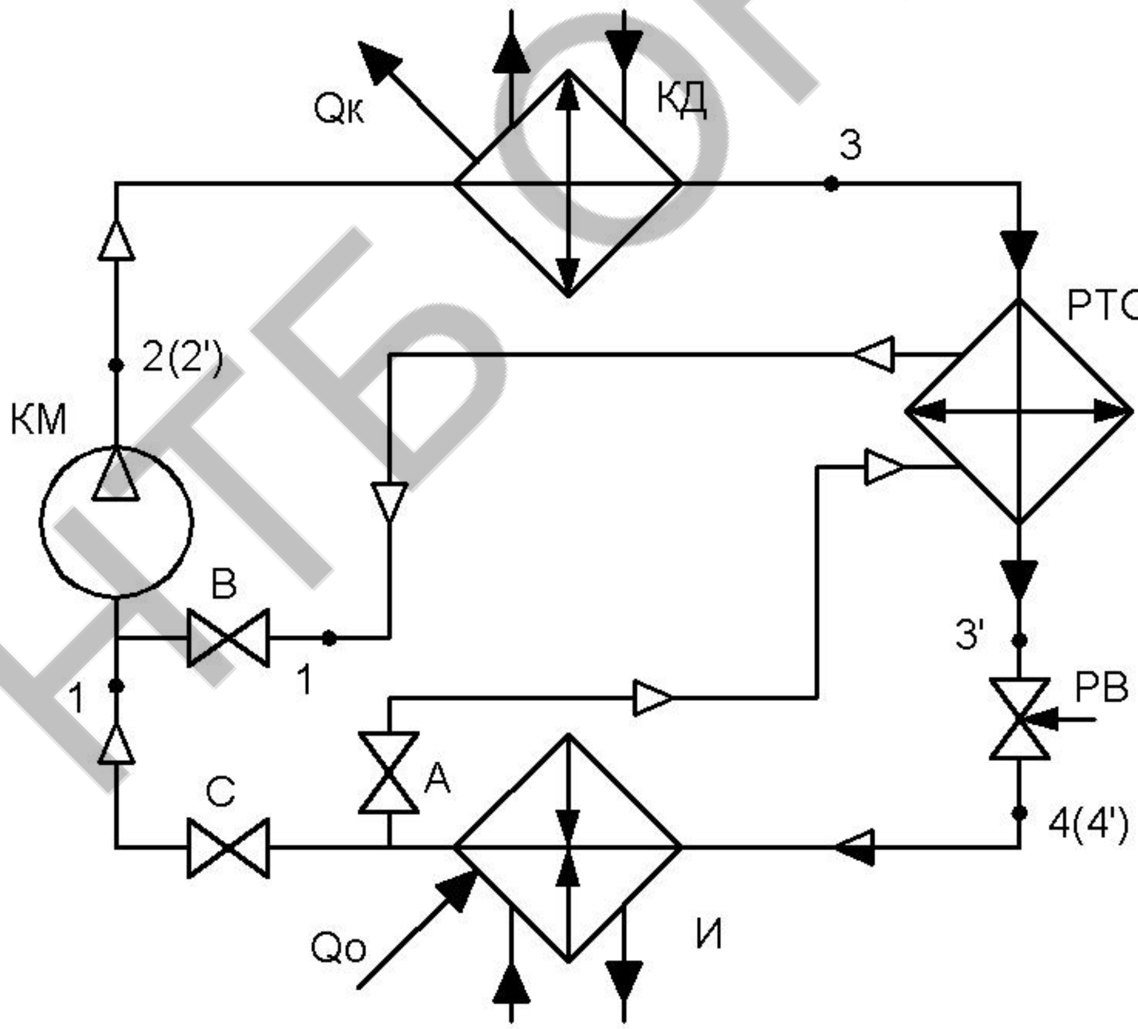

Fig. 2.4. Scheme of a single-stage steam compression installation

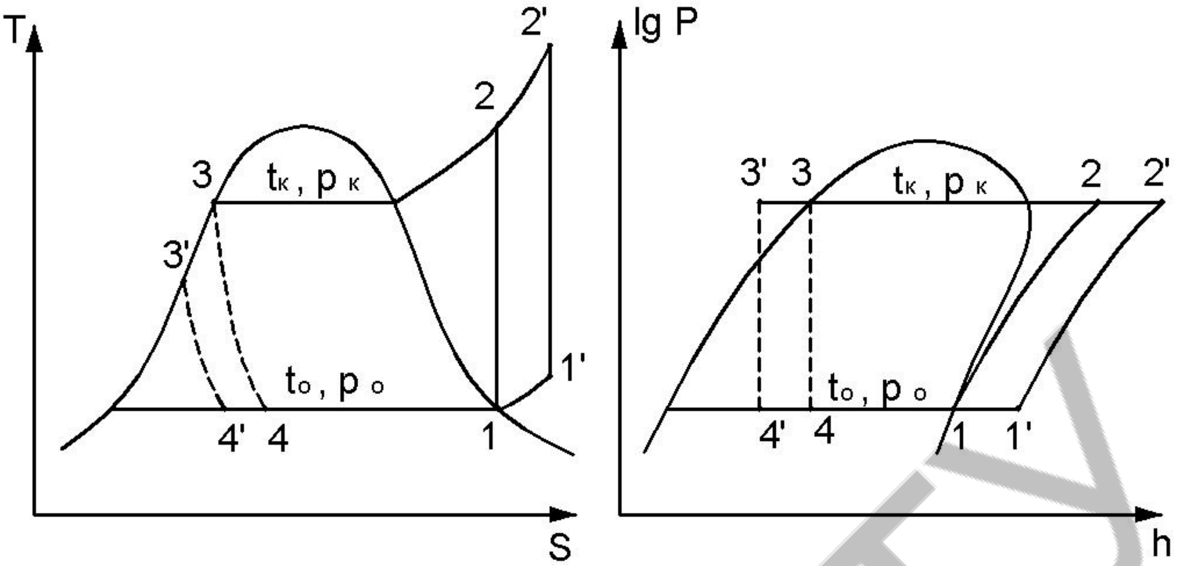

Fig. 5. The cycle of a single-stage vapor compression plant.

bodies - condensation (process 2-3). Then, the condensate is throttled in the control valve (RV) to the pressure p_o (process 3-4) and in the form of a vapor -liquid mixture enters the evaporator (B), where it boils due to heat removal from the object being cooled (process 4-1). The vapors formed are sucked off by the compressor (KM) and the cycle is closed.

According to the above schemes and installation cycles, the parameters at the nodal points of the cycles are determined depending on the selected working body. Calculation of the main thermal characteristics RM was carried out according to known methods.

2.3. Formation of modules of the third level

To carry out engineering calculations and scientific research, it is necessary to have information about the thermodynamic and thermophysical properties of the working body. Designing in an interactive mode of operation causes certain difficulties when using standard tables and diagrams of the properties of substances common in engineering assessments of the thermodynamic efficiency of RM cycles. The most effective is the use of aggregated models of equations of state, knowing the parameters of which, you can reproduce all the necessary thermodynamic information in a minimum time.

2.3.1. Module "Thermodynamic properties of R134a"

According to the Montreal Protocol, signed in 1987, it is envisaged to reduce the production and use of ozone-depleting refrigerants and mixtures based on them [27]. In this regard, the alternative refrigerant R134a (1, 1, 1, 2 – tetrafluoroethane / CH_2FCF_3), belonging to the ethane group of derivatives, was used as a simulating working medium in TT cycles. The use of R134a instead of R12 allows, in the case of high boiling temperatures, to increase the energy efficiency of thermotransformers and, under certain conditions, even at relatively low boiling temperatures [17].

The main parameters of 134a [1,4,..,17}: normal boiling point, $T_{н.к} = 247{,}02\,°К$; molecular weight, $M = 102{,}032$ *кг / кмоль* ; critical temperature, $T_{кр} = 374{,}18\,°К$; critical pressure $p_{кр} = 4{,}059$ *МПа*; critical density, $\rho_{кр} = 508$ *кг / м*^{3} . To calculate the thermodynamic properties of the working medium chosen for modeling R134a, empirical equations from the works of Kabelec and Cleland [24] were used as the most adapted for computer calculations.

Saturation pressure by saturation temperature

$$p_{sat} = \exp\left(21{,}51297 - \frac{2200{,}9809}{246{,}61 + t_{sat}}\right). \qquad (2.1)$$

Saturation temperature saturation pressure

$$t_{sat} = \left(\frac{-2200{,}9809}{\ln(p_{sat}) - 21{,}51297}\right) - 246{,}61, \qquad (2.2)$$

where t_{sat} is the saturation temperature ^oC,; p_{sat} - t isk of saturation, *Pa*.

The enthalpy of a saturated or supercooled liquid depends on its temperature, moreover

$$\Delta t_в = t_{sat} - t_L, \qquad (2.3)$$

where $\Delta t_в$ is the degree of supercooling of a saturated liquid, *degrees*;

t_L - temperature of supercooled liquid, ^oC.

Provided that $\Delta t_в \geq 0$, the enthalpy will be

$$h_L = 200000 + 1335{,}29\, t_L + 1{,}70650\, t_L^2 + 7{,}6741 \cdot 10^{-3} t_L^3, \qquad (2.\ 4)$$

where h_L = 200,000 *J/kg* at 0 ^oC.

The enthalpy of saturated steam is described by a polynomial dependence

$$h_{i1} = 249455 + 606{,}163\, t_{sat} - 1{,}05644\, t_{sat}^2 - 1{,}82426 \cdot 10^{-2} t_{sat}^3; \qquad (2.5)$$

$$h_V = h_{i1} + 149048,\ J/kg. \qquad (2.\ 6)$$

The enthalpy of superheated steam is calculated using a polynomial equation depending on the amount of steam superheat

$$\Delta t_s = t_s - t_{sat}, \qquad (2.\ 7)$$

where Δt_s - The amount of steam overheating, *degrees* ; t_s - Temperature of superheated steam, ^oC.

Then the enthalpy:

$$h_{i2} = h_{i1}(1 + 3{,}48186 \cdot 10^{-3} \Delta t_s + 1{,}6886 \cdot 10^{-6} \Delta t_s^2 + 9{,}2642 \cdot 10^{-6} \Delta t_s t_{sat} -$$

$$- 7{,}698 \cdot 10^{-8} \Delta t_s^2 t_{sat} + 1{,}7070 \cdot 10^{-7} \Delta t_s t_{sat}^2 - 1{,}2130 \cdot 10^{-9} \Delta t_s^2 t_{sat}^2); \quad (2.9)$$

$$h_s = h_{i2} + 149048,\ J/kg. \qquad (2.9)$$

The specific amount of steam on the saturation line is described by the following polynomial:

$$v_V = \exp\left(-12{,}4539 + \frac{2669{,}0}{273{,}15 + t_{sat}}\right)(1{,}01357 + 1{,}06736 \cdot 10^{-3} t_{sat} -$$

$$-9{,}2532 \cdot 10^{-6} t_{sat}^2 - 3{,}2192 \cdot 10^{-7} t_{sat}^3),\ м^3 / кг. \qquad (2.11)$$

The specific amount of superheated steam is calculated using the following polynomial depending on the amount of superheated steam Δt_s (see equation 2.7):

$$v_s = v_V (1 + 4{,}7881 \cdot 10^{-3} \Delta t_s - 3{,}965 \cdot 10^{-6} \Delta t_s^2 + 2{,}5817 \cdot 10^{-5} \Delta t_s t_{sat} -$$

$$-1{,}8506 \cdot 10^{-7} \Delta t_s^2 t_{sat} + 8{,}5739 \cdot 10^{-7} \Delta t_s t_{sat}^2 - 5{,}401 \cdot 10^{-9} \Delta t_s^2 t_{sat}^2),\ м^3 / кг. (2.12)$$

Enthalpy difference during adiabatic (isentropic) compression of steam in a compressor without overheating

$$\Delta h = \frac{c}{c-1} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{c-1}{c}} - 1\right],\ Дж / кг, \qquad (2.1\ 1)$$

where p_1 - t is suction, Pa ; p_2 - t injection pressure, Pa ;

v_1 – specific volume of steam at suction $м^3 / кг$ (see (2. 09);

c - n exponent of compression polytropy.

When calculating the index of compression polytropy, the difference in saturation temperatures is taken into account

$$\Delta t_c = t_{sat2} - t_{sat1},\ city\ \_ \qquad (2.12)$$

Then

$$c_{i1} = 1{,}06469 - 1{,}6907 \cdot 10^{-3} t_{sat1} - 8{,}560 \cdot 10^{-6} t_{sat1}^2 - 2{,}135 \cdot 10^{-5} t_{sat1} \Delta t_c -$$

$$-6{,}1730 \cdot 10^{-7} t_{sat1}^2 \Delta t_c + 2{,}0740 \cdot 10^{-7} t_{sat1} \Delta t_c^2 + 7{,}720 \cdot 10^{-9} t_{sat1}^2 \Delta t_c^2 -$$

$$-6{,}103 \cdot 10^{-4} \Delta t_c; \qquad (2.15)$$

$$c = c_{i1}.$$

The difference in enthalpies during isentropic compression of steam in the compressor, taking into account their overheating on suction, is determined by equation (2.11).

The index of compression polytropy is preliminarily calculated, taking into account the amount of steam overheating Δt_s, as in equation (2.6)

$$c = c_{i1} (1 + 1{,}1757 \cdot 10^{-3} \Delta t_s - 1{,}814 \cdot 10^{-5} \Delta t_s^2 + 4{,}121 \cdot 10^{-5} \Delta t_s t_{sat1} -$$

$$-8{,}093 \cdot 10^{-7} \Delta t_s^2 t_{sat1}), \qquad (2.16)$$

where c_{i1} - from the equation (2.12).

The entropy of the working body, depending on its temperature and density, can be generally calculated as follows:

$$S(T, v) = S_o + \int_{T_o}^{T} c_v^0(T) \frac{dT}{T} + \int_{\infty}^{v} \left[\left(\frac{\partial p}{\partial T}\right)_v - \frac{R}{v}\right] dv + R \ln\left(\frac{v}{v_o}\right), \qquad (2.1\ 4)$$

where $$c_v^0(T) = C_p^0(T) - R \tag{2.15}$$

The entropy of a saturated and superheated R134a vapor can be determined by the following empirical equation [16]:

$$S(T_r, v_V) = 0{,}7808 + (c_1 - 0{,}081489)\ln T_r + c_2 T_r + \frac{c_3}{2} T_r^2 + 0{,}081489 \ln v_V + \frac{1}{T_{cr}} \times$$

$$\times \left\{ \frac{1}{6} \frac{B}{T_r^{8/3} v_V} - \frac{1}{20} \left[C_1 - 3{,}2 C_2 \exp(-3{,}2 T_r) \right] \frac{1}{v_V^2} + \frac{1}{45} \frac{D}{T_r^{5/3} v_V^3} \right\}, кДж/(кг \cdot К), (2.19)$$

where $T_r = \dfrac{T}{T_c}$ is the specified temperature; T_c=374.205^oK – critical temperature R 134 a ; c_1=0.19014; c_2=0.94817; c_3=-0.177953; B= -821.538;
D= -274.861; C_1=318,032; C_2=14052.44; v_V - p ith volume of saturated steam, $дм^3/кг$; S_o=1.00 *kJ/(kg K)* for 0°C.

The entropy of a saturated liquid is defined as follows:

$$S'_L(T_r) = S(T_r, v''_V) - \frac{1}{10}\left[v''_V(T_r, p_s) - v'_L(T_r) \right] \frac{1}{T_c} \frac{dp_s}{dT_r}, \tag{2.16}$$

where $$\frac{dp_s}{dT_r} = -p_s(T_r)\left[\frac{a_1}{T_r^2} + a_2 \frac{(1-T_r)^{1/3}}{T_r}\left(\frac{1}{3} + \frac{1}{T_r} \right) + a_3 \frac{(1-T_r)^2}{T_r}\left(2 + \frac{1}{T_r} \right) \right],$$

(2.17)

where a_1= -7.7842; a_2=1.4040; a_3= -3.11419.

To determine the entropy in the two-phase region, the author developed a polynomial equation by approximating the array of tabular data by the least squares method, which, depending on the enthalpy and saturation temperature in the regions t_s =–30...95 $^\circ C$ and h=160...350 *kJ/kg,* allows obtaining the entropy value with a maximum error of up to ±0.001 *kJ/(kg ·K)* :

$$S = (F_{11} t_s^2 + F_{12} t_s + F_{13}) h + F_{14} t_s + F_{15}, кДж/(кг \cdot К), \tag{2.18}$$

where $F_{11} = 0{,}4464 \cdot 10^{-7}$ $F_{12} = -13{,}512 \cdot 10^{-6}$; $F_{13} = 36{,}7 \cdot 10^{-4}$;

$F_{14} = 0{,}262 \cdot 10^{-2}$; $F_{15} = 0{,}265$.

It is also important to correctly determine the parameters at the end of the isentropic compressor compression process. The author proposed such an analytical method.

Fig. 8 shows a simplified representation of the isentropic compression process in compressor 1–2. The amount of compression work in the case is determined as:

$$l_{ад} = \Delta h = h_2 - h_1, \tag{2. 19}$$

from where $$h_2 = h_1 + \Delta h \tag{2.20}$$

It also compresses work $l_{ад}$ can be determined analytically by equation (2.9), then

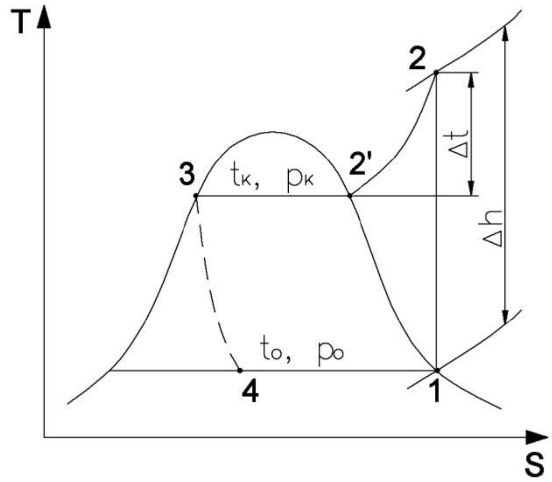

Fig. 8. Determining the temperature at the end of compressor compression

$$h_2 = h_1 + \frac{c}{c-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{c-1}{c}} - 1 \right]. \quad (2.21)$$

In equation (2.21), all parameters are known, except for the index of compression polytropy c, which is calculated using equations (2.11) or (2.10). Now solving equation (2.19) , taking into account the parameters of equations 2.11) or (2.12), with respect to *Δt* we get:

$$\Delta t_{1,2} = -\frac{1}{2}\frac{A}{B} \pm \sqrt{\left(\frac{1}{2}\frac{A}{B}\right)^2 - \frac{C - h_2 + 149048}{CB}}, \quad (2.22)$$

where

$$3{,}48186 \cdot 10^{-3} + 9{,}2642 \cdot 10^{-6} t_{sat} + 1{,}7070 \cdot 10^{-7} t_{sat}^2 = A; \quad (2.23$$

$$1{,}6886 \cdot 10^{-6} - 7{,}698 \cdot 10^{-8} t_{sat} - 1{,}2130 \cdot 10^{-9} t_{sat}^2 = B; \quad (2.24)$$

$$249455 + 606{,}163\, t_{sat} - 1{,}05644\, t_{sat}^2 - 1{,}82426 \cdot 10^{-2} t_{sat}^3 = C. \quad (2.2\ 5)$$

The real root Δt must meet the conditions: $0 \le \Delta t < t_c$.

Another root has no physical meaning.

And, in the end, the temperature after isentropic compression in the compressor will be determined as:

$$t_2 = t_{sat2} + \Delta t,\ °C. \quad (2.\ 6)$$

Determination of the final temperature of the working body after the compressor during the polytropic compression process taking into account the adiabatic (isentropic) efficiency. is carried out in a similar way. At the same time, it is important to correctly define first $h_2,\ p_2, t_{sat2}, v_{sat2}$:

It is also important to correctly determine the parameters at the point of the cycle after subcooling the liquid working medium in an industrial refrigerator or a regenerative heat exchanger.

The enthalpy at point 3' is determined from the heat balance of RTO:

$$h_{3'} = h_3 - (h_{1'} - h_1),\ J/kg. \quad (2.\ 27\ )$$

Also, the enthalpy at point 3' can be calculated using equation (2.4). Solving the equation (2.4) with respect to the temperature t_L taking into account h_L, neglecting, at the same time, the fourth term of this equation as an infinitesimally small value, which is approximately 0.004% compared to the smallest third term, we obtain:

$$t_{L1,2} = -\frac{1}{2}\left(\frac{1335{,}29}{1{,}7065}\right) \pm \sqrt{\left[\frac{1}{2}\left(\frac{1335{,}29}{1{,}7065}\right)\right]^2 - \frac{200000 - h_L}{1{,}7065}}\,. \quad (2.\ 28\ )$$

The real root of this equation must meet the conditions: $t_o < t_L \le t_к$. Another root has no physical meaning.

To determine this temperature, the author also developed the following polynomial equation with a determination error of ±0.05 $^\circ C$ in the temperature range of saturated and supercooled working fluid (–30...95) $^\circ C$ and enthalpies (160...350) *kJ/kg*

$$t_s = B_5 h_L^2 + B_6 h_L + B_7,\ ^\circ C \quad (2.\ 29\ )$$

where $B_5 = -10{,}3675 \cdot 10^{-4}$_ $B_6 = 1{,}1942$; $B_7 = -197{,}151$.

2.3.3. Module "Thermophysical properties of R 134a"

Density of the liquid working medium [24]

$$\rho_R = 1 + C_1(1 - T_R)^{1/3} + C_2(1 - T_R)^{2/3} + C_3(1 - T_R) + C_4(1 - T_R)^{4/3}, \quad (2.55)$$

where $\rho_R = \dfrac{\rho_{\text{ж}}}{\rho_c}$, $кг/м^3$- The given density; $\rho_c = 515{,}3$,$кг/м^3$- Critical density R 134a [28,29,36]; $T_R = \dfrac{\bar{T}_{\text{ж}}}{T_{cr}}$- The specified temperature; $T_{cr} = 374{,}205\ ^oK$- Critical temperature R 134a [26 , 27 , 29]; $C_1 = 1{,}7127$; $C_2 = 1{,}045143$; $C_3 = -0{,}2276586$; $C_4 = 0{,}1836428$- coefficients.

$$\rho'_{\text{ж}} = \rho'_R \cdot \rho_c,\ кг/м^3. \quad (2.56)$$

Specific heat capacity of liquid, *kJ/(kg K)* [29]

$$c_{p\text{ж}} = M_o + M_1(1 - T_R)^{1/9} + M_2(1 - T_R)^{2/9} +$$
$$+ M_3(1 - T_R)^{3/9} + M_4(1 - T_R)^{6/9}\ , \quad (2.57)$$

where $M_o = 395{,}19033$; $M_1 = -1588{,}637$; $M_2 = 2233{,}8111$; $M_3 = -1120{,}361$; $M_4 = 81{,}256634$.

The coefficient of thermal conductivity of the liquid [29]

$$\lambda_{\text{ж}} = j_o + j_1 t,\ W/(m \cdot K), \quad (2.58)$$

where $j_o = 94{,}21 \cdot 10^{-3}$_ $j_1 = -0{,}42784 \cdot 10^{-3}$.

Coefficient of dynamic viscosity of liquid [29]

$$\ln\left(\frac{\eta_{\text{ж}}}{10^{-3}}\right) = H_o + H_1 t + H_2 t^2 + H_3 t^3,\ Pa \cdot s\ , \quad (2.59)$$

where $H_o = -1{,}29909$; $H_1 = -0{,}0129286$; $H_2 = 4{,}9223 \cdot 10^{-6}$; $H_3 = -1{,}9860 \cdot 10^{-7}$.

Coefficient of thermal conductivity of a vaporous working fluid [29]

$$\lambda_{\Pi} \cdot 10^3 = L_o + L_1 t + L_2 t^2 + L_3 t^3 + L_4 t^4,\ W/(m \cdot K) \tag{2.60}$$

where $L_o = 11{,}84$; $L_1 = 0{,}08305$; $L_2 = 1{,}33741 \cdot 10^{-4}$; $L_3 = 0$; $L_4 = 0$.

Coefficient of dynamic vapor viscosity [29]

$$\eta_{\Pi} \cdot 10^6 = \frac{1}{\zeta \cdot 10^{-5} \sum\limits_{n=0}^{3} I_n \left(\frac{T}{T_c}\right)^n},\ Pa \cdot s, \tag{2.61}$$

where $\zeta = 39721{,}0\,(\Pi a \cdot c)^{-1}$;

$$\zeta = \frac{N_o^{1/3} (T_c \cdot R)^{1{,}6}}{M^{1/2} \cdot P_c^{2/3}}; \tag{2.62}$$

where $N_o = 6{,}023 \cdot 10^{26}$ $кмоль^{-1}$ is Avogadro's number; $R = 8{,}314\ кДж/(кмоль \cdot K)$- Universal gas constant; $M = 102{,}032\ кДж/кмоль$ – molecular weight R 134a; $P_c = 4060000\ H/м^2$- Critical pressure R 134a; $I_o = 4{,}378104 \cdot 10^{-1}$; $I_1 = -3{,}8781499 \cdot 10^{-1}$; $I_2 = 3{,}7866105 \cdot 10^{-1}$; $I_3 = -3{,}29657 \cdot 10^{-3}$.

Specific heat capacity of a vaporous working fluid [29]

$$c_{p_{\Pi}} = D_1 + D_2 t + D_3 t^2 + D_4 t^3 + \frac{D_5}{t};\ kJ/(kg\ K), \tag{2.63}$$

where $D_1 = -5{,}257455 \cdot 10^{-3}$; $D_2 = 3{,}29657 \cdot 10^{-3}$;

$D_3 = -2{,}017321 \cdot 10^{-6}$; $D_4 = 0$; $D_5 = 15{,}8217$.

Density of the vaporous working fluid [23]

$$\frac{1}{\rho_{s_{\Pi}}} = \frac{1}{\rho_{V_{\Pi}}}(1 + 4{,}7881 \cdot 10^{-3} \Delta t_s - 3{,}965 \cdot 10^{-6} \Delta t_s^2 + 2{,}5817 \cdot 10^{-5} \Delta t_s t_o -$$

$$-1{,}8506 \cdot 10^{-7} \Delta t_s^2 t_o + 8{,}5739 \cdot 10^{-7} \Delta t_s t_o^2 - 5{,}401 \cdot 10^{-9} \Delta t_s^2 t_o^2),\ м^3/кг, \tag{2.64}$$

where $\Delta t_s = \bar{t}_{\Pi} - t_o$, ^oC; t_o- t boiling temperature in the cycle, ^oC;

$$\frac{1}{\rho_{V_{\Pi}}} = \exp\left(-12{,}4539 + \frac{2669{,}0}{273{,}15 + t_o}\right) \times$$

$$\times (1{,}01357 + 1{,}06736 \cdot 10^{-3} t_o - 9{,}2532 \cdot 10^{-6} t_o^2 - 3{,}2192 \cdot 10^{-7} t_o^3),\ м^3/кг. \tag{2.65}$$

2.4. Formation of modules of the fourth level

2.4.1. Module "Piston steam compressor"

In R M, compressors are designed to compress and move gas or steam of working bodies.

According to the principle of action, compressors are divided into two groups: volumetric action (piston, screw, various rotary and others) and dynamic action (centrifugal, axial and others). Reciprocating compressors have become the most widely used for installations of various performance.

The formed model of the piston compressor must take into account the factors that affect the operation of a real compressor in contrast to a theoretical one. The influence of many factors affects the actual working processes of the compressor, including the following main ones: 1) presence of dead space; 2) hydraulic losses in the suction and discharge cavities; 3) steam heating on the suction line; 4) heat exchange in the cylinder during compression and reverse expansion; 5) pressure pulsation; 6) leakage through gaps, leaks and piston ring locks; 7) friction in mechanical elements.

Real work processes are noticeably different from theoretical ones. The suction process occurs at a variable pressure lower than the pressure in the suction pipe. The difference in pressure in the nozzle and at the beginning of compression is called suction depression. The injection process takes place under variable pressure. The difference between the final pressure in the cylinder and at the outlet of the compressor is called the compression pressure. The most important factor is the actual performance and capacity of the compressor. The main indicator of the decrease in actual productivity V_∂ compared to the theoretical V_T one is the feed rate λ

$$V_\partial = \lambda \cdot V_T . \tag{2.66}$$

Compressor delivery ratio

$$\lambda = \lambda_c \lambda_{\partial p} \lambda_w \lambda_{пл} . \tag{2.67}$$

The operation of the compressor with valid processes of compression, discharge, reverse expansion and suction corresponds to the indicator power N_i, which is determined by the valid indicator diagram

$$N_i = p_i \cdot V_T , \tag{2.68}$$

where p_i is the average indicator pressure.

The isentropic (adiabatic) capacity of the compressor evaluates the energy efficiency of all its actual working processes.

$$N_{a\partial} = G_{a\partial} \cdot l_{a\partial} = \frac{\lambda V_T l_{a\partial}}{v_1} , \tag{2.69}$$

and the effective power is supplied to the compressor shaft N_e.

The compressor module of a single-stage installation can be provided as a structural model shown in Fig. 9a.

Structural models of compressor modules I and II stages of the two-stage installation are shown in Fig. 9b and 9c, respectively.

2.4.2. Module "Screw oil-filled steam compressor

When conducting a thermal calculation of an oil-filled screw

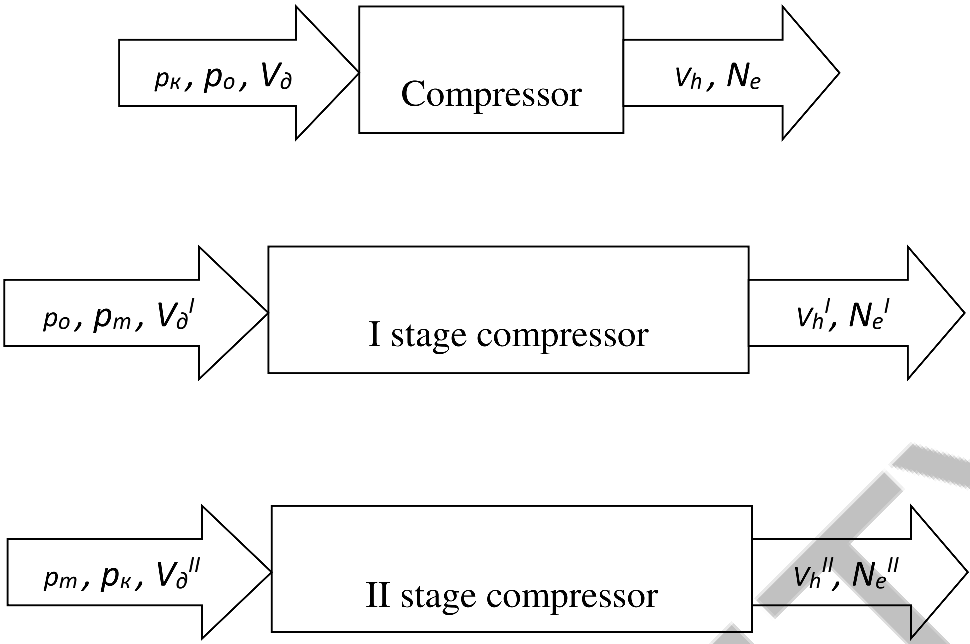

Fig. 9. Structural diagram of the compressor model

output parameters are set for the spring compressor as for the piston compressor. At the same time, the corresponding modules of the second and third levels are initiated.

The actual volumetric productivity is determined V_{∂}.

Theoretical volume performance of the compressor:

$$V_h = \frac{V_{\partial}}{\lambda},\ м^3/с,$$

where λ - the supply coefficient of the oil-filled screw compressor, which, depending on the degree of compression $\pi_к$, can be determined analytically according to the generalized data (at $2 \le \pi_к \le 18$):

$$\pi_к = \frac{p_2}{p_1}; \qquad \lambda = -0{,}0188\pi_к + 0{,}914, \qquad (2.70)$$

All other parameters are identical to the parameters of the piston compressor.

2.4.3. "Condenser" module

The condenser is one of the main heat exchange devices in R M. Liquids and gases are used as coolants for condensers.

most often used in R M, in which the coolant (water) is passed inside the pipes, and the working fluid is condensed on the outside of the pipes in the inter-pipe space. On the basis of the above, a model of the shell-and-tube structure of the condenser with the cooling medium - water was formed. For this, the initial data are pre-set: geometric and design parameters of the heat exchange element of the condenser - the tube. A fragment of the tube with the main geometric dimensions is shown in Fig. 13. Modeling of heat exchange elements of any standard size is possible.

The outputs used in this module are identified in the first and second level modules: G_a, $Q_к$, $t_к$, t_{w_1}, t_{w_2}.

The capacitor is calculated according to known methods.

hydromechanical calculation of the capacitor forms the basis of the modules of the seventh level.

The structural model of the capacitor module is shown in Fig. 10.

2.4.4. "Evaporator" module

In R M, the evaporator is one of the main heat exchange devices and is designed to remove heat at low temperature from the environment or from the coolant. Water and water-salt solutions based on brine $NaCl$ are mainly used as heat carriers $CaCl_2$.

Most often in R M, shell-and-tube evaporators of the submerged type are used, inside the pipes of which the coolant passes, and the working medium boils inside the casing of the device, cooling the coolant at the same time.

Let's create a model of a flooded shell-and-tube evaporator with boiling of the working fluid in the intertube space.

Initial parameters from the modules of the first and second levels are pre-entered: Q_o, G_a, t_o, t_{s1}, t_{s2}. The design characteristics of the heat exchange element of the evaporator - the tube - are also pre-set (the same tubes are used in submerged shell-and-tube evaporators as in submerged shell-and-tube condensers). The choice is made based on the data of the modules of the fifth level. An aqueous solution of calcined salt ($CaCl_2$) with a composition of 23.8 *kg/kg was chosen as the heat carrier.* Its thermophysical properties depend on the average temperature $\bar{t}_s$ in the form of appropriate modules presented in the sixth level.

transfer coefficient

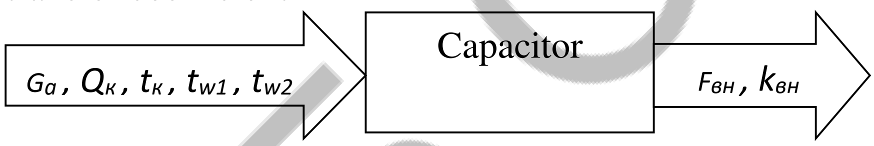

Fig. 10. Structural diagram of the capacitor model

and the size of the heat exchange surface of the evaporator is carried out in the same way as for the condenser (clause 2.4.3).

The hydromechanical calculation of the evaporator forms the basis of the modules of the seventh level.

The structural model of the evaporator module is shown in Fig. 11.

2.4.5. Module "Regenerative heat exchanger"

PTOs belong to R M auxiliary devices. In them, heat exchange takes place between the liquid working fluid that enters the throttle valve after the condenser, and the vapor-like agent that leaves the evaporator. For modeling, a vertical type of shell-and-tube RTO, which has the largest heat exchange surface, was chosen.

The initial data for modeling are modules of the first and second levels, as well as:

- heat load RTO, Q_{PTO}, kW;
- the temperature of the liquid working medium at the entrance to the RTO, $t_{жс1}$, ^oC;

- the temperature of the supercooled liquid working fluid at the outlet of the RTO, $t_{ж2}$, ^{o}C;

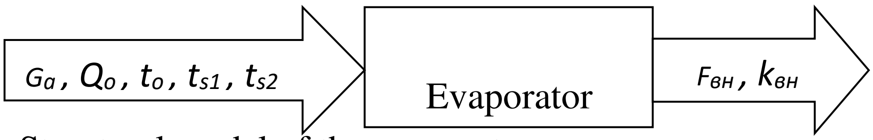

Fig. 11. Structural model of the evaporator

- the temperature of the saturated vapor of the working medium at the entrance to the RTO, t_{n1}, ^{o}C;
- the temperature of the superheated vapor of the working medium at the outlet of the RTO, t_{n2}, ^{o}C.

The main structural dimensions characterizing the heat transfer surface of the device are set: heat exchange elements - copper tubes, smooth; $d_{вн}$ - Inner diameter of the tube; $d_{н}$ - Outer diameter of the tube; n - Number of tubes connected in parallel; $D_{б}$ - The outer diameter of the core (displacer), on which the heat exchange tubes are wound; m – the number of winding rows; S_1 - The distance between adjacent rows along the center of the pipes; S - The thickness of the distance plates. Along the core, the tubes are wound without a gap.

Thermophysical properties of refrigerants at their average temperatures are determined using modules of the third level.

Calculation of RTO is carried out according to known methods.

Hydraulic calculations of RTO in the pipe and inter-pipe space are carried out with the help of the corresponding modules of the seventh level.

The structural model of the regenerative heat exchanger module is shown in Fig. 12.

2.5. Formation of modules of the fifth level

Modules of the fifth level are a description of the geometric and design parameters of the heat exchange elements of the main and auxiliary devices [21, 24, 27, 29], as well as a formalized series of pipes

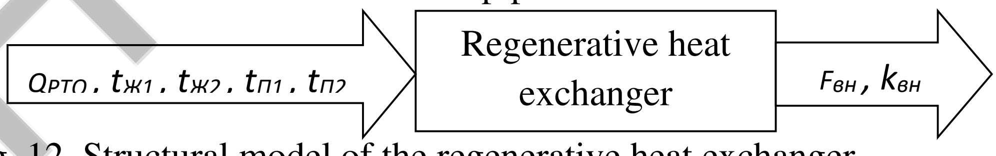

Fig. 12. Structural model of the regenerative heat exchanger for the etiquette of RTO.

2.5.1. Module "Heat exchange tubes"

The main geometric parameters of the heat exchange tube are shown in Fig. 13. Conventional designations of tube parameters are as follows:
$d_{вн}$ - Inner diameter of the pipe, m; d_o - Diameter of the pipe along the circumference of the depressions, m; $d_н$ - Diameter of the pipe around the protrusions, m; S_p - Rib

step, m ; $F'_н$ - Outer surface, $м^2/м$; $F'_{вн}$ - Inner surface, $м^2/м$; $\varphi = F'_н / F'_{вн}$ - Rib coefficient; δ_p - Rib thickness, m ; h_p - Rib height, m ; $h_p = \dfrac{d_н - d_o}{2}$; λ_p - Coefficient of thermal conductivity of the rib, *W / (m K).*

According to the above designations, the size range of heat exchange tubes is integrated into the TT – RH modular system.

2.5.2. The module "Pipes for communication to RTO ".

Hot-rolled seamless steel pipes (D ST U 8732-70).

Depending on the purpose, the pipes are supplied in groups (D ST U 8731-66). The standard size of the corresponding pipes for the PTO promises is integrated into the TT - RH modular system.

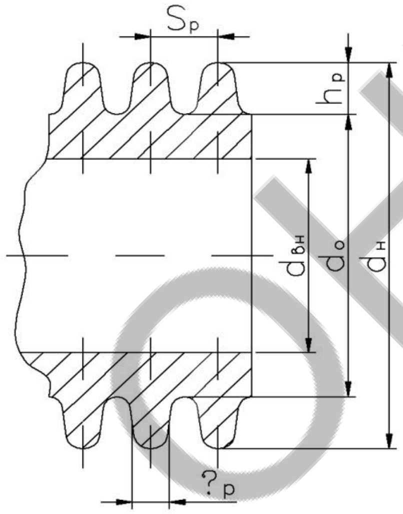

Fig. 13. The main geometric parameters of the heat exchange tube

2.6. Formation of modules of the sixth level

The modules of the sixth level include the thermophysical properties of coolants used in the main R M devices: water and brine (of different chemical composition), as well as dry air (as a coolant for an air condenser). For better algorithmization of the model on the computer, all thermophysical properties of the coolant were processed by the method of least squares and presented in an analytical form.

2.6.1. Module "Thermophysical properties of water"

Heat carrier: water. Temperature range 0...100 ^oC [3,5].

Heat capacity, *kJ/(kg ·K)*

$$c_{pw} = 0{,}1647 \cdot 10^{-4} t_w^2 - 0{,}1488 \cdot 10^{-2} t_w + 4{,}212. \qquad (2.7\ 0)$$

Density , $кг/м^3$

$$\rho_w = -0{,}356 \cdot 10^{-2} t_w^2 - 7{,}09 \cdot 10^{-2} t_w + 999{,}9. \qquad (2.71)$$

Coefficient of kinematic viscosity, $м^2/c$

$$v_w \cdot 10^6 = 0{,}1931 \cdot 10^{-3} t_w^2 - 3{,}229 \cdot 10^{-2} t_w + 1{,}682. \qquad (2.72)$$

Prandtl number

$$\Pr_{жсw} = 0{,}1623 \cdot 10^{-2} t_w^2 - 0{,}263\, t_w + 12{,}409. \qquad (2.73)$$

Thermal conductivity coefficient, *W/(m ·K)*

$$\lambda_w = -0{,}708 \cdot 10^{-5} t_w^2 + 0{,}1929 \cdot 10^{-2} t_w + 0{,}561. \qquad (2.74)$$

2.6.2. Module "Thermophysical properties of an aqueous solution $CaCl_2$"

The heat capacity of the brine $CaCl_2$ composition is 23.8 *kg/kg* depending on $\bar{t}_s$ [22]:

$$c_s = 4{,}1868\,[c_o + 0{,}19 \cdot 10^{-2} \bar{t}_s (\sqrt[4]{1 - \lambda_{15}} - 0{,}409)],\ KJ/(kg \cdot K), \qquad (2.75)$$

where c_o=0.7 *kcal/kg ·degrees* – the heat capacity of brine of the specified composition at $0^o C$;

λ_{15}=0.558 *W/m K* is the thermal conductivity of brine of the specified composition at $15^o C$.

The values of the densities of the brine of the specified composition, depending on the temperature $\bar{t}_s$, as well as other thermophysical properties λ_s, v_s, and $\Pr_s$ were processed by the method of least squares and presented in an analytical form:

$$\rho_s = 0{,}6264 \cdot 10^{-2} \bar{t}_s^2 - 0{,}4382\, \bar{t}_s + 1227,\ кг/м^3. \qquad (2.7\ 6)$$

Thermal conductivity coefficient of brine

$$\lambda_s = -0{,}1904 \cdot 10^{-5} \bar{t}_s^2 + 0{,}1381 \cdot 10^{-2} \bar{t}_s + 0{,}5383,\ W/m\ To. \qquad (2.77)$$

Coefficient of kinematic viscosity of brine

$$v_s \cdot 10^6 = 0{,}5032 \cdot 10^{-2} \bar{t}_s^2 - 0{,}1319\, \bar{t}_s + 3{,}1222,\ м^2/c. \qquad (2.78)$$

Prandtl number of brine

$$\Pr_{жсs} = 0{,}0527\, \bar{t}_s^2 - 1{,}0129\, \bar{t}_s + 21{,}2. \qquad (2.79)$$

If necessary, you can generate the indicated properties of brine based on $CaCl_2$ any composition, depending on $\bar{t}_s$ and, after appropriate processing, obtain generalized dependences of thermophysical properties, depending on the composition of the brine and its temperature:

$$c_s, \rho_s, v_s, \lambda_s, \Pr_{жсs} = f(\xi, \bar{t}_s).$$

2.6.3. Module "Thermophysical properties of air"

Air temperature range (-30... 50) ^oC [3,5].

Air density

$$\rho = 0{,}1583 \cdot 10^{-4} t^2 - 0{,}475 \cdot 10^{-2} t + 1{,}2932, \quad кг/м^3. \qquad (2.81)$$

Coefficient of kinematic air viscosity

$$\nu \cdot 10^6 = 0{,}249 \cdot 10^{-3} t^2 + 0{,}8227 \cdot 10^{-1} t + 13{,}2929, \quad м^2/c. \qquad (2.82)$$

Coefficient of thermal conductivity of air

$$\lambda \cdot 10^2 = 0{,}83 \cdot 10^{-5} t^2 + 0{,}7724 \cdot 10^{-2} t + 2{,}4343, \quad Вт/(м \cdot К). \qquad (2.83)$$

Heat capacity of air in the temperature range (–20...–1)^oC

$$c_p = 1{,}009, \quad кДж/(кг \cdot К). \qquad (2.84)$$

Heat capacity of air in the temperature range (0...60)^oC

$$c_p = 1{,}005, \quad кДж/(кг \cdot К). \qquad (2.85)$$

2.7. Formation of modules of the seventh level

2.7.1. Module "Hydromechanical calculation of condenser, evaporator, regenerative heat exchanger"

The hydromechanical calculation of the condenser , evaporator, and regenerative heat exchanger is based on the determination of pressure losses due to hydro- and aerodynamic resistances that occur during the movement of working media in the heat exchanger.

2.8. Formation of modules of the eighth level

Modules of the eighth level are catalogs of the main functional elements of RM: compressors, condensers, evaporators and PTO. These modules are autonomous under the conditions of operation of the TT- RH dialogue system. They can be accessed by modules of the ninth level in order to generate the composition RM for the specified operating conditions. The composition of the modules of the eighth level is as follows.

1. Unified range of piston compressors (OST26.03-943-77) [20,21]. The selection criteria are: the volume described by the pistons V_h, $м^3/c$ and the limit pressure difference $(p_к - p_o)$, *MPa.*
2. Horizontal shell-and-tube condensers of cold air [18,19]. The selection criterion is: the area of the heat transfer surface (internal) $F_{вн}$, $м^2$.
3. Evaporators are shell-and-tube horizontal khladon submerged type [18,19]. The selection criterion is: the area of the heat transfer surface (internal) $F_{вн}$, $м^2$.
4. Regenerative refrigerant heat exchangers [18,19]. The selection criterion is: the area of the heat transfer surface (internal) $F_{вн}$, $м^2$.

The parametric series of the relevant industrial equipment are integrated into the TT–RH modular system.

3. CONCLUSIONS ON THE WORK

conclusions can be drawn from the conducted research :

1. an open universal multi-level modular system TT–RH was developed for the formation of thermotechnical models of thermotransformers operating according to the combined RH scheme ;
2. the developed TT-RH system allows generating heat engineering models RM of

arbitrary schemes; calculate the characteristics under the given conditions of changing the parameters; document the results of calculations, store reference and other information;

3. for calculations of thermodynamic and thermophysical properties of R134a, R717, $CaCl_2$, water and air under appropriate conditions, approximation dependencies were developed.

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