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A turbocompressor refrigeration machine characteristics' calculation

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Abstract. A performance analysis method for a centrifugal compressor chiller is developed. The source data used for the analysis are geometric parameters of the compressor stages, heat exchangers, refrigerant properties, and temperature regimes in the condenser and evaporator. The stationary mode is determined by iterative calculation as a point where the performance characteristics of the compressor, condenser, and evaporator cross. The methodology is tested using two refrigerants, R134a and R404a. The calculated performance characteristics for a two-stage chiller are presented.

1. Introduction

Freon vapor compression chillers with a centrifugal compressor are widely used by petrochemical enterprises to chill liquid coolants. Currently, chillers intended for media subject to replacement based on environmental standards continue to be operated. Therefore, when retrofitting such chillers and converting them to modern environmentally friendly refrigerants, the need arises to assess changes in their performance characteristics. The creation of such a methodology is also of theoretical interest in the design and analysis of the efficiency of centrifugal compressor chillers. Basic approaches and methods for calculating centrifugal compressor chillers and gas compressors are described quite extensively in the papers of Yu. Galerkin [1], I. Sukhomlinov [2], G. Den, etc.

2. Subject of study

The subject of study is a chiller comprising a centrifugal compressor with an electric motor and lubrication system, an evaporator, a shell-tube condenser, and an economizer.

The centrifugal compressor is a two-section type, in which both sections are one-stage; the impellers are of a closed type. The multiplier is integrated in the compressor's body.

The chiller operates through a two-stage cycle with dual throttling and partial intercooling (Fig. 1).

The refrigerant compressed in two sections of the centrifugal compressor is fed under condensation pressure in the condenser's tube space. Refrigerant chilling until saturation and its condensation is the result of removing the condensation heat toward the cooling circulated water in the tube space. The liquid refrigerant, in saturated or somewhat overchilled (by 2–3°C) state, flows to the upper chamber of the economizer from where it passes the first throttling through the float-type throttle and then flows to the interstage pressure chamber. Vapors produced in the first throttling are sucked by the second section of the turbocompressor for vapor intercooling between the sections. The liquid is re-throttled in the second throttle until reaching boiling pressure and then goes to the lower part of the tube space of the shell-tube evaporator. In the evaporator, the liquid refrigerant boils, removing heat from the chilled coolant circulating in the evaporator's tubes. Vapor produced by boiling is sucked in by the first section of the turbocompressor.

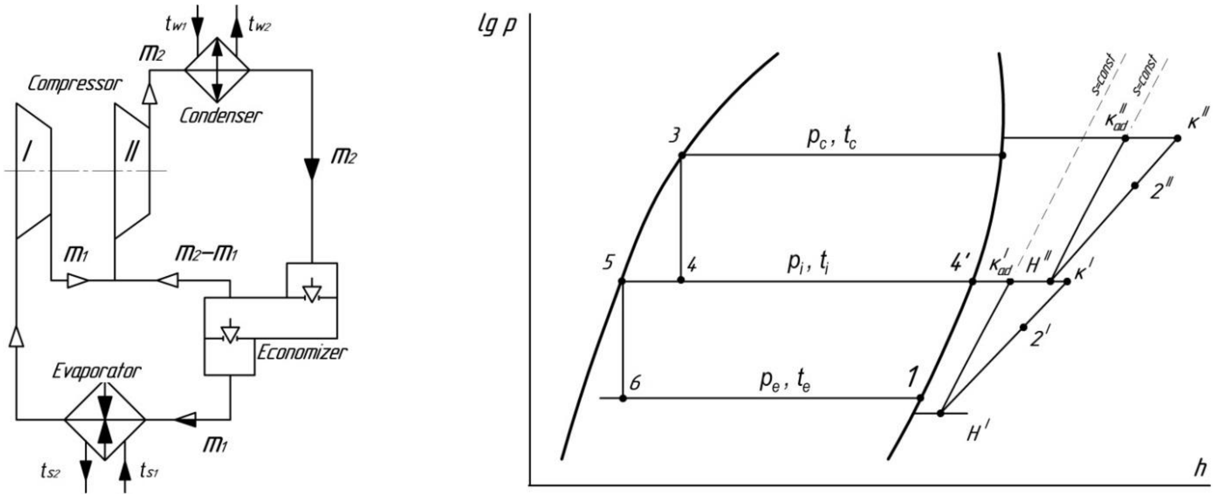


Figure 1. Principal diagram and operating cycle of the chiller

3. Calculation method

A particular feature of performance analysis of the chiller is that the parameters of the refrigerant being sucked in and pressurization of the centrifugal compressor are determined by the temperature and pressure of the refrigerant in the heat exchangers, which in turn rely on the compressor's productivity and the conditions of heat exchange of the refrigerant with the coolant in the evaporator and with the cooling water in the condenser. Therefore, the confirmatory analysis of the entire chiller is of an iterative nature. During analysis, the parameters of the refrigerant in the nodal points of the thermodynamic cycle are adjusted in each iteration. As a result, a working point is determined on the compressor, condenser, and evaporator characteristics, which fulfills the thermal balance of the chiller as a whole.

The initial data for the analysis are the following: temperature t_{w1} and flow rate V_w of cooling water at the inlet of the condenser; temperature t_{s1} and flow rate V_s of the coolant at the inlet of the evaporator; refrigerant; thermal-physical properties of the coolant and cooling water; geometric parameters of the impellers: D_2 is the outer diameter of the impellers, b_1, b_2 is the width of the impellers at the inlet and outlet, respectively, β_1 и β_2 are the blade angles at the inlet and outlet of the impeller, respectively, z_1 и z_2 are the number of blades at the inlet and outlet of the impeller, δ are the thickness of the blades; rotation frequency n of the turbocompressor's rotor; geometric parameters of the heat exchange surfaces of the condenser and evaporator.

The algorithm of the confirmatory analysis of the chiller is reduced to the following basic calculation steps:

1. We enter the initial data.
2. We specify the preliminary values of the temperature of boiling t_{e} and condensation t_{c}.
3. Based on the accepted values of t_{e} and t_{c}, we determine the corresponding pressure of boiling p_e and condensation p_c and carry out confirmatory analysis of the wet part of the compressor. The analysis of the wet part of the compressor is also iterative and carried out using the methodology given in [3]. We preliminarily specify the coefficients of flow φ_{r2} by the stages based on the recommendations depending on angle β_2. During the analysis, the values of φ_{r2} are iteratively refined, the productivity of the compressor's stages and thermal loads upon the heat exchangers in the specified regime (with accepted t_{e} and t_{c}) are determined. The polytropic efficiency η_{pol} of the stages was defined on the basis of the operation and design of centrifugal compressor chillers [4].

4. Based on the initial data for the condenser and the data obtained from the calculation of the thermal load upon the condenser under item 3, we calculate the heat transfer coefficient in the apparatus and refine the value of t_c, at which removal of the required heat with the obtained value of the heat transfer coefficient is possible. If the obtained temperature t_c deviates from the earlier accepted values by more than 0.1°C, we repeat the analysis starting from item 3, and the newly-obtained value of condensation temperature t_c is accepted.

5. Based on the initial data for the evaporator and the data obtained from the calculation of the cooling capacity under item 3, we calculate the heat transfer coefficient in the evaporator and refine the value of t_e, at which removal of the required heat with the obtained value of the heat transfer coefficient is possible. If the obtained value of t_e deviates from the earlier accepted value by more than 0.1°C, we repeat the analysis starting from item 3 including item 4, and the newly-obtained value of boiling temperature $\mathrm{t_e}$ is accepted.

The basic equations of the confirmatory analysis of the turbocompressor are given below.

In the first approximation, an optimal value of the flow coefficient in the first stage $\varphi_{r2}^{\mathrm{I}}$ is accepted for this type of impellers.

Correction to the theoretical coefficient of head by the finite number of blades on the first impeller

$$\mathrm{k_z^I} = 1 - \frac{3{,}14}{\mathrm{z_2^I}}\sin\beta_2^{\mathrm{I}}. \quad (1)$$

Coefficient of theoretical head of the first impeller

$$\varphi_{\mathrm{u2}}^{\mathrm{I}} = \mathrm{k_z^I} - \varphi_{\mathrm{r2}}^{\mathrm{I}} \cdot \mathrm{ctg}\beta_2^{\mathrm{I}}. \quad (2)$$

Peripheral speed of the impeller

$$u_2 = \frac{3{,}14 D_2 \cdot n}{60}. \quad (3)$$

Enthalpy change in the first stage

$$\Delta\mathrm{h^I} = 1{,}05 \cdot \mathrm{u_2^2} \cdot \varphi_{\mathrm{u2}}^{\mathrm{I}}, \quad (4)$$

where 1,05 is the coefficient, accounting for the losses related to disk friction in the stage and internal overfills of refrigerant through labyrinths.

Degree of reaction of the first impeller

$$\Omega^{\mathrm{I}} = 1 - \frac{\left(\varphi_{\mathrm{r2}}^{\mathrm{I}}\right)^2 + \left(\varphi_{\mathrm{u2}}^{\mathrm{I}}\right)^2}{2{,}1 \cdot \varphi_{\mathrm{u2}}^{\mathrm{I}}}. \quad (5)$$

Enthalpy change in the first impeller

$$\Delta\mathrm{h_2^I} = \Omega^{\mathrm{I}} \cdot \Delta\mathrm{h^I}. \quad (6)$$

Adiabatic efficiency of the centrifugal compressor

$$\eta_{ad} = \frac{\Pi^{\frac{k-1}{k}} - 1}{\Pi^{\frac{k-1}{k \cdot \eta_{pol}}} - 1}, \quad (7)$$

where Π is the pressures ratio in the compressor, and k is the heat capacity ratio.

Adiabatic change of the vapor enthalpy in the first stage

$$\Delta\mathrm{h_{ad}^I} = \eta_{\mathrm{ad}} \cdot \Delta\mathrm{h^I}, \quad (8)$$

Vapor enthalpy beyond the first stage at isentropic compression

$$\mathrm{h_{\kappa.ad}^I} = \mathrm{h_{H^I}} + \Delta\mathrm{h_{ad}^I}\,. \quad (9)$$

Actual enthalpy of vapor beyond the first stage at isentropic compression

$$\mathrm{h_\kappa^I} = \mathrm{h_{H^I}} + \Delta\mathrm{h^I}. \quad (10)$$

Enthalpy beyond the first impeller

$$\mathrm{h_2^I} = \mathrm{h_{H^I}} + \Delta\mathrm{h_2^I}. \quad (11)$$

Factor of blockage of the outlet section of the impeller by blades

$$\tau_2^{\mathrm{I}} = 1 - \frac{0{,}5 z_2^{\mathrm{I}} \delta}{\pi D_2 \sin\beta_2^{\mathrm{I}}}. \quad (12)$$

Mass flow rate in the first section (stage)

$$m^{\mathrm{I}} = \frac{\pi D_2 b_2^{\mathrm{I}} \tau_2^{\mathrm{I}} u_2 \varphi_{r2}^{\mathrm{I}}}{v_2^{\mathrm{I}}}. \quad (13)$$

Amount of vapor produced in the first throttling

$$m_v = \frac{m_1 x_4}{1 - x_4}, \quad (14)$$

where x_4 is the degree of dryness of the refrigerant after the first throttling (Fig. 1)

Refrigerant enthalpy at the inlet of the first stage

$$h_{\text{н}^{\mathrm{II}}} = \frac{m^{\mathrm{I}} h_{\text{к}}^{\mathrm{I}} + m_v h_v}{m^{\mathrm{II}}}. \quad (15)$$

Vapor parameters in the compression process in the second stage (process $H^{II} - K^{II}$) are calculated similarly to the vapor parameters in the first stage (process $H^{I} - K^{I}$).

Refined coefficient of flow in the second stage

$$\varphi_{r2}^{\mathrm{II}} = \frac{m^{\mathrm{II}} v_2^{\mathrm{II}}}{\pi D_2 b_2^{\mathrm{II}} \tau_2^{\mathrm{II}} u_2}, \quad (16)$$

If the refined coefficient of flow deviates from its value accepted in the beginning of the calculation of the second stage by an amount exceeding the allowed value, further approximation is required. If the pressure at the end of compression in the compressor $p_{\text{к}}^{\mathrm{II}}$ deviates from the discharge pressure by an amount exceeding the allowed value, the entire analysis must be started over with the first stage with the refined value of coefficient of flow $\varphi_{r2}^{\mathrm{I}}$.

Confirmatory analysis of the evaporator and condenser was performed using dimensionless equations describing heat exchange between the media in apparatuses [3, 5].

Cooling capacity of the compressor

$$Q_{\mathrm{e}} = m^{\mathrm{I}}(h_1 - h_6). \quad (17)$$

Effective power of the turbocompressor

$$N = \frac{1}{\eta_m}\left(m^{\mathrm{I}} \Delta h^{I} + m^{\mathrm{II}} \Delta h^{II}\right) \quad (18)$$

where η_m is the mechanical efficiency.

COP factor is

$$\varepsilon = Q_e / N. \quad (19)$$

This analysis was implemented as a macro in Microsoft Office software suite using the library for calculating the properties of refrigerants programmed by the Department of Energy Engineering, Technical University of Denmark. The calculations have been done according to the block-diagram which is presented on fig. 2.

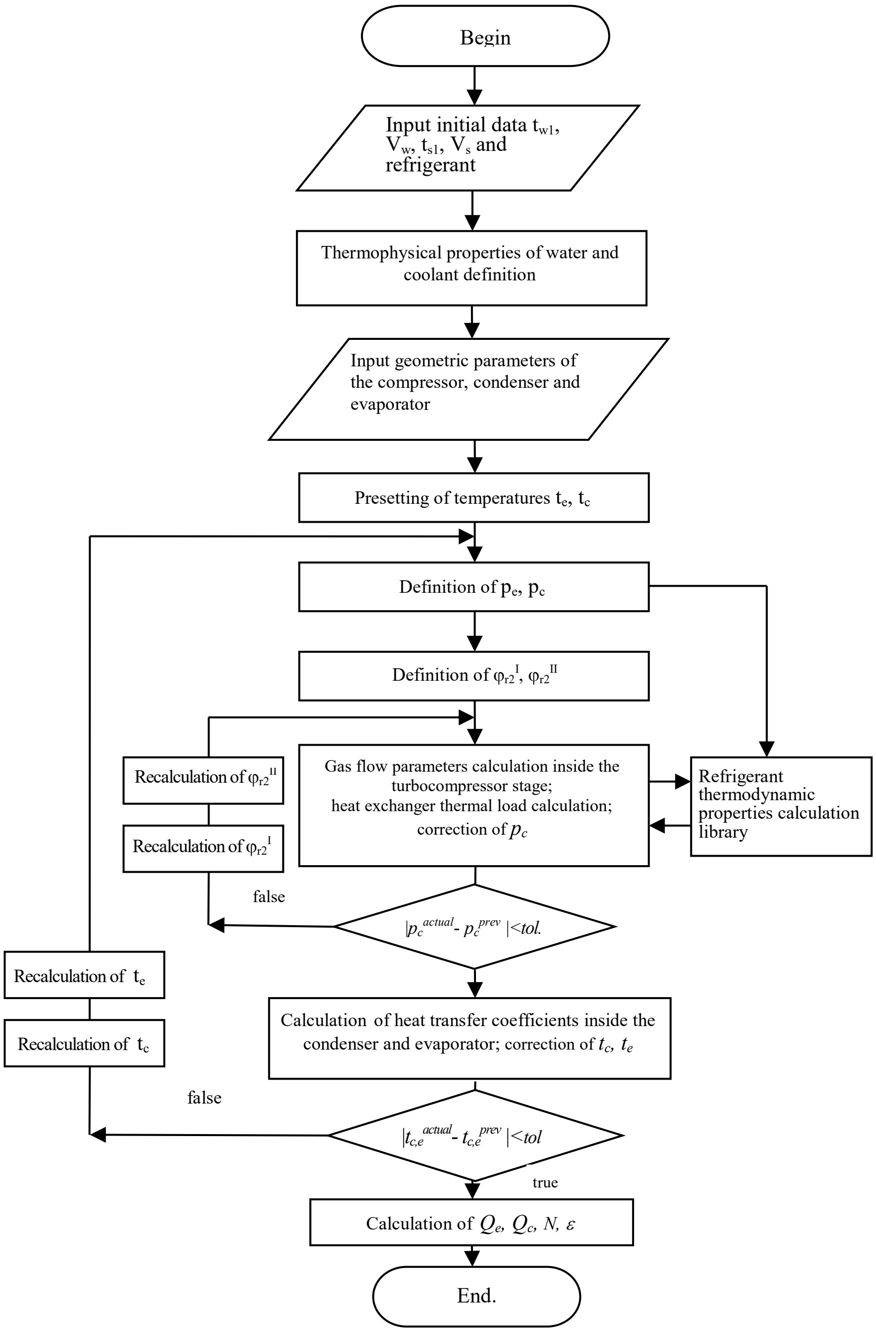
Presented refrigeration machine working conditions are characterized by the peripheral Mach number of 1…1,25. Those values are acceptable for the refrigeration centrifugal compressors.

4. Results of performance analysis

The calculations were carried out for the chiller with the following parameters of impellers by stages: $D_2^I = D_2^{II} = 350mm$, $b_1^I = 40mm$, $b_1^I = 32mm$, $b_2^I = 17{,}5mm$, $b_2^I = 9{,}5mm$, $\beta_1^I = 48°$, $\beta_2^I = 42°$, z_1^I=$z_2^I = 18$, z_1^{II}=$z_2^{II} = 16$, $\delta = 6mm$.

We set the following regime parameters: $V_s = 400m^3/h$, $V_W = 500m^3/h$, $t_{w1} = 24°C$. We studied two types of refrigerants: R134a and R404a. We calculated the performance characteristics of the chiller with changing rotation frequency n of the rotor and temperature of the coolant at the inlet of the evaporator t_{s1}.

The results of the calculation Q_{e} and N at different rotation frequency of the rotor in operation with R134a are given in Fig. 3. The designed rotation frequency of the rotor in the given chiller is 10,300 rpm. It can be seen that an increase of the rotation frequency by 1,000 rpm leads to virtually linear growth of the cooling capacity by 20% on average. Consequently, power intake *N* also linearly grows. The increase of *n* also leads to a decrease in boiling temperature t_0, increase in condensation temperature t_c, and decrease in temperature of the coolant at the outlet of the evaporator t_{s2} (Fig. 4).



Figures 2. The block-diagram refrigeration machine characteristics' calculation

Figures 5 and 6 show the results of the performance analysis of the chiller at different values of temperature at the inlet of the evaporator t_{s1} when operating on R134a (Fig. 5) and R404a (Fig. 6) with fixed $n = 10300\ rpm$. When operating on R404a, the chiller has a larger cooling capacity than when operating on R404a; however, the temperature and, therefore, the pressure of condensation are also notably larger. This can be explained by a larger specific cooling capacity of the R404a refrigerant and also by a decline of heat exchange at the condensation side causing the thermal head in the condenser to grow.

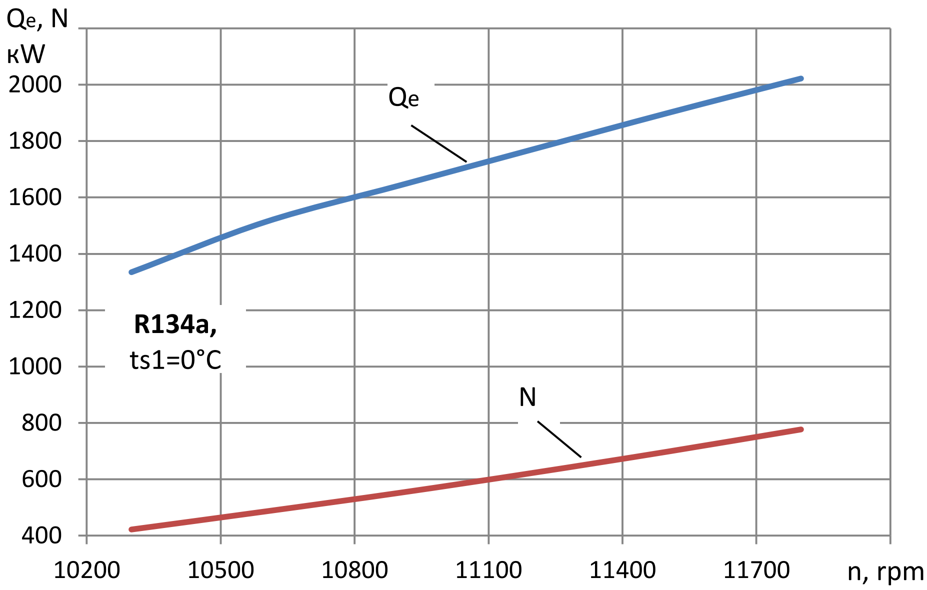


Figure 3. Change of Q_e and N as a function of rotation speed (refrigerant R134a, $t_{s1} = 0°C$)

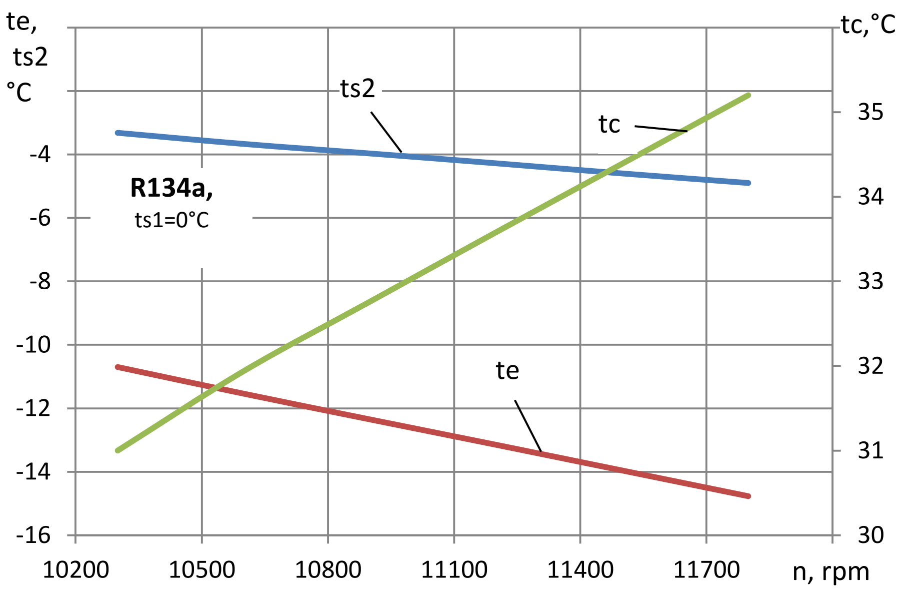


Figure 4. Change of t_e, t_c, and t_{s2} as a function of rotation speed (refrigerant R134a, $t_{s1} = 0°C$).

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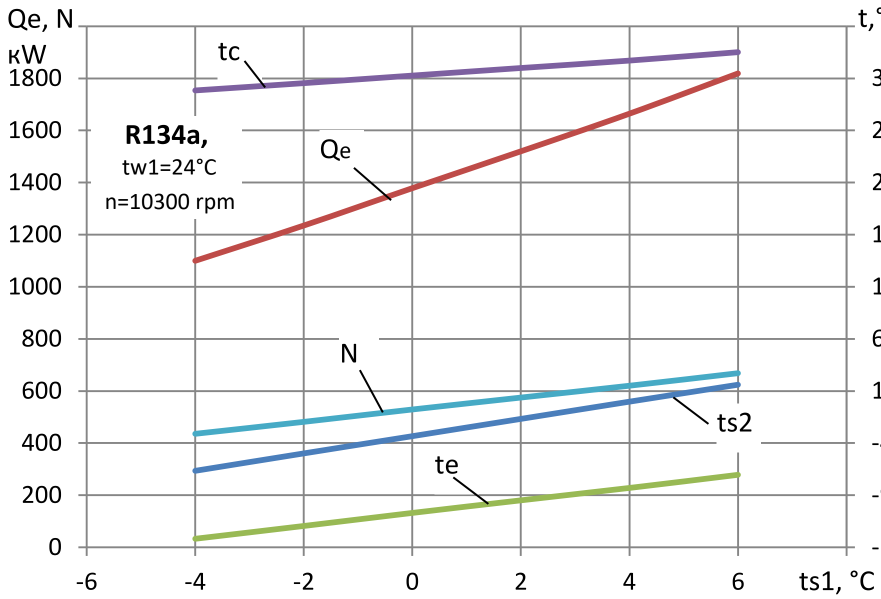

Figure 5. Change of Q_e, N, t_e, t_c, and t_{s2} as a function of t_{s1} (refrigerant R134a, n = 10,300 rpm)

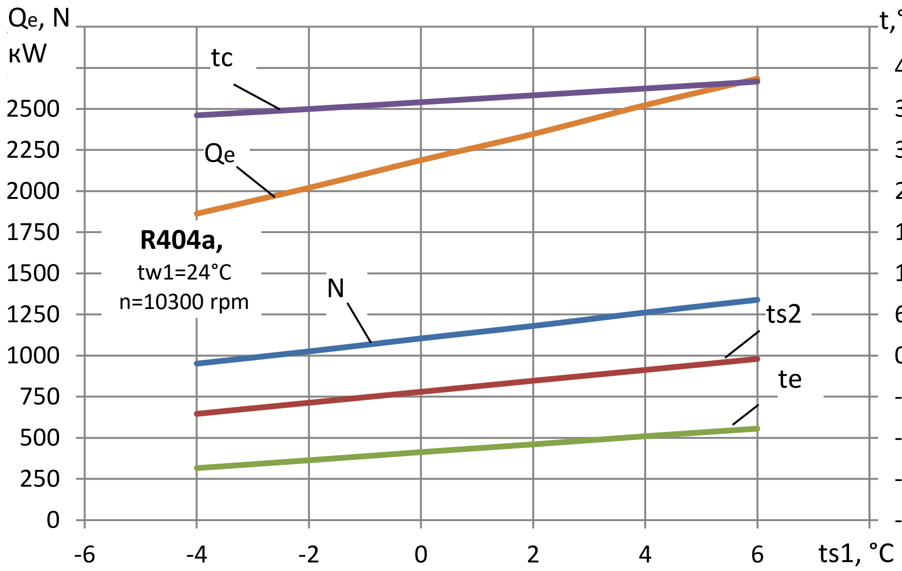

Figure 6. Change of Q_e, N, t_e, t_c, and t_{s2} as a function of t_{s1} (refrigerant R404a, n = 10,300 rpm)

5. Conclusions

We developed an analysis model that makes it possible to calculate the performance characteristics of the centrifugal compressor chiller depending on the geometric parameters of the centrifugal compressor and regime parameters of the chiller as a whole. We presented the results of the calculation of dependences of the chiller's performance characteristics on some regime parameters. It should be noted that the given analysis must be viewed along with the analysis of rotor dynamics with the chosen refrigerants, since replacement of a refrigerant and change in the rotor rotation frequency may result in a change in critical frequencies and forces acting upon the rotor.

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The presented methodic allows to predict new refrigeration machine characteristics in case of refrigerant changing or changing of other working condition parameters. Manufactures members can easily use this during exploitation process.

Nomenclature

k_z – theoretical coefficient;
h – enthalpy, kJ/kg;
Ω - degree of reaction;
η – efficiency;
Π – pressures ratio;
k – heat capacity ratio;
m – mass flow rate, kg/s;
u – impeller peripheral speed, m/s;
c_{u2} – tangential velocity, m/s;
$\varphi_{u2}=c_{u2}/u_2$ – coefficient of theoretical head;
c_{r2} – radial velocity, m/s;
$\varphi_{r2}=c_{r2}/u_2$ – coefficient of flow;
τ – factor of blockage;
v – specific volume, m3/kg;
x – degree of dryness;
p – pressure, Pa;
t – temperature, °C;
tol – tolerance, °C or Pa;
V – flow rate, m3/c;
Q_e – cooling capacity, kW;
Q_c – condensation heat, kW;
N – effective power, kW;
n – designed rotation, rpm;
D – impeller diameter, mm;
b – width of the impellers, mm;
β – blade angles, °;
z – number of blades;
δ – thickness of the blades, mm;
R – individual gas constant, J/(kg K);
k – heat capacity ratio;
$M_u=u_2/\sqrt{kRT1}$ – peripheral Mach number.

Subscripts

I – first stage;
II – second stage;
1 – inlet of the impeller;
2 – outlet of the impeller;
s1 – coolant at the inlet of the evaporator;
s2 – coolant at the outlet of the evaporator;
e – evaporate (boiling);
c – condensation;
ad – adiabatic;
pol – polytropic;
m – mechanical;
v – vapour;
w – water.

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