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CALCULATION OF HEAT TRANSFER IN CASE OF FREON CONDENSATION IN PLATE CONDENSER CHANNELS

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ABSTRACT

The article deals with the process of R407C refrigerant vapor condensation in brazed plate condensers of vapor compression refrigerating machines, operating as part of small-scale generation objects performing centralized autonomous refrigeration supply of the group of consumers. The research is conducted with respect to different types of corrugated plates with V-shaped profile, the angle of expansion φ of which equals to 60° and 120° on the assumption that condensation occurs on the entire surface of the plate. As a result of the undertaken studies semi-empirical calculation characteristics for the calculation of heat transfer in the process of R407C refrigerant vapor condensation were received.

Keywords: heat transfer, heat exchanger, plate condenser.

INTRODUCTION

Climate warming observed today in the world and the increase in energy consumption in society leads to a change in consumer and industrial enterprises demand, in particular, to an increase in demand in the cold. According to expert estimates the refrigeration equipment market is growing at a rate of about 5% per year (both in the sector of air-conditioning and commercial refrigeration sector).One way to improve the energy efficiency of energy-supply is the creation of small-scale power generation objects operating on the principle of tri-generation (simultaneous generation of electricity, heat and cold). This concept implies two possible directions of implementation.

The first of them is the creation of autonomously working tri-generation facilities of low power, sufficient to supply power to one or more consumers. At the same time consumers and small-scale power facility are completely isolated from other sources of energy [1].

The second direction is the construction of additional energy sources in the vicinity of the consumers connected to networks of centralized power supply [1]. Thus, the consumer and the small-scale power facility have links with networks of electricity and heat supply, but do not have connections in the cold supply system. The power of such sources is chosen basing on the expected power of consumer taking into account the available restrictions, and may vary within wide limits (from two or three to ten thousands of kilowatts). Therefore, improving the energy efficiency of refrigerators as components of small-scale power facilities becomes an essential aspect of sustainable development of society [2].

Much of the cold is produced by means of vapor compression cold producing machines, structurally consisting of a compressor, a condenser, a throttle and an evaporator. Traditionally shell-and-finned-tube or shell-and-smooth-tube heat exchangers were used as condensers. Production and operation of such devices are debugged and optimized, but they have poor overall performance. Recently plate heat exchangers have been becoming more common as they have larger heat transfer coefficients and smaller dimensions.

This paper deals with brazed plate condensers of cold producing machines (evaporators of cold producing machines are identical to condensers, except for the physical processes occurring in them).Brazed plate condenser is a welded unit, channels for the movement of operating medium of which are formed by corrugated stainless steel plates. The channels are made by welding two adjacent plates along the side generating line and joining them to the appropriate collector. At that the direction of the corrugations of two adjacent plates differs by 180°. The channels for the flow of the medium are formed by two adjacent pairs of plates.

The performance of brazed plate heat exchangers running in the mode of the condenser, it is little studied. All plate heat exchangers, including the brazed plate condensers are calculated by specialized computer programs developed using the results of tests of full-scale specimen of heat exchangers.The type of heat exchange dependencies at the forced movement of single-phase heat-bearing agents and at Freon vapor condensation incorporated in the program, as a rule, is unknown to the user. Unavailability of calculation dependencies to determine heat transfer coefficients limits the overall development of brazed plate condensers.

The determinative feature of the construction of plate heat exchangers [3] is the design and shape of the surface of heat transfer and operating medium channels. Heat exchange surface is formed ofseparate corrugated plates and the channels for the operating medium have a variable slit-like cross-sectional shape in the flow direction with a hydraulic diameter less than 6 mm. The pronounced turbulence also promotes intensification of heat exchange.
The comparison of the plate devices with other types of devices in the case of condensation of ammonia is made in [4] and is presented in Table 1. The small thickness of the plates and their parallel alignment with the small distance between the plates allow placingoperating medium of the heat exchanger in the space more compact with the "density" which is not achievable in other types of vapor-liquid heat exchangers. This fact together with the high degree of intensification of the heat transfer process, eventually leads to the fact that the plate heat exchangers have in much smaller dimensions and metal content at the equal thermal load than the apparatus of "pipe in pipe" exchanger, shell and tube exchanger and other types [4].

Table-1. Comparing the effectiveness of different types of condensers.

Type of heating element	Filmheat-transfer coefficient, kW /(m^2·K)	Pressure loss in cooling loop, КPa	Component density, m^2/m^3	Clearance volumetric efficiency of heat transfer, kW/(m^3·K)
Shell and tube	2.7	20	71	192
Shell and tube with surface enhancement	5.7	21	69	393
Plate	4.2	103	191	802

The most fundamental dependence describing the process of heat transfer at stationary steam condensation on the vertical plate is the Nusselt number [5]:

$$\alpha(x)=\sqrt[4]{\frac{\lambda_c^3\cdot\rho_c^2\cdot g\cdot r}{4\cdot\mu c\cdot(t_n-t_{st})\cdot x}}. \quad (1)$$

Normally, the study ends at the stage of empirical dependence [6-10]. The authors [5] suggest using the following equations to calculate heat transfer at the situation when the moving vapor condensates inside vertical winding channels of plate heat exchangers:

Where $\Delta t=\overline{t}_c-\overline{t}_{st}>10°C$

$$\mathrm{Nu}=\mathrm{A}_1\cdot\mathrm{Re}_c^n\cdot\mathrm{Pr}^m \quad (2)$$

Where $\Delta t=\overline{t}_c-\overline{t}_{st}<10°C$ and low speed of vapor movement.

$$\alpha=A_2\cdot\sqrt[4]{\frac{g\cdot\rho_c\cdot r\cdot\lambda_c^3}{\nu_c\cdot L_t\cdot\Delta t}}. \quad (3)$$

The following dependencies are presented in works [6-10]:

$$\mathrm{Nu}_c=3{,}371\mathrm{Re}_c^{0,55}\mathrm{Pr}_c^{0,3}\left(\frac{D^2}{\rho_c^2 c_{pc}\Delta t_1}\right)^{1,3}\left(\frac{\rho_c^2 h_c}{D^2}\right)^{1,05}\left(\frac{\rho_c\sigma_c}{\mu_c D}\right)^{0,05}\left(\frac{\rho_c}{\rho_c-\rho_v}\right)^{2} \quad (4)$$

$$\alpha_1=0{,}2092\frac{\lambda_c}{d}\mathrm{Re}_c^{0,78}\mathrm{Pr}_c^{0,33}\left(\frac{\mu_c}{\mu_{cst}}\right)^{0,14} \quad (5)$$

$$\alpha_1=Ge_1\cdot\frac{\lambda_c}{d}\cdot\mathrm{Re}_c^{\mathrm{Ge}_2}\cdot\mathrm{Pr}_c^{0,33}\cdot\left(\frac{\mu_c}{\mu_{cst}}\right)^{0,14}, \quad (6)$$

$$\alpha_1=4{,}118\cdot\frac{\lambda_c}{d}\cdot\mathrm{Re}_c^{0,4}\cdot\mathrm{Pr}_c^{0,33}, \quad (7)$$

METHODOLOGY

The aim of this work is to obtain generalized goal dependencies of heat transfer during Freon R407C vapor condensation for different types of corrugated plates with a V-shaped profile of the corrugations, the angle of expansion φ of which equals to 60° and 120° (Figure-1). At the angle of expansion of the corrugations φ = 60° side spacing of corrugations is S_1 = 26 mm, normal pitch is S_N

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= 12 mm. Atφ = 120° - S_1 = 10 mm, S_N = 9 mm. The corrugation height is h = 3 mm. The thickness of plates is δ = 0.5 mm. Geometrical characteristics of the plates and the channels formed by them are presented in Table-2.

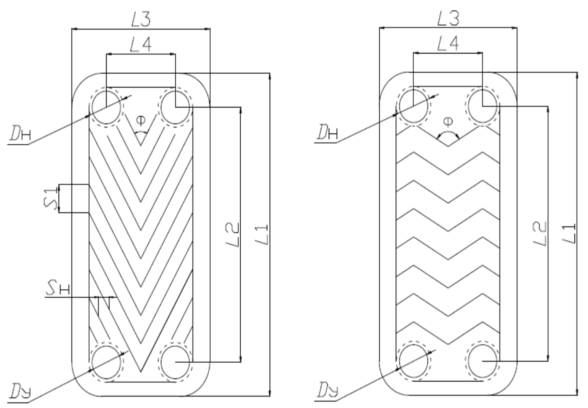

Figure-1. Basicgeometrical characteristics of the plates.

All calculations necessary for the task were performed with the usage of the available programs for the choice of brazed plate condensers. The scheme of the refrigerant vapor and cooling heat-bearing agent (water) movement is countercurrent: the vapor moves downward, the water - up from the bottom. The incoming steam has a saturated state. In the condenser comes. A complete condensation of the vapor on the entire surface area of heat transfer takes place in the condenser.

Table-2. Geometrical characteristics of the plates*.

Type	D_y, mm	F_0, m^2	L_2, mm	L_4, mm
1	**20**	**0.024**	**40**	**278**
2	20	0.033	50	250
3	100	0.0347	239	862
4	25	0.036	65	242
5	50	0.147	150	520
6	25	0.073	65	446

*d = 0.00465 mwhereφ = 60°, d = 0.0487 mwhereφ = 120°иd = 0.00476 m. For the channels, formed by the plateswith the *inclination angle of corrugations*φ = 60°andφ = 120°

Initial data for calculation included:

- The heated coolant type;
- Flow of the heated coolant, kg/s;
- Initial and final temperatures of the heated coolant, °C;
- Allowable pressure losses kPa;
- Vapor temperature, °C
- Standard size of the plate;
- The material of the plate;
- The maximum heat-bearing agent pressure, MPa;
- Maximum heat-bearing agent temperature °C.
- For all calculations, the following conditions were:
- The type of vapor – Freon R407C;
- The type of heated heat-bearing agent – water;
- Allowable pressure losses - 50kPa;
- Plate material – stainless steel *AISI* 316;
- The maximum heat-bearing agent pressure – 2.5 MPa;
- The maximum heat-bearing agent temperature- 150°C.

The study was performed on the assumption that the vapor is saturated, the condensation occurs on the entire length of the plate (the condensate does not supercool); there are no heat losses to the environment. The temperature of the Freon vapor at the condenser inlet was set in the range of 45 to 70°C. Thermal power of the unit was varied by changing the flow rate of the heated heat-bearing agent and its initial and final temperatures. Only one-way condensers were considered.

Dependencies are laid into the algorithm to account for changes in thermal properties of the heat-bearing agent according to temperature:

Freon R407C:

- Density of the condenser and vapor:

$$\rho_c = -0{,}0007t^3 + 0{,}0413t^2 - 4{,}7477t + 1240{,}2;$$

$$\rho_v = 0{,}0006t^3 - 0{,}0343t^2 + 1{,}6058t + 16{,}374;$$

- Specific enthalpy of the condenser and vapor:

$$\mathrm{h_c} = 0{,}0057\mathrm{t}^2 + 1{,}2404\mathrm{t} + 201{,}26;$$

$$\mathrm{h_v} = -2 \cdot 10^{-6} \cdot \mathrm{t} + 0{,}0002\mathrm{t}^3 - 0{,}0104\mathrm{t}^2 + 0{,}5891\mathrm{t} + 409{,}31;$$

- Specific thermal capacity of the condenser and vapor:

$$\mathrm{c_{pc}} = 5 \cdot 10^{-10} \cdot \mathrm{t}^6 - 10^{-7} \cdot \mathrm{t}^5 + 8 \cdot 10^{-6} \cdot \mathrm{t}^4 - 0.003\mathrm{t}^3 + 0{,}0047\mathrm{t}^2 - 0.0225\mathrm{t} + 1.4304;$$

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$c_{pv} = 4 \cdot 10^{-6} \cdot t^6 - 8 \cdot 10^{-8} \cdot t^5 + 6 \cdot 10^{-6} \cdot t^4 - 0.0002t^3 + 0.0036t^2 - 0.0143t + 0.9675;$

- Thermal conductivityof the condenser and vapor:

$\lambda_c = -0.005t + 0.0958;$

$\lambda_v = 2 \cdot 10^{-9} \cdot t^4 - 2 \cdot 10^{-7} \cdot t^3 + 9 \cdot 10^{-6} \cdot t^2 - 2 \cdot 10^{-5} \cdot t + 0.0123;$

- Shear viscosity of the condenser and vapor:

$\mu_c = 0.0055t^2 - 2.2598t + 208.58;$

$\mu_v = 5 \cdot 10^{-7} \cdot t^4 - 5 \cdot 10^{-5} \cdot t^3 + 0.0022t^2 + 0.0111t + 11.323;$

- Prandtl number of the condenser and vapor:

$Pr_c = 4 \cdot 10^{-10} \cdot t^6 - 7 \cdot 10^{-8} \cdot t^5 + 6 \cdot 10^{-6} \cdot t^4 - 0.0002t^3 + 0.0034t^2 - 0.0329t + 3.1104;$

$Pr_v = 2 \cdot 10^{-10} \cdot t^6 - 4 \cdot 10^{-8} \cdot t^5 + 3 \cdot 10^{-6} \cdot t^4 + 0.0015t^2 - 0.0061t + 0.8989;$

- Surface tension coefficient of the condenser and vapor:

$\delta_c = 4 \cdot 10^{-7} \cdot t^2 - 0.0002t + 0.0107;$

$\delta_v = 3 \cdot 10^{-7} \cdot t^2 - 0.0002t + 0.0111.$

Water:

- Surface tension coefficient σ_c, N/m

- $\sigma_c = 0{,}0797 - 0{,}0002 \cdot t_{11};$

- Water densityρ_w, kg/m^3

$$\rho_w = \frac{997}{0.99534 + 0.466 \cdot 10^{-3} \cdot \frac{1}{2} \cdot (t_{21} + t_{22})};$$

- Water thermal capacityc_{pw}, kJ /(kg·°C)

$$c_{pw} = 4.20511 - 0.00136578 \cdot \frac{1}{2} \cdot (t_{21} + t_{22}) + 0.152341 \cdot 10^{-4} \cdot \frac{1}{4} \cdot (t_{21} + t_{22})^2;$$

- Water thermal conductivityλ_w, W/(m·°C)

$$\lambda_w = 0.5678 - 0.0017 \cdot \frac{1}{2} \cdot (t_{21} + t_{22}) - 6 \cdot 10^{-6} \cdot \frac{1}{4} \cdot (t_{21} + t_{22})^2;$$

- Water shear viscosityμ_w, Pa·s

$$\mu_w = \rho_w \cdot 10^{-6} \left(\exp\left(\exp 33.22999 - 5.93043 \cdot \ln\left(\frac{1}{2} \cdot (t_{21} + t_{22}) + 273 \right) \right) - 0.87 \right);$$

The program generates an extended list of calculated parameters of which the following data were selected:

- Log mean temperature difference, °C;
- Resultant temperatures of heat-bearing agents, °C;
- Heat losses in collectors and channels, kPa;
- Velocity ofheat-bearing agentsin collectors and channels, m/s;
- Number of plates;
- Vapor discharge, kg/s;
- Heat power of the device, kW;
- Thermal-physical properties of heat-bearing agents;
- Heat-transfer resistance of contaminations, (m^2·°C)/W;
- Heat exchange surface area, m^2;
- Heat transfer coefficient for clean and contaminated surface, W/(m^2·K).

RESULTS AND DISCUSSIONS

The programs were used for calibration calculations of brazed plate condensers of Freon. As a result, thermal power, Freon steam consumption, the final temperature of the water, the mean temperature difference and heat transfer coefficients of condensers were identified for all heat exchangers. Next, using convective heat transfer dependence at a forced coolant flow in the channels of such heat exchangers were calculated heat transfer coefficients for water

$$\alpha_2 = A_3 \cdot \frac{\lambda_2 \cdot Re_2^n \cdot Pr_2^{0.4}}{d}$$

temperature difference between the vapor and the heat transfer surface

$$\Delta t_1 = \Delta t_л - \frac{Q}{F} \cdot \left(\frac{1}{\alpha_2} + \frac{\delta}{\lambda} + 2 \cdot R_{dif} \right)$$

and the coefficients of heat transfer of condensing steam were defined

$$\alpha_1 = \frac{1}{\frac{1}{k} - \frac{1}{\alpha_2} - \frac{\delta}{\lambda} - 2 \cdot R_{dif}}.$$

In [11-13] factors affecting the heat transfer intensity were identified: the Reynolds number Re_c, the Prandtl number Pr_c, the Froude number Fr_c, the Weber number We_c, the Galileo number Ga of condensate film,

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phase transition K, as well as the condition to account for the effect of the density ratio of steam and condensate $((\rho_c / \rho_v)^{0.5} + 1)$. A peculiarity of the condensation process of the moving steam is the flow stratification: in the vicinity of the longitudinal axis of the channel the vaporor (at higher speeds) fog mixture moves, there is condensate on the walls. That is, there is a dispersed-annular regime of flow, the presence or absence of which is directly determined by the Kutateladze number Ku, whose values are ranged from 3.72 to 27.14 in the test modes.

That is why in generalizing the results of calculation of heat exchange it was required to introduce the geometric ratio $2F_0/f_0$ as the determining factor.

Therefore, the generalized dependence of heat transfer, during the condensation of vapor of in shaped channels is of the form:

$$\mathrm{Nu_c} = A_c \cdot \mathrm{Re}^n \cdot \mathrm{K}^{m_\kappa} \cdot \mathrm{Ku}^{n_1} \cdot \mathrm{We}^{n_2} \cdot \mathrm{Fr}^{n_3} \cdot \mathrm{Ga}^{n_4} \cdot \left(\left(\frac{\rho_c}{\rho_v} \right)^{0.5} + 1 \right)^{l_\kappa} \cdot \left(\frac{2F_0}{f_0} \right)^b \cdot \mathrm{Pr}_c^{0.4}.$$

The analysis of the data obtained showed little effect of the Froude and Weber numbers. The analysis of the Kutateladze number revealed the existence of dispersed-annular regime of flow in channels [3]. However, its effect on the heat exchange process is not essential.

The final results of the calculation of heat transfer during the condensation of vapor were presented in the form of dependence

$$\mathrm{Nu_c} = A_c \cdot \mathrm{Re}_c^{n_\kappa} \cdot \left(\frac{2F_0}{f_0} \right)^b \cdot K^{m_c} \cdot \left(\left(\frac{\rho_c}{\rho_v} \right)^{0.5} + 1 \right)^{l_\kappa} \cdot \mathrm{Pr}_c^{0.4} \cdot X_{t0}.$$

It is worth noting that the process of condensation is strongly dependent on the temperatures of the heated medium. Therefore the generalization was performed individually for several values of the inlet and outlet temperatures of the cooling medium. The values of the A_κ proportionality coefficient for each standard size of the plates are shown in Table-2. The value of the indices at the chosen criteria for each configuration of plates (φ = 60°, φ = 120° and the mixed configuration) are constant. Table 3 also shows the studied ranges of influencing factors.

Table-3. Basiccalculation data.

№ plate*	A_κ	m_κ	l_κ	b	n	δNu_c, %
φ = 60°						
3	0.0527	-0.068	0.0578	0.1	0.9	16.9
5						19.7
6						7.8
φ = 120°						
1	11.43	0.4	-0.35	-0.55	0.8	16.9
2						12.0
3						19.1
4						13.7
5						19.5
6						8.73
φ = 60°and φ = 120°						
3	0.07	-0.078	-0.16	0.22	0.8	13.3
5						12.0
6						18.2

Thermal properties of Freon R407C condensate in the calculation of Nu_κ, Re_κ, Ku and K were taken at the saturation temperature, which was 45-70° C. The Prandtl number varied from 1.5 to 1.7. The maximum supercooling of the condensate for the given initial conditions did not exceed 24.3°C.

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The average speed of the heated water was calculated by the continuity equation and ranged from 0.2 to 1.2 m/s. The hydraulic diameter of the channel *d* was used as a typical size.

The Reynolds number in the forced flow of condensate film subject to a friction force at the phase interface was calculated as follows:

$$\mathrm{Re}_{\mathrm{c}} = \frac{Q \cdot L_{\mathrm{p}}}{F \cdot (h_{\mathrm{c}} - c_{\mathrm{pc}} \cdot t_{\mathrm{c}}) \cdot \mu_{\mathrm{c}}}.$$

CONCLUSIONS

The obtained deviations of the calculated Nusselt numberfrom the generalized results at vapor condensation of the Freon R407C are quite significant at first glance. However, it should be noted that in the case of evaporation and condensation deviations up to 30% are considered acceptable in view of the enormous number of factors that affect the condensation process.

The obtained generalized dependencies relating to heat transfer for the channels of intensified heat exchange surfaces are most valuable because they allow identifying the main factors affecting the intensity of the processes occurring in them, and should be considered in the modeling of heat transfer equipment. The numerical values of the constants in these dependencies are largely determined by the peculiarities of the geometry of the heat transfer surfaces and their production technology.

Symbolic notations

*D*and*G* – vapor and water consumption; *k* – heat transfer coefficient; α – film heat transfer coefficient; δNu – mean square error for the Nusselt numbers; R_{foul} – fouling resistance; λ – thermal conductivity of plate material; λ_c, c_{pc}, ν_c, μ_c – thermal conductivity, heat capacity at constant pressure and kinematic and dynamic condensate viscosity at saturation temperature; δ – the plate thickness; Q – heat capacity of the heat exchanger; F_0 – heat exchange surface area of one plate; F – heat exchange surface area of the heat exchanger; f_0 – cross section area of an interplate channel; φ – the expansion angle of the plate corrugations; h_v– vapor enthalpy; *r*– latent heat of vaporization; L_1, L_3– the plate height and width; L_2, L_4– vertical and horizontal distances between centers of adjacent nozzles for heat-bearing agent inlet and outlet; L_t– the plate total length; D_o– the nozzle outer diameter; D_y–the nozzle inner diameter; *g*– gravitational acceleration; ρ_v– the vapor density; W_v– the vapor speed; $K = \dfrac{r}{c_{pc} \cdot \Delta t_1}$ –phase transition number; $K = \dfrac{\rho^{0,5} w_v}{\left[\sigma g(\rho_c - \rho_v)\right]^{0,25}}$ –the Kutateladze number.

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