Experimental Investigation of a Chiller with Cold Accumulator Using the Vertical-Tube Evaporator Water Chiller

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Abstract: *Freezing has been successfully employed for the long-term preservation of many foods, some medical and pharmaceutical substances and equipment, providing a significantly extended shelf life. This process involves lowering the product temperature generally to a level that will prevent spoilage and for some food substances to -18 °C or below. For some food substances the physical state is changed when energy is removed by cooling below freezing temperature. The extreme cold simply retards the growth of microorganisms and slows down the chemical changes that affect quality or cause food to spoil. Introduction of cold accumulators in refrigerating machines has become one the modern ways of reducing the cost of refrigeration***.**

Keywords: *Shelf life, chemical change, cold accumulator, cost of refrigeration*

Introduction

Freezing is considered one of the oldest and most widely used methods of food preservation; it allows preservation of taste, texture, and nutritional value in foods better than any other method. The freezing process is a combination of the beneficial effects of low temperatures at which microorganisms cannot grow, chemical reactions are reduced, and cellular metabolic reactions are delayed [6].

The international trade of fruits and vegetables is dominated by developed countries, mostly the U.S. The United States is ranked first accounting

for the highest percent as both importer and exporter, of fresh produce in world trade. However, many developing countries still lead in the export of fresh exotic fruits and vegetables to developed countries [11].

For developing countries, the application of freezing preservation depends on several favorable main considerations. From a technical point of view, the freezing process is one of the most convenient and easiest of food preservation methods ten plagued with lack of power supply. However, when compared with other commercial preservation techniques, is preferred. The availability of several choices of equipment with proper application for several different food products results in a flexible process in which degradation of initial food quality is minimal. High capital investment of the freezing industry usually plays an important role in terms of economic feasibility of the process in developing countries.

As for cost distribution, the freezing process and storage in terms of energy consumption constitute approximately 10 percent of the total cost [15]. The selection and appropriate application depending on the government regulations, especially in developing countries, energy cost for producers can be subsidized by means of lowering the unit price or reducing the tax percentage in order to enhance production. Therefore, in determining the economical convenience of the process, the cost related to energy consumption (according to energy tariffs) should be considered.

1.1 System process description

For the purpose of studying the processes occurring in the layout of vertical-tube evaporator water chiller, Kuban state technological University (Krasnodar, Russia) [7] designed and manufactured an experimental stand (figure 2.1), where 1 - evaporator; 2 - capacitor; 3 - cooler condenser; 4,5 – measuring tanks; 6 - gauge; 7 - container with heat carrier; 8 – flow pump; 9 – coolant volumetric flow meter ; 10-heater; 11 – refrigerant cylinder; 12 - vacuum pump; 13 - branch pipe; 14 - manometer;

15 - thermometer; 16 - safety valve; 17 – liquid collector; 18 - steam refrigerant pipe; 19 - liquid refrigerant pipe; 20 - liquid coolant pipe; 21 - glass thermometer; 22 - control flap.

Figure 2.1 is a schematic diagram of the experimental stand. The designed values of the experimental stand were determined by standard formulas according to [8].

The stand was equipped with necessary measuring and control devices [4, 23]

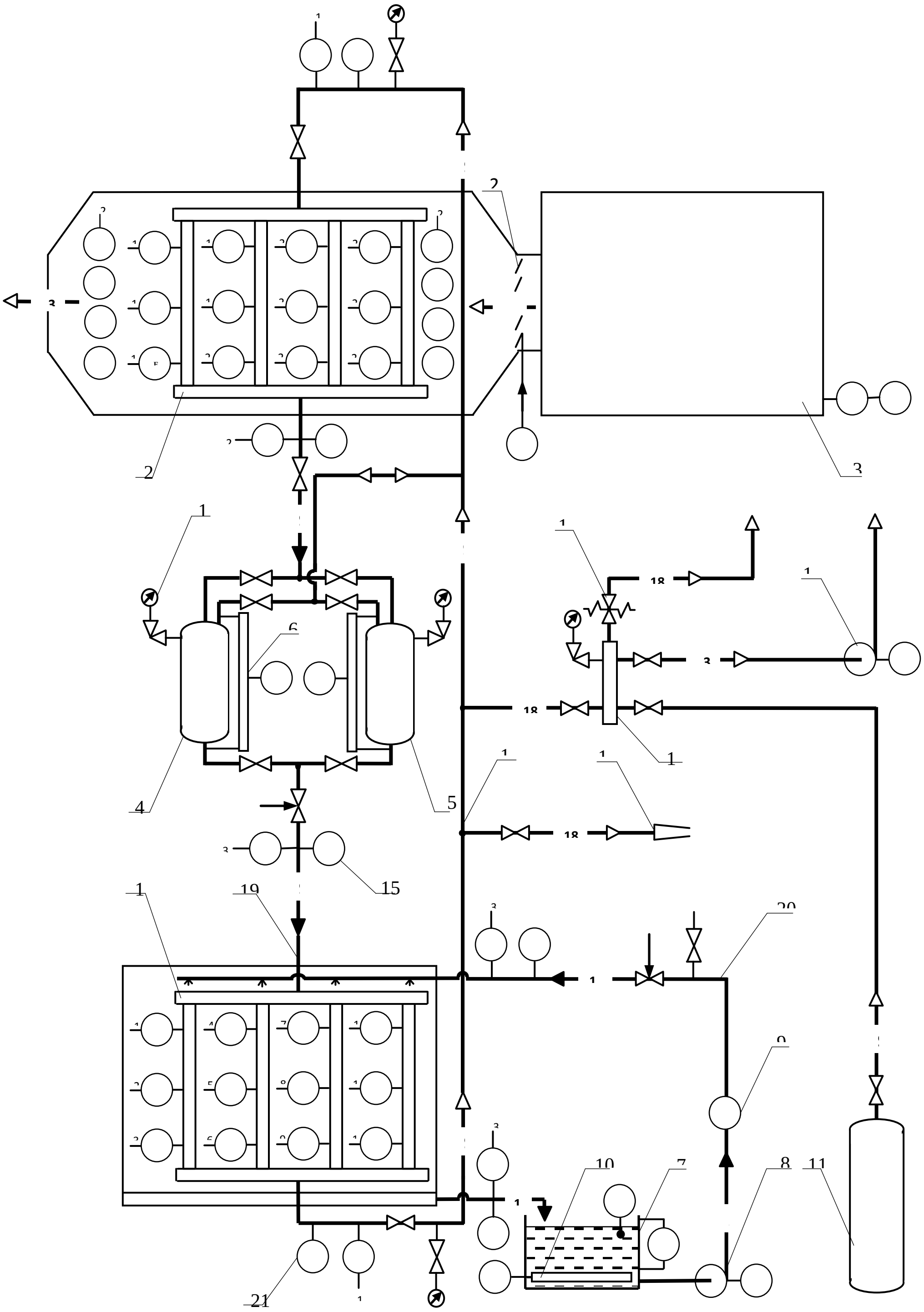


Figure 1.1 - EXPERIMENTAL SETUP (DIAGRAM)

2.0 Methodology for experimental studies

2.1 Determining the operating parameters of the refrigeration machine

The experimental test was carried out in thermosyphon condition on its stand. The evaporator 1 was flooded with water heated by the electric heater 10, mounted in the tank 7. Vapor from the boiling refrigerant in the evaporator 1, enters the condenser 2 through steam conduit 18, cooled by stream of cold air coming from the air conditioner 3. From the condenser 2, the condensate is poured into one of the measuring containers (e.g., 4), which simultaneously performed functions of liquid refrigerant collection and liquid from another tank entering from the evaporator 1. This allowed us to simultaneously measure the mass flow rate of refrigerant through the evaporator and the condenser.

The mass of liquid refrigerant supplied to the evaporator was controlled by regulating the flow valve. The flow of coolant (water) through the evaporator was controlled by a flow valve installed on the pipeline supplying the coolant to the evaporator. Regulation of air flow through the condenser was carried out by changing the position of the regulating flap and (or) changing the condenser cooling fan speed.

In operation, on the stand, the following parameters were measured during the study:

- The pressure difference of the refrigerant at the evaporator inlet and at its outlet - ΔP_{io}, MPa;

- the vapor pressure of the refrigerant at the outlet of the evaporator p_{ia2}, MPa;

- the vapor pressure of the coolant at the evaporator inlet – P_{TH}, MPa;

- the vapor pressure of the refrigerant entering into the condenser - $P_{a.\kappa 1}$, MPa;

- the pressure of the liquid refrigerant at the outlet from the condenser - $P_{a.\kappa 2}$, MPa;

- the pressure difference of the refrigerant at the inlet and at the outlet from the condenser - $\Delta P_{k.io}$, MPa;

- the aerodynamic resistance of the capacitor - $\Delta P_{w.\kappa}$, MPa;

- the mass flow of refrigerant through the condenser - $\dot{m}_{a.\kappa}$, kg/s;

- volume of air flow through the condenser -- V_a, m^3/s;

- the mass flow of refrigerant through the evaporator - $\dot{m}_{a.i}$, kg/s;

- relative humidity of air at condenser - φ_1,%;

- power consumed by water heater - Q_{w}, kW;

- volumetric flow rate measured by the flow meter - V_n, m^3/s;

- temperature of vapor refrigerant at the outlet from the condenser -$\theta_{a.\kappa 2}$, ^{0}C;

- temperature of the liquid refrigerant before the expansion valve - $\theta_{a.lw}$; ^{0}C

- temperature of the coolant at the inlet to the evaporator - θ_{tc1}, ^{0}C;

- temperature of the coolant at the outlet of evaporator - θ_{tc2}, ^{0}C;

- temperature of vapor refrigerant entering into the condenser - $\theta_{a.k1}$, ^{0}C;

- air temperature before the condenser - θ_{wz1}, ^{0}C;

- air temperature after the condenser - θ_{wz2}, ^{0}C;

- relative humidity of air after the condenser - φ_2,%;

- temperature of liquid refrigerant entering the evaporator - $\theta_{a.i1}$, ^{0}C

- temperature of the refrigerant vapor at the outlet of the evaporator - $\theta_{a.i2}$, ^{0}C;

- current in the air conditioning circuit -I_{kd}, A;

- voltage in the air conditioning circuit - U_{kd}, V.

The tests were carried out on the layout of the vertical -pipe water chiller [23], constructed as a tube bundle of four pipes in front and in depth from the top to feed the liquid refrigerant pipe 700 mm long and 25 mm in diameter, with a film providing; fluid flow along the pipes walls and discharge steam from the lower reservoir. Tests were carried out on Freon R22 and R134a, altering heat loads from maximum to minimum and from minimum to maximum (top-down and down-up) and these results of the experiments were processed separately.

Temperature values were measured simultaneously at three different points along the length of the pipe and the average computed and recorded for a number of tubes and bundle of tubes.

2.2 Measuring and recording physical quantities technically

Copper - constantan thermocouple which in the temperature region from 0 ^{0}C to + 100 ^{0}C with an average sensitivity (conversion coefficient) 41 μv/ ^{0}C [2] was used to measure temperature with errors of ± (0.1 – 0.2) °C, which is acceptable when testing heat exchangers [3]. A simple means of measuring the temperature, is the mercury in glass laboratory thermometers with a scale division of 0.1 ^{0}C.

Pressure measurement was carried out with gauge models within measurement range of 0 - 2.5 MPa and an accuracy class of 0.4.

To measure the flow of coolant (water) passing through the evaporator, we used hot water counter with sensitivity threshold of 0,03 m^3 /hr, the smallest scale interval value of 0.0002 m and limiting average integral of relative error of not more than ± 1.8 %.

The measurement of air flow through the condenser was done by multi-range manometer inclined tube, whose sensor uses a Pitot tube mounted in the center of the duct before the condenser. The pressure drop ΔP of the coolant as it passes through the condenser was also measured by taking the static pressure before and after the condenser [4].

The aspiration psychrometer, at a flow rate of 2.0 m/s was used to measure the humidity in the experimental setup and the environment.

A laboratory wattmeter of accuracy class of 0.5 was used to measure the power consumed by the heater and the coolant cooling condenser.

2.2.1 Planning of Experiment

In order to investigate vertical-tube cooler accumulator as in many other investigations, it was necessary to conduct experimental studies. The experiment involved significant expenditure of time and material resources. Therefore, to achieve representative results within the given timeframe and with small material costs, the solution to the set tasks depended on proper planning of experiment.

The response function of this experiment took the characteristics of the chiller referred to in paragraph 2.1

2.2.2 The Choice of factors and levels of study

The design of the experimental stand allows you to change many parameters of the chiller over a wide range, therefore, pre-select a number of variable parameters, which are expected to have greater effect and show limits of their change.

Since the number of factors of variations in the experiment is limited, the following input values were assumed constant:

-temperature θ_{oc}, ^{0}C ;

- relative humidity φ, %;

- barometric pressure P, Pa of ambient air;

- temperature of the refrigerant entering in heat exchangers - $\theta^c_{a.i}$‘ $\theta^c_{a.k}$‘ ^{0}C;

- internal diameter of the heat exchanger tubes d_{tube}, m;

- height of the heat exchanger tubes h_{tube}, m

The factors of variation adopted:

- Coolant temperature (evaporator under study - water) entering the heat exchanger- θ^{cx}_{tn}, ^{0}C;

- coolant flow (evaporator under study - water) through the corresponding heat exchanger- V_{tn}, m^3/s;

- refrigerant flow through the heat exchanger - V_a, m^3 /s;

The variation levels of the factors were given the following values:

factor x_1 in the evaporator under study:

x_1^- = 40 ^{0}C, x_1^0 = 50 ^{0}C, x_1^+ = 60 ^{0}C;

factor x_2 in the evaporator under study:

x_2^- - = 3 x 10^{-5} m^3/s, x_2^0 = 4.5 x 10^{-5} m^3/s, x_2^+ = 6 x 10^{-5} m^3/s;

factor x_3 in the evaporator under study:

x_3^- = 2 x 10^{-6} m^3/s, x_3^0 = 3 x 10^{-6} m^3/s, x_3^+ = 4 x 10^{-6} m^3/s;

It should be noted that the factors X_1, X_2, X_3 - controlled, driven and independent, so it was possible to maintain each of the selected levels. The values of the maximum absolute and relative errors of measurement factors of variation were taken as the following:

for the factor X_1: Δx = 0.5 ^{0}C, δx = 0.6 %;

for factor X_2: Δx = 0.015 m /s, δx = 8.5 %;

for factor X_3: Δx = 4. 10^{-5} m^3/s, δx = 8.8 %;

The output variables of the research process were determined by:

temperature of the refrigerant at the outlet of heat exchangers $\theta_{a.i}^{вых}$, $\theta_{a.к}^{cix}$, ^{0}C;

coolant temperature at the outlet of the heat exchangers $\theta_{тн.и}^{вых}$, $\theta_{тн.к}^{cix}$, ^{0}C;

- power consumed by air conditioning W_{kd}, W;

- heating power - coolant heater - water- W_{ten}, W;

The planning matrix of the full three-factor experiment is presented in a tabular form. The average values of the output variables are designated y_{cp}.

Table 2.1: - Matrix of experimental planning

№ of site	Factors			Process output	№ of site	Factors			Process output
	x_1	x_2	x_3	$\bar{y}$		x_1	x_2	x_3	$\bar{y}$
1	+	+	+	$\bar{y}_1$	15	0	0	-	$\bar{y}_{15}$
2	+	+	0	$\bar{y}_2$	16	0	-	+	$\bar{y}_{16}$
3	+	+	-	$\bar{y}_3$	17	0	-	0	$\bar{y}_{17}$
4	+	0	+	$\bar{y}_4$	18	0	-	-	$\bar{y}_{18}$
5	+	0	0	$\bar{y}_5$	19	-	+	+	$\bar{y}_{19}$
6	+	0	-	$\bar{y}_6$	20	-	+	0	$\bar{y}_{20}$
7	+	-	+	$\bar{y}_7$	21	-	+	-	$\bar{y}_{21}$
8	+	-	0	$\bar{y}_8$	22	-	0	+	$\bar{y}_{22}$
9	+	-	-	$\bar{y}_9$	23	-	0	0	$\bar{y}_{23}$
10	0	+	+	$\bar{y}_{10}$	24	-	0	-	$\bar{y}_{24}$
11	0	+	0	$\bar{y}_{11}$	25	-	-	+	$\bar{y}_{25}$
12	0	+	-	$\bar{y}_{12}$	26	-	-	0	$\bar{y}_{26}$
13	0	0	+	$\bar{y}_{13}$	27	-	-	-	$\bar{y}_{27}$
14	0	0	0	$\bar{y}_{14}$					

The remaining parameters are functions that depend on these factors.

Thus, we have obtained that the experiment is foreseen to change the three external factors at three levels that have the greatest impact on the output parameters of the chiller-accumulator. The resulting scheme of the experiment corresponds to a three-factor experiment of type 3^3. The number of measurements in each replica is planned to be held not less than three. The model for this experiment according to [2, 3, 8, 14, 16, 19, 22], has the form:

$$y_{ijkl} = \mu + \tau_i + \beta_j + \gamma_k + (\tau\beta)_{ij} + (\tau\gamma)_{ik} + (\beta\gamma)_{jk} + (\tau\beta\gamma)_{ijkl} + (\epsilon)_{ijkl}, \qquad (2.0)$$

where y_{ijkl}, is the response function of the experiment;

μ - the mathematical expectation of the average effect;

τ_i, β_j, γ_k the true effects of levels of factors; (Tr), β γ effects of interaction with appropriate factors at all levels;

ϵ_{ijkl} - random error.

2.2.3 Determining the sequence of the statistical analysis of the experiment

Selection of external parameters levels, affecting the response variable and their variation were provided in the course of the experiment. In order to eliminate systematic errors in successive change factors, they should vary randomly.

Accumulated experimental data were subjected to statistical analysis to eliminate errors in the experiment and to check the adequacy of data obtained to the real process.

The adequacy test was performed using Fisher $F_т$, which for three-factor experiment, with a 95% probability of obtaining reliable data and degrees of freedom of varied factors is, according to [9]

$F_т$ 0. 05, 2, 54 = $F_т$ 0. 05, 4, 54 = 2.55

The calculated F-value must satisfy the inequality F < Fт. Statistical analysis was performed according to [9].

It is known [16] that results of the experiments in the n dimensional factor space, i.e. by varying n variables can be represented in the form of a polynomial - segment of Taylor series. For the considered experiment, the polynomial has the form

y = $b_0 + b_1.x_1 + b_2.x_2 + b_3.x_3$, (2.1)

where y - the response function;

b_{ij} - polynomial coefficients;

X - external variables in the code.

According to [20], the expression for any coefficient has the form

$$b_i = b_{\ddot{y}} = \frac{\sum_{g=1}^{N} x_{gi} \cdot \bar{y}_{gCP}}{N}, \quad (2.2)$$

where y_{gCC} - the average value of the response variable in the experiment;

N - number of experiments.

A property of the matrix full-factorial experiment is that the variances of all coefficients of the polynomial are the same and equal

$$S^2\{b_i\} = \frac{S^2\{y\}}{N.n}, \quad (2.3)$$

where $S^2\{y\}$ is the variance of the reproducibility of the experiment;

n - the number of iterations.

The results of the experiment have a random error, so the polynomial coefficients may be insignificant and, without prejudice to the accuracy of the model, you can omit them, i.e., to equate to zero.

The standard significance tests of the coefficients are the variance of experiment $S^2\{y\}$. The value of $S^2\{y\}$ is determined by the results of the first three duplicate experiments for the first line of the matrix according to the formula

$$S^2\{b_i\} = \sum_{l=1}^{n} \frac{(y_{gl} - \bar{y}_g)^2}{n-1}, \quad (2.4)$$

where y_{gl} - the value of the response variable;

$\bar{y}_g$ - the average value of the response variable.

The significance of the coefficients of the polynomial is tested using Student's t test, which is determined by the formula

$$t_i = \frac{|b_i|}{S\{b_i\}}, \quad (2.5)$$

where b_i- any polynomial coefficient, often the smallest;

S $\{b_i\} = \sqrt{S^2\{b_i\}}$ - mathematical deviation of b_i .

The value of t_i is compared with the selected tabular value t_{kp}.

The choice of the values of t_{kp} was based on two variables: q and n_f .

The q-Value is the significance level, which characterizes the probability of making a wrong decision was 5%. The value of n_f is the number of degrees of freedom determined by the volume of the experimental sample, determined by the formula

$$n_f = \mathrm{N}. (\mathrm{n} - 1). \quad (2.6)$$

For this experiment

n_f = 27 • (3 - 1) = 54.

At q = 5%, n_f = 54, the value of t_{kp} = 1.67 [1]. The coefficient is important when the following inequality is satisfied $t_i > t_{kp}$.

To check the adequacy of the polynomial the object under study, compared two quantities - $\bar{y}$ and $\hat{y}$. The value $\bar{y}$ is obtained experimentally, $\hat{y}$ is the response function, obtained by calculation on the polynomial.

The adequacy test was carried out using the statistical Fisher test, which is calculated according to [9]

$$F = \frac{S^2_{a\partial}}{S^2\{y\}}, \quad (2.7)$$

where S^2 is the variance of inadequacy.

$$S^2_{a\partial} = \frac{1}{N-d} \sum_{g=1}^{N} (\bar{y}_g - \hat{y}_g)^2, \quad (2.8)$$

where d - number of significant coefficients of the polynomial.

The calculated value of F was compared with F_{kp} taken from the statistical tables [9] taking into account the level of significance - q, the number of degrees of freedom - $n_{f2} = n_f$, and the number of degrees of freedom of the approximated polynomial - n_{f1} = N - d. Estimated value of Fisher's exact test was 1.67, and the table value - 1.73. Thus, we can conclude on the adequacy of the resulting polynomial.

The results of the experiment for the relative flow of the cooling refrigerant Ψ_{m0} and the flow coefficient λ of the compressor obtained the following polynomials.

The transition to natural units is carried out according to the formula

$$x_i = \frac{X_i - X_{iCP}}{X_{iCP} - X_{i\,min}}, \qquad (2.9)$$

where x_i - the value of the factor in the codes for the i — th point of space;

X - the value of the factor in natural units for the i — th point of the factor in space;

X_{iCP} - base value factor in natural units;

X_{imin} - the minimum value of the factor in natural units.

2.3 Experimental Results from the studies

The experimental results are shown in Figures 2.2 and 2.3 in a dependency study form of *Y = f (X).*

Figure 2.2 presents the results of the dependence of heat transfer coefficient α W/m^2*degree from the values of the specific heat load q W/m^2 in the temperature range of boiling point t_0 = 263 – 258 K for R22; figure 2.4 for R134a in the temperature range t_0 = 263 - 258 K. Each point in the figures 2.2 and 2.3 corresponds with 4-6 measurements; differences in the measurements does not exceed +/- 10 %.

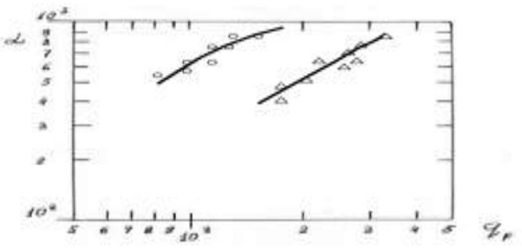
In the process of the experiments; conditions were created close to the conditions of operation of the apparatus for water cooling with a sharp increase of the heat load. Vertical-tube apparatus with upper feed tubes of liquid refrigerant practically does not change the mode of your work. This is due probably to the lack of steam locks and the ability to remove heat film from the entire area of heat transfer surface.

Data on the boiling of refrigerants in vertical channels and tubes are extremely small, and, moreover, are not summarized in relation to the real devices. Figures 2.2, 2.3, 2.4 present data on the variations of average heat transfer coefficients α and the heat flux q.

It is obvious that the evaporation from the film surface, flowing under gravity contributes to the intensification of heat exchange and increasing heat flux Q may reduce the refrigerant flow to ensure complete evaporation of the liquid at the bottom of the tube.

Thus, with respect to vertical-pipe coolers in determining the heat transfer intensity it is mainly hydrodynamics of flowing films that will provide increase in the value of heat transfer coefficient two to three times greater than the value during boiling in large volume.

Using the above apparatus and instrumentation, and test procedures allowed us to obtain representative data on heat transfer and processes in the layout of vertical-tube evaporator and, thus, to use these data in calculation methods for industrial cold accumulators.



1 - T_o = 263 K; 2 - T_o = 258 K

Figure 2.2 - Dependence of the heat transfer coefficient α W / m^2 * degrees of heat flux density q W / m^2 and the boiling temperature T_o for R22.

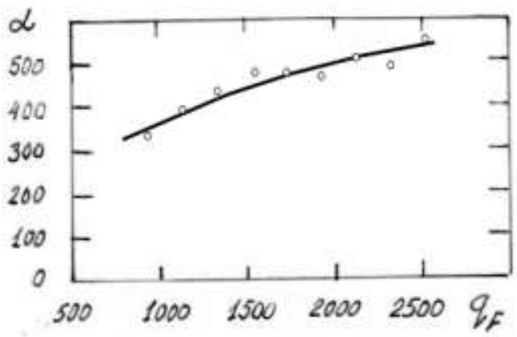


Figure 2.3 - Dependence of the heat transfer coefficient α W / m^2 * degrees of heat flux density q W / m^2 and the boiling temperature of T_0 = 263 K for R134a.

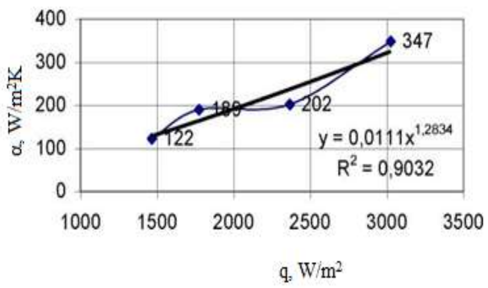


Figure 2.4 - Dependence of the heat transfer coefficient α W/m^2 K on heat flux density q W/m^2 and the boiling point of R134a at t_0 = 258 K

In the experiment, we obtained sufficient convergence of results with the theoretically obtained values when implementing the mathematical model.

3.0 TECHNICAL RESULTS OF THE PERFORMED EXPERIMENT

3.1 Technical and Economic Evaluation of Vertical Tubular Heat Exchange Accumulator with Intensifiers

The purpose of the feasibility calculation [18, 20] of vertical-pipe cold accumulator is the selection of the optimal mode of its operation, characterized by the mean log temperature difference θ_m, the speed ω of the cooling medium and the amount of frozen ice. When calculating variants with different values of θ_m, ω and δ_l is determined by the portion of the annual cost, which depends on the mode of operation of the device. The optimal mode corresponds to the option with the least variable part of the annual costs. The existence of the minimum is due to the nature of the influence on the economic efficiency of the evaporator parameters θ_m, ω and the value of δ_l. With an increase in θ_m, reduces the area of heat transfer surface of the evaporator F and the cost, but increases the temperature t_0 in the evaporator. The change in the temperature t_0 leads to an increase in irreversible thermodynamic losses due to finite temperature difference between the evaporating refrigerant and the coolant. The result is an increase in the specific power of the compressor Ne/Q_0 in the refrigeration machine.

The existence of the optimum value ω is associated with the fact that increasing this parameter intensifies heat exchange and reduces θ_m, but at the same time increases the power of the pump that provides circulation of the coolant.

The specific reduced annual costs Φ_{ca}, rub. / (year-kW) is determined by [5, 16] as:

$$\Phi_{ca} = [K(E_н + C_d + C_r) + \tau . C_E(N_c + N_н)] / Q_0 \quad (3.0)$$

where K is the cost of the evaporator.;

E_H = 0.15 - regulatory coefficient of effective capital expenditures;

C_d = 0.128; C_r. = 0.055 - the share of the annual budget earmarked respectively for depreciation of the evaporator and its repair;

C_E - The cost of electricity, rub. / (kWh);

N_c and N_H - power required for driving respectively the compressor and the water pump, kW;

Q_0- refrigerating capacity of the machine;

τ - time of operation of the refrigeration machine.

The option with the minimum value of the Φ_{ca} will match the optimal values θ_m, ω and Δh. Take five groups of options A, B, C, D, E respectively the values of the boiling temperature t_0 = 5, 0, -5, -10, -15^{0}C.

Each group includes four sub options value δ_l: in the first δ_l = 0.001 m; in the second δ_l = 0,004 m; in the third δ_l = 0.006 m; in the fourth δ_l = 0.008 m.

Each subgroup in turn includes five options for the speed u of the water in the annular space of the evaporator: in the first, ω = 0.1 m/s; in the second, ω = 0.2 m/s; in the third ω = 0.3 m/s; in the fourth, ω = 0.5 m/s; in the fifth, ω = 0.7 m/s. Thus, we obtained the100 options with different values.

For each group of options define the following options: θ_0, Q_0, Ne.

Cooling capacity Q_0 is determined by [18, 20]:

$$Q_0 = V_T \lambda_{км}(i_1 - i_{вых})/V_1, \quad (3.1)$$

where i_1 is the enthalpy of R22 refrigerant vapor at the outlet of the evaporator and entering the compressor, kJ/kg;

The calculation results are shown in table 3.1

Table 3.1 - Parameters of the operation of the refrigeration machine

Number of options	t_0, ^{0}C	θ_m, ^{0}C	Q_k, kW	Q_0, kW	N_e, kW
A	5	2.79	107.1	91.0	20.38
B	0	3.99	106.2	89.7	20.98
C	-5	5.10	105.3	88.4	21.57
D	-10	6.17	104.4	87.1	22.16
E	-15	7.21	103.6	85.9	22.74

To calculate the surface area of the evaporator; heat transfer equations are solved together in the heat flux from the evaporating refrigerant q_a = $f(\theta_a)$ and cooled medium q_w = $f(\theta_w)$, where θ_a is the temperature difference between the refrigerant and the tube wall, θ_w is the temperature difference between the tube wall and coolant.

The volumetric flow rate of water is determined by [5]:

$$V_w = Q_0 / [(\theta_2 - \theta_1) C_w \rho_w] \quad (3.2)$$

where θ_1 and θ_2 - water temperature respectively at the inlet and outlet, ^{0}C

C_w - heat of water, kJ (kg-K);

ρ_w - the density of water, kg/m^3.

Hydraulic resistance for water is Δp, Pa, is defined as [1, 11]:

$\Delta p = 1.34 \text{x} Re_{в}^{-0,182} \text{x} n_R \text{ x } n_{rn}$,
(3.3)

$Re_в$ - Reynolds number of the flow around smooth pipes (chess beam).

n_R - the number of rows of pipes;

n_{rn} - number of runs in a cross flow water tube bundle formed by the partitions.

The power expended in driving the water pump is defined as [17]:

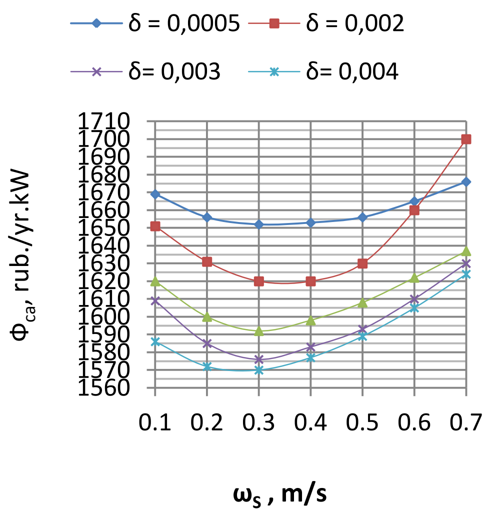


Figure 3.3: – Determination of the optimal speed of the water in the accumulator

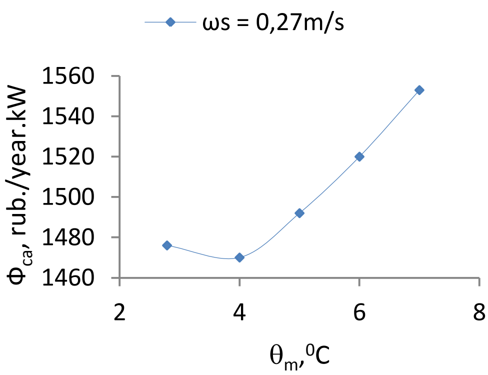


Figure 3.4 - determining the optimal values of the average logarithmic temperature difference in the evaporator.

$N_H = V_w \, \Delta p / \eta_p$

where η_p - pump efficiency (adopted η_p = 0.6).

Analysis of the data shows that the minimum value of Φ_{ca}= 1470 rub. (year/kW) at δ_l = 0,004 m. According to the obtained calculations, we plotted a graph of the dependence of Φ_{ca} on ω_s (figure 3.3) for all values of θ_m at δ_l = 0,004 m. The relationship analysis in figure 4.3 shows that optimal water velocity ω_s, = 0.27 m/s. the minimum of the function Φ_{ca} = (ω_s, θ_m) is plotted the dependence of the $\boldsymbol{\Phi}_{ca}$ on θ_m (figure 3.4), the analysis of which shows that the optimal value θ_m = 3.8 ^{0}C.

Figure 3.5 shows the dependencies Φ_{ca} on ω_s, for different thickness values of frozen ice at θ_m = 3.5 – 4 ^{0}C, which suggests that the cost of frozen ice depends on the ω_s,, as follows: at a velocity of 0.25 – 0.27 m/s the cost of ice formation is minimal regardless of the thickness of freezing ice, in general, minimal ice thickness in the formation is 4 mm.

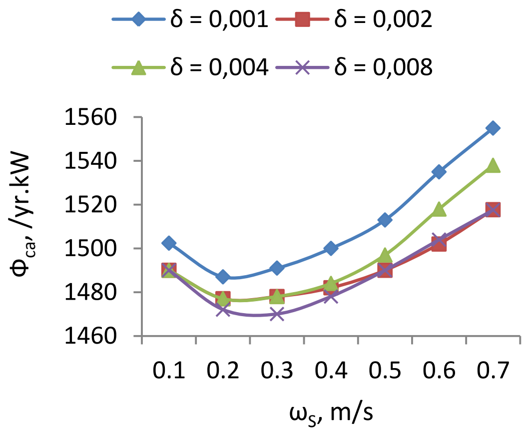


Figure 3.5 - Determination of the optimal value of the thickness of frozen ice.

3.3 Method of calculating circulation in the heat pipe the cold accumulator

Initial data for calculating the mode of operation of the heat pipe accumulator:

1. The amount of heat transmitted by the heat pipe;

2. The height of the heat pipe;

3. The length of the evaporation zone and condensation of the heat pipe; δ = 0.004

4. The maximum temperature difference between the outer wall of the evaporator and the condenser;

5. The working substance of the heat pipe;

Radial heat flow: The boiling in the wick may lead to the blockage of steam fluid formations access to all parts of the evaporator. In arterial heat pipes, the appearance of bubbles in the artery can lead to even more serious problems. It is therefore desirable to use a working fluid with superheat in order to reduce the probability of formation of bubbles. The value of the overheating required for the formation of bubbles is determined by the ratio

$$\Delta T_s = \frac{3.06 \, . T_w \, . \sigma_l}{L \, . \, \rho_w \, . \, \delta}, \quad (3.5)$$

where δ is the thickness of the thermal layer.

To determine the minimum flow section of the fluid that provides power transmission, it is necessary to equate the maximum amount of the capillary pressure of the hydraulic fluid flow resistance and gravitational pressure (pressure drop in the steam channel is neglected)

$$\Delta P_l + \Delta P_g = \Delta P_c, \quad (3.6)$$

$$\Delta P_c = \frac{2\sigma_l cos\theta}{r_c}; \quad (3.7)$$

$$\Delta P_g = P_l g l cos\Phi; \quad (3.8)$$

$$\Delta P_l = \frac{\mu_l}{\rho_l L} \frac{Q l_{eff}}{A_w K}. \quad (3.9)$$

Fluid properties are taken at 80 ^{0}C. Select the 400 grid mesh (pore size of about 0,036 mm). Assuming that $l sin\Phi = 1$ cm (the difference between the levels of the ends plus the diameter of the pipe), l_{eff} = 100cm, cos θ =1, and calculating K using equation of Baena - Kozani,

$$K = \frac{d_w^2 (1-\epsilon)^2}{66.6 \, \epsilon^2}, \quad (3.10)$$

where ϵ is the volume fraction of the solid phase (0.314),

d_w is the wire diameter.

We get:

The diameter of the artery: Equation (3.9) characterizes the ability of the artery to be filled with liquid; it gives the maximum value of the size of any artery

$$d_a = \left[\frac{1}{2} \sqrt{h^2 + \frac{8\sigma_l cos\theta}{(P_l - P_w)g}} - h\right]. \quad (3.11)$$

Using this equation, we find d_a at 30 °C (for convenience, the filling of the artery can be done at room temperature). The maximum permissible value of d_a = 0.59 mm. By considering the error of the tables of physical properties of liquids, characteristics of the wetting (angle of θ is assumed equal to zero) as well as manufacturing tolerances, acceptable practically allowable value d_a = 0.5. When determining d_a the value of h, is assumed equal to 1 cm to meet the conditions near the top of the vapor space.

The distribution of the liquid along the perimeter of the pipe: The thickness of the wick, providing the distribution of fluid on the perimeter, is limited to a specified maximum value of the temperature difference between the steam space and the outer surface of the heat pipe (or Vice versa), which should not exceed 3°C. Considering temperature drop in the aluminum wall is negligible, one can determine the coefficient of thermal conductivity of the wick structure and using the steady state equation of heat conduction to find the thickness of the wick

$$K_{u.l.c.k} = \left(\frac{\beta-\epsilon}{\beta+\epsilon}\right) \quad (3.12)$$

where $\beta = \left(1 + \frac{K_s}{K_l}\right) / \left(1 - \frac{K_s}{K_l}\right)$

It is desirable to have several arteries in order to provide some margin. The restriction on the height of capillary rise is achieved in the case when the sum of the pressure losses in the liquid and the pore phases, and gravitational pressure equal to the maximum capillary pressure, i.e.

$$\Delta P_{la} + \Delta P_{lm} + \Delta P_g + \Delta P_v = \Delta P_c \quad (3.13)$$

where ΔP_{la} – pressure drop in the artery

ΔP_{lm} - pressure loss in the distributing wick.

Axial flow on the grid has little effect and is negligible.

Mac Adams proposed the equation for calculating the pressure loss in the case of laminar flow in a rectangular channel. It shows that the equation agrees well with the experiment in the relationship of channel depth to its width a_a / b_a = 0.05 - 1.0. This equation can be written as

$$\Delta P_{la} = \frac{4 K_l L_{eff} Q}{a_a^2 \, b_a^2 \Phi_c N,} \quad (4.14)$$

where N- number of channels;

Φ_c - function relationship of the channel,

$K_l = \mu_l / \rho_w \mathrm{L}$
(3.15)

The total pressure loss in the evaporator and condenser is determined by the formula

$$\Delta P_{lm} = \frac{K_l L_{eff} \mathrm{Q}}{2\mathrm{K}\, A_c} \qquad (3.16)$$

where L_{eff} - effective length of the distributing wick with the liquid flow rate m/4, approximately equal πr_{w} /4; here m is the mass flow rate of the liquid;

A_c - the flow section of the distributing wick (thickness x length of the evaporator and condenser);

K - Permeability grid 400 mesh = 0.314 x 10^{-10} m^2.

Seeder wick orifice formed by two layers of grid 400 mesh, is: Ac = 8 x 10^{-2} x 0.1x 10^{-3} = 8 x 10^{-6} m^2.

Then the hydraulic resistance of the condenser and evaporator will be

For each section

Thermal load, corresponding to the transition from laminar to turbulent flow is determined from the condition that Re_{z} = 1000 in determining the Re_{z} for the hydraulic radius.

Initial data for calculation are the length of pipeline sections of the circuit. The difference in density of the medium in the standpipe and the lift pipe creates the driving pressure, which for vertical channels is calculated according to [12], the ratio

$$\Delta P_{\partial \delta} = g.\ \varphi.\ (\rho' - \rho'')\ .\ h \qquad (3.22)$$

where φ is the true volumetric vapor content;

ρ' - density of saturated liquid, kg/m^3;

ρ'' - density of saturated vapor, kg/m^3;

h - height of the portion of the pipe M.

The total driving pressure is defined as the sum of all the pressures in the pipeline sections. The driving pressure arising in portions of the path on for the portions of the contour according to the methods set forth in [10].

$$\Delta P_{lm} = \frac{\pi 2{,}5.10^{-3}}{2.4} \cdot \frac{1}{8.10^{-6}} \cdot \frac{K_l}{0{,}314.10^{-10}} \qquad (3.17)$$

For the six channels

$$\Delta P_{lm} = \frac{4 \text{ x } 0.92.K_l \mathrm{Q}}{(0.5)^2\ (1.0)^2\ 10^{-12} \text{ x } 0.115 \text{ x } 6} = 21.3 \text{ x } 10^{12} K_l \mathrm{Q}$$
(3.18)

For the four channels

$$\Delta P_{lm} = \frac{4.0{,}92.K_l \mathrm{Q}}{(0{,}5)^2\ 1{,}0^2\ 10^{-12}.0.088 \text{ x } 4} = 18{\cdot}59 \text{ x } 10^{12} K_l \mathrm{Q};$$
(3.19)

The pressure loss in the steam flow in two approximately semicircular channels can be found from the equation of Hangen-Poiseuille

$$\Delta P_v = \frac{1}{2}\left(\frac{8K_v L_{eff} \mathrm{Q}}{\pi r^4 H}\right), \qquad (3.20)$$

Where $K_w = \mu_w / \rho_w \mathrm{L}$

The axial Reynolds number Re_{z} is determined by the formula

$$Re_z = \frac{Q}{\pi r H \mu_l L}. \qquad (3.21)$$

which the moving liquid-vapor mixture, is spent on overcoming local resistance, and the total pressure is determined by the expression (3.23)

$$P_{tot.} = \Delta P_{l-v} - \Delta P_{res}. \qquad (3.23)$$

According to the recommendations in [12], the steady state circulation should meet the following condition

$$\Delta P_{tot.} = \sum \Delta P_{loss}, \qquad (3.24)$$

where $\sum \Delta P_{loss}$ - the sum of hydraulic losses in the lower part of the circuit, Pa.

At the intersection of the curve $\Delta P_{tot.} = f\,(W_0)$ and $\sum \Delta P_{loss} = f\,(W_0)$, is determined by the circulation rate of the refrigerant in a predetermined path.

After determining the speed of circulation of the refrigerant, we calculate the pipeline diameters

4.0 CONCLUSION

Theoretical and experimental studies have shown:

Based on the analysis of heat transfer in the evaporator the mathematical model of the cold accumulator, takes into account the peculiarities of its structural design, as well as changing the actual parameters of the refrigerant and the storage medium. The obtained data on the values of heat transfer coefficients and the accumulation of the values of heat flow ranged from 55 to 520 W/m^2 K at heat fluxes of 1000 to 4000 W/m^2.

1. The experimental set up proved that refrigerating machine with cold accumulator of heat pipes can work in conditions of multi-rate electricity tariffs.

2. A mathematical model based on heat pipes of the cold accumulator was developed;

3. A methodology for calculating the modes of a cold accumulator was developed;

4. We performed analysis of energy and economic efficiency of the application of cooling machines with cold accumulator;

5. We obtained experimental data on heat and mass transfer process in the cold accumulator (the heat transfer coefficients during boiling of the refrigerant in the heat pipe, the mass of freezing ice and cycles of charging and discharging the accumulator).

6. Temperature range from t_0 = -10 to -15 ^{0}C for effective work of the accumulator was identified when working in the recommended R134a refrigerant in the cold accumulator;

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