UDC 621.56
DOI: 10.15587/1729-4061.2022.251834

IMPROVEMENT OF REFRIGERATING MACHINE ENERGY EFFICIENCY THROUGH RADIATIVE REMOVAL OF CONDENSATION HEAT

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In regions with a continental climate, refrigerators with air-cooled condensers operate at high condensation pressures during the summer season which reduces their efficiency and accelerates the wear of compressors. To reduce condensation pressure, it was proposed to use radiative cooling which is a way of heat removal through the planet's atmosphere to outer space in a form of infrared radiation. A refrigerating machine with an assembly of condensation heat removal including air and liquid cooling condensers connected in series has been developed. To reduce the condensation temperature, a pre-cooled heat-transfer agent is fed to the liquid cooling condenser during the day hours at high atmospheric temperatures. At night, the heat-transfer agent is cooled by radiative cooling.

An experimental study of the operation of a 600 W refrigerating machine including a sealed piston compressor was conducted. R134a refrigerant was used. Supply of pre-cooled heat-transfer agent at +33.1 °C has provided a reduction of condensation temperature from +47.0 to +39.1 °C. The study was conducted at an atmospheric air temperature of +38.0 °C. The degree of pressure growth was decreased by 30 %. The refrigeration coefficient was increased by 11 %. In comparison with the conventional scheme with an air-cooled condenser, energy consumption by the system did not change in the daytime.

The offered scheme of condensation heat removal reduces the pressure of condensation and provides stability for refrigerating machine operation. It can be used in stationary refrigerating machines at high daytime temperatures

Keywords: radiative cooling, energy efficiency, condensation pressure reduction, refrigerating machine

Received date 08.11.2021
Accepted date 01.02.2022
Published date 25.02.2022

How to Cite: *Tsoy, A., Titlov, O., Granovskiy, A., Koretskiy, D., Vorobyova, O., Tsoy, D., Jamasheva, R. (2022). Improvement of refrigerating machine energy efficiency through radiative removal of condensation heat. Eastern-European Journal of Enterprise Technologies, 1 (8 (115)), 35–45. doi: https://doi.org/10.15587/1729-4061.2022.251834*

1. Introduction

In vapor-compression refrigerating machines (VCRM), heat from the cooled object is removed to the environment through a condenser. The refrigerant condensation temperature depends on the temperature of the medium used to remove the condensation heat. Condensation pressure depends on condensation temperature.

In regions with a hot continental climate, e.g., Republic of Kazakhstan steppes, the daytime temperature of atmospheric air in its surface layers reaches +35...45 °C. If the heat of condensation is removed to atmospheric air, the refrigerant condensation temperature rises to +50...+60 °C. The efficiency of the refrigerating machine cycle is significantly reduced at such temperatures. Compressors may experience increased wear. Emergency shutdown of the refrigerating machine by protective automation may occur. In this regard, seeking ways to the reduction of condensation temperature under given conditions is the actual problem.

Depending on the aggregate state of the substance to which heat of refrigerant condensation is transferred, there are:

– air-cooled condensers (ACC);

– liquid-cooled condensers (LCC) in which heat is transferred to water or another heat-transfer fluid;

– evaporative cooling condensers in which heat is removed to the evaporating water and the water vapor is removed by an airflow [1].

ACCs are the most common due to their simple design and low cost. However, in the case of using ACC, the condensation pressure of refrigerants increases with an increase in the temperature of atmospheric air since the condensation temperature in them must be 10 to 20 K higher than ambient temperature.

Liquid-cooled condensers require a source of cooled water. In some cases, water may be taken from natural reservoirs [2]. However, the water heated in condensers is more often cooled in cooling towers or dry coolers [3]. Cooling towers require fresh water. This limits their use in regions where there is a shortage of freshwater. In dry coolers, the water temperature will always be higher than ambient temperature, consequently, condensation temperature in the VCRM will be higher than in the ACC scheme. Thus, the energy efficiency of refrigerating machines will decrease.

Evaporative condensers also consume fresh water in their operation [4]. Therefore, the use of such condensers is not always possible in regions where water resources are limited. To solve these problems, it is proposed to consider the possibility of removing condensation heat through radiative cooling (RC). Radiative cooling is a method of removing heat to the environment by transferring it through the planet's atmosphere into outer space in a form of infrared radiation. Theoretical principles of implementation of radiative cooling are systematized in [5, 6]. The RC advantage consists in the theoretical possibility of obtaining temperatures lower than the atmospheric air temperature, without losing freshwater. Therefore, studies aimed at using radiative cooling to remove condensation heat can be considered relevant.

2. Literature review and problem statement

Conventionally, the possibility of using RC for direct removal of heat from cooled objects was considered in [7]. However, this approach to using RC is of limited applicability because RC is dependent on environmental conditions. Thus, obtaining of required temperatures is only possible for a limited part of the year. Also, when using RC for direct removal of heat from cooled objects, it is difficult to obtain negative temperatures since the amount of radiated heat decreases sharply with a decrease in temperature of the radiating surface.

Because of the limited efficiency of RC, a number of refrigeration systems have been developed that use RC in conjunction with other cooling methods [8].

A scheme was proposed in [9] for removing the heat of refrigerant condensation through RC. In this scheme, an additional LCC is installed in the VCRM refrigerant circulation loop downstream the ACC to which heat-transfer agent precooled by RC is supplied. The use of additional RC-cooled condensers makes it possible to increase overcooling and/or reduce condensation pressure resulting in an increase in the VCRM energy efficiency.

It is indicated in [10] that the scheme of removing the condensation heat due to RC can be used mainly for air conditioning in multi-story buildings. However, the scheme itself in which condensation heat is removed by RC has a number of theoretical advantages. In such a scheme, the temperature of the radiating surface can be much higher than the ambient temperature. In this case, the flux of thermal radiation increases substantially. At the same time, the radiating surface transfers heat to the environment by convection. Accordingly, there is no need to take measures preventing convective heat transfer. Additionally, condensation and freezing of water from atmospheric air do not occur on the radiating surface.

A preliminary theoretical assessment of the effectiveness of such a scheme was made in [11] according to which this approach will save up to 21 % of electricity. An experimental study of radiative cooling is presented in [12]. In experiments, the level of energy-saving was from 5 to 10 %. Thus, the shown experimental level of energy-saving turned out to be less than the results of preliminary theoretical calculation. The presented data for one daily cycle is not enough to evaluate the effectiveness of such systems in various climatic conditions.

Radiators with a selective coating used in studies [11, 12] cooled the heat-transfer agent both at night and during the day. Radiating surface in such radiators has a high emissivity in the infrared part of the spectrum as well as a high reflectivity in the visible part. This combination of optical properties increases amount of heat transferred by radiative cooling. Selective coating for radiators was first presented in [13]. Early versions of such coatings are relatively difficult to manufacture and are expensive. The possibility of using such coatings for round-the-clock cooling of the radiating surface to a temperature below that of atmospheric air was shown later in [14, 15]. Also, the coating presented in [14] is made of a polymer film with a silver coating and is easier to manufacture than the previously proposed coating options. However, such coatings are not durable and, therefore, we used a radiator without selective coating in this study. It can only cool at night.

It was previously noted that high condensation temperatures reaching 60 °C cause a decrease in the refrigerating machine efficiency [16]. Therefore, we considered the possibility of using a scheme with RC not to increase the overcooling of the refrigerant as was done in [12] but to reduce the condensation temperature. To do this, it is necessary to increase the LCC area in order to increase the heat flux to the heat transfer fluid.

Since the system radiators will cool only at night, it is necessary to use a cold accumulator [17]. Storage of cooled heat-transfer agent is created in this cold accumulator at night. During the daytime, the storage of heat-transfer agent will be used to reduce the condensation pressure.

A preliminary theoretical study of the system of the proposed type is presented in [18]. However, because of the comparative complexity of theoretical calculation of the combined scheme of condensation heat removal, the indicated theoretical calculation procedure was based on a number of simplifications. Their correctness must be confirmed by experimental studies.

The foregoing shows that further study is required on schemes of removal of condensation heat by means of radiative cooling.

3. The aim and objectives of the study

The study objective consisted in finding a way to improve refrigerating machine efficiency through the removal of condensation heat using radiative cooling. The study must confirm the efficiency of the proposed scheme of condensation heat removal, verify the correctness of the previously developed theoretical model and identify ways for further improvement of such systems.

To achieve the objective, the following tasks were set:

– conduct experimental studies of the refrigerating machine cycle at various temperatures of air and heat-transfer agent delivered to the condensers;

– analyze the energy efficiency of the experimental unit when condensation heat is removed only in an air-cooled condenser and when condensation heat is removed due to combined action of air-cooled and liquid-cooled condensers;

– study how the use of a particular refrigerant affects the energy efficiency of the refrigeration cycle in the temperature regime provided by the proposed scheme of condensation pressure reduction.

4. The study materials and methods

4. 1. Experimental refrigerating machine

To conduct the study, a refrigerating machine was designed. Its schematic diagram is shown in Fig. 1, *a*. It contains a loop of a single-stage refrigerating machine with a compressor CM and loops of heat-transfer agent circulation with pumps P1 and P2. The refrigerating machine operates according to the standard scheme. The only feature consists in presence of a liquid-cooled condenser (LCC) installed downstream the air-cooled condenser (ACC). The LCC can be manually connected to the scheme using ball valves BV1, BV2, and BV3. A heat-transfer agent is supplied to the LCC from the cold accumulator (CA). In the nighttime, the heat-transfer agent in the cold accumulator can be cooled down in a radiator (RAD).

The unit uses a small hermetically sealed piston compressor Wansheng WQ15HF (China) with refrigeration capacity Q_0=332.9 W at vaporization temperature t_0=–23.3 °C and condensing temperature t_c=+54.4 °C (low-temperature operation according to ASHRAE LT Standard). By all characteristics, this compressor is an exact analog of the Secop (Danfoss) SC15F compressor.

A finned-tube heat exchanger Kaideli FNHM 12/4 (China) with a heat exchange surface area of 4 m^2 is used as an air-cooled condenser. Its fan consumes 30 W and has a rated thermal emission capacity of 1200 W.

The liquid-cooled condenser is a cylindrical metal container (diameter: 220 mm; height: 625 mm) made from a receiver, model GVN VLR.33b.21.B6.C6.F4 (Turkey). Inside the container, there is a spiral wound from a copper pipe (outside diameter: 9.52 mm; length: 7 m). Area of the heat exchange surface of the spiral: 0.2 m^2. Refrigerant is fed to the coil via a copper pipe. The heat-transfer agent fills the space between the vessel and the coil.

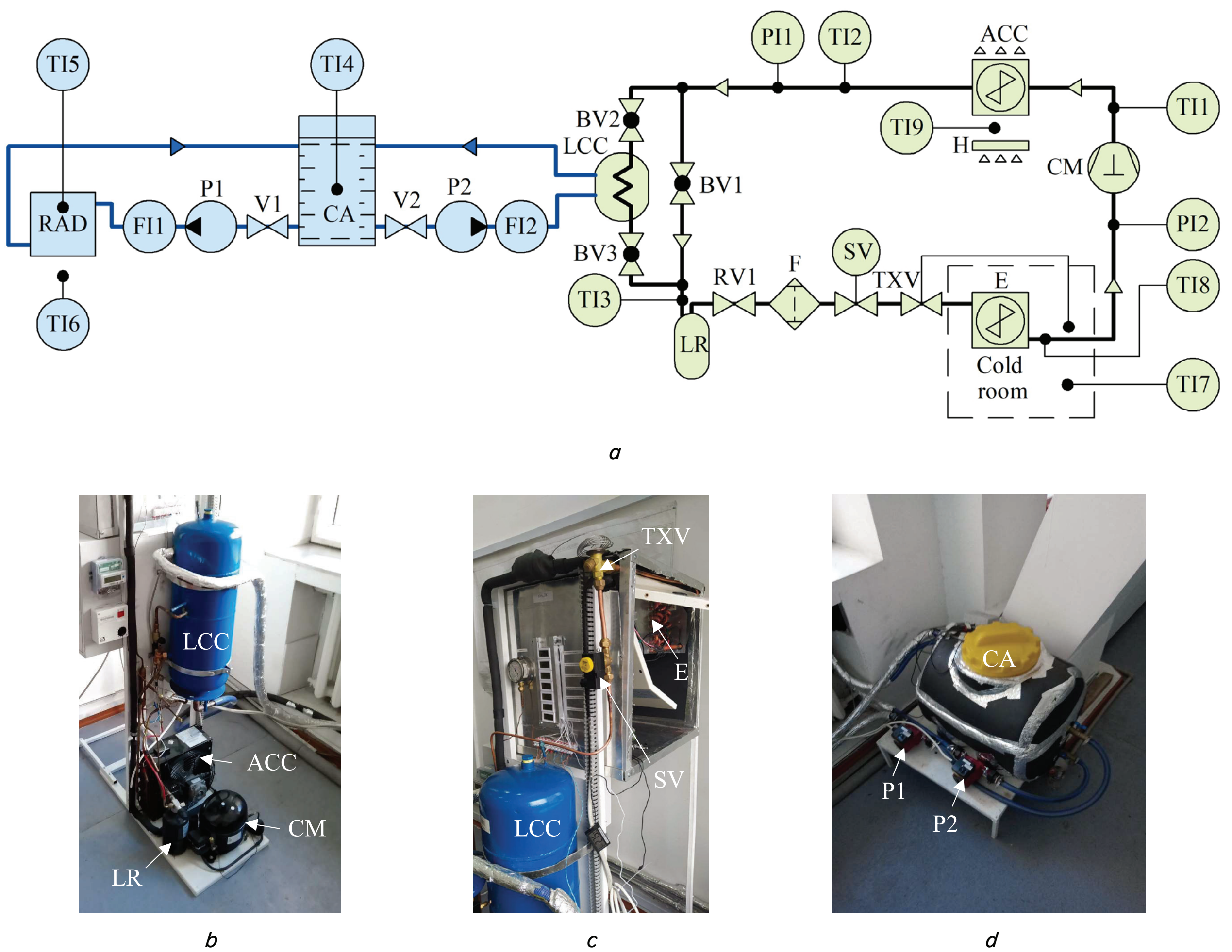


Fig. 1. Refrigerating machine: *a* – schematic hydraulics diagram; *b*, *c* – refrigerating machine; *d* – cold accumulator CA; ACC – air-cooled condenser; BV1, BV2, BV3 – ball valves; CA – cold accumulator; CM – compressor; E – evaporator; F – filter; FI1, FI2 – flow meters; H – heater; LCC – liquid-cooled condenser; LR – liquid receiver; P1, P2 – pumps; PI1, PI2 – pressure gauges; RAD – radiator; RV1 – Rotaloc valve; SV – solenoid valve; TI1...TI9 – temperature sensors; TXV – thermostatic valve; V1, V2 – valves

A linear receiver with an internal volume of 2.3 l is installed at the LCC outlet. A Rotaloc Frigopoint FP-RV-1-038 valve (Russia) is installed at the receiver outlet.

Thermostatic valve TXV Danfoss TN2, design 068Z3346, (Denmark) without external pressure equalization line and without MOP function with valve assembly No. 01 (068-2010) was used.

DE-0.45/2.5 (China) ribbed-pipe air cooler with heat exchange surface area of 2.5 m^2, electric power consumption of the fan: 30 W, and nominal refrigeration capacity of 450 W was used.

A radiator with a radiating surface area of 2 m^2 is used in the unit. The heat-transfer agent is distributed in it along parallel channels connecting distributing and collecting manifolds (Fig. 2). Radiating surface is made of a 0.8 mm thick aluminum sheet painted with white enamel PF-115 (Republic of Kazakhstan). Channels for the heat carrier are made from a round copper pipe.

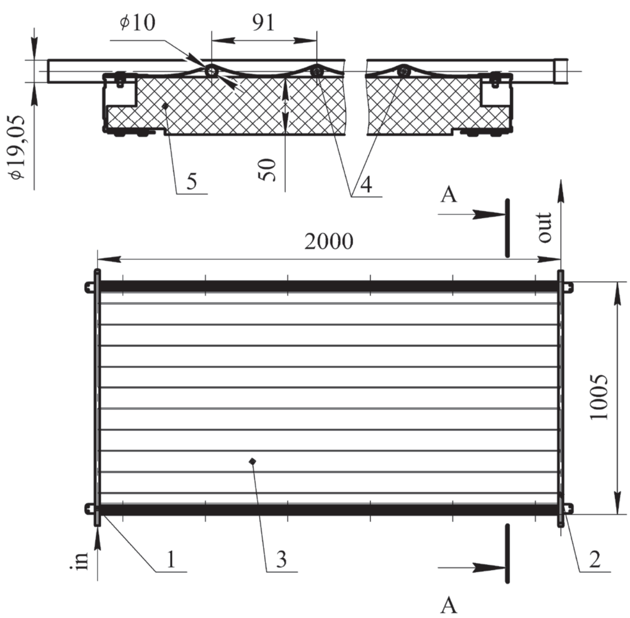


Fig. 2. The radiator: 1 – distributing manifold; 2 – collecting manifold; 3 – radiating surface; 4 – channels for heat-transfer agent; 5 – thermal insulation

Pumps P1 and P2 are circulation pumps with a wet rotor and a single-phase electric motor, model Unitech GPD 25-4S (Hungary). They are set to the minimum (first) rotational speed and consume 40 W.

KSC 40-281 cold accumulator (Republic of Kazakhstan) is a polymer horizontal cylindrical container with an internal volume of 100 l. To reduce heat inflows, the container is thermally insulated with 19 mm thick foamed rubber Misot flex ST-LR (Republic of Kazakhstan).

To shut off the refrigerant flow, Nantong 3/8" taps (China) are used.

The refrigerating machine is charged with Sanmei R134a refrigerant (China). The cold accumulator is filled with 100 kg of the heat-transfer agent which is an aqueous solution of propylene glycol with a mass concentration of 45 %.

The unit controller periodically turns on the compressor and maintains the temperature in the cooling chamber in the range from 8 to 5 °C.

All elements, except the radiator, are installed indoors. The radiator is installed outdoors.

4. 2. Climatic conditions in which the unit was used

The experimental unit considered in this study was designed for operation in a sharply continental climate. The study was conducted in the city of Almaty (43° north latitude). The climate of the city region is characterized by the significant daily fluctuation of atmospheric air temperature exceeding 10 °C. Data on atmospheric air temperature for 2018 were used for calculations (Fig. 3)

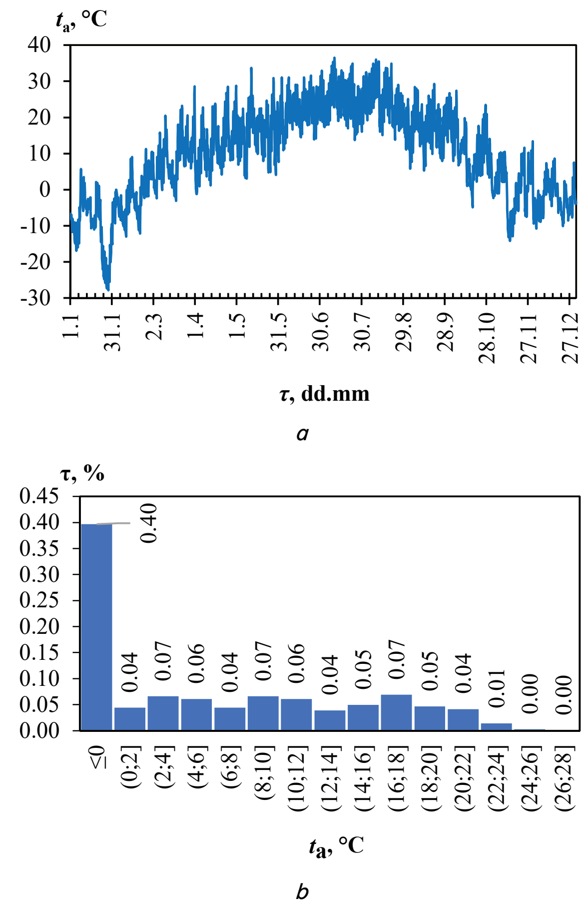


Fig. 3. Atmospheric air temperature (t_a, °C) in the city of Almaty for 2018: *a* – change during the year; *b* – a part of the total days in a year when minimum air temperature per day fell within the specified range

4. 3. Measurement of operating parameters of the experimental unit

Since there are no computer models to calculate the energy efficiency of refrigerating machines of a given condenser design, we conducted an experimental study and collected data on the unit operation in various temperature conditions.

In this study, the heat-transfer agent was not cooled in radiators at night since the dependence of the heat-transfer agent temperature on atmospheric air temperature by the end of the night was studied in [19–22].

To determine operating parameters of the refrigerating machine under various conditions, four experiments were carried out with the LCC on and off as well as at low or high atmospheric air temperature t_a at the ACC inlet (Table 1).

The essence of the experiments consisted in turning on the refrigerating machine for 1 hour with periodic measurement of its main operating parameters. Experiments 1 and 2 were performed with the LCC off to simulate the operation of a conventional refrigerating machine. To do this, valves BV2 and BV3 were closed and valve BV1 was opened. Pumps P1, P2 were off. Compressor CM, ACC fan, and air cooler fan E were on. In experiment 1, low-temperature air was supplied to the ACC since the heater H was turned off. In experiment 2, heater H was turned on which resulted in the air entering the ACC at increased temperature.

Experiments 3 and 4 were carried out with the LCC turned on. At the same time, valve BV1 was closed and valves BV2 and BV3 were open. Pump P2 was on. Compressor CM, fan ACC, and air cooler fan E were on. In experiment 3, the ACC was supplied with air at low temperature as the heater H was turned off and in experiment 4, the ACC was supplied with air heated in the heater H.

The ACC fan was turned on simultaneously with the compressor CM in all experiments.

When the pump P2 was on, the heat-transfer agent flow through the LCC was 0.454±0.01 m^3/h.

Table 1

The parameters varied in experiments

No.	Is the heat-transfer agent supplied to the LCC?	Indoor air temperature t_a, °C	Is the heater H on?	Air temperature at the ACC inlet, °C	The initial temperature of the heat-transfer agent supplied to the LCC, °C
1	No	+27.0	No	+27.0	–
2	No	+27.3	Yes	+38.6	–
3	Yes	+27.6	No	+27.6	+25.9
4	Yes	+27.6	Yes	+38.0	+31.9

The layout of measuring instruments used to monitor the operation of the refrigerating machine is shown in Fig. 1.

The heat-transfer agent flow rate was measured using EcoMeter Du15 1/2' meters (Russia) produced in metrological class B according to ISO 4064/1-77 with a relative measurement error of 2 % in the range of heat-transfer agent flow from 0.15 to 3 m^3/hour.

The temperature was measured in the range from −10 to +85 °C by means of Dallas Semiconductor DS18B20 temperature sensors (China) with an error of ±0.5 °C.

The surface temperature of the copper pipelines through which refrigerant moves was measured with sensors TI1, TI2, TI3, TI8. The sensor TI4 was immersed in the heat-transfer agent in the center of the cold accumulator. The temperature sensor TI7 measured air temperature in the center of the cooling chamber. The sensor TI9 measured the temperature of the air entering the ACC.

Condensation pressure p_c (PI1) was measured using a ColdGauge 202 mechanical pressure gauge (China) in an overpressure measuring range from 0 to 30 Bar with an error of ±0.5 Bar. Refrigerant boiling pressure (PI2) was measured using a P&M 68 mm mechanical manometer (China) in an overpressure measurement range from −1 to +15 Bar with an error of ±0.1 Bar.

Pressure gage PI2 measured pressure at the compressor inlet and pressure gage PI1 measured pressure at the ACC outlet. Using readings of these manometers, boiling point t_0 and condensation point t_c of the refrigerant were determined.

Energy consumption was measured by means of ORMAN CO-E711 R TX IP P RS 220V 5 (60) A watthour meter (Republic of Kazakhstan).

4. 4. Determination of the unit energy characteristics

Accurate measurement of compressor refrigeration capacity is a substantial challenge. However, with data on operating temperature conditions of the refrigerating machine, refrigeration capacity Q_0 and heat of condensation Q_c can be obtained based on the compressor manufacturer's data using Danfoss Coolselector 2 software (version 3.8.0, base 55.55.2.23.9.36) [23].

Refrigeration coefficient *COP* of the unit was calculated as a ratio of theoretical refrigeration capacity Q_0 of the compressor to measured energy consumption N_e for all elements of the unit (compressor, condenser fan, air cooler fan, solenoid valve).

4. 5. Calculation of energy consumption by the unit per day

Based on experimental data, it is possible to construct dependence of the refrigeration coefficient *COP* of the refrigerating machine on the temperature t_a of air entering the ACC. In this case, for the case when the heat-transfer agent does not enter the LCC,

$$COP_{\mathrm{ACC}}=a_{\mathrm{ACC}}-b_{\mathrm{ACC}}\cdot t_a, \quad (1)$$

where a_{ACC} and b_{ACC} are empirical coefficients.

When a heat-transfer agent is fed to the LCC, we also have the following when ignoring the change in its temperature:

$$COP_{\mathrm{LCC}}=a_{\mathrm{LCC}}-b_{\mathrm{LCC}}\cdot t_a, \quad (2)$$

where a_{LCC} and b_{LCC} are empirical coefficients.

Energy consumption by the unit at a given time can be calculated as a ratio of required refrigeration capacity to refrigeration coefficient plus energy consumption by the pump P1 (kW):

$$N_{\mathrm{e}}=Q_0/COP\,(t_a)+N_{\mathrm{e.p1}}, \quad (3)$$

where Q_0 is the refrigeration capacity of the unit at a given time, kW.

$N_{\mathrm{e.p1}}$ is energy consumption by the pump P1 supplying heat-transfer agent to the radiators, kW.

Let us assume for simplicity that required refrigeration capacity Q_0 does not change during the day. In this case, (1) or (2) is substituted in (3) depending on whether the heat-transfer agent is currently supplied to the LCC.

If the refrigerating machine operates all day without heat-transfer agent supply to the LCC, energy consumption per day, kWh, is:

$$E=\sum_{i=1}^{n}\tau_i\cdot N_{e.\mathrm{ACC}.i}\left(t_{a.i}\right), \quad (4)$$

where τ_i is the duration of time interval, hrs;

$N_{\mathrm{e.ACC}.i}\left(t_{a.i}\right)$ is energy consumption by the unit calculated at atmospheric air temperature $t_{a.i}$ at a given time, kWh.

n is the number of time intervals in one day.

If the heat-transfer agent supply to the LCC is turned on during the day, the energy consumption must be calculated separately for the periods when the LCC is turned off and when it is running. It is also necessary to take into account the energy consumed at night by the pump that supplies heat-transfer agent to the radiators. Then energy consumption by the unit per day (kWh) is:

$$E=\sum_{i=1}^{n}\tau_i\cdot N_{e.\mathrm{ACC}.i}\left(t_{a.i}\right)+\sum_{j=1}^{m}\tau_j\cdot N_{e.\mathrm{LCC}.j}\left(t_{a.j}\right)+\tau_n\cdot N_{e.\mathrm{p1}}, \quad (5)$$

where n is the number of time intervals per day when LCC is off;

m is the number of time intervals per day when LCC is on;

$N_{e.\mathrm{LCC}.j}\left(t_{a.j}\right)$ is the energy consumed by the unit, kW, at a given time when the heat-transfer agent is supplied to the LCC;

τ_n is night duration, hrs.

The dependence of changes in atmospheric air temperature t_a on time is based on climate data on the hottest summer day.

4. 6. Thermal power transferred to the liquid cooling condenser

The average value of thermal power coming to the heat-transfer agent in the liquid cooling condenser LCC is calculated on the basis of data on changes in temperature of the heat-transfer agent in the cold accumulator (W):

$$q_c=\frac{m_{\mathrm{HTL}}c_{\mathrm{HTL}}\Delta t_{\mathrm{HTL}}}{\tau}-N_{\mathrm{e.p2}}, \quad (6)$$

where m_{HTL} is a mass of heat-transfer agent in the unit, kg;

c_{HTL} is the thermal capacity of the heat-transfer agent, J/(kg·°C). For the given conditions, c_{HTL}=3656 J/(kg·°C);

Δt_{HTL} is a change in heat-transfer agent temperature in the unit for the considered period of time determined from thermometer TI4 readings, °C;

$N_{\mathrm{e.p2}}$ is energy consumption by the pump P2 supplying the LCC with a heat-transfer agent, kW.

5. The results obtained in studying the refrigerating machine

5. 1. Experimental study of the refrigerating machine cycle

Fig. 4 shows dynamics of changes in the refrigerating machine cycle parameters with a working air cooling condenser at different temperatures of atmospheric air.

Dynamics of temperature changes in the unit with a combined scheme of heat removal in condensers of air and liquid cooling at different temperatures of air entering the ACC are presented in Fig. 5.

Average values of the main parameters of the refrigerating machine with the compressor turned on for the period of observations in all experiments are presented in Table 2.

In Table 2, the absolute error in measuring the boiling point t_0 and the condensation temperature t_c was ±1.5 *°C*. Error in measuring the heat-transfer agent temperature t_{htl} was ±0.5 *°C*. When calculating the degree of pressure increase defined as a ratio of absolute condensation pressure p_c to absolute boiling pressure p_0, (p_c/p_0), the relative error was 10 %. When measuring overheating Δt_{sh} and overcooling Δt_{sc}, an absolute error was ±1.9 *°C*. Taking into account errors in temperature and pressure measurements, refrigeration capacity Q_0 of the compressor and the heat of condensation Q_c may change by 10 %. The relative error in calculating the *COP* was 12 %.

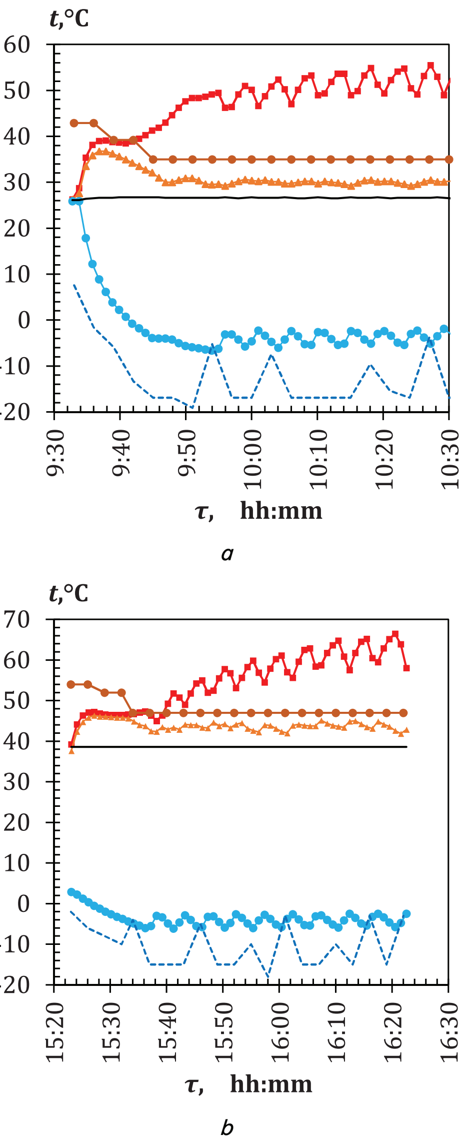


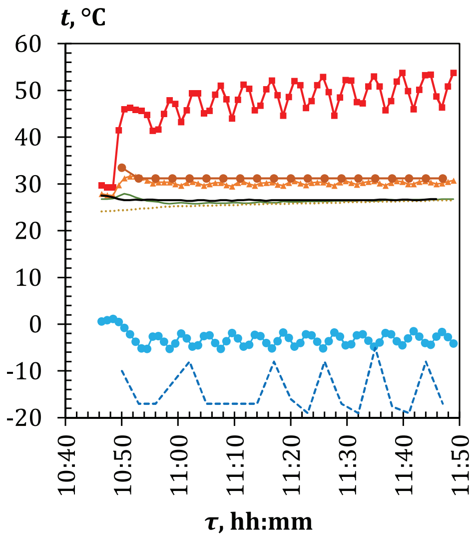
Fig. 4. Dynamics of temperature changes in the unit depending on time τ: *a* – experiment 1 at a low temperature of the air supplied to the ACC; *b* – experiment 2 at high temperature of the air supplied to the ACC; – delivery temperature (TI1); – refrigerant temperature at the outlet of the ACC (TI2); – air temperature in the cooling chamber (TI7); – air temperature at the ACC inlet (TI9); – boiling point of refrigerant, t_0; – refrigerant condensing temperature, t_c

The results of the experimental study of the refrigerating machine cycle at different air temperatures show that the difference between condensation temperature and temperature of incoming air was 7...8 *°C*.

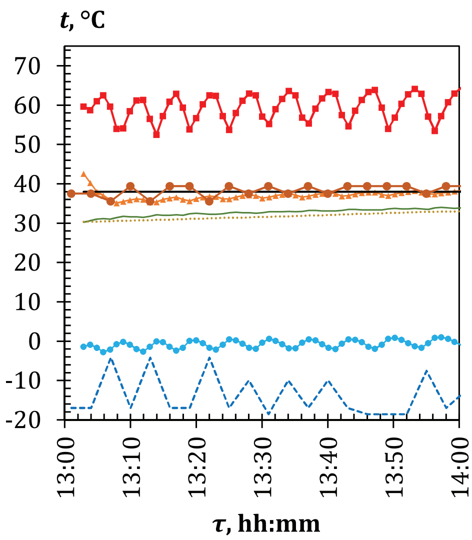
Table 2

Average parameters of the refrigerating machine for the observation period

No.	t_a	t_{htl}	t_c	t_0	p_c	p_0	p_c/p_0	Δt_{sc}	Δt_{sh}	N_e	Q_0	Q_c	*COP*
Meas. unit	°C	°C	°C	°C	Bar(g)	Bar(g)	–	K	K	W	W	W	W/W
1	+27.6	–	+35.0	-16.7	8.0	0.5	6.0	5.0	14.2	402	573.6	910	1.43
2	+38.6	–	+47.0	-16.7	11.9	0.5	8.6	2.5	10.4	421	452.7	809.8	1.08
3	+27.3	+27.0	+31.2	-17.1	7.0	0.5	5.3	4.3	14.3	425	585.7	914.9	1.38
4	+38.0	+33.1	+39.1	-17.4	8.9	0.5	6.6	1.3	13.9	417	501.5	838.9	1.20



a



b

Fig. 5. Dynamics of temperature changes in the unit depending on time τ: *a* – experiment 3 with the LCC turned on at a low temperature of air supplied to the ACC; *b* – experiment 4 with the LCC turned on at a high temperature of air supplied to the ACC; – delivery temperature (TI1); – refrigerant temperature at the ACC outlet (TI2); – refrigerant temperature at the LCC outlet (TI3); – heat-transfer agent temperature in the cold accumulator (TI4); – air temperature in the cooling chamber (TI7); – air temperature at the ACC inlet (TI9); – boiling point of refrigerant t_0; – condensation temperature of the refrigerant t_c

Refrigerant boiling point t_0 was 9 *°C* lower than the required temperature in the cooling chamber in all experiments.

According to the results of comparison of experiments 1 and 2, an increase in the temperature of the air entering the ACC for a conventional refrigerating machine led to an increase in condensation temperature to +47.0 °C. The degree of pressure rise increased by 43 %. Refrigeration capacity decreased by 27 % and refrigeration coefficient decreased by 32 %.

When air supplied to the ACC had a temperature of about +27.6 °C, turning on of heat-transfer agent supply to the LCC (in experiment 3) led to a 3.6 % increase in the refrigeration coefficient of the system compared to experiment 1. Accordingly, in this unit, the use of the scheme of condensing heat removal through the LCC is of low efficiency at a temperature of the air entering the ACC below +30 °C.

Comparison of the results of experiments 2 and 4 conducted at high temperatures of air entering the ACC showed the following. At an air temperature of +38.0 °C, the supply of heat-transfer agent at a temperature of +33.1 °C to the LCC led to a change in refrigeration coefficient from 1.08 to 1.20 W/W (or 11 %). Condensation temperature decreased from +47.0 to +39.1 °C. The degree of pressure growth has dropped from 8.6 to 6.6 (by 30 %).

5. 2. Analysis of energy efficiency of the refrigerating machine

A study of the dependence of refrigeration coefficient *COP* of the refrigerating machine on temperature t_a of air supplied to the air cooling condenser showed the following.

If the heat of condensation is removed only by ACC, then according to the results of experiments 1 and 2,

$$COP=2.309-0.032\cdot t_a. \qquad (7)$$

If LCC works, then according to the results of experiments 3 and 4,

$$COP=1.8258-0.0164\cdot t_a. \qquad (8)$$

It follows from a comparison of (7) and (8) that turning on the LCC in the unit increases the refrigeration coefficient at ambient temperatures above +31 °C.

Estimation of energy consumption by the refrigerating machine in a diurnal cycle of operation at different scenarios has shown the following.

In the first scenario, the unit operation was considered when condensation heat is removed only in the ACC. Change in the refrigeration coefficient during the day was determined (curve 1) based on the graph of changes in atmospheric air temperature (curve t_a in Fig. 6, *a*).

It was assumed in this and subsequent scenarios that the refrigeration capacity of the unit is constant during the day (Q_0=1 *kW*). The result of the calculation of energy consumption by the refrigerating machine is shown by curve 1 in Fig. 6, *b*. The unit consumed 17.83 *kW*h per day.

In the second scenario, the operation of the unit in which a cooled heat-transfer agent was supplied to the LCC from 13:00 to 17:00 was considered. During this period, the refrigeration coefficient of the unit increased compared to the first scenario (curve 2 in Fig. 6, *a*). To charge the cold accumulator all night (from 21:35 to 5:15), the pump P1 supplying heat-transfer agent to the radiators was switched on. Given the increased refrigeration capacity of the unit, the pump P1 has consumed 67 W. Because of its switching on at night, the refrigeration coefficient decreased (curve 2 in Fig. 5, *a*), and energy consumption increased (curve 2 in Fig. 6, *b*). The total amount of energy consumed per day was 18.10 kWh (which is 1.5 % more than in scenario 1).

In the third scenario, the operation of the refrigerating machine that worked in the same way as in the second scenario but with the use of energy-efficient pumps was considered. To compare, the pumps used in the unit had an efficiency of about 20 % in the most efficient mode of operation, and pumps of efficient models operated at low consumption of the heat-transfer agent can have an efficiency of about 35 % (according to the manufacturer). The pumps consumed 40 W in the considered experimental unit (Table 2) while more efficient pumps would consume 23 W. Then power consumption in experiments 3 and 4 would be 407.9 and 399.9 W, respectively. As a result, *COP* will be 1.44 W/W in experiment 3 and 1.25 W/W in experiment 4. Then change in the refrigeration coefficient of the unit when the heat-transfer agent is supplied to the LCC using efficient pumps is as follows:

$$COP=1.9-0.017 \cdot t_a. \tag{9}$$

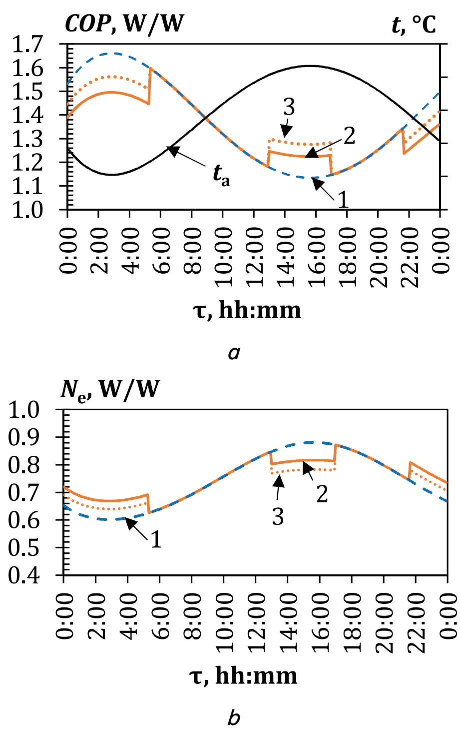


Fig. 6. Change of energy parameters during the day: *a* – refrigeration coefficient of units; *b* – energy consumption; 1 – operation with only ACC; 2 – operation with ACC and LCC; 3 – operation with ACC and LCC and also with highly effective pumps; τ – time

Energy consumption by pumps P_1 and P_2 was 38 W in a refrigerating machine with a refrigeration capacity of 1 kW. Thus, the change in the refrigeration coefficient occurred as shown by curve 3 in Fig. 6, *a*, and change in energy consumption occurred as shown by curve 3 in Fig. 6, *b*. In total, the unit consumed 18.07 kW per day which is 1.3 % more than in the first scenario.

5. 3. Calculation of the refrigeration coefficient in a cycle with different refrigerants used

Using the data on temperature conditions of operation of the refrigerating machine in experiment 4, the refrigeration coefficient of the sealed piston and volute compressors was calculated for operation with the use of different refrigerants. Table 3 presents the result of the calculation performed in Danfoss Coolselector 2 software.

Of all refrigerants listed in Table 3, it is not recommended to use R404a and R410a refrigerants in the long run as they have a high global warming potential. In Table 3, the refrigeration coefficient was calculated for the R410 refrigerant used in the Danfoss HRH032U5 volute compressor. For other refrigerants, calculations were performed using Danfoss MTZ018-4 piston compressor and Danfoss MLZ-015T2A volute compressor.

Table 3

Calculation of the refrigeration coefficient *COP* of compressors (t_0=–17.4 °C, Δt_{sc}=2 K, Δt_{sh}=14 K)

Refrigerant	Piston compressor		Volute compressor	
	t_c=+39 °C	t_c=+50 °C	t_c=+39 °C	t_c=+50 °C
R134a	1.32	0.90	1.73	1.21
R404a	1.44	1.03	1.71	0.95
R407c	1.42	0.96	1.96	1.40
R407f	1.48	1.09	–	–
R410a	–	–	1.58	1.28
R448a	1.61	1.15	1.73	1.07
R449a	1.61	1.15	1.73	1.06
R452a	1.64	1.16	1.79	1.17
R513	1.41	1.09	1.72	1.20

6. Discussion of the results obtained in the study of energy efficiency of the refrigerating machine

This study features the use of radiative cooling in the combined scheme of condensation heat removal to reduce pressure.

Comparison of the results obtained in the calculation of energy consumption in scenarios 1 and 2 has shown that the considered experimental refrigerating machine did not provide a reduction of energy consumption in a diurnal cycle under specified conditions. These calculations were performed under the condition that ordinary circulating pumps are used in the unit. If we suppose that high-efficiency circulating pumps will be used (scenario 3), the proposed scheme of condensation heat removal would theoretically give approximately the same level of energy consumption as with the conventional scheme using an air-cooled condenser. Therefore, the proposed unit scheme can be only used to reduce the risk of an emergency shutdown of the refrigerating machine in the daytime at high ambient temperatures.

Data on the daily level of energy consumption are consistent with the results of calculation according to the method proposed in [18] which confirms its correctness.

The obtained data on temperature conditions of the refrigerating machine operation (Table 2) are based on the results obtained in observing the values of at least eight compressor on/off cycles after stabilization of changes in boiling and condensing temperatures. Thus, the results of the calculations are based on stabilized operating parameters of the refrigerating machine.

In this study, the heat-transfer agent was cooled in radiators only at night because of the lack of selective coating on the radiating surface enabling heat removal during the daytime. The use of radiators without selective coating in scenarios 2 and 3 did not provide the energy-saving level compared to the scheme with radiators having a selective coating on the radiating surface, as in the study [12]. In the future, radiators with a selective coating applied on the radiating surface should be used to improve the energy efficiency of the system.

It follows from Fig. 5, *a*, that there was a 2 K change in temperature in the cold accumulator during 1 hr (experiment 3). The temperature in the cold accumulator changed by 2.5 K in experiment 4 (Fig. 5, *b*). According to formula (6), thermal power q_c transferred from refrigerant to heat-transfer agent in the LCC was 163±23 W in experiment 3 and 213.8±23 W in experiment 4. Comparison of transferred thermal power q_c and heat Q_c of refrigerant condensation (Table 2) has shown that no more than 26 % of total condensation heat is transferred to the heat-transfer agent in the LCC and 74 % is removed to air in the ACC and through the pipeline walls. It was assumed in the theoretical study [18] that most condensation heat will enter the heat-transfer agent.

Based on the data in [19], it is expected that by the end of the night, the temperature in the cold accumulator will be 5 °C higher than the minimum ambient temperature. Thus, in the conditions considered (Fig. 6, *a*), the temperature of the heat-transfer agent in the cold accumulator will be no more than +25.5 °C by the end of the night. To raise the temperature of the heat-transfer agent to the value from experiment 4, it must be heated to 7.6 °C. In experiment 4, heat power q_{HTL} of 253.8±23 W was applied to the heat-transfer agent in the cold accumulator taking into account the heat power of the pump. Thus, a minimum time during which the condensation temperature can be maintained by the LCC is:

$$\tau=\frac{m_{\mathrm{HTL}}c_{\mathrm{HTL}}\Delta t_{\mathrm{HTL}}}{3600\cdot q_{\mathrm{HTL}}}=\frac{100\cdot 3656\cdot 7.6}{3600\cdot(253.8+23)}=2.8\,\mathrm{hr}. \tag{10}$$

We obtained the weight of the heat-transfer agent required for one hour of operation by dividing the weight of the heat-transfer agent in the experimental unit by the minimum time during which the scheme of condensation heat removal will work. A maximum of 35 kg of the heat-transfer agent is required to maintain the condensation temperature for one hour. For a system with a refrigeration capacity of 1 kW, it is necessary to have 70 kg of heat-transfer agent per 1 hr of operation (or 65 kg of water).

It was previously established that by the end of the night, the temperature in the cold accumulator will be 5 °C higher than the minimum night temperature. In the climate of the city of Almaty in 2018 (Fig. 3, *a*), the night air temperature was never higher than +24 °C (Fig. 3, *b*). Thus, we can expect that the temperature in the cold accumulator will never exceed +29 °C by the end of the night in a year. If it is assumed that temperature in the cold accumulator will rise by 7.6 °C during the day cycle, then it will reach +36.6 °C by the end of the day. At the same time, such a high temperature in the cold accumulator will be observed for no more than three days a year (≈1 % of the year time). The temperature in the cold accumulator will not exceed +35 °C in the remaining 362 days (99 % of the year time). Thus, the heat-transfer agent temperature set in the experiments corresponds to that which can be expected when cooling the heat-transfer agent with RC.

The proposed scheme of reducing the condensation pressure can be applied if the following conditions are taken into account:

1. Atmospheric air temperature in summer often rises above +30 °C during the day.

2. There is a shortage of fresh water in the region where the unit is located, thus, the use of water and evaporative cooling condensers is limited.

3. There is space at the unit location for mounting of radiators (not less than 1.6 m^2 per 1 kW of refrigeration capacity taking into account the resulting weight of heat-transfer agent in the cold accumulator and required area of radiators according to [18]).

When operating the proposed unit in the winter, it is necessary to take measures to prevent freezing of the heat-transfer agent. To this end, it will be necessary to drain all radiators and pipelines installed outdoors. It is not recommended to use aqueous solutions of glycols as a heat carrier because this will substantially increase the unit cost.

Also, in the cold period of the year, there may be problems in the operation of the refrigerating machine in a connection with an excessive reduction of condensation pressure if the heat-transfer agent is not drained from the unit elements. If the heat-transfer agent is not drained, care must be taken to prevent it from circulating in a natural way through the LCC heat exchanger or supply of refrigerant to the LCC may be stopped by simultaneous closure of valves BV2 and BV3 (Fig. 1, *a*). If one of these valves is missing, the refrigerant will enter the LCC by natural circulation.

It is impossible to use the proposed scheme of reduction of condensation pressure if a capillary tube is used as a throttling device. This is explained by significant changes in the condensation pressure that lead to a substantial change in the amount of refrigerant supplied to the evaporator.

The growth of refrigeration capacity of the system makes it possible to use more efficient circulating pumps which can improve the energy efficiency of the proposed scheme. For the smallest models of pumps with a wet rotor, e.g., Wilo Star-RS 25/4), the highest value of efficiency (about 22 %) is reached when consumption of heat-transfer agent is about 1.8 m^3/hr. Thus, the specified value of efficiency will be reached at a fourfold increase in refrigeration capacity of the experimental unit that will result in a fourfold increase in consumption of the heat carrier passing through the LCC (from 0.45 to 1.8 m^3/hour). Therefore, the proposed scheme of condensation heat removal is recommended for refrigerating machines with a refrigeration capacity of more than 2,000 W.

When using the obtained regularities, it should also be borne in mind that they are valid for refrigeration systems with small sealed piston compressors having refrigeration capacity up to 10 kW. The obtained dependences (7) and (8) can be used to calculate the refrigeration coefficients of units if:

– temperature of atmospheric air is in the range from +25 to +40 °C;

– temperature of the heat-transfer agent is in the range from +25 to +35 °C.

To increase the accuracy of calculation of energy consumption per day according to the proposed method in further studies, it is recommended to use more complex polynomial dependences on ambient air temperature t_a to calculate the refrigeration coefficient *COP*.

The study did not take into account the amount of heat transferred from the heat-transfer agent to the environment through the pipeline walls. These pipelines were insulated, so heat loss through their walls was considered negligibly small. However, a significant deviation from air temperature of +27 °C in the room where the refrigerating machine is installed can affect the unit operation.

The procedure of calculating the refrigeration coefficient according to formula (2) does not take into account changes in temperature of the heat-transfer agent supplied to the LCC. Accordingly, the obtained results of the calculation of energy consumption are correct, provided that the change in heat-transfer agent temperature in the cold accumulator does not exceed 10 °C in a diurnal cycle.

The considered experiment scheme involves the use of a heat-transfer agent with a pre-set temperature and also makes it possible to control the temperature of the air entering the ACC. This minimizes the impact of environmental conditions on the unit ensuring reproducibility of experimental results.

Analysis of the data in Table 3 has shown that the choice of R134a refrigerant for the experimental unit in this study was not optimal. When using LCC, the condensation temperature drops to +39 °C. At this temperature, the refrigeration system with piston and volute compressor and R134a refrigerant will be comparable in energy efficiency with the systems running on environmentally friendly refrigerants such as R407c, R448a, R449, R452a. Taking into account the high consumption of R134a refrigerant, the use of R134a refrigerant is not recommended for the proposed type of the unit as it increases the compressor cost but does not increase the refrigeration coefficient of the unit.

To forecast annual energy consumption by refrigeration systems with a given design of condensers based on the obtained data, it is necessary to construct a detailed computer model. The construction of such a model is connected with a substantial theoretical complexity of calculation of heat transfer processes if the refrigerant is condensed in two series-connected condensers.

7. Conclusions

1. Operation of a refrigerating machine in which condensation pressure is reduced by supplying the heat-transfer agent pre-cooled by radiation cooling to an additional condenser was experimentally studied. At an ambient temperature of +38.0 °C, the supply of the heat-transfer agent with a temperature of +33.1 °C to the liquid cooling condenser has resulted in a 7.9 °C decrease in condensation temperature. At the same time, the degree of pressure growth decreased by 30 %. The above temperature of the heat-transfer agent can be reached at night by removing heat in radiators without selective coating on their radiating surface. Therefore, the obtained results confirm the action of the proposed scheme of reducing the condensation pressure when using radiators without selective coating on the radiating surface.

2. When supplying air at about +27.6 °C to the air cooling condenser, switching on of heat-transfer agent supply to the liquid cooling condenser led to a 3.6 % increase in the refrigeration coefficient of the system.

When air with a temperature of +38.0 °C was supplied to the air cooling condenser, the switch on of supply of heat-transfer agent to the liquid cooling condenser has resulted in an 11 % increase in refrigeration coefficient. Thus, the use of the proposed scheme of condensation heat removal during daylight hours leads to a decrease in energy consumption.

Energy consumption by the refrigeration system with refrigeration capacity Q_0=1 kW using a combined scheme of removal of condensation heat was calculated for climatic conditions of the city of Almaty. For these conditions, daily energy consumption by the considered refrigerating machine will be 1.3 % higher than for a standard system with an air-cooled condenser. Thus, the proposed scheme of removal of condensation heat did not provide an increase in energy efficiency during the diurnal cycle.

3. Comparison of refrigeration cycles with different refrigerants for the considered unit has shown that refrigerants such as R407c, R448a, R449, R452a should be used to improve energy efficiency.

Acknowledgment

The study was funded by the Science Committee of the Ministry of Education and Science of the Republic of Kazakhstan (Grant No. AP09258901).

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