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COOLING CAPACITY OF EXPERIMENTAL SYSTEM WITH NATURAL REFRIGERANT CIRCULATION AND CONDENSER RADIATIVE COOLING

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The surface of the Earth is a source of radiation of thermal energy, which, passing through the atmosphere, is partially absorbed while the bulk of the energy is released into the surrounding outer space. A cooling technique based on this physical phenomenon is known as radiative cooling (RC). It is possible to reduce the consumption of electricity for cooling, as well as to reduce capital costs, by integrating the unit with radiative cooling directly into the circulation circuit of the refrigerant of the refrigeration machine. An experimental refrigeration system has been designed, in which in the cold periods of the year the removal of heat from the cooled object is carried out due to the mode of natural circulation of the refrigerant from the evaporator to the heat exchanger, cooled by radiative cooling. A refrigeration system with natural circulation and radiative cooling of the refrigerant R134a was experimentally studied during the autumn period in Almaty. The experimental study established that the chamber is cooled with the help of the examined system while the temperature in the cooled volume is maintained by 5...7 K above ambient air temperature at night. The dependence of the air temperature in the refrigerating chamber on the temperature of the atmospheric air has been determined. A procedure for assessing the cooling capacity of the system has been devised.

The study reported here demonstrated the possibility of using radiative cooling to remove heat under the mode of natural circulation of the refrigerant.

The refrigeration system reduces energy consumption in the cold seasons by diverting heat to the environment without the compressor operating

Keywords: radiative cooling, effective radiation, natural circulation, refrigeration machine, thermosiphon system, energy saving

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1. Introduction

Refrigeration systems consume up to 17 % of the world's electricity. At the same time, their number continues to grow. A significant part of the electricity consumed by refrigeration systems is generated by burning fossil fuels. Accordingly, an increase in energy consumption by refrigeration systems leads to an increase in the adverse impact on the planet's climate. In this regard, an urgent task is a search for ways to improve the energy efficiency of refrigeration systems. Reducing the level of energy consumption can be achieved through the use of alternative cooling techniques, implemented without electricity use.

Objects located on the surface of the Earth are a source of radiation of thermal energy, which, passing through the atmosphere, is partially absorbed; the bulk of the energy is released into the surrounding outer space. As a result of studying the Earth in the infrared range of wavelengths (effective radiation), objects are cooled below the temperature of the atmosphere in the near-surface layer [1–4].

A cooling technique, which involves removing heat from a cooled object by means of thermal radiation through the planet's atmosphere into the surrounding outer space, is widely known as radiative cooling (RC).

Radiative cooling makes it possible to obtain temperatures below the temperature of atmospheric air without electricity use. Therefore, it can be used to design refrigeration systems with improved energy efficiency.

2. Literature review and problem statement

A standard refrigeration system using RC includes a device for transferring heat to the environment, which is called a radiator. The heat carrier that took the heat off the cooled object is transferred to the radiator. This is where it cools down. As a rule, a liquid is used as a heat carrier. Pumps are used to supply a liquid heat carrier to radiators [5]. Transportation of a liquid heat carrier requires energy expenditure. To reduce energy losses while maintaining a minimum cost, various radiator designs are built, characterized by minimal hydraulic resistance. For example, a radiator proposed in [6] is made of two metal sheets, the space between which is filled with liquid.

In work [7], a significant reduction in cost was achieved by designing a radiator of an open structure, in which the heat carrier was cooled by flowing on the surface of the metal sheet. This approach leads to significant contamination of the heat carrier, and its losses, and is also associated with large consumption of electricity for the transportation of the heat carrier. Therefore, radiators with an open structure are used quite rarely.

In some cases, atmospheric air is used as a heat carrier [8]. However, this significantly increases the internal volume of the channels for transporting the heat carrier, which significantly increases the overall dimensions of the refrigeration system. At the same time, electricity is still consumed to transport the gaseous heat carrier.

On the other hand, radiators have limited performance and usually remove no more than 100 watts from 1 m^2 of the radiating surface. To reduce their payback period, work [9] proposed using radiators in the daytime as solar collectors for heating the heat carrier. In addition, in the cited work, in order to minimize the cost of the system and reduce energy consumption, it is proposed to place batteries for the heat carrier under the ceiling of the cooled room in the immediate vicinity of the radiators. For such an arrangement of components, it is required to design the building taking into consideration the structural features of the cooling system. The system reported in [10] also uses radiators for heating and cooling the heat carrier. In it, the elements of the system can be placed at a considerable distance from each other, which simplifies the arrangement of the system in the building. However, in any case, in order to heat the heat carrier, radiators must possess low emissivity in the infrared range while for cooling their emissivity must be high. Therefore, it is difficult to design radiators that simultaneously effectively cool the heat carrier at night and heat it in the daytime. In addition, such a combination of heating and cooling systems leads to an increase in the complexity of the piping system through which the heat carrier is circulated. Therefore, in practice, it is difficult to use radiators simultaneously for cooling and heating the heat carrier.

If the required temperature of the object to be cooled is below the ambient air temperature, it is impossible to maintain the temperature continuously throughout the year with the help of RC alone. As a result, for a year-round refrigeration supply, it is necessary to use refrigeration systems that combine the use of radiative cooling with other cooling techniques. As a rule, a traditional steam-compression refrigeration machine is used as the main cooling technique. The refrigeration machine as part of the refrigeration system operates during periods when RC does not provide the required temperature regime. In some cases, the refrigeration machine can directly cool the air supplied to the premises [11]. In other cases, both the refrigeration machine and radiators cool the intermediate heat carrier, which is used to lower the temperature of the air [12] or the refrigerated product [13]. In another variant of the joint use of the refrigeration machine and radiators, it is proposed to use the heat carrier cooled in the radiators to reduce the condensation pressure of the refrigeration machine in the daytime [14]. In some cases, the heat carrier cooled in radiators can be used periodically both to remove heat from the cooled object and to reduce the condensation pressure [15]. In all the above options for the joint use of the refrigeration machine and radiators, two circuits are actually built, one of which serves to circulate the heat carrier, and the other to circulate the refrigerant. The presence of two circuits leads to a significant increase in capital costs for designing such a refrigeration system and leads to a complication in its automation scheme.

The refrigeration system can be significantly simplified if the circulation circuit of the heat carrier is excluded while the refrigerant can be directly cooled in the radiators. Thus, intermediate heat exchangers are excluded, which should reduce the temperature difference between the cooled object and the radiating surface of the radiator.

In a system proposed in [16], the removal of heat from the cooled object is carried out due to the natural circulation of the refrigerant through radiators with a change in its aggregate state. It includes a radiator and an evaporator. The refrigerant evaporates in the evaporator due to the heat coming from the object being cooled. It then rises into the radiator where it condenses due to radiative cooling. Next, the condensed refrigerant drains back into the evaporator. The reported experimental data show the performance of the radiator at the level of about 20 W/m^2 in summer conditions at an atmospheric temperature of about +20 °C. At the same time, the data in the cited study refer to the daily period only, and, therefore, it is difficult to predict the behavior of the system under other environmental conditions.

The system described in [16] can maintain the temperature of the cooled object only in those periods of the year when the temperature of the atmospheric air is lower than the required temperature of the cooled object.

Our review of the scientific literature reveals that the possibility of cooling due to the natural circulation of a refrigerant that removes heat into the environment due to RC has not been studied in detail. Therefore, further research in this area is needed.

3. The aim and objectives of the study

The aim of this work is to determine the performance characteristics and identify the features of the functioning of the refrigeration system in which cooling occurs due to the natural circulation of the refrigerant, which transmits heat to the environment due to radiative cooling. This will make it possible to identify further ways to improve such systems.

To accomplish the aim, the following tasks have been set:

– to monitor the dynamics of temperature changes in the refrigeration system of the proposed design;

– to determine the cooling capacity of the designed system.

4. The study materials and methods

4. 1. Experimental refrigeration system

To implement cooling due to the natural circulation of the refrigerant with its condensation in the radiator, an experimental refrigeration system is proposed, the scheme of which is shown in Fig. 1.

The experimental refrigeration system includes an AC air cooler, an R radiator, a C2 steam manifold, a CM compressor, V1...V4 valves, CT throttling device. At the outlet of the air cooler, there is a steam pipeline C1 going to the steam collector through the V2 valve. The suction pipe of the CM compressor is connected by pipeline to the steam collector C1. The discharge pipe of the CM compressor is connected to the C2 steam manifold by a pipeline. The C2 steam manifold is connected to the radiator R. At the outlet of the radiator R, there is a receiver LR, from which two pipelines exit. The first pipeline goes to the CT throttle device and then to the AC heat exchanger. The second pipeline is connected via a V1 valve to the AC air cooler inlet.

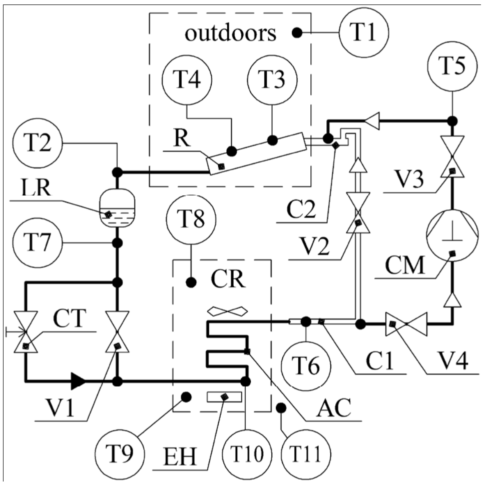


Fig. 1. Structural diagram of the refrigeration system: AC — air cooler; C1, C2 — collectors; CM — compressor; CR — refrigerating chamber; CT — capillary tube; EH — electric heater; LR — liquid receiver; R — radiator; T1...T11 — thermometers; V1...V6 — valves

If the ambient air temperature is 5...10 °C below the required temperature of the cooled object, the system operates in the mode of natural circulation of the refrigerant. In this case, the refrigerant evaporates in the AC air cooler, rises through the V2 valve into the radiator R, and condenses. It then drains through the open V1 valve back into the heat exchanger.

If the ambient air temperature is higher than the required temperature of the refrigerated object, the CM compressor is turned on, and the V1 and V2 valves are closed. As a result, the hot refrigerant is injected into the radiator R. There, it is cooled and condensed. Then it enters the het exchanger through the CT throttling device where it boils at the pressure generated by the compressor corresponding to the required temperature in the chamber. The degree of an increase in the compressor pressure is about 3–5 for the R134a refrigerant. For a given case, the boiling point of the refrigerant can be from −5 to +15 °C. Thus, maintaining the required temperature of the object under cooling is due to the implementation of natural circulation or a vapor-compression refrigeration cycle.

Technical results from the use of the proposed solution are as follows:

– stable maintenance of the temperature of the cooled object during the year;

– reduction of energy consumption over the annual cycle of operation by eliminating the need to turn on the compressor in the cold period of the year;

– reduction of the annual operating time of the compressor, which would prolong its service life.

The proposed design bears significant similarities with the well-known thermosiphon cooling systems [17–19] in which the refrigerant circulates in winter through the refrigeration circuit due to natural convection through the evaporator and condenser. Similar designs have previously been tested by such manufacturers of refrigeration equipment as Mayekawa [20], Carrier (Aquasnap Puron chillers, USA), and Gea (installation at the Kez Cheese Factory, Russia [21]). However, the RC system theoretically provides a lower refrigerant temperature at the outlet of the radiator compared to the output from the air-cooled heat exchanger [22, 23].

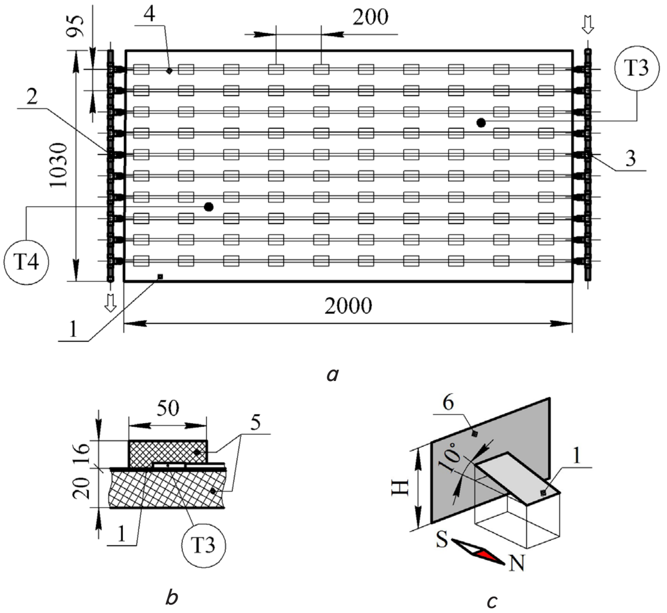
The experimental refrigeration system uses the R134a refrigerant. Polyester oil is used to lubricate the compressor.

The estimated cooling capacity of the system is 150 W. In this case, the speed of movement of the vapor and liquid refrigerant in the pipelines during the implementation of the natural circulation mode is not more than 0.3 m/s. Accordingly, the liquid pipeline at the outlet from the receiver has a diameter of 12 mm, and the steam pipeline rising to the radiator is 19 mm.

The AC air cooler is a ribbed tube heat exchanger with a fin surface area of 5.5 m^2. The AC air cooler is designed to remove heat from the object being cooled. Two XD1238A2HST (China) fans mounted on a diffuser are used to circulate air through AC. When two fans operate, 36.86±17.24 W of heat is generated.

The radiator (Fig. 2) is a plate-tube heat exchanger with a radiating surface area of 2 m^2. The radiating surface is made of a 0.8 mm thick aluminum sheet and painted with the white paint PF-115 (Russian Federation). 10 channels for the refrigerant, made of copper pipe with a diameter of 12.5 mm, are attached to the radiating surface. The distribution and receiving collectors are made of copper pipe with an outer diameter of 19 mm. The radiator is located outdoors. Its radiating surface faces the night sky and is arranged at an angle of 10° to the horizontal plane (Fig. 2, *c*). In this case, it is oriented in the northern direction. On the south side of the radiator, there is the wall of the building, the height of which H is about 30 m. This wall eliminates the possibility of direct sunlight hitting the radiating surface of the radiator.

To simulate the thermal load on the air cooler, a resistive electric heater EH is installed in the refrigeration chamber with the regulation of thermal power by changing the supply voltage.



a

b *c*

Fig. 2. Radiator: *a* — top view; *b* — scheme of fixing the thermometer on the radiating surface; *c* — scheme of radiator placement in space; 1 — radiating plate; 2 — distribution manifold; 3 — receiving collector; 4 — parallel channels for the heat carrier; 5 — thermal insulation; 6 — a wall of the building; T3, T4 — thermometers

The CT capillary tube as a throttling device is used only when the compressor is turned on.

In addition, there is a compressor in the scheme for the implementation of the refrigeration cycle during periods when it is impossible to maintain the required temperature in the cooled volume due to RC. The compressor is a small reciprocating compressor in a sealed housing.

As a refrigerating chamber, the freezer Snezh MLG-500 (Russian Federation) is used, the overall dimensions of which are 1.4×0.8×0.6 m; the internal volume is not more than 0.73 m^3.

The experimental refrigeration system is located in the city of Almaty, in the northern hemisphere at a latitude of 43°.

The current consumed by the EH heater is measured with the Appa A9 clamp meter (Taiwan). The voltage supplied to the heater is measured using the Uni-T UT50C multimeter (China) with an error of ±5.8 V.

The temperature of the refrigerant in the pipelines of the system, as well as the air temperature inside and outside the cooled volume, is measured using electronic thermometers Dallas Instruments DS18B20 (China). The measurement error is ±0.5 °C.

Thermometers T2, T5, T6, T7 for measuring the temperature of the refrigerant are fixed on the surface of the pipelines under the layer of thermal insulation.

Thermometers T3 and T4 are fixed on the radiating surface of the radiator (Fig. 2, *b*). They are in contact with the radiating surface, and they are closed from above with a layer of thermal insulation.

The ambient air temperature is measured using the thermometer T1, which is fixed at a height of 1.5 m above ground level. From above, the thermometer is covered with a canopy, which excludes radiative heat exchange between this thermometer and the night sky.

The T9 thermometer is fixed at the bottom of the cooled volume at the air inlet to the air cooler fan. The T8 thermometer is fixed at the top of the cooled volume.

The T11 thermometer measures the air temperature in the room where the CR refrigeration chamber is located. It is fixed at a height of 0.5 m above the floor level.

4. 2. Calculation of cooling capacity

The cooling capacity of the system is calculated by an indirect method based on the thermal balance of the refrigeration chamber.

The thermal balance of the refrigeration chamber (Fig. 3) consists of the thermal cooling capacity of the air cooler Q_0, the power of the heater Q_h, the thermal power of the fan motors of the air cooler Q_f, the heat flow through the wall of the refrigeration chamber Q_w:

$$Q_0+Q_w+Q_f+Q_h=0. \quad (1)$$

Hence, the cooling capacity of the air cooler:

$$Q_0=-(Q_f+Q_h-Q_w). \quad (2)$$

In (2), the thermal power of the fan motors Q_f, and the thermal power of the heaters Q_h are taken to be equal to their power consumption. The electrical power consumed can be directly measured.

The heat flow through the walls of the cooled volume Q_w depends on the difference between the temperature of the air in the cooled volume and the temperature of the air surrounding the cooled volume. The dependence of Q_w on Δt_w was determined experimentally. To that end, we turn on the fans of the air cooler in the refrigerator, and, by changing the power supply voltage of the heater EH, we set its thermal power. Leave the air cooler disconnected, blocking the supply of refrigerant to it. Then, after stabilizing the temperature in the refrigerator:

$$Q_f+Q_h=Q_w. \quad (3)$$

Since the air temperature at different points of the cooled volume is different, we calculate the average temperature difference on the enclosure of the cooled volume from the expression:

$$\Delta t_w=\frac{\sum_{i=1}^{n} t_{\text{in}.i}}{n}-\frac{\sum_{j=1}^{m} t_{\text{out}.j}}{m}, \quad (4)$$

where $t_{\text{in}.i}$ is the temperature of the temperature sensor in the refrigerating chamber, °C;

$t_{\text{out}.j}$ is the temperature of the temperature sensor outside the refrigerating chamber, °C;

n is the number of temperature sensors inside the refrigeration chamber;

m is the number of temperature sensors outside the refrigeration chamber.

The layout of the temperature sensors is shown in Fig. 3.

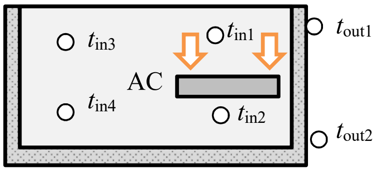


Fig. 3. Schematic arrangement of thermometers in the vertical cross-section of the refrigeration chamber

By setting the heat output of the heater, it is possible to determine the average temperature difference on the enclosure of the cooled volume. Next, based on the data obtained, it is possible to calculate the dependence of the average temperature difference on the thermal power entering the refrigeration chamber. By changing the value of Q_h and measuring Δt_w, we obtain:

$$Q_w=0.0802\cdot\Delta t_w^2+2.7994\cdot\Delta t_w, \quad (5)$$

where Δt_w is the difference between the average temperature within the refrigerated volume and the ambient temperature, K.

Dependence (5) holds if the Δt_w value is in the range from 0 to +50 K.

The absolute error ΔQ_{w1}, due to the procedure of conducting the experiment, when calculated from (5), can be determined from the expression:

$$\Delta Q_{w.1}=0.310\cdot\Delta t_w+15.275. \quad (6)$$

In addition, when calculating the amount of heat passing through the walls, it is worth considering the error when calculating Δt_w.

Absolute error in calculating the temperature difference Δt_w, K [24]:

$$\Delta\left(\Delta t_w\right)=1{,}1\cdot\sqrt{\Delta t_{\text{in}}^2+\Delta t_{\text{out}}^2}, \quad (7)$$

where Δt_{in} is the absolute error in measuring the temperature in the volume to be cooled, K;

Δt_{out} is the absolute error in measuring the ambient temperature, K.

The relative error in calculating the temperature difference Δt_w:

$$\delta(\Delta t_w)=\frac{\Delta(\Delta t_w)}{\Delta t_w}. \tag{8}$$

Given that the Δt_w value has an absolute error $\Delta(\Delta t_w)$, let us determine the absolute error in calculating the amount of heat entering through the enclosures, due to the uncertainty of the Δt_w value:

$$\Delta Q_{w.2}=1.1\cdot\sqrt{\left[\delta(\Delta t_w)\right]^2}\cdot\left[\sqrt{2}\cdot\left(\begin{array}{l}0.0802\cdot\Delta t_w^2+\\+2.7994\cdot\Delta t_w\end{array}\right)\right]. \tag{9}$$

The total absolute error in calculating the amount of heat entering through the enclosure:

$$\Delta Q_w=1{,}1\cdot\sqrt{\left(\Delta Q_{w.1}\right)^2+\left(\Delta Q_{w.2}\right)^2}. \tag{10}$$

Fig. 4 shows the dependence of the difference Δt_w between the average temperature inside the volume being cooled and the ambient temperature on the amount of heat entering the cooled volume (curve 1). Taking into consideration possible measurement errors, the total thermal power may be in the range between curves 2.

Curve 1 was built on the basis of empirical data using the methodology described above. Curves 2 show the boundaries of possible changes in Q_w values depending on Δt_w taking into consideration measurement errors.

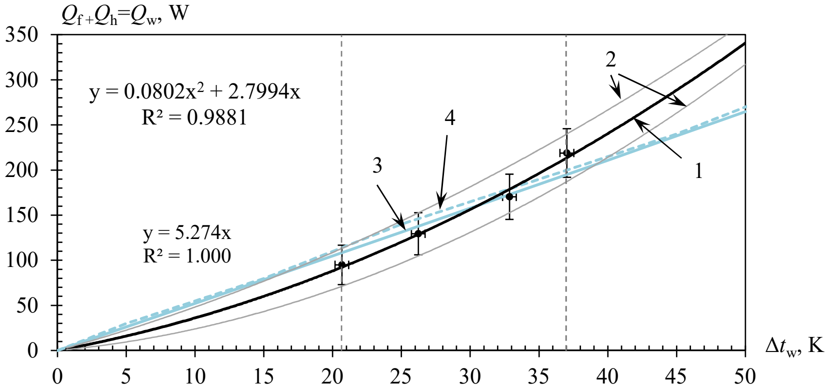


Fig. 4. Dependence of the total thermal power Q_w generated inside the cooled volume on the difference between the temperature inside the cooled volume and the ambient air temperature Δt_w

Curve 3 was built using a computer model which calculates the actual values of the heat transfer coefficients. In this case, the following conditions are met: the outer surface of the wall of the cooled volume is blown with air at a speed of 1.5 m/s, and the fans generate a mass airflow in the cooled volume at the level of 1.0 kg/s.

In addition, the amount of heat passing through the walls of the cooled volume was calculated using the software "Inflow 2.1 free" (curve 4). The software is used to calculate the thermal balance of refrigeration chambers. It does not perform detailed calculations of the heat transfer coefficients and accepts their standard values characteristic of refrigeration chambers. Based on the results of the experiment, a nonlinear relationship is observed between the thermal power and the temperature difference Δt_w. In this case, according to all known theoretical models (curves 3, 4), the dependence should be close to linear.

The discrepancies between theoretical modeling and our experiment can be explained by the fact that in theoretical modeling, the temperature of the air inside the entire cooled volume is considered to be the same. In addition, when calculating, the speed of air movement along all internal surfaces also remains the same, although, in reality, it would differ. Nevertheless, dependences 3 and 4 fall within the range of possible values of experimental data taking into consideration errors, provided Δt_w<40 K. Accordingly, it can be considered that the empirical curve coincides, in general, with the results of the theoretical calculation.

Fig. 5 shows the result of calculating the absolute error ΔQ_w depending on the temperature difference Δt_w.

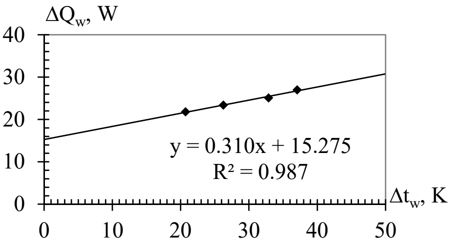


Fig. 5. Dependence of the absolute error ΔQ_w in determining the amount of heat passing through the walls of the freezer on the difference Δt_w

Fig. 5 demonstrates that the absolute error in determining the amount of heat ΔQ_w increases with increasing Δt_w.

5. Results of the experimental study of the refrigeration system

5. 1. Results of monitoring the dynamics of temperatures in the system

To determine the characteristics of the designed experimental refrigeration system during its operation under the mode of natural circulation of the refrigerant, we experimentally observed temperature changes at its individual points. The compressor of the refrigeration system was turned off during the experiment. The observations were conducted from August 28 to September 6, 2019, in the city of Almaty. In the study period, the average daily temperatures of atmospheric air were close to +22 °C (Fig. 6). The overall level of cloudiness in some periods rose to 100 %.

The daily plot of temperature changes in the refrigeration system for September 3 is shown in Fig. 7.

The surface temperature of the radiator (T3 and T4) at night was 1 K lower than the ambient air temperature.

The surface temperature of the air cooler (T10) was 3.5 K lower than the air temperature inside the cooled volume; at night, it fell below the air temperature in the room where the refrigeration chamber is located (T11).

At night, the temperature in the cooled volume was maintained by 5...7 K above ambient temperature.

The radiator is located on the north, shadow side of the building, and, therefore, during the daytime the surface temperature of the radiator (T3 and T4) remained below the ambient air temperature (T1). The cooling process continued in the chamber and the temperature was set to 2...3 K above ambient air.

The temperature inside the cooled volume was higher than the air temperature in the room due to heat generation from the fans. At night, the temperature inside the cooled volume was 1.5...2 K higher than indoor air temperature.

The change in temperatures during the observation period from 28.08.2019 to 06.09.2019 is illustrated in Fig. 8.

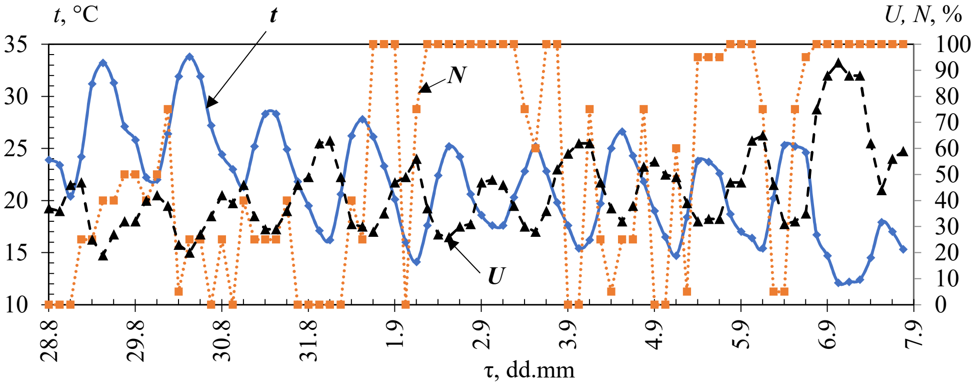


Fig. 6. Weather data during the experiment: *t* – atmospheric temperature; *U* – relative humidity of atmospheric air; *N* – general cloudiness level

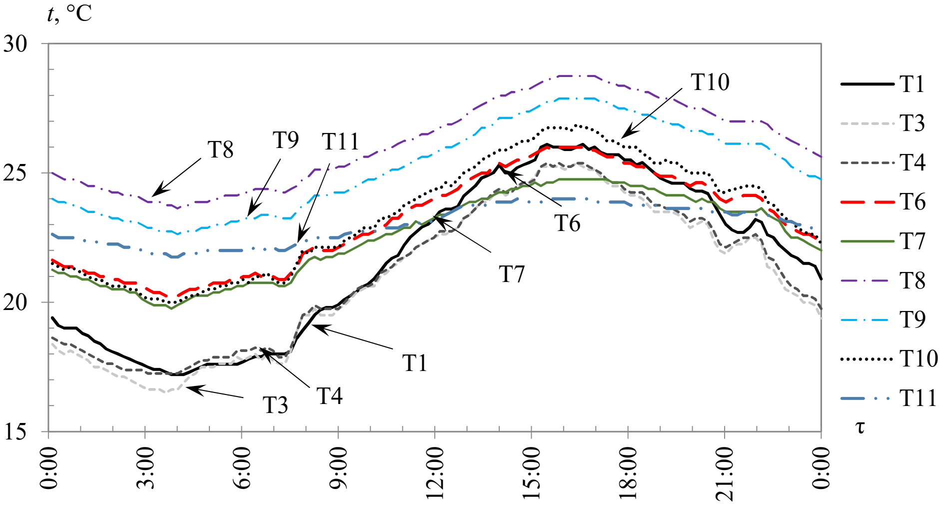


Fig. 7. Temperature change in the experimental refrigeration system for 03.09.2019:
T1 – atmospheric temperature; T3 – the surface temperature of the radiator in its upper part; T4 – the surface temperature of the radiator in its lower part; T6 – refrigerant outlet from the air cooler; T7 – refrigerant outlet from the receiver; T8 – air temperature at the top of the cooled volume; T9 – air temperature at the bottom of the cooled volume; T10 – surface temperature of the AC air cooler; T11 – air temperature in the room in which the refrigerating chamber is located

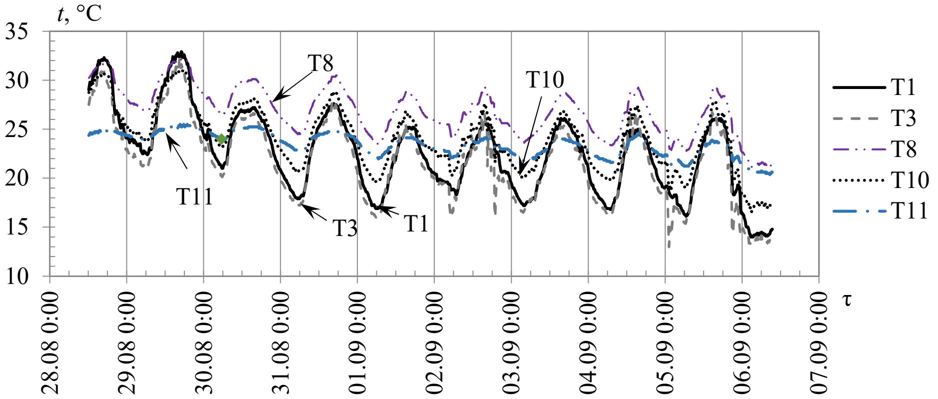


Fig. 8. Temperature changes in the experimental refrigeration system in the period from 28.08.2019 to 06.09.2019:
T1 – ambient air temperature; T3 – radiator surface temperature; T8 – temperature at the top of the cooled volume; T10 – surface temperature of the AC air cooler; T11 – room air temperature

On no night did the temperature inside the refrigerated (T8) volume fall below the temperature in the room in which the refrigeration chamber was located (T11). The surface temperature of the radiator (T3) at night was most of the time about 1 K below the ambient air temperature (T1). A significant decrease in the temperature of the radiating surface at about 18:00 on 02.09.2019 and at 5:00 on 05.09.2019 is associated with rain falling on the surface of the radiator.

The dependence of the temperature inside the cooled volume on the temperature of the atmospheric air is shown in Fig. 9.

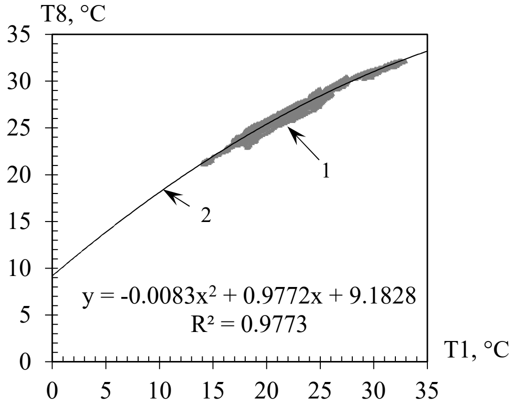


Fig. 9. Dependence of the temperature inside the cooled volume T8 on the temperature of the atmospheric air T1: 1 — experimental data; 2 — extrapolated curve

At a street air temperature of +14 °C, the temperature in the cooled volume was +21 °C. That is, the temperature in the cooled volume was 7 K higher than the street air temperature. At the same time, the temperature head on the air cooler was 3.5 K.

5. 2. Calculation of cooling capacity

Fig. 10 demonstrates the result of calculating the cooling capacity of the experimental refrigeration system over the observation period.

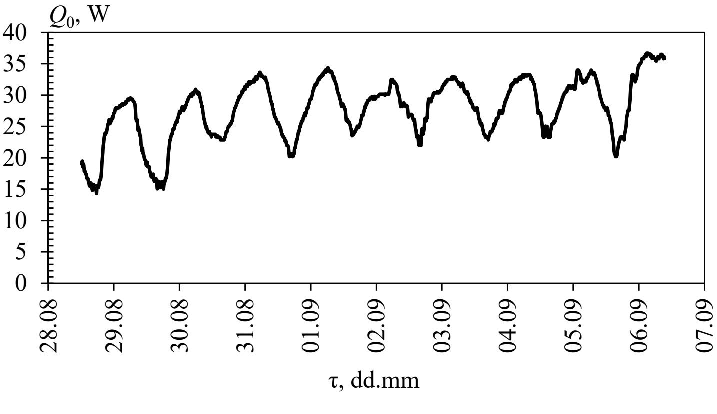


Fig. 10. Changing the cooling capacity of the experimental refrigeration system

The cyclicity of changes in cooling capacity is due to the daily cycle of the system. At night, the cooling capacity of the system increased due to radiative cooling.

The average value of the cooling capacity at night over the observation period was 27.7 W. The absolute error in the calculation of the cooling capacity is ±27.5 W. In this case, the highest value of the cooling capacity of the radiator was observed by the end of the night at 5:00. From dawn until 18:30, there was a gradual decrease in cooling capacity. After 6:30 p.m., there was a gradual increase in cooling capacity. The specific cooling capacity of the radiator was 13.9 W/m^2.

6. Discussion of results of studying the refrigeration system operation

Our experiments show the fundamental operability of the refrigeration system with the natural circulation of the refrigerant and its condensation in the radiator. Our observations of temperatures in the experimental refrigeration system (Fig. 7, 8) demonstrate that heat is transferred from the cooled volume and then enters the radiator.

The resulting value of the cooling capacity of the system (Fig. 10) was lower than the expected cooling capacity of the radiators. Under these conditions, according to the theoretical model proposed in [4], up to 75 W per 1 m^2 of the radiative surface should be transmitted to the environment due to radiative cooling. In other words, when using a radiator of the specified design, its cooling capacity could theoretically reach 150 watts. In addition, the resulting cooling capacity is 30 % less than the value reported in [16]. The reduced cooling capacity of the system may be due to the insufficient mass flow of refrigerant from the air cooler to the radiator; a low value of heat transfer coefficients in the air cooler and radiator. In addition, the decrease in cooling capacity could be caused by the presence of an oil film over the refrigerant in the air cooler. The density of polyester oil under the specified conditions is less than the density of the refrigerant R134a. The oil film on the surface of the liquid refrigerant can prevent the formation of refrigerant vapors.

To increase the heat transfer coefficients, as well as to increase the amount of heat carried by the refrigerant, it is recommended that the use of refrigerants with higher saturated vapor pressures be considered in the future. The refrigerant R134a used, in comparison with other refrigerants applied in refrigeration machines at present, has a relatively low saturated vapor pressure. The low operating pressure has made it possible to reduce the required mechanical strength of the radiator pipelines. However, when the ambient air temperature in the system decreases, the operating pressure would decrease, which could ultimately cause a decrease in vapor density and a decrease in cooling capacity.

In addition, to increase the cooling capacity of the system, it is necessary to ensure that the air cooler is filled with refrigerant in such a way that its entire inner surface is in contact with the liquid refrigerant. In this case, filling the entire air cooler with liquid refrigerant is not desirable as this could lead to a significant increase in the amount of refrigerant filled into the system. In this regard, it is necessary to develop special heat exchangers in which the refrigerant, for example, is distributed over the heat exchange surface due to the capillary effect or other physical phenomena.

The applied method for measuring the cooling capacity has relatively low accuracy. In the future, it is recommended to use more accurate methods involving calorimeters.

In the experimental refrigeration system, manual ball valves are used to close the channels for the refrigerant. In the future, the process of switching the operating mode of the system can be automated using ball valves with a motor drive. The use of solenoid valves in the scheme is not desirable since they generate excessively large pressure losses of the refrigerant. To enhance the cooling capacity of the system, further research is required to improve the design of heat exchangers, select the optimal refrigerant and oil, and reduce the hydraulic resistance of pipelines for refrigerant circulation.

7. Conclusions

1. During the autumn period, experimental studies of the refrigeration system with natural circulation and radiative cooling of the refrigerant were carried out in the city of Almaty. In the course of the experimental study, it was established that the air temperature in the cooled volume is maintained 5...7 K above the ambient air temperature at night. Thus, in order to maintain the air temperature in the refrigeration chamber under the storage mode of +5 °C, the ambient air temperature of about 0 °C is necessary.

Radiative cooling was also induced during the daytime due to the location of the radiators on the northern side of the building where direct sunlight does not fall.

2. A methodology for assessing the cooling capacity of the system, based on the calculation of the thermal balance of the refrigeration chamber, has been devised. The estimated value of the cooling capacity over the observation period was only 27.7 W, which is significantly less than the expected theoretical cooling capacity of radiators under the specified conditions.

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