Development and experimental researches of refrigerating unit with a radar station

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Abstract. The antennas of modern radar devices need effective cooling, which is carried out by forced circulation of the coolant. A new pneumohydraulic diagram of a refrigerating unit, which allows achieving effective heat removal from an antenna of a radar station with minimal overall dimensions and weight, is considered in this paper. The paper presents the results of experimental research related to an operation of the refrigerating unit at ambient temperatures equal to +50 ºС and -50 ºС within the temperature range from -50 ºС to 10 ºС.

1. Introduction
The antennas of modern radar stations (RS) need effective cooling [1], which can be conducted by forced circulation of a coolant – a non-freezing liquid (antifreeze agent, Tosol, etc.). The heat from cooled elements of an antenna is removed to the environment (especially into the air) as well as heat from other elements of a system. When ambient temperature is less than 10-15 ºС, heat extracting is carried out in liquid-gas heat exchangers, which are blown with air fans. When the ambient temperature is higher than 10-15 ºС, it is necessary to use a refrigerating unit, most commonly a compressor, to cool the coolant pumped through the antenna. In industry, such units, often called chillers, are widely used [2,3]. This unit has a closed-circuit system with a low-boiling liquid – usually, a mixture of hydrocarbons called freons or refrigerants (refrigerating agent) [4,5]. This refrigerant boils in one of the heat exchangers with a closed-circuit system, called an evaporator, collecting heat from the coolant (in a form of latent heat of vaporization and heating). After that, it is compressed in a compressor and directed into another heat exchanger, where this refrigerant gives up heat accumulated in the evaporator and the compressor. Here, in the second heat exchanger, the refrigerant of a closed-circuit system undergoes a phase transition from gas to liquid during cooling, which enters the evaporator through a device that lowers its pressure. The refrigerant of a closed-circuit system condenses (returning vaporization heat) in the second heat exchanger. That’s why it is called a condenser. The functions of the condenser are usually performed by an additional heat exchanger, which is also blown by fans, but it has a large thickness of heat transfer walls since it must resist the increased pressure of the refrigerant compressed by a compressor with a closed-circuit system. A distinctive feature of heat exchangers that give heat to the atmosphere is their large dimensions: a large heat transfer surface of the heat exchanger on the airside is required to discharge a large amount of heat into the atmosphere due to the small temperature difference between the refrigerant or coolant and the air. To intensify heat exchange, the cooling surface is blown by a fan, which requires additional energy costs. The presence of two air-cooled heat exchangers, one of which
is also designed for high pressure, increases the size and weight of the refrigeration unit, so it is difficult to install it on mobile RS platforms.

2. Principle diagram of a plant
Researchers at the Joint Institute for High Temperatures of the RAS (JIHT RAS) and specialists of PJSC ALMAZ R&P Corp. offered a new hydraulic installation diagram to reduce mass and dimensions characteristic of a refrigerating unit [6]. Its simplified schematic diagram is presented in figure 1.

![Principle diagram of a refrigerating unit](image)

**Figure 1.** Principle diagram of a refrigerating unit: 1 – primary cooling circuit pump for cooling fluid circulation; 2 – evaporator; 3 – the consumer – RS antenna; 4 – liquid-air heat exchanger; 5 – locking element for cooling fluid transport into the pump 1 through the liquid-air heat exchanger 4; 6 – locking element for cooling fluid transport into the pump 1; 7 – compressor of the refrigerating unit; 8 – secondary cooling circuit pump for cooling fluid circulation; 9 – condenser; 10 – throttle of the refrigerating unit.

The new hydraulic diagram contains the primary and secondary circulation circuits of the cooling fluid–cooler, as well as the circuit of the refrigerating unit. The primary circuit includes a pump 1. Its output is connected through the evaporator 2 with the cooling system input of the RS antenna 3. In turn, the outlet from the antenna’s cooling system 3 is connected with the inlet of the pump 1 through a liquid-air heat exchanger 4. The primary circuit also includes two locking elements 5 and 6. The locking element 5 mounted on the coolant supply line to the primary cooling circuit pump 1 through the liquid-air heat exchanger 4. The locking element 6 is on the line which connects the output of the customer 3 with the inlet of pump 1 and parallel coolant supply line through the liquid-air heat exchanger 4.

The secondary circuit of the coolant circulation also includes a pump 8. Its input and output are connected respectively to the output and input of the liquid-air heat exchanger 4, moreover, the latter is connected to the output of the pump 8 through the condenser 9.

The circuit of the refrigerating unit includes a compressor 7, whose output is connected through the condenser 9 and the throttle 10 of the refrigerating unit with the evaporator 2, which, in turn, is connected with the input of the compressor 7.

The refrigeration unit works as follows. Depending on the air temperature, there are two operation modes of the refrigeration unit. The first operation mode of the device at an air temperature varied within the range from -40 ºС to +15 ºС includes the operation of the primary cooling circuit pump 1, which pumps coolant through the evaporator 2 to the consumer 3—the cooling system of RS antenna. The coolant is heated in the antenna cooling system according to the RS operating mode. It enters the liquid-air heat exchanger 4 with increased temperature. In the liquid-air heat exchanger 4, the coolant is cooled by air flow. In this case, the locking element 5 of the coolant supply to the pump 1 through
the liquid-air heat exchanger 4 is in the open position, and the locking element 6 of the coolant supply to the pump 1 is closed. Then, from the liquid-air heat exchanger 4, the cooling fluid enters the inlet of the pump 1. In the first mode, the compressor 7 of the refrigerating unit does not work and, as a result, there are no heat exchange processes in the evaporator 2.

The second operation mode of the refrigeration unit at the heat discharge from the consumer 3 and at the air temperature varying within the range from +15 °C to +50 °C, includes the operation of the primary cooling circuit pump 1, which pumps the coolant through the evaporator 2 to the consumer 3. In this case, the locking element 5 is in the closed position, and the locking element 6 is open, the pump 8 of the secondary coolant circulation circuit also works. By means of the secondary cooling circuit pump 8, the coolant is directed through the condenser 9 to the liquid-air heat exchanger 4, and then it returns to the input of the same pump 8. Freon circulates in the circuit of the refrigerating unit due to the operation of the compressor 7. Thus, the heated coolant in the cooling system of the consumer antenna 3 is pumped through the evaporator 2, where the coolant is cooled to the desired temperature due to heat exchange with the refrigerant – freon, which circulates in the circuit of the refrigeration unit. In the secondary circuit of the coolant circulation, the heat generated by the refrigerating unit is removed in the condenser 9. The coolant heated in the condenser 9 is directed by the pump 8 to the liquid-air heat exchanger 4, where the heat of the refrigerating unit is discharged into the environment. At the same time, the compressor 7 of the refrigerating unit creates freon pressure in the evaporator 2 and the condenser 9, which are necessary for evaporation and condensation, respectively. The cold, which is obtained during the evaporation of the refrigerant, is transferred to the primary circuit of the coolant circulation to cool the consumer 3. And the heat of freon condensation throughout the secondary circuit of the coolant circulation is dissipated into the atmosphere by a liquid-air heat exchanger 4.

The diagram and operating modes of the consumer thermal maintenance unit, which are presented above, are implemented by specialists of the PJSC ALMAZ R&P Corp. together with JIHT RAS scientists during the modernization of the RS refrigeration unit to reduce its mass and dimension characteristics, as well as also reduce the electrical power consumed. Significant simplification of the modernized refrigeration unit and reduction of its dimensions was achieved by using a new liquid-liquid type heat exchanger in the condenser. Also, the weight and dimension reduction of the refrigeration unit is provided by the use of the same low-pressure liquid-gas heat exchanger for the discharge of heat into the environment of the refrigeration cycle and the direct cooling circuit (for the first operation mode in winter).

3. Test rig
To check the efficiency of the refrigeration unit and confirm the claimed characteristics, a test rig is designed and manufactured based on the proposed diagram (Fig.1). Its pneumohydraulic diagram is shown in figure 2.

During the design process of refrigerating unit test rig, a necessity to create conditions for maximum versatility is taken into account to provide the following tests:

- testing of elements and equipment (electrical equipment and automatic control systems) of a refrigeration unit;
- testing of experimental and serial samples of the refrigerating unit;
- developmental and control testing of an experimental refrigeration unit;
- conducting simulations within a wide range of load and climate conditions.
Figure 2. Pneumohydraulic diagram of the refrigerating unit test rig.

Diagram of the refrigerating unit: C – compressor; CN – condenser; FR – freon receiver; FD – filter drain; TCV – temperature control valve; E – evaporator; IV – isolation valves; SLV – solenoid valve; SV – safety valve.

Coolant system: P– pump; HE, HE2, EOH – heaters; EV – electromagnetic valve; T – tank; TF – Tosol filter; R – radiator; F – fans; IBV – isolation ball valve.

Measuring system: ND – narrowing device; ΔP – differential pressure sensor; PS – pressure sensor; LI – level indicator; LS – lever sensor (alarm); PDS – pressure drop sensor (alarm); T – thermometer, thermocouple; TR – thermal resistance; PSW – pressure switch.

Meeting the requirements for the test rig involved mutually exclusive solutions. Thus, the necessity to test almost all elements of the refrigeration unit and related provision of the possibility of using aggregates with different dimensions and joint designs determined the free placement of design elements with a reservation of space around them for the convenient work on their replacement and connection. This required considerable dimensions. The opposite requirements for dimensions resulted from the necessity to ensure testing of the refrigeration unit in various climatic conditions. It is offered to use the existing climate chamber for the testing. Therefore, the test rig must have been mobile, with dimensions that would ensure its placement in the climate chamber. Also, it must have had remote control systems, measurements, and operational data processing.

Determining the main components of the test rig, it was taken into account that the amount of heat, which must be removed in the evaporator \( (Q_e) \), is a sum of the RS thermal power \( (Q_p) \) and the power consumed by the primary circuit circulation pump \( (N_{n1}) \):

\[
Q_e = Q_p + N_{n1}
\]

And the amount of heat that needs to be removed in the liquid-air heat exchanger \( (Q_r) \) is the sum of the heat power taken away in the evaporator \( (Q_e) \), the power consumed by the secondary circuit circulation pump \( (N_{n2}) \), and the power consumed by the compressor \( (N_c) \):

\[
Q_r = Q_e + N_{n2} + N_c
\]

Analysis of these equations and the diagram shows that the dimensions of the liquid-air heat exchanger allow optimizing the parameters of the refrigeration unit, since the bigger the heat exchange surface (mass) the lower is the cost of electricity for providing the necessary air flow velocity, and otherwise.
4. Experimental research

Designed according to the requirements mentioned above and pneumohydraulic diagram, the test rig was produced at the PJSC ALMAZ R&P Corp. for experimental research.

At the first stage, the characteristics of the refrigeration unit were determined at the most unfavorable operating mode where the ambient temperature was +50 ºС. The following stages were considered: cooling of the Tosol in the primary circuit after the evaporator to +20 ºС (the temperature after which the RS can be switched on); working with a connected heat load (110±1 kW); working at an ambient temperature of +55 ºС without requirements for cooling capacity. Figures 3-5 present time-dependent temperature curves at the outlet (average value of TR4 and TR5) and the inlet (value of TR6) of the unit.

Figure 3. Cooling of the Tosol to the temperature of 20 ºС.

Figure 3 shows that as the temperature decreases at the outlet of the unit, the rotation frequency of the compressor rotor decreases since its cooling capacity is regulated by a frequency-regulating drive. The unevenness of the speed change is associated with a feature of the control algorithms (PD regulator is implemented), which allows maintaining the temperature of the Tosol within acceptable limits in the conditions of rapidly changing heat load. The temperature difference between the inlet and the outlet of the unit is associated with a significant difference between the dimensions and thermal insulation of the cavities where the sensors are located.

As shown in figures 4–5, at an ambient temperature of +50 ºС, the refrigeration unit provides a temperature at the inlet of the RS which equals to 22 ºС, and at +55 ºС it equals to 24 ºС, which confirms the ability of the unit to operate in a hot climate.

Figure 4. Performance of the refrigerating unit in the climate chamber at 50 ºС. 1 – temperature in the climate chamber (TR7); 2 – temperature at the inlet of the refrigerating unit (TR6); 3 – temperature at the point between the heaters HE и HE2; 4 – temperature at the outlet of the refrigerating unit (TR4, TR5).
Figure 5. Performance of the refrigerating unit in the climate chamber at 55°C. 1 – temperature in the climate chamber (TR7); 2 – temperature at the inlet of the refrigerating unit (TR6); 3 – temperature at the point between the heaters HE и HE2; 4 – temperature at the outlet of the refrigerating unit (TR4, TR5).

At the second stage of experimental research, the possibility of starting and operating of the unit at an ambient temperature of -50 °C, as well as in the temperature range from -50 °C to +5 °C is determined. The main difficulty of the refrigeration unit starting at -50 °C is the increased viscosity of the Tosol. This circumstance makes difficult to start the circulation pumps, and it can lead to an accident. So that algorithms have been developed for starting circulation pumps at low ambient temperatures, allowing maintaining the starting currents within acceptable limits.

Figure 6 shows that up to the air temperature of -35 °C, the operation of the refrigerating unit occurs in a pulse mode. When the Tosol temperature reaches 20 °C, the fans that blow air through the liquid-air heat exchanger are switched on at a minimum speed. Then the Tosol is cooled to a temperature of less than 15 °C, and the fans are switched off. When the ambient temperature increases, the Tosol can no longer be cooled in the liquid-air heat exchanger with the temperature below 15 °C at the minimum fan speed. With a further increase in the ambient temperature up to 8 °C, the required temperature of the Tosol is maintained by increasing the fan speed. The refrigeration compressor must be switched on at a higher ambient temperature.

Figure 6. Performance of the refrigerating unit within a range of ambient temperatures from –50 to 8°C: 1 – fan rotation speeds F1 and F2; 2 – temperature in the climate chamber (TR7); 3 – temperature at the outlet of the refrigerating unit (TR4, TR5).
Also, as a result of experimental researches, values of the refrigerating unit cooling factor are determined at various ambient temperatures (see table 1). The cooling factor is defined as a ratio of the cooling capacity to the total electrical power consumed by the elements of the refrigerating unit. Five values of an ambient temperature are considered: -20 °C (because at lower temperatures the refrigerating unit operates in a pulse mode); 8 °C; 10 °C; 30 °C and 50 °C.

Table 1. Values of the cooling factor and electrical power consumed by the refrigerating unit at various ambient temperatures.

| Parameter                        | Value  |
|----------------------------------|--------|
| Ambient temperature, °C          | -20    |
| Power consumed by pump P1, kW    | 9,5    |
| Power consumed by pump P2, kW    | 0      |
| Power consumed by fans, kW       | 1,0    |
| Power consumed by compressor, kW | 0      |
| Cooling factor                   | 10,5   |

The results of experimental studies allow confirming the efficiency of the proposed diagram and design of the refrigerating unit with a refrigerating capacity of 110 kW. Its configuration is shown in figure 7.

**Figure 7.** Configuration of the refrigerating unit.

**Conclusions**

Successful tests of the refrigeration unit have experimentally confirmed the efficiency of the cooling diagram with a single low-pressure liquid-air discharge heat exchanger and a flexible automated control system. The transition temperature between the modes, which is +8 °C according to the results of experimental studies, was clarified. Based on the results of the research, the refrigerating unit with a cooling capacity of 110 kW is designed for ambient temperatures ranged from -50 °C to +50 °C.
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