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Original Study Article

The HVAC system engineering method for a new family of unified cabins of combine harvesters

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ABSTRACT

BACKGROUND: Designing a combine harvester cabin climate system is a complex and multi-stage process that includes solving a set of tasks. To solve them, a design engineer must have knowledge and skills in various fields, starting with thermal engineering calculations and ending with experimental research methods and computer modeling, which requires a large amount of intellectual and time resources. Therefore, the task of creating a unified method of engineering of combine harvester cabin HVAC system is highly relevant.

AIM: Development of the HVAC system for a unified cabin of a grain harvester and a forage harvester. The designed HVAC system is purposed to create a comfortable microclimate in the cabin of the combine for 2 people in summer and winter operating conditions.

METHODS: The methodology of calculation and selection of the combine harvester cabin HVAC system, which includes the development of engineering calculation methods and mathematical modeling of thermodynamic and ventilation processes in the cabin, is considered.

RESULTS: The main parameters were determined: heat intakes and heat losses for the grain harvester and forage harvester cabins, which amounted to 2.8 and 2.2 kW for the grain harvester, 2.9 and 2.35 kW for the forage harvester; the required air flow rate supplied to the cabin to ensure a comfortable temperature — 740 m^3/h; the percentage of air recirculation regarding the conditions of absence of fogging and creation of excessive pressure in the cabin — 75%; the cooling and heating capacity of the HVAC system, taking into account the operating conditions of the combine, are 7.8 and 6.3 kW respectively. The selection of the main equipment of the HVAC system for a unified cabin for a new family of unified cabins of combines — the BUHLER 1000 MFWD evaporator-heater and the Valeo TM16 compressor are chosen.

CONCLUSION: The HVAC system designed in accordance with the presented methodology is capable of ensuring a comfortable cabin air temperature in the range of 22–24 °C at various operating modes of the combine harvester in different regions. In addition, the HVAC system parameters eliminate fogging of the cabin windows and the penetration of dusty air inside due to the created excessive pressure.

Keywords: combine harvester; HVAC system; engineering calculation; design; computer modeling.

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Оригинальное исследование

Методология проектирования климатической системы для нового семейства унифицированных кабин комбайнов

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АННОТАЦИЯ

Введение. Проектирование климатической системы кабины комбайна — сложный и многоэтапный процесс, включающий в себя комплекс задач. Для их решения инженер-конструктор должен обладать знаниями и навыками в различных областях, начиная с теплотехнических расчетов и заканчивая методиками проведения экспериментальных исследований и компьютерным моделированием, на что требуется большое количество интеллектуальных и временных ресурсов. Поэтому задача создания унифицированной методологии проектирования климатической системы кабины комбайна является весьма актуальной.

Цель исследования. Целью настоящей работы является разработка климатической системы для унифицированной кабины зерноуборочного и кормоуборочного комбайна. Проектируемая климатическая система предназначена для создания комфортного микроклимата в кабине комбайна для 2-х человек в условиях летнего и зимнего режима эксплуатации.

Материалы и методы. Рассматривается методология расчета и выбора климатической системы кабины комбайна, включающая в себя разработку методики инженерных расчетов и математическое моделирование термодинамических и вентиляционных процессов в кабине.

Результаты. Определены основные параметры: теплопритоки и теплопотери для кабины зерноуборочного и кормоуборочного комбайна, которые составили 2,8 и 2,2 кВт для зерноуборочного комбайна; 2,9 и 2,35 кВт для кормоуборочного комбайна; необходимый расход воздуха, подаваемый в кабину для обеспечения комфортной температуры — 740 м^3/ч; процент рециркуляции воздуха из условий отсутствия запотевания и создания избыточного давления в кабине — 75%; холодо- и теплопроизводительность климатической системы с учетом условий эксплуатации комбайна — 7,8 и 6,3 кВт соответственно. Осуществлен подбор основного оборудования климатической системы для унифицированной кабины для нового семейства унифицированных кабин комбайнов — испаритель-отопитель BUHLER 1000 MFWD; компрессор Valeo TM16.

Заключение. Спроектированная в соответствии с представленной методикой климатическая система позволит обеспечить комфортную температуру воздуха в кабине в пределах 22–24 °C при различных режимах эксплуатации комбайна в различных регионах. Кроме того, параметры климатической системы исключат запотевание стекол кабины и за счет созданного избыточного давления — проникновение внутрь запыленного воздуха.

Ключевые слова: комбайн; климатическая система; инженерный расчет; проектирование; компьютерное моделирование.

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INTRODUCTION

Designing the heating, ventilation, and air conditioning (HVAC) system [1-5] for a combine harvester cab involves solving numerous challenging problems. Key tasks include developing engineering calculation methods using Excel and verifying these calculations with experimental test data. Furthermore, scientific problems related to finite element ventilation modeling must be addressed using modern ANSYS software.

Selecting the correct main equipment for the HVAC system requires linking (adapting) the developed engineering methodology to match the technical characteristics of suppliers' equipment, which may have been tested under different test conditions than those of the harvester cab. This article presents the methodology for designing an HVAC system, using the new cabin of the SAV 3 combine harvester as an example. It summarizes the calculation methodology and discusses the results obtained.

A review of relevant publications and accomplishments in this field reveals that choosing the right HVAC system relies primarily on accurately calculating heat exchange within the operator's cabin, using various methods.

The following methods and techniques have been identified:

1. An experimental method with parameter measurements. For example, in [6], this involves measuring the temperature difference both outside and inside the cabin first with the heater off and on.
2. Engineering calculation. This is an approximate method that calculates the total heat transfer coefficient of various surfaces during convective and radiant heat exchange [7]. This technique is presented in more detail in [1, 4].
3. Mathematical modeling of heat and mass transfer. This complex method uses a mathematical approach to analyze the cabin's thermal regime systematically. It describes the cabin micro-climate using a system of differential equations [8].
4. Computer simulation. Widely used in modern scientific research, this approach allows for three-dimensional modeling of non-stationary heat and mass transfer processes. It enables subsequent analysis of physical phenomena through modern software like ANSYS and NX CAE.

INITIAL DATA AND REQUIREMENTS FOR THE DESIGN OF THE COMBINE HARVESTER CAB HVAC SYSTEM

The initial data and requirements for designing the HVAC system in combine harvester cabs were formulated by technical specialists at KZ Rostselmash LLC, located in Rostov-on-Don. They provided information on the cabin dimensions and materials for the grain harvesting ZUK and forage harvesting KUK combine harvesters. Thermal insulation properties of cabin materials were referenced from SP 50.13330.2012 "Thermal protection of buildings" or according to suppliers.

The surface areas of cabin elements were determined using 3D models. The thicknesses and types of cabin walls were selected either from 3D models and drawings of the designed elements, or by analogy with existing cabins, considering their layered structures if not explicitly designed during calculations.

Outdoor air parameters were based on the operating conditions of the combine harvesters: in summer, the outdoor air temperature can reach up to +45°C with a relative humidity of $\varphi = 30\%$; in winter, it can drop to 20°C with a relative humidity of $\varphi = 80\%$. The maximum solar radiation intensity is 950 W/m^2. The HVAC system must operate with partial cabin air recirculation within the cabin, calculated to prevent fogging of the cabin windows.

The internal air parameters in the cabin should align with the effective temperature diagram according to GOST ISO 14269-2-2003 under maximum summer and minimum winter operating conditions.

Given the complexity and multi-factor nature of the problem, the initial data on the dimensions and materials of the cabin walls are input into engineering calculation files (Excel). This data can be adjusted during later design stages, which helps automate calculations and reduces the designer's workload.

To achieve the above indicators, it is necessary to adhere to a certain algorithm, presented below in stages:

1. Calculating heat flows and determining the parameters and volume of air supplied to the cabin.
2. Calculating and selecting the main elements for the air conditioning system, considering air dustiness and heat exchange equipment pollution.
3. Drawing up an air treatment scheme and determining thermal loads on the main equipment of air conditioners, considering air recirculation.
4. Calculating and selecting the main equipment of the air conditioning system for combine harvester cabins, considering the length of the cooling ducts.
5. Developing a methodology for selecting climate system equipment adapted to the technical characteristics of suppliers' equipment and the conditions of their protocol tests.
6. Developing a mathematical model of heat and mass transfer and calculating thermodynamic parameters in a full-fledged 3D cabin model using the finite element method performed in the ANSYS software.
7. Verifying the calculation process using experimental data by comparing engineering calculation methods, results from finite element modeling

of cabin ventilation, and HVAC system performed conducted by specialists of KZ Rostselmash LLC and manufacturers of individual system elements.

DETERMINATION OF THERMAL LOADS AND LOSSES ON THE COMBINE HARVESTER CABIN

In previous studies, the authors extensively analyzed the heat flows and heat loss in the SAV 3 combine harvester [9–11]. The analysis revealed during summer, the most significant heat flow is attributed to solar radiation, amounting to 1393.2 watts. It is recommended to calculate this using P.Y. Hamburg, which incorporates shading coefficients, or according to [12]. Furthermore, transmission heat input, measured at 914.1 W, constitutes a larger share of the total heat input, while operational heat input plays a less important role. In winter, major heat losses are primarily attributed to transmission heat gain. This data is crucial for determining the required air flow and load on the heat exchange equipment. For calculations, the highest value is selected with an outdoor air velocity of 2.7 m/s (10 km/h).

DETERMINATION OF THE REQUIRED AIR FLOW RATE SUPPLIED TO THE CABIN

The air consumption required to assimilate excess heat and reduce the temperature to an optimal +24°C is determined from the thermal balance of the cabin, according to the following equation:

$$L = \frac{\sum Q}{\rho \cdot c \cdot \Delta t_{\text{р}}}, \tag{1}$$

where Q is the thermal load on the cabin, kW; ρ — denotes the air density, kg/m^3; c indicates the specific heat of air, J/(kg·K); $\Delta t_{\text{р}}$ signifies the permissible operating temperature difference equal to 10°C.

The permissible operating temperature difference $\Delta t_{\text{р}}$ was selected based on recommendations for air supplied to the working area, adjusted according to experimental data from KZ Rostselmash LLC. This data pertains to the air temperature at the outlet of deflectors in combine harvesters during summer operations. Since the air conditioning scheme in the combine cabin does not provide for changing airflow across different modes, it is necessary to determine the required temperature of hot air at the deflectors' outlet for winter conditions. Based on the air consumption for the summer mode, we convert Eq. 1 to the following form:

$$t_{\text{наг}} = \frac{\sum Q}{c \cdot L} + t_{\text{к}}, \tag{2}$$

where $t_{\text{к}}$ is the temperature of the internal air, °C.

Thus, to maintain a comfortable cabin temperature of 24°C under winter design conditions, it is necessary to supply air at a rate of 0.227 m^3/s with a temperature of 31.8°C.

CALCULATION OF THE THERMAL LOAD ON THE EVAPORATOR AND CABIN HEATER

To determine the mixing point of return air and supply air, we use the mixing rule based on the following equation:

$$t_{\text{с}} = \frac{L_{\text{р}} \cdot t_{\text{р}} + L_{\text{п}} \cdot t_{\text{п}}}{L_{\text{об}}}, \tag{3}$$

where $L_{\text{р}}$, $L_{\text{п}}$ are the flow rate of recirculating and supply air, m^3/s; $t_{\text{р}}$, $t_{\text{п}}$ are the temperature of recirculating and supply air, °C, and $L_{\text{об}}$ is the total air flow, m^3/s.

In summer mode with 75% recirculation in the cabin, the mixing temperature will be 29.3°C. For winter mode, under the same recirculation level, it will be 13°C.

To calculate thermal loads, we will determine the heat and humidity relationships and construct i-d diagrams for both summer and winter modes. These diagrams will illustrate the parameters of the nodal points (Fig. 1 and 2).

The heat and humidity ratios for both periods are determined using the following equation:

$$\varepsilon_{\text{п}} = \frac{\sum Q_{\text{я}}}{\sum W} + i_w, \tag{4}$$

where $\sum Q_{\text{я}}$ is the total inflow of sensible heat, kJ/h; $\sum W$ represents the total consumption of moisture exchanged in the process of changing the state of the air, kg/h; i_w denotes the specific enthalpy of water vapor (i_w = 2500). During the summer, the value of $\varepsilon_{\text{п}}$ was 29684 J/kg, while during the winter, it reached 21917 J/kg.

Thermal load on the evaporator in the recirculation mode:

$$Q_0 = L_{\text{н}} \cdot \rho \cdot (i_{\text{с}} - i_{\text{п}}), \tag{5}$$

where $i_{\text{с}}$, $i_{\text{п}}$ indicate the specific enthalpy of mixed and supply air, respectively.

The calculated thermal load on the evaporator was initially 6 kW. However, considering recirculation, contamination levels, and a coefficient compensating for pressure loss in the return line to the compressor, this increases by 30% compared to that determined by Eq. 5, resulting in 7.8 kW. Similarly, the heater load was calculated at 6.3 kW.

The effect of dust contamination on the heat transfer surfaces of the HVAC systems in combine harvester

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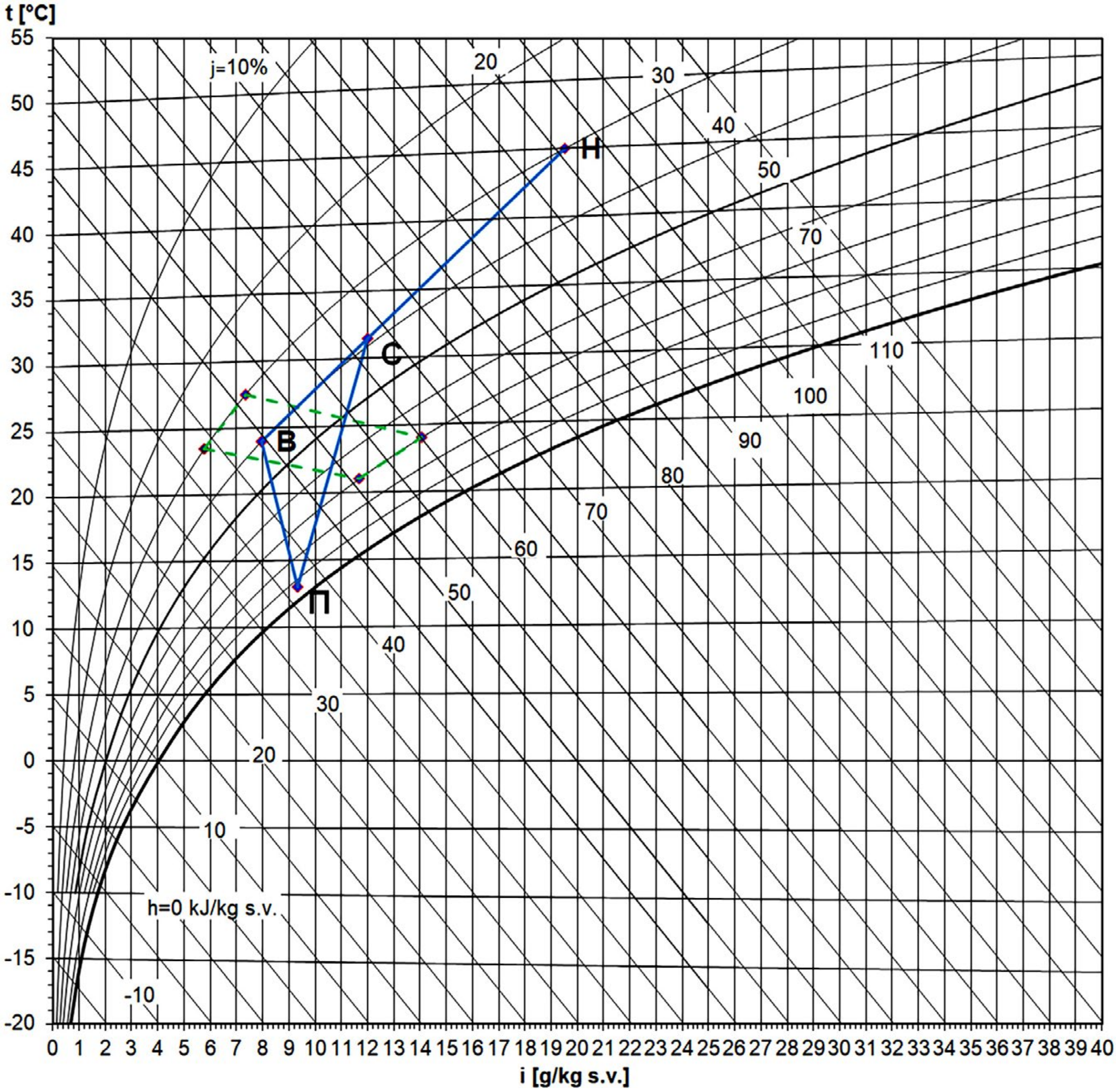

Fig. 1. The process of changing the air condition in the i-d diagram for the summer conditions.
Рис. 1. Процесс изменения состояния воздуха в i-d диаграмме для летнего режима.

cabins, particularly its effect on calculating the thermal load on the evaporator and heater (using a coefficient of 1.3) is detailed in [13].

These calculations were performed under operating conditions in a very hot and dry climate (Kazakhstan, Uzbekistan) at an ambient temperature of +45°C and 30% humidity.

Further, using the same method, the thermal load on the evaporator was determined for other regions based on their climate reports (SP 131.13330.2020 "SNiP 23-01-99* Construction climatology"):

1. Krasnodar Krai, characterized by a hot and humid climate with an ambient temperature of +37°C and humidity of 67%;
2. Amur region, known for its hot and very humid climate, with an ambient temperature of +35°C and humidity of 80%.

For winter conditions, real indicators of outdoor temperature and humidity for any region are specified.

Table 1 presents the final calculated data for all regions during the summer.

CALCULATION OF THE COOLING CYCLE OF THE COMBINE CABIN AIR CONDITIONING SYSTEM

The selection and calculation of the main components of the cooling equipment, such as the compressor and

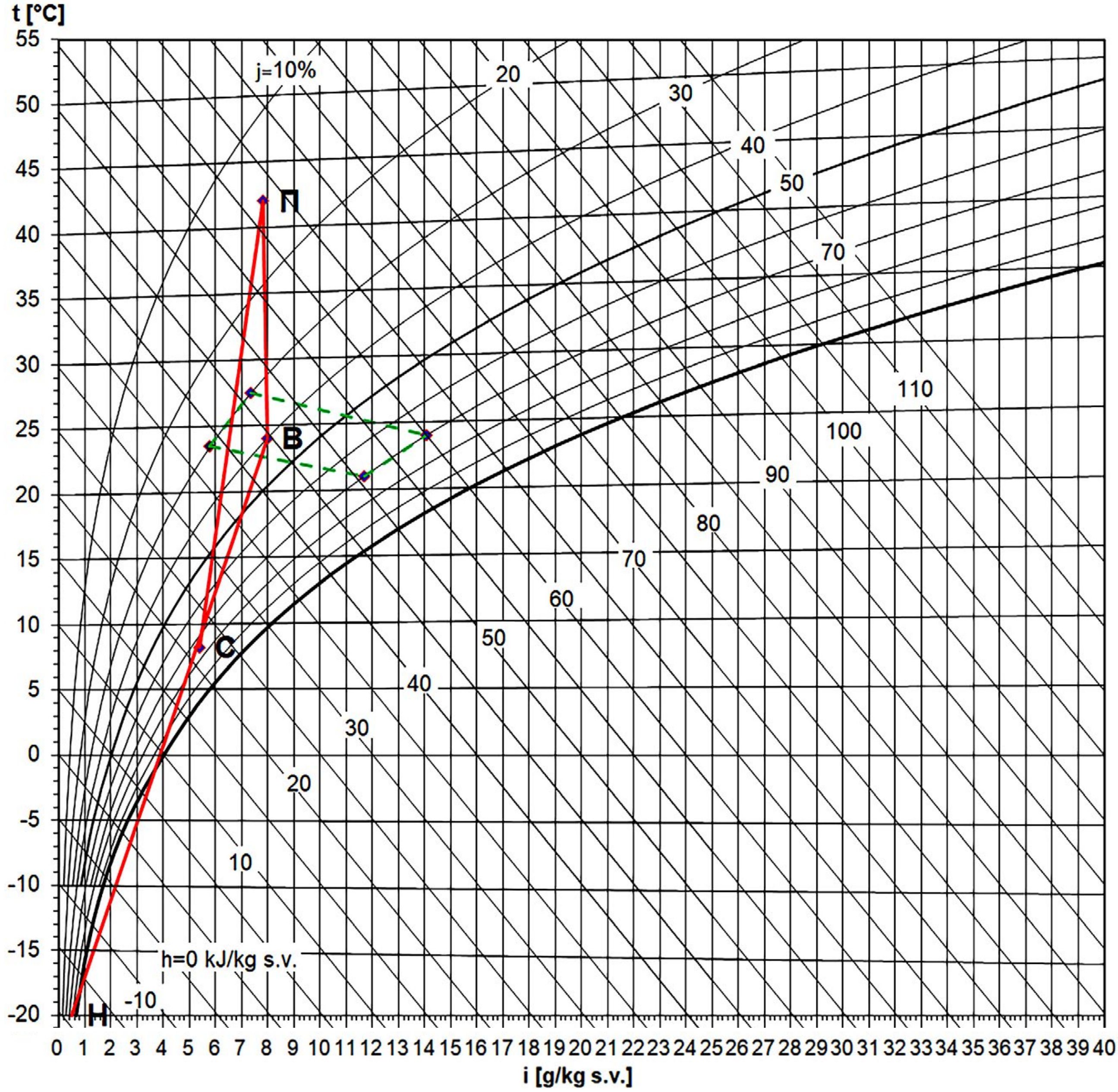

Fig. 2. The process of changing the air condition in the *i-d* diagram for the winter conditions: *H* — the state of the outdoor air; *B* — the state of the air inside the cabin; *C* — the state of the mixture of recirculating and outdoor air; *П* — the state of the supply air.
Рис. 2. Процесс изменения состояния воздуха в *i-d* диаграмме для зимнего режима: *Н* — состояние наружного воздуха; *В* — состояние воздуха внутри кабины; *С* — состояние смеси рециркуляционного и наружного воздуха; *П* — состояние приточного воздуха.

condenser, were based on experimental data developed by KZ Rostselmash LLC together with the August air conditioning plant. Table 2 presents the operating parameters of the cooling cycle derived from this data. Principal feature of these studies include pressure losses in the suction, discharge, and heat exchanger circuits, as along with temperatures of boiling, discharge, superheating, and subcooling. Experts from KZ Rostselmash LLC anticipate similar pressure losses and operating temperatures in the new SAV3 cabin system. Therefore, the cycle compilation for selecting key equipment will utilize existing experimental data from the cooling circuits of the combine harvesters of KZ Rostselmash LLC.

The main operating parameters of the system, such as condensation and boiling pressure, overheating, and supercooling, are defined based on recommendations [12, 13] and available experimental data. The boiling pressure for a freon cooling machine for air cooling is assumed to be 5°C. The condensation pressure is 10°C above the ambient temperature, resulting in 55°C under the selected design conditions. Pressure losses,

Table 1. Final calculations of heat loads for the summer conditions
Таблица 1. Итоговые расчеты тепловых нагрузок для летнего режима

Region	Thermal load on the cabin, kW	Required cabin air flow rate, m^3/h	Thermal load on the evaporator with recirculation of 75%, kW	Thermal load taking into account the coefficient of 1.3, kW
Kazakhstan, Uzbekistan	2,8	817	6	7,8
Krasnodar Krai	2,4	688	6	7,8
Amur region	2,2	656	5,9	7,6

Table 2. Parameters of nodal points of the theoretical cycle of a cooling machine with the R134a according to the p-i diargam and tests
Таблица 2. Параметры узловых точек теоретического цикла ХМ с R134a согласно p-i диаграммы и испытаниям

	1’	1	2	2’	3’	3	4
p, MPa	0,345	0,28	1,69	1,6	1,6	1,52	0,345
t, °C	4,72	15	60,24	57,94	57,94	52,94	4,72
T, К	277,87	288,15	333,39	328,15	331,09	326,09	278,15
i, kJ/kg	402,49	412,63	429,66	428,73	282,49	275,0649	275,0649
v, m^3/kg	0,061572	0,079619	0,013276	0,014045	–	–	0,022

overheating temperature, and supercooling temperatures are taken from available studies of the operating cooling circuits of the system and specified at key points in the cooling circuits.

The compressor selection was made from Valeo’s range of automotive compressors, which are already used in the HVAC installations of Rostselmash combine harvesters. According to the technical characteristics of Valeo compressors, the recommended condensation temperature during evaporator boiling ranges from 55°C to 67°C, aligning with existing guidelines [14, 15]. However, it is important to note that the compressor performance data, depending on the clutch speed, provided by the manufacturer assumes conditions of 10°C overheating and 5°C supercooling, without pressure loss in the cooling circuit during real conditions. Therefore, when designing the cooling cycle, it is crucial to account for these losses and adjust the required compressor frequency from the standard level declared by the manufacturer. The method of calculating the cooling cycle is indicated below, considering the actual conditions of existing tests of cooling circuits.

We define the operating parameters of the cooling cycle from tests carried out on the operating cooling circuit of the combine at the August air conditioning plant under conditions that closely resemble real-world scenarios.

The specific mass cooling capacity of the cooling agent was 127.43 kJ/kg according to the following equation:

$$q_0 = i_{1'} - i_4 . \quad (6)$$

The specific heat of compression in the compressor determined by the following equation:

$$L_t = i_2 - i_1 , \quad (7)$$

will be equal to 17.03 kJ/kg.

The specific thermal load on the condenser reaches value of 147.17 kJ/kg and is calculated according to:

$$q_к = i_2 - i_{3'} . \quad (8)$$

The mass flow of the heat removal agent according to the following equation

$$M_т = \frac{Q}{q_0} , \quad (9)$$

will be 0.0612 kg/s.

The required theoretical volumetric capacity of the compressor was determined as 0.008 m^3/s according to the following equation:

$$V_т = \frac{M_т \cdot v_1}{\lambda} , \quad (10)$$

where v_1 is the specific volume of the sucked steam, m^3/h; and λ is the compressor actual volumetric efficiency.

Based on the obtained value of $V_т$ and recommendations regarding compressor speed, the "Valeo TM16" compressor was selected. Its technical characteristics are presented in Table 3.

According to compressor operation guidelines [13], the optimal operating condition is a coupling rotation speed of 3100 rpm. At this selected frequency, the cylinder’s working volume is equal to 0.008 m^3/s using the following equation:

$$V_p = \frac{0,7854 \cdot D^2 \cdot S \cdot n \cdot z}{1000}, \qquad (11)$$

where D is the cylinder diameter, cm; S is the piston stroke, cm; n is the crankshaft rotation speed, 1/s; and z is the number of cylinders.

Table 3. Technical specification of the Valeo TM16 compressor

Таблица 3. Технические характеристики компрессора «Valeo TM16»

Parameter	Value
Compressor type	Heavy Duty Swash Plate
Type	Valeo TM16
Displacement, cm^3/rev	163
Cylinder diameter, cm	3,6
Piston stroke, cm	2,67
Number of cylinders, pcs	6
Revolution speed, rpm	6000

The resulting flow rate matches the theoretical value, confirming the evaporator ability to deliver a capacity of 7.8 kW at a coupling speed of 3100 rpm, thereby validating the selection of the compressor. Verification and further testing are necessary, as a decrease in suction pressure requires higher volumetric performance, requiring an increase in compressor speed.

The actual mass flow rate of the cooling agent in the compressor, calculated using Eq. 10, is 0.0615 m^3/s.

The actual cooling capacity of the compressor, calculated by Eq. 9, is 7.83 kW.

All conditions are valid for a loss of suction pressure of no more than 1 bar. Should this loss increase, the compressor speed will need to be adjusted, which will be confirmed by bench tests at the August air conditioning plant.

HVAC SYSTEM COMPONENT SELECTION METHODOLOGY

After the capacity of the main working components of the air conditioning system was determined, a request was made to the August air conditioning plant about the need to provide components with a capacity according to the calculated one.

EVAPORATOR-HEATER

The August air conditioning plant proposed an evaporator-heater and provided calculations for both cooling and heating capacities in accordance with the parameters calculated using this methodology. These results are listed in Tables 4 and 5.

Verification of engineering methods of calculation by means of universal

From the calculations provided by the August air conditioning plant for the evaporator and heater, it appears that air consumption, temperature difference, cooling capacity, and heating capacity are sufficient to ensure cooling and heating requirements according to this methodology. However, as with any theoretical calculations, validation in real operating conditions is essential. The calculations were performed under certain standard calculation conditions. In this case, it is necessary to draw up a test procedure confirming the calculations of the evaporator and heater before installing the components on the combine harvester. This procedure should closely mimic the real working conditions of the combine harvester cabin HVAC system. Only after successful testing should the components be installed on the real combine harvester.

COMPRESSOR

When determining the necessary volumetric capacity and coupling speed of the processor, real operating conditions from serial combine harvesters were considered. However, similarly to the evaporator, it is necessary to confirm these calculations through bench tests that closely mimic real-world conditions. This evaluation ensures the system operates effectively based on the selected operating parameters. The main criterion for assessing the effectiveness of the combine harvester cabin HVAC system is achieving the proper system balance, maintaining the required temperature difference and managing air flow and rotation of the compressor coupling.

CONDENSER

The factory manufacturer of climatic equipment “August” offered a serial condenser that, according to calculations based on this method and those by “August” (see Table 5), adequately provides the necessary heat removal. This still needs to be confirmed later by bench tests.

ANSYS PACKAGE ON THE BASIS OF FINITE ELEMENT MODELING

Using modern mathematical and computer modeling increases the reliability of designers’ conclusions and expands the scope of process studies.

To address the objectives, a mathematical model of heat and mass transfer was developed, along with

Table 4. Results of the calculations of the BUHLER 1000 MFWD evaporator-heater
Таблица 4. Результаты расчетов испарителя-отопителя BUHLER 1000 MFWD

Temperature, humidity of air at the inlet	Air flow	Temperature, humidity of air at the inlet	Performance, kW
Summer mode			
28,2 °C; 60 %	720	13,5 °C; 93,2 %	6,72
	1000	14,8 °C; 91,2 %	8,4
29,3 °C; 65 %	720	14 °C; 95,9 %	7,82
	1000	15,4 °C; 94,1 %	9,79
27,6 °C; 70 %	720	13,6 °C; 97,1 %	7,38
	1000	15 °C; 95,6 %	9,23
Winter mode			
-20 °C; 85 %	720	8 °C; 8 %	7,84
	1000	4,7 °C; 10 %	9,62
2 °C; 85 %	720	23 °C; 21,3 %	5,44
	1000	20,6 °C; 24,7 %	6,7
20 °C; 50 %	720	47 °C; 11 %	6,6
	1000	44 °C; 13 %	8,13
8,6 °C; 75 %	1000	27 °C; 23,5 %	6,47

Table 5. Results of calculations of the condenser
Таблица 5. Результаты расчетов конденсатора

Condensing temperature	Subcooling/ superheating	Length, mm	Number of pipes	Number of rows	Number of circuits	Temperature, humidity of air at the inlet
55 °C	5	855	24	2	2	40 °C; 50%

calculations of thermodynamic parameters and fair flow mobility within the unified cabins of ZUK and KUK harvesters. Both numerical and analytical calculations of the cabin's gas dynamics were performed. The problem formulation involved setting heat transfer coefficients, external temperature, and solar radiation as explicit boundary conditions.

In this form, the model most accurately reflects the real air flow behavior in the cabin and can be used to calculate various characteristics and parameters. The developed numerical model allows to calculate the desired characteristics and select the necessary flow rates for any geometry and boundary conditions. In particular, for this task, it enables changing the geometry and location of inlet and outlet air deflectors, setting different flow rates, temperatures, and flow directions for each deflector. For example, it can direct air to the windshield and driver at different speeds, track air flows throughout the cabin, adjust flow distribution, o and modify solar flows for any sun location and radiation.

Figure 3 shows some results of the model calculation.

RESULTS

Based on the calculation results and selection of key HVAC system equipment for the unified SAV 3 combine harvester cabin, using the above-described methodology, the following conclusions were reached:

1. Initial data, including dimensions and materials of combine harvester cabin walls, and external climatic conditions with an outside air velocity of 10 km/h, led to these design characteristics:
   - heat flows amounted to 2.8 kW for the ZUK cabin and 2.9 kW for the KUK cabin;
   - heat losses amounted to 2.2 kW for the ZUK cabin and 2.35 kW for the KUK cabin.
2. The estimated required air flow rate needed to maintain a comfortable cabin temperature of 24°C was 817 m^3/h. Further modeling of heat and mass transfer using the ANSYS software package adjusted this to 740 m^3/h for both cooling and heating.
3. To prevent fogging and overpressure in the cabin, air circulation was set at 75%, with 25% of fresh air intake.

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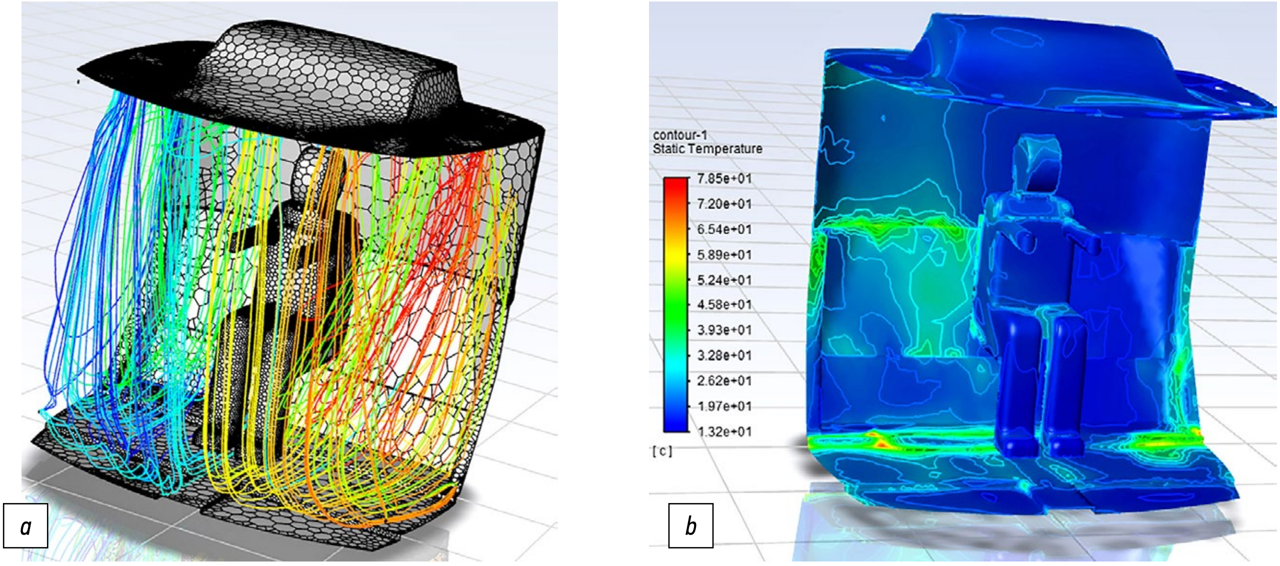

Fig. 3. The results of computer simulation of microclimate parameters in the cabin with the working HVAC system of the combine harvester cabin: *a* — lines of flow rate; *b* — air temperature.

Рис. 3. Результаты компьютерного моделирования параметров микроклимата в кабине при работающей КСКК: *a* — линии тока скоростей; *b* — температура воздуха.

4. Considering the operating conditions of the HVAC system (air dustiness), a correction factor of $K = 1{,}3$ was introduced..
5. The cooling and heating capacities of the combine harvester cabin HVAC system are as follows:
- cooling capacity is 7.8 kW;
- heating capacity is 6.3 kW.
6. The Valeo TM16 type compressor was selected for a coupling speed of 3100–3500 rpm based on the cooling cycle calculations.
7. The condenser was designed to handle a thermal load of 12.7 kW, with an evaporator cooling capacity of 7.8 kW and a condensing pressure of 15 bar.
8. In the future, when selecting climate equipment, suppliers should include thermal engineering calculations and/or bench tests of specific units under the specified external conditions with verified output parameters for temperature, humidity and air flow.

CONCLUSION

Through collaboration with the designers at KZ Rostselmash LLC, a 3D finite element model of the SAV3 combine cabin was developed. This digital copy enabled modeling of heat and mass transfer processes, allowing for precise calculations of gas dynamics and thermodynamic parameters both inside the cabin and on its surface [13]. For example, using the ANSYS software package for modeling heat and mass transfer, the flow rate and temperature parameters of the air supplied for cooling and heating the cabin were clarified, something not achievable with standard engineering calculations [1, 10]. Numerical calculations were carried out in both winter and summer modes, which helped in fine-tuning the equipment characteristics for the HVAC system. It should also be noted that the accuracy of HVAC parameter calculations was improved by validating engineering methods using the universal ANSYS package based on finite element modeling.

ADDITIONAL INFORMATION

Authors' contribution. V.V. Maslensky — analysis of research topic publications, formulation of the article idea, verification of main calculations; Yu.I. Bulygin — expert opinion, approval of the final version of man-uscript; A.V. Pavlikov — performing the calculations, writing and editing the text of the manuscript. All authors confirm their authorship compliance with the ICMJE international criteria (all authors made a significant contribution to the conceptualization, research and preparation of the article, read and approved the final version before publication).

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ДОПОЛНИТЕЛЬНАЯ ИНФОРМАЦИЯ

Вклад авторов. В.В. Масленский — поиск материалов, формализация идеи статьи, проверка основных расчетов; Ю.И. Булыгин — экспертная оценка, утверждение финальной версии; А.В. Павликов — написание и редактирование текста рукописи, проведение расчетов. Авторы подтверждают соответствие своего авторства международным критериям *ICMJE* (все авторы внесли существенный вклад в разработку концепции, проведение исследования и подготовку статьи, прочли и одобрили финальную версию перед публикацией).

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