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Electric Heating System with Thermal Storage Units and Ceiling Fans for Cattle-Breeding Farms

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Abstract: A combined energy–saving heat supply system was proposed that included a combined ETS unit and a ceiling fan, and provided the normative air parameters in a livestock room, with an air temperature of −17 °C and air relative humidity (ARH) of −75%. A heat supply system of a preventive maintenance premises for calves was chosen as the subject of the study. Comparative analysis of the temperature and ARH distribution with height in the preventive maintenance premises, was carried out, with and without a ceiling fan. The study showed that, during the heating period, application of the ceiling fans helped to raise the air temperature and to reduce ARH, in the areas where young stock is located, in accordance with the normative indicators. The energy-saving effect was achieved by supplying warmer ventilation air, which accumulated in the upper zone of the premises from the ceiling fan to the locations of the animals. At the same time, there was a decrease in the consumption of electric energy for the heat supply system of up to 14%.

Keywords: microclimate; ceiling fans; electric thermal storage unit; energy–saving; heat supply system

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1. Introduction

In the case of group housing for calves, animals must be kept in a rather limited space, which may cause critically bad environment conditions, leading to the loss of livestock that, in some circumstances, attains 30% to 40%. Besides, it is associated with productivity reduction of 15% and with the growth of specific fodder consumption by 10% to 15% per product unit [1,2]. This is why, in conditions of intensive animal housing, major attention must be paid to maintaining the optimal microclimate (first of all, temperature and ARH).

Technical–economic studies show that, in case of group housing for calves (i.e., in boxes, preventive maintenance premises, cages, etc.), it is advisable to use storage-type electric units for air heating to serve as internal heat sources designed for small premises operating on the principle of convective heat exchange [3].

Applications of electric thermal storage (ETS) units in combination with energy generating installations in a net-zero multi-energy system, including those designed for farms, belong to the most effective solutions, as well. Such installations must be based mainly on various renewable energy sources (RES), first of all, solar- and wind-power installations for heat and water supply [4–8].

Simulation and tests on an electric thermal storage heating system with solid-state heat storage materials (SS-ETSHSM) using electric energy generated by coal combined heat and power (CHP) units [9] and wind, and solar power stations have been carried out [10,11]. ETS unit are charged during the minimum load period of the energy system. Then, this energy is transferred to the heat-carrier into the water-heating system with the help of 'air-water' heat-exchangers. Sun Y. et al. [9] investigated the characteristics of thermal processes and the regularities of temperature changes in different parts of the heating system occurring in the modes of charging and heat emission ETS under quantitative and qualitative regulation. Zhao H. et al. [10] investigated the heat transfer characteristics of SS-ETSHSM. The thermal calculation method for key parameters of the electric heating system was proposed. Based on the experimental data, the amount of accumulated heat was calculated along with the heat-exchange coefficient, in air channels of various geometries. Amounts of thermal energy transferred from the heat-storage unit to the heated air were estimated as well. The perational effectiveness of ETS unit installations can be improved by a reasonable selection of the air channels number in ETS units, as well as by optimizing their shape and dimensions. Correctly defining the optimal values of air velocity, in channels, and location of air inlets and outlets is also important [11]. Xu G. et al. [12] developed a static-type ETS unit with a heat-storage phase change material (PCM) in which heat-exchange occurs in air channels, owing to natural convection. The proposed device is charged by the off-peak electricity. The dynamic process of the thermal state under a fully charging/discharging cycle of the device was tested. The amount of heat transferred in the air channels to the heated air and the heat given off by the ETS casing by natural convection and radiation were determined as a percentage.

Klymchuk et al. [13] considered various schemes of arrangement of tubular heating elements for the charging of heat storage cells. The temperature distribution over the cross-section of the thermal storage battery at the key points as a result of mathematical simulations was obtained. The obtained simulation results were tested on an experimental installation consisting of four heat storage cells during the charging and discharging of the thermal storage unit. The most effective arrangement of tubular heating elements was determined, allowing for maximum use of the volume of heat storage material. Dependences for determining the index and averaging coefficient of the heat flux were also found. An algorithm for defining the design parameters of the thermal storage depending on the heat supply system operating conditions was described by Klymchuk et al. [14] which made it possible to calculate the optimal position of the heating elements in the heat-storage core, their number, the thickness of the heat-storage layer, dimensions of the heat-storage core and the specific heat flux upon the heated surface. Beknazarian et al. [15] developed a method for selecting and optimizing the high-temperature thermal insulation that can be applied in ETS units.

Theoretical studies of the heat-storage core thermal state of the dynamic-type ETS units were performed by Janssen et al. [16], for both heat charging and heat emission operation modes. Three-dimension models for the heat-exchange processes were developed and the calculations for the non-stationary temperature field were carried out using the finite element method.

When selecting a reasonable method of heat and air distribution in premises, one must consider the requirements for providing the specified air–thermal conditions in work areas where calves are kept, including those of cages.

Various design options for inflow-exhaust ventilation installations and systems, can be implemented enabling solutions for the following assigned tasks, to a significant degree: providing natural ventilation, conventional air distribution equipment (air-ducts for uniform air distribution); sputtering heads, inflow exhaust installations of AHU type), ejection-axifugal air terminal units, conventional air distribution units, etc. [17]. Also, in order to improve the energy efficiency of heat supply systems when using the heat of low-potential energy sources and environmental protection, it is possible to use heat pumps with an eco-friendly refrigerants such as CO_2 [18,19].

However, in such systems, non-homogeneous air temperature distribution with the height takes place. Warm air, having lower volumetric density, tends to concentrate directly under the ceiling, thus forming an airlock. On the other hand, cold and excessively wet air is accumulated in the lower layers where young stock is located, creating unfavorable conditions for animals. Air velocity in the area where calves are located should not exceed 0.3 m/s, during the cold and period of the year [20].

Based on the results of experimental studies, Bodrov [21] developed a method for defining the required value of the heat transfer resistance, for unheated livestock premises depending on the individual biological characteristics of the particular animal type. Methods for calculating the natural ventilation, on an annual basis, were developed as well.

We assume that the maintenance and energy-related parameters of heat- and air-distribution installations and systems can be improved substantially by application of ceiling fans. Such fans, installed in addition to the conventional heating equipment, ensure effective air circulation (within the permissible air velocity range, in areas where animals are located). As a result, the air temperature distribution becomes more uniform with the height, thus reducing the heat energy loss through the enclosing structures and increasing the heat-exchange coefficient of heating devices. The ceiling fan speed control function makes it possible to maintain its specified performance and, consequently, the air flow velocity. Fans can be mounted in any area of premises making it possible to produce their effect on the animals locally depending on their age and physical status.

A ceiling fan installed in the upper area of the premises draws in air from above and directs the fan-twisted air jets having an internal vortex core towards the floor. These air jets reach the ceiling in form of an overlapping flux that gets spread over the premises and moves towards the walls, penetrating the areas where animals are kept.

In these conditions, the required air mobility is provided, and warm air is supplied to the space where animals are located, including in zones under the cages, thus insuring temperature equalization along the vertical axis. This technical solution makes it possible to reduce the required heating system capacity and to save considerable amounts of energy.

Shah et al. [22] reported the results of a technical-economic effectiveness evaluation, for various ceiling fan options, having optimized fan blade design and for wider application of DC electric motors. Babich et al. [23] developed and verified a three-dimensional time-dependent implicit model of the standard ceiling fan by simulation results comparison with the experimental data. Present et al. [24] analyzed the results of the ceiling fans practical application, in commercial premises. Liu et al. [25] and Raftery et al. [26] carried out a series of experimental studies of air velocity fields formed with the use of the ceiling fans, in enclosed areas. Based on the results of experimental data, parameters were specified that, to a major extent, defined the air flow rate in premises, for the case when the ceiling fans operate. Li W. et al. [27] have carried out numerical simulations and experimental studies of air conditioning system and ceiling fans combined operation. Application of a ceiling fan made it possible to reduce the density of airborne participates by more than 20% in the area of the human breathing zone, owing to a better dispersion of airborne participates over the premises. Omrani et al. [28] have performed an incisive analysis of the ceiling fan effects on the microclimate, i.e., air flow rate, thermal acceptability and air quality and energy consumption. It was pointed out that the thermal acceptability and air the flow rate belong to the most extensively studied parameters. Major attention was paid to the air quality and electric energy consumption. Besides, that analysis has brought to light the gap in our knowledge concerning the specifics of natural ventilation and ceiling fans combined operation including their influence on the air quality, in premises, which is vitally important during pandemic periods.

2. Materials and Methods

In this research area, previous works were devoted to simulations of the heat storage cells thermal state of the ETS unit [29] and calculation of the ETS unit thermal characteristics [30,31]. The results of experimental studies of the thermal state of heat-storage cells in both heat-charging and heat-emission modes of ETS unit have been reported [32,33]. Also, an energy-saving infrared heater (IR) for calves was developed for preventive maintenance premises of cattle-breeding farm with an adjustable heat flux depending on the position of the animal and an evaluation of the parameters of its operation was carried out [34].

The heat supply system for the preventive maintenance of an 8 m long, 4 m wide and 4 m high premises, for housing calves, was chosen as the subject of research. The recommended air temperature value in the premises was 17 °C, while the maximum permissible relative humidity (ARH) of the air was 75% [20].

In our research, the air heating combined type ETS unit [31] designed for operation under the electric energy time of use price plan was chosen. A ceiling fan of the type MR-1 [35] was applied for temperature equalization and uniform air flow distribution in the premises.

As part of this work, field studies of the thermal and humidity parameters of the air were carried out in two similar preventive maintenance premises of cattle-breeding farm. Two variants of the operation of the heat supply system were investigated. In the first room, air heating is carried out from the combined ETS unit without the use of ceiling fans, and in the second, with their use. Experimental studies of the proposed heat supply system were carried out in winter from mid-December to the end of March. The temperature and ARH were measured along the height of the preventive maintenance premises at several characteristic control points.

During the experimental studies, the consumption of electrical energy for heat supply of the preventive maintenance premises was recorded for two variants of the system operation.

The results of measurements of the thermal and humidity parameters of the air in the preventive maintenance were processed using the probabilistic-statistical method. A calculation method was also used to justify the parameters and selection of the ceiling fan.

3. Calculating and Selecting the Ceiling Fan Parameters

At the present time, the most commonly applied air heating systems in cattle-breeding farms are those with heated air distribution into the upper area of premises. In this case, the air temperature becomes uniform, in the upper zone, within a short period of time (there is no sensible temperature gradient) owing to the intensive turbulent air mixing. At the same, air flows do not practically reach the lower ('sluggish') zone. It happens because the temperature of air exhausted from the outlet into an unlimited area differs from that of the ambient air in premises with anisothermal air jets. Therefore, air jet parameters, as well as their trajectory, depend on not only inertial forces but also on those of gravity [33].

The following factors have an essential effect on the circulation of air flows produced by the ceiling fans: the height of the premises, the length of suspension, the number fans and their installation points, etc.

The reasonable length of fan suspension h_f (see Figure 1) insuring the maximum air flow rate in the operating area, can be defined from the following equation:

$$h_f = \frac{H_p - L}{4}, \tag{1}$$

where H_p is distance between the floor and the ceiling (m), L is that between the floor and the animal shoulder (m).

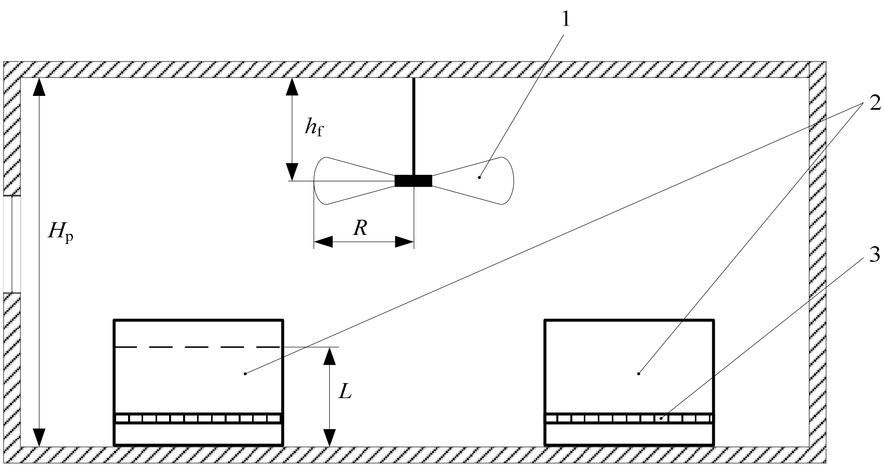

Figure 1. Diagram for optimal ceiling fan installation: 1—ceiling fan; 2—cages for calves; 3—slatted floor of the cage.

In this case, fan blades diameter (D = 2R) shall not exceed 1/3 of the premises width [3]. As applied to up-to-date premises designed for calves housing, fans must be mounted along the longer premises axis.

The number of fans N_f capable to ensure the required values of air velocity and flow rate, for reasonable fan suspension length, can be found from the following expression deduced using experimental methods [3]:

$$N_f = c_f P^{2.91} H_p^{-2.14} \omega_{av}^{4.43}, \quad (2)$$

where P is room perimeter (m), ω_{av} is average air flow rate, in operation area, i.e., standing or rest zone, (m/s); c_f is coefficient depending on the ceiling fan design and its operation conditions.

Since the air velocity field has practically uniform profile, we can assume ω_{ave} = 0.5 ω_0 where ω_0 is initial value of air velocity (m/s).

It has been found out that coefficient c_f = 0.07, for premises having 4 × 8 m, in plain view, and 4 m in height, for ceiling fans type MR-1. By substituting the variables in formula (2) with their known values we obtained N_f = 1.3. One 60 W fan unit type MR-1 with controlled air flow performance rate was chosen, having three 620 mm long blades [35].

4. Calculating Basic Thermal Characteristics of the Electric Thermal Storage Unit

Combined type ETS unit comprises two independently operating heaters, i.e., heat storage core and convector heater (see Figure 2). The heat storage core is designed to be charged with thermal energy during the lower electric power price rate period in order to supply heat during periods of higher electric power price rates. The electric convector-type heater serves as a direct heating energy source supplying heat mainly during the period when the heat storage core is being charged. Technical parameters of the installation are presented in Table 1.

Figure 2. Electric thermal storage unit.

Table 1. Technical characteristics of ETS unit.

Input voltage (V)	380/220
Heat storage capacity (kW)	4.8
Electric convector heater capacity (kW)	2.4
Minimum heat charging period (h)	4
Heat emission period (h)	48
Weight (kg)	200

The required volume of ETS unit heat storage core can be calculated using the following expression:

$$V_{\mathrm{HSC}} = \frac{3600 Q_{\mathrm{ch\_ave}} \tau_{\mathrm{he}}}{\rho_{\mathrm{HSM}} c_{\mathrm{HSM}} (T_{\mathrm{HSC,max}} - T_{\mathrm{HSC,min}})}, \quad (3)$$

where $Q_{\mathrm{ch\_ave}}$ is average heat capacity of ETS unit (W); τ_{he} is duration of ETS unit heat emission period (h); ρ_{HSM} is volumetric density of the heat storage core material (kg/m^3); c_{HSM} is heat capacity ratio of the core material (kJ·kg^{-1}·K^{-1}); $T_{\mathrm{HSC,max}}$ and $T_{\mathrm{HSC,min}}$ are temperature values of ETS unit heat storage core, in the initial (600 °C to 650 °C) and the final (50 °C to 100 °C) moments of the heat charging period, respectively [36,37].

The maximum temperature of the casing outer surface, by the end moment of the ETS unit charging mode is $T_{\mathrm{sh,max}}$ = 60 °C to 70 °C, while at the end moment of the heat emission period it is in the range of $T_{\mathrm{sh,min}}$ = 25 °C to 30 °C which is in the reasonable compliance with the requirements for ETS unit casing outer surface temperature specified in [36].

In view of the fact that there is, practically, no heat loss, in the beginning of the core charging period, the average value of thermal energy Q_{loss} emitted from the surface of the ETS unit casing into surrounding space, in the course of charging, should be calculated for the casing surface temperature value T_{sh} = ($T_{\mathrm{sh,min}}$ + $T_{\mathrm{sh,max}}$)/2, using the following formula:

$$Q_{\mathrm{loss}} = k_{\mathrm{rad}} \alpha_{\mathrm{sh\_ave}} F_{\mathrm{ETS}} (T_{\mathrm{sh}} - T_{\mathrm{ap}}), \quad (4)$$

where $\alpha_{\mathrm{sh\_ave}}$ is average value of the coefficient of heat-exchange between the electric thermal storage outer surface and ambient air (premises), (W·m^{-2}·°K^{-1}); T_{ap} is ambient air temperature in premises (°C); F_{ETS} is surface area of the ETS casing (m^2); k_{rad} is coefficient taking into account the heat loss by radiation from the surface of the ETS casing (value is 1.25).

When calculating thickness δ_{ins} of the thermal insulation layer, it should be considered as a single-layer plain wall. The required thickness of ETS unit thermal insulation layer insuring compliance of the heat loss from the thermal storage unit with the specified Q_{loss} value can be defined from the following equation:

$$\delta_{\mathrm{ins}} = \frac{\lambda_{\mathrm{ins}} F_{\mathrm{ave}} (T_{\mathrm{ins,int}} - T_{\mathrm{sh,max}})}{Q_{\mathrm{loss}}}, \quad (5)$$

where λ_{ins} is thermal-conductivity coefficient of the thermal insulation material (W·m^{-1}·°K^{-1}); $T_{\mathrm{ins,int}}$ is temperature of the thermal insulation internal layer, by the end of the charging period (°C).

$$F_{\mathrm{ave}} = \frac{F_{\mathrm{ins}} + F_{\mathrm{ETH}}}{2}, \quad (6)$$

where F_{ins} is surface area of the thermal insulation (m^2).

Electric heater capacity W_{unit} of ETS unit is defined from the following equation:

$$W_{\mathrm{unit}} = k_{\mathrm{r}} Q_{\mathrm{ch\_ave}} \frac{\tau_{\mathrm{he}}}{\tau_{\mathrm{ch}}} + Q_{\mathrm{loss}}, \quad (7)$$

where τ_{ch} is duration of ETS unit charging period (h); $\boldsymbol{k}_{\mathrm{r}}$ is power reserve coefficient that takes into account the aging of electric heating elements and changes in the supply voltage (value is 1.2).

Quantity of the accumulated heat can be calculated from the following expression:

$$\begin{gathered} Q_{\mathrm{st}} = Q_{\mathrm{ins}} + Q_{\mathrm{HSM}} = \\ = c_{\mathrm{ins}} \rho_{\mathrm{ins}} V_{\mathrm{ins}} (T_{\mathrm{ins,max}} - T_{\mathrm{ins,min}}) + c_{\mathrm{HSM}} \rho_{\mathrm{HSM}} V_{\mathrm{HSM}} (T_{\mathrm{HSC,max}} - T_{\mathrm{HSC,min}}) \end{gathered} \quad (8)$$

where c_{ins} is heat capacity ratio of the thermal insulation (kJ·kg^{-1}·°K^{-1}); ρ_{ins} is its volumetric density (kg/m^3); $T_{\mathrm{ins,max}}$ and $T_{\mathrm{ins,min}}$ are temperature values of the thermal insulation in the end moments of ETS unit charging and heat emission periods, respectively.

Quantity of heat Q_{st} accumulated in ETS unit makes it possible to define time period τ_{warm} required for its heating to temperature value $T_{HSC,max}$:

$$\tau_{warm} = \frac{Q_{st}}{3600(0.8W_{unit?} - Q_{loss,max})}, \quad (9)$$

where $Q_{loss,max}$ is heat loss, for the maximum temperature of the casing surface $T_{sh,max}$ of ETS unit, at the end moment of the heat charging period (W).

In paper by [31] the method of the thermal and aerodynamic calculation of the basic thermal characteristics has been described, for dynamic type ETS unit.

When calculating the average value of heat-exchange coefficient α_{sh_ave}, for the outer ETS unit casing surface, similarity criterion Gr should be defined from Formula (10), following which similarity criterion Nu is calculated in accordance with expression (11). After that, α_{sh_ave} value can be determined [30]:

$$\text{Gr} = \frac{\beta h^3 g(T_{sh} - T_{ap})}{v^2}, \quad (10)$$

where $\beta = \frac{1}{273+T_{ap}}$ (K^{-1}); h—is ETS unit height (m); g is gravity factor (m/s^2); ν is air kinematic viscosity (m^2/s).

$$\overline{\text{Nu}} = 0.695 Gr_{dliq}^{0.25}. \quad (11)$$

In order to define the average value of the heat-exchange coefficient α_{ch_ave}, in air channels of ETS unit heat storage cells, similarity criterion Nu has to be calculated using Equation (12), with the account of the temperature difference θ_{ch} = T_{wch}/T_{air_ch} [38–40]:

$$\overline{\text{Nu}} = 0.023\text{Pr}^{0.4}\text{Re}^{0.8}\left(\frac{T_{wch}}{T_{air\_wch}}\right)^{-0.55}. \quad (12)$$

Thermal-physical properties of solid heat storage materials (HSM) have been analyzed. Formulas for calculating temperature dependent values of the thermal-conductivity coefficient and heat capacity ratio are presented in Table 2 [41].

Table 2. Thermal-physical properties of solid-state HSM.

Material	c (kJ·kg^{-1}·°K^{-1})	λ (W·m^{-1}·°K^{-1})	ρ_{ave} (kg/m^3)
Magnesium oxide	$1.05 + 0.29 \cdot 10^{-3}\ T_{HSM}$	$4.7–1.7 \cdot 10^{-3}\ T_{HSM}$	3000
Chamotte	$0.88 + 0.23 \cdot 10^{-3}\ T_{HSM}$	$0.84 + 0.58 \cdot 10^{-3}\ T_{HSM}$	2200
Corundum	$0.79 + 0.42 \cdot 10^{-3}\ T_{HSM}$	$2,1 + 1.9 \cdot 10^{-3}\ T_{HSM}$	3300
Dinas	$0.837 + 0.25 \cdot 10^{-3}\ T_{HSM}$	$0.93 + 0.69 \cdot 10^{-3}\ T_{HSM}$	2200

Figures 3 and 4 shows the change of the thermal-conductivity coefficient and heat capacity ratio, in temperature range from 50 °C to 650 °C. These are the minimum and the maximum temperatures of the heat storage elements, in ETS unit heat emission and charging modes, respectively.

Analysis of the obtained results provides the reason to conclude that chamotte, corundum and dinas feature linear temperature dependence of the thermal-conductivity coefficient (see Figure 3). Magnesium oxide has the highest values of the thermal-conductivity coefficient and heat capacity ratio in the course of heating and cooling processes, compared to other materials in the study (see Figures 3 and 4). It should also be noted that the thermal-conductivity coefficient of magnesium oxide is inversely proportional to the temperature [42].

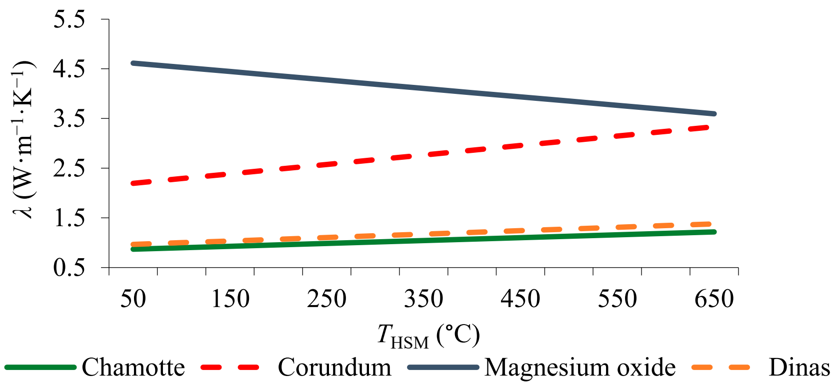

Figure 3. Dependence of thermal-conductivity coefficient on temperature T_{HSM}.

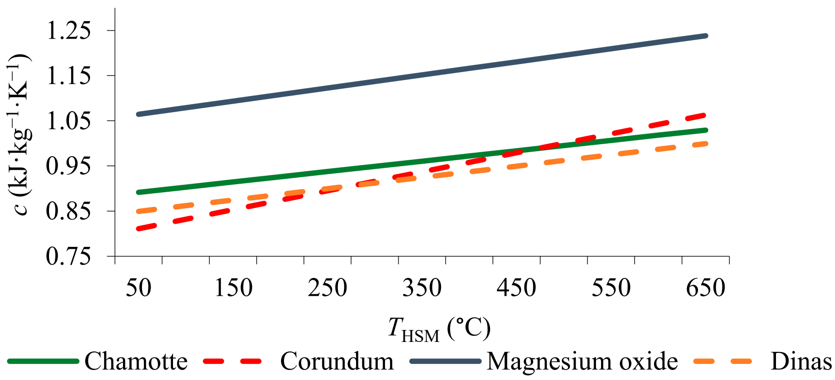

Figure 4. Dependence of heat capacity ratio on temperature T_{HSM}.

5. Discussion

A comparative analysis of temperature and ARH distribution with height of the preventive maintenance premises was made for the ceiling fan switching on and off operation modes in this study. An initial evaluation of the energy efficiency for the system with ceiling fans was also carried out.

5.1. Experimental Studies of Thermal and Humidity Parameters of Air

Two system operation modes applied to preventive maintenance premises were studied. One of these operation modes involves inflow ventilation with heating internal air from electric thermal storage units thus implementing the combined air heating method (without the use of ceiling fans). In this case, there exists a temperature drop in the lower areas and under the cage, i.e., directly under the slatted floor, and 'no flow areas' occur. Besides, minimum required air temperature of 17 °C is not achieved in areas where animals are located (area under the cage [20]). This means that either a heating installation of higher capacity should be used during or the period of its operation should be extended.

In the other operation mode, air flows driven with the use of the ceiling fans circulate with permissible velocity limited in accordance with the recommendations for technological production design (not exceeding 0.3 m/s) [20]. Avoiding forming a heat cushion directly under the ceiling helps to inject a certain amount of heat into the work area, thus maintaining a required temperature environment. It also contributes to reduce the heat loss through the enclosing structures, first of all through the ceiling.

Recommendations for technological production design [20] stipulate the optimal air temperature for housing calves that must be maintained within the cage and under the slatted floor.

Temperature conditions and the aggregate thermal energy consumption for heating depend essentially on the temperature distribution over the space of the premises.

Figure 5 shows vertical air temperature profile, in preventive maintenance premises for calves.

It should be noted that operating ceiling fans ensure ARH reduction by up to 5% in areas where animals are located because of the warmer air supplied into these areas (see Figure 6).

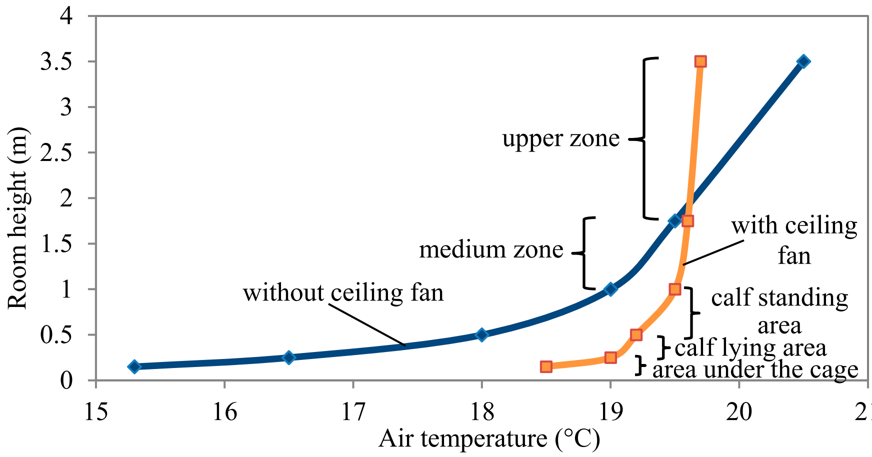

Figure 5. Air temperature distribution, in preventive maintenance premises for calves.

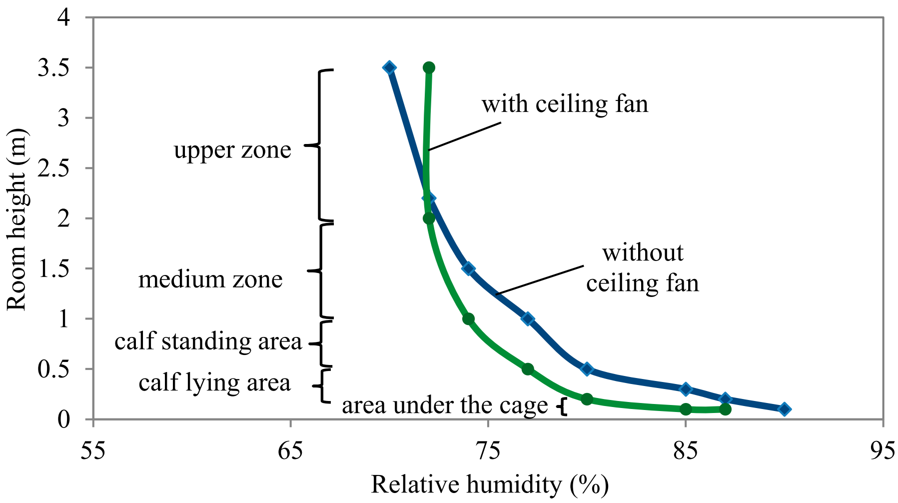

Figure 6. ARH distribution, in preventive maintenance premises for calves.

5.2. Evaluation of the Energy Efficiency of the Combined Heat Supply System

Normally, internal air temperature is kept within a specified range with the use of the on-off action control.

In case that heating-ventilation system is not equipped with ceiling fans the time period within which air gets heated to a specified temperature equals to [43]:

$$\tau_{p1} = \frac{c_{air} G \Delta T}{Q_p}, \tag{13}$$

where c_{air} is air heat capacity ratio (kJ·kg^{-1}·°K^{-1}), G is mass of air (kg), ΔT is specified air temperature control interval in work zone (°C), Q_p is thermal energy production by the heating installation (W).

In case of the ceiling fans application, time period τ_{p2} within which air gets heated to a specified temperature will be equal to:

$$\tau_{p2} = \frac{c_{air} G \Delta T}{k_{te} Q_p}, \tag{14}$$

where k_{te} is coefficient that takes account of additional thermal energy incoming as a result of the ceiling fans operation.

It was experimentally found out that the values of k_f fall into the interval between 1.2 and 1.25.

Then it is follows from Expressions (13) and (14) that $\tau_{p2} = 0.8\tau_{p1}$.

Let us consider the effect of the ceiling fans operation on the air-cooling rate in premises. Evidently, air cooling time period is defined by the heat dissipation rate:

$$Q_{out} = \alpha_{es}(T_{ap} - T_{es})F_{es} + Q_{air}, \quad (15)$$

where α_{es} is heat-exchange coefficient of the enclosing structure internal surface (W·m^{-2}·°K^{-1}), T_{ap} is ambient air temperature in premises (°C), T_{es} and F_{es} are, respectively, temperature (°C) and area (m^2) of the enclosing structure internal surface, Q_{air} is thermal energy loss with exhausted air (W).

Heat-exchange coefficient αfen comprises both convective α_{con} and radiant α_{rad} energy components:

$$\alpha_{es} = \alpha_{con} + \alpha_{rad}, \quad (16)$$

where the surfaces of the premises are blown by air flows there the forced and combined convection operation mode takes place in which case the following expression is valid, with regard to [44]:

$$\alpha_{con} = 3.38\left(\frac{\omega}{l}\right)^{-0.5}, \quad (17)$$

where ω is air flow velocity (m/s), l is distance between the floor and arbitrary cross-section (m).

Studies carried out in [3] have shown that, in case of ceiling fans application, velocity of air flows blowing onto the surfaces increases by the average rate from 0.15 m/s to 0.18 m/s, i.e., increases by a factor of 1.2. If we assume l = 1 m (area where animals are located), then the coefficient of heat convective exchange is 1.1 times higher, for ceiling fans active state, compared to their idle status. Coefficient α_{rad} = const since it does not depend on the air flow velocity and it can be assigned a value in the range of 4 W·m^{-2}·K^{-1} to 4.5 W·m^{-2}·K^{-1}, for animal-housing premises [44].

Having performed the relevant transformations, we can deduce the relationship between the time of air cooling, in premises, in case of ceiling fans application τ_{o2} and without ceiling fans τ_{o1} (in the assumption that Q_{air} = 0):

$$\tau_{o2} = \tau_{o1}\frac{\alpha_{con} + \alpha_{rad}}{1.1\alpha_{con} + \alpha_{rad}}. \quad (18)$$

With the account of optic and thermal-technical parameters of standard buildings [44] designed for young stock housing, Expression (18) can be reduced to the following form $\tau_{o2} = 0.95\tau_{o1}$.

Therefore, in the case when no ceiling fan is applied, the period of self-sustained oscillating process of maintaining a required air temperature in areas where animals are kept is equal to:

$$T_1 = \tau_{p1} + \tau_{o1}. \quad (19)$$

With operating ceiling fans, this parameter can be calculated as follows:

$$T_2 = 0.8\tau_{p1} + 0.95\tau_{o1}. \quad (20)$$

The relative switching frequency of heaters without ceiling fans defined from Expressions (19) and (20) equals to:

$$n_1 = \frac{\tau_{p1}}{\tau_{p1} + \tau_{o1}}. \quad (21)$$

The same parameter defined for the case when ceiling fans are applied has the following form:

$$n_2 = \frac{0.8\tau_{p1}}{0.8\tau_{p1} + 0.95\tau_{o1}}. \quad (22)$$

It is clear from Formulas (21) and (22) that $n_1 > n_2$. Consequently, the average heat energy income, for $Q_{p1} = Q_{pn1}$, exceeds that, for $Q_{p2} = Q_{pn2}$. It means that the thermal energy consumption is greater, in the first case.

Electric power consumption by ceiling fans is insignificant (its maximum input power is just 0.06 kW). Besides, thermal energy dissipated by the electric motor remains in premises thus contributing to the positive component of the thermal balance.

Results of a number of tests and initial technical-economic evaluation of the newly designed heat supply system of preventive maintenance premises for calves have shown that electric power consumption is 1760 kWh for a month-long heating period without ceiling fans, while in case of ceiling fans application it is 1520 kWh. Therefore, electricity consumption can be reduced by 14%.

6. Conclusions

The combined energy–saving heat supply system, which includes a combined ETS unit and a ceiling fan, allows the provision of normative air parameters in the livestock premises: air temperature and ARH.

Experimental studies of the heating–ventilation system carried out for preventive maintenance premises of cattle-breeding farmin the winter period have shown that the application of ceiling fans makes it possible to reduce the heating and cooling times of air in the premises. At the same time, there was a decrease in the consumption of electric energy for the heat supply system by up to 14%. The electric capacity of the ETS unit or the overall time of its operation can be also reduced. The energy-saving effect is achieved by using the heat of the air that accumulates in upper zone of the premises, when it is supplied by a ceiling fan, to the locations of the animals.

Based on the results of tests, a comparative analysis of temperature and ARH distribution with height of the preventive maintenance premises was made for the ceiling fan switching on and off operation modes. An initial evaluation of the energy efficiency for the system with ceiling fans was performed.

Application of combined type ETS unit for air heating makes it possible to reduce the current end user's annual expenditures on electricity by up to 30%, provided that the time of use price plan for electricity is adhered to.

Moreover, it ensures more uniform daily electric power load schedules in power networks, reduction of the equipment installed capacity, as well as that of nighttime electric energy loss. Besides, a mass-scale implementation of ETS units in farming will not require putting into operation considerable additional power-generating capacities.

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