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NATURAL VENTILATION OF ANIMAL SHEDS DUE TO THERMAL BUOYANCY AND WIND

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Abstract. In designing a natural ventilation system for animal sheds it is necessary to assess the ventilation induced by thermal buoyancy and wind forces during different seasons and under different animal housing conditions. By applying analytical and experimental investigation a methodology was prepared to establish ventilation intensity caused by thermal buoyancy and wind and data were achieved on thermal buoyancy and wind values and their relationship. The innovation of the methodology can be described by the fact that a simple equation was formed to calculate the air speed in inlet and outlet openings, a mathematical expression of thermal buoyancy and wind ratio was achieved and the required inlet opening area to let in fresh air compared with the outlet opening area to let out polluted air was substantiated to ensure that all polluted air is removed through a rooftop open in winter. It was calculated that the average air speed in the rooftop outlet opening of a typical cold-type cowshed is 1.3 m/s (when there is no wind, this speed decreases to 0.3 m/s), thermal buoyancy and wind ratio is 0.27 and in order to have all polluted air removed through the rooftop open in winter the inlet opening area in the walls must not exceed 40% of the rooftop opening area. The accuracy of the prepared methodology was tested under natural conditions of barn operation when the distance between air inlet openings and outlet openings was 6.5 m. During the investigation indoor and outdoor temperatures, air speed in the outlet and wind speed were measured. During the experiments the difference of indoor and outdoor temperatures varied from –2 to +16°C and air speed in the outlet – from 1.2 to 1.9 m/s. The analytical results reflect the mean values of experimental data under natural conditions of operation rather accurately. The difference between the experimental and calculated air speed values in the outlet opening was insignificant and was within 0–8% range.

Keywords: natural ventilation, thermal buoyancy, wind, ratio, parameters.

1. Introduction

Ventilation is a key element in the successful operation of any animal production facility. There are two basic types of ventilation system: mechanical and natural. Both require a good design and management.

Of late years it is stressed that natural ventilation of buildings can save a lot of energy, CO_2 emissions are reduced, therefore, a natural ventilation system is environmentally friendly (Allocca *et al.* 2003). A natural ventilation system is by half cheaper compared with a mechanical ventilation, and maintenance makes only one third of the maintenance of a mechanical ventilation system. If natural ventilation is not sufficient, fans have to be installed for mechanical ventilation to complement the natural one.

The study of airflow through building inlet and outlet openings has been the subject of papers by a number of authors. Natural ventilation is induced by thermal buoyancy and wind. At least two methods have appeared in literature to combine the effects of wind with thermal buoyancy (Natural ventilation…2008). One method superimpose wind and thermally induced pressure differences across openings, and use Bernoulli's equation to develop expressions for the speed as a function of vertical in the opening. An alternative method is to calculate natural ventilation due to wind separately and then combine them using the equation: square of total air flow rate is equal to the sum of squares of air flow rates induced by thermal buoyancy and wind forces.

It is proposed that the intensity of natural ventilation in premises is determined by many factors such as wind speed, direction and its turbulence, the size of ventilation openings and their location, heat sources, thermal conduction of outer walls, solar irradiance, etc. However, the most important thing is to evaluate ventilation induced by thermal buoyancy and wind speed (Li and Delsante 2001). Thermal buoyancy pressure is usually smaller than wind pressure, therefore, wind induced ventilation is also greater. However, it is difficult to forecast as wind speed and direction change invariably.

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Theoretical and experimental research of natural ventilation of buildings is broadly published in literature of late years (Zhang *et al.* 1989; Li *et al.* 2000; Yang *et al.* 2005; Luo *et al.* 2007; Andersen 2003; Gebremedhin and Wu 2003). A mathematical model of pigsty ventilation integrating thermal buoyancy and wind was created. According to this model it is possible to find indoor air temperature and ventilation intensity depending on outdoor air temperature, wind speed and direction, the size of ventilation openings and building configuration (Zhang *et al.* 1989). Methodology to calculate air speed in a building depending on wind speed was developed (Aynsley 1999). A computerized model was formed to demonstrate the dependencies of a building zone ventilation on wind pressure (Li *et al.* 2000). The theoretical analysis of the equations of air movement in a building suggests that thermal buoyancy and wind are directed towards one side or directed towards opposite side. Therefore, depending on the arrangement of ventilation openings, wind can assist or oppose the thermal buoyancy force in natural ventilation (Andersen 2003). At proper installation and adequate maintenance of a ventilation system, square of total ventilation intensity is equal to the sum of squares of ventilation intensities induced by thermal buoyancy and wind forces (Luo *et al.* 2007).

So natural ventilation is an effective measure to save energy consumed in buildings and improve indoor air quality. This increased use of mostly fossil-based energy leads to atmospheric pollution and global warming. It is important to fully understand and analyse animal shed ventilation under combined wind and thermal buoyancy effects in order to make its design more effective. This paper reports our efforts in providing such an information to designers.

The objectives of this study were: first to work out a methodology for establishing ventilation intensity induced by thermal buoyancy and wind in designing an animal shed ventilation system, then to test the methodology in production situation of animal shed.

Analytical and experimental investigation was applied.

2. Methods

2.1. Concepts and theory

Natural ventilation is induced by thermal buoyancy and wind (Fig. 1).

Due to thermal buoyancy air comes through the openings in both side walls of a building and is removed through the opening installed above. Wind forces air to come through the openings of one side wall and the air is removed through the higher opening and through the openings of the opposite side wall.

It was established that the total value square of building ventilation intensity is equal to the sum of squares of ventilation intensities induced by thermal buoyancy and wind (Luo *et al.* 2007). By analogy it can be written that mass resultant air discharge in inlet and outlet openings is:

$$G_{1(2)} = \sqrt{G_{t1(2)}^2 + G_{v1(2)}^2}, \quad (1)$$

where $G_{1(2)}$ – mass total airflow rate induced by thermal buoyancy and wind in inlet opening (Index 1) or outlet opening (Index 2), kg/s; $G_{t1(2)}$ – mass airflow rate induced by thermal buoyancy in inlet or outlet opening, kg/s; $G_{v1(2)}$ – mass airflow rate induced by wind in inlet or outlet opening, kg/s.

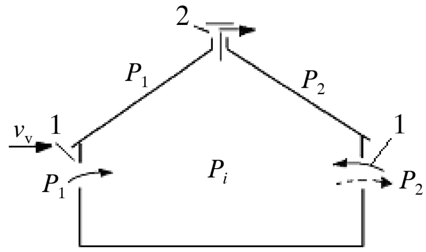

Fig. 1. Schematic view of animal shed with inlet opening (1) and outlet opening (2): P_1, P_2, P_i – wind pressure on windward wall, leeward wall and indoors

By applying Bernoulli's equation it is found out that mass airflow rate induced by thermal buoyancy in inlet and outlet openings are respectively:

$$G_{t1} = A_1 C_{d1} \rho_o \sqrt{\frac{gH\Delta t}{T_o}}; \quad (2)$$

$$G_{t2} = A_2 C_{d2} \rho_i \sqrt{\frac{gH\Delta t}{T_i}}, \quad (3)$$

where: A_1, A_2 – total area of inlet and outlet openings, m^2; C_{d1}, C_{d2} – air discharge coefficient for inlet and outlet openings; ρ_o, ρ_i – outdoor and indoor air density, kg/m^3; g – gravitational acceleration, m/s^2; H – height from midpoint of air inlet opening to air outlet opening, m; Δt – difference between indoor and outdoor temperatures, °C; T_o, T_i – thermodynamic indoor and outdoor temperature, K.

While calculating wind-induced ventilation intensity it is taken that outdoor and indoor air densities are the same and they are indicated by ρ, kg/m^3. By using Bernoulli's equation it is achieved that wind-induced air discharge in inlet and outlet openings is, respectively:

$$G_{v1} = A_1 C_{d1} \sqrt{2\rho(P_1 - P_i)}; \quad (4)$$

$$G_{v2} = A_2 C_{d2} \sqrt{2\rho(P_i - P_2)}, \quad (5)$$

where P_1, P_2, P_i – wind pressure on inlet and outlet openings and indoors, P_a.

As air discharges in inlet and outlet openings are equal, i. e. $G_{v1} = G_{v2}$, and after solving equations (4) and (5) it is achieved that wind pressure induced indoors is:

$$P_i = \frac{C_{d1}^2 A_1^2 P_1 + C_{d2}^2 A_2^2 P_2}{C_{d1}^2 A_1^2 + C_{d2}^2 A_2^2}. \quad (6)$$

Wind pressure on wall P is directly proportional to pressure coefficient k, i.e.

$$P = k\frac{\rho v_v^2}{2}, \quad (7)$$

where v_v – wind speed, m/s.

By applying dependence (7), equation (6) is solved as follows:

$$k_i = \frac{k_1 C_{d1}^2 A_1^2 + k_2 C_{d2}^2 A_2^2}{C_{d1}^2 A_1^2 + C_{d2}^2 A_2^2}, \tag{8}$$

where k_1, k_2, k_i – coefficient of wind pressure on the wall near inlet and outlet openings and indoors.

Solution of equations (4), (5), (7) and (8) results in

$$G_{v1} = G_{v2} = C_{d1} C_{d2} A_1 A_2 \rho v_v \sqrt{\frac{k_1 - k_2}{C_{d1}^2 A_1^2 + C_{d2}^2 A_2^2}}. \tag{9}$$

When all indoor ventilating air circulates through the openings of the ventilation system, $C_{d1}A_1 = C_{d2}A_2$, and equation (9) is simplified:

$$G_{v1} = G_{v2} = C_{d1(2)} A_{1(2)} \rho v_v \sqrt{\frac{k_1 - k_2}{2}}. \tag{10}$$

Having solved equations (1), (2), (3) and (10) the indoor ventilation intensity G is achieved, which is induced by thermal buoyancy and wind and which is equal to air mass discharge in inlet and outlet openings:

$$G = G_1 = G_2 = C_{d1(2)} A_{1(2)} \rho_{0(i)} \sqrt{\frac{gH\Delta t}{T_{o(i)}} + v_v^2 \frac{k_1 - k_2}{2}}. \tag{11}$$

Air speed in inlet (Index 1) and outlet (Index 2) openings $v_{1(2)}$, m/s is:

$$v_{1(2)} = C_{d1(2)} \sqrt{\frac{gH\Delta t}{T_{o(i)}} + v_v^2 \frac{k_1 - k_2}{2}}. \tag{12}$$

For a traditional animal shed (building is one-storey, $T_i = 283$ K and $k_i = 0.7$, $k_2 = -0.3$) equation (12) is simplified:

$$v_{1(2)} = 0.19 C_{d1(2)} \sqrt{H\Delta t + 14 v_v^2}. \tag{13}$$

Thermal buoyancy pressure is lower than wind pressure. Perhaps the ventilation intensity induced by wind is higher than that induced by thermal buoyancy. The ratio of ventilation intensity induced by thermal buoyancy and ventilation intensity induced by wind is;

$$\beta = 0.27 \frac{\sqrt{H\Delta t}}{v_v}. \tag{14}$$

Due to thermal buoyancy air comes through the inlet openings located in both side walls of the building and is removed through the outlet openings above. Wind forces air to come only through the inlet openings of one side wall. In order to avoid draughts at cold weather, it is recommended to have all polluted air removed from the building's upper outlet openings. To achieve that the ventilation intensity induced by thermal buoyancy has at least to be equal to the amount of air blown in by a window through one side, i.e. $G_{t2} \geq 0.5 G_{v1}$. By solving equations (2) and (10) we find out how much inlet openings have to be reduced in winter compared with outlet openings to have air flow upwards in a stable way:

$$A_1 \leq 0,5 A_2 \frac{C_{d2}\sqrt{H\Delta t}}{C_{d1} v_v}. \tag{15}$$

Equation (15) shows that air flow stability increases with an increase of the vertical distance between inlet and outlet openings.

Fig. 2 shows dependence of calculated values of the air speed in the outlet on temperatures between indoor and outdoor air difference and wind, when the distance between the inlet and outlet openings H = 4 m and air discharge coefficient C_{d2} = 0.5 (equation 13). At normal wind (speed 3.5 m/s) and increase of the difference between indoor and outdoor air temperatures from 0 to 20 °C, air speed in the outlet increases from 1.26 to 1.52 m/s, i. e. by 21%. However, if the wind speed increases from 0 to 10 m/s (when Δt = 3 °C), air speed in the outlet increases from 0.33 to 3.6 m/s, i.e. 11 times.

It was calculated that in a typical animal shed (when Δt = 3 °C, H = 4 m, C_{d1} = 0.5, C_{d2} = 0.65, v_v = 3.5 m/s) air speed in the outlet opening of the rooftop is 1.3 m/s (when there is no wind, this speed decreases to 0.3 m/s), the ratio of ventilation intensity induced by thermal buoyancy and wind – 0.27; and in order to have the entire air removed through the outlet opening in the rooftop in winter, the area of inlet openings in the walls does not have to exceed 40% of the rooftop opening area.

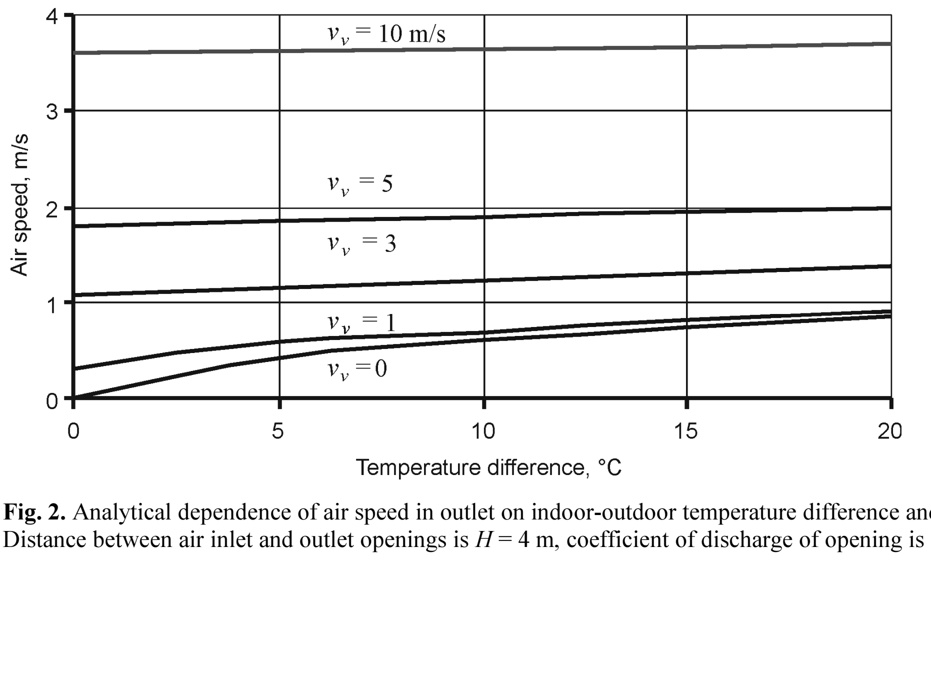

Fig. 2. Analytical dependence of air speed in outlet on indoor-outdoor temperature difference and wind speed v_v. Distance between air inlet and outlet openings is H = 4 m, coefficient of discharge of opening is C_{d2} = 0.5

2.2. Experimental testing of air flow rate through ventilation system openings

The objective of the experimental investigation is to evaluate the influence of thermal buoyancy and wind while ventilating animal sheds and test the accuracy of the equations obtained.

The experiments were carried out in a pigsty with fattening pigs. The number of pigs was 600. In the pigsty animals were grown from 30 kg up to 110 kg. The pig stall was scattered with straw. The room width was 18 m, length – 60 m, and height – 3.5 m. The pigsty was oriented in a longitudinal south-north axis.

The pigsty was naturally ventilated. Clean air flows in the pigsty through gaps (inlet) of 30 mm width at the top of longitudinal wall boards, and the polluted air is removed through one chimney (outlet). The distance between inlet and outlet (top of the chimney) was 6.5 m. The total area of the inlet in winter was 3 m^2. When the average outdoor temperature increased over 5 °C, the windows were opened. Cross-sectional area of the chimney was 4 m^2.

During the experiments measurements of air speed at the outlet, wind speed, temperature indoors and outdoors were carried out. Air speed and temperature were measured by the instrument ALMEMO 2290-3. The values of air speed gained were analysed statistically and compared with the results calculated according to equation (12).

3. Results and discussion

The measuring results are given in Fig. 3.

The investigation was carried out from July to December. During the experiments the air flow in the outlet was stable. Outdoor air temperature varied from –7 to +24 °C, and indoor air temperature was 9 – 22 °C. Thus temperature difference between indoor and outdoor air was from –2 to +16 °C. The average wind speed varied from 3.6 to 4.2 m/s and was higher in the cold season, i.e. when the indoor and outdoor temperature difference was higher. Air speed in the outlet v_2, m/s, and the dependence on the indoor and outdoor temperature difference Δt, °C, are expressed by the following regression dependence:

$$v_2 = 0.042\,\Delta t + 1.25 \quad R^2=0.45, \tag{16}$$

when indoor-outdoor air temperature difference Δt = –2 – 16 °C, wind speed v_v = 3.6 – 4.2 m/s, and distance between air inlet and outlet openings H = 6.5 m.

In the warm season, when indoor and outdoor temperatures were equal (Δt = 0 °C), air speed in the outlet was 1.15 m/s. In the cold season, when indoor and outdoor temperature difference increased up to 16 °C and wind speed increased from 3.7 to 4.2 m/s, air speed in the outlet opening increased up to 1.92 m/s, i.e. 1.5 times.

Comparison of experimental and analytical air speed in the inlet, when indoor and outdoor temperature differs and wind speed is 3.6 (Δt = 0 °C) – 4.2 (Δt = 16 °C) m/s, is given in Fig. 4. The difference between the experimental and analytical values varies from 0 to 8%. When Δt = 3 °C, this difference is equal to 0. It can be stated that the predictive results reflect the experimental data rather accurately.

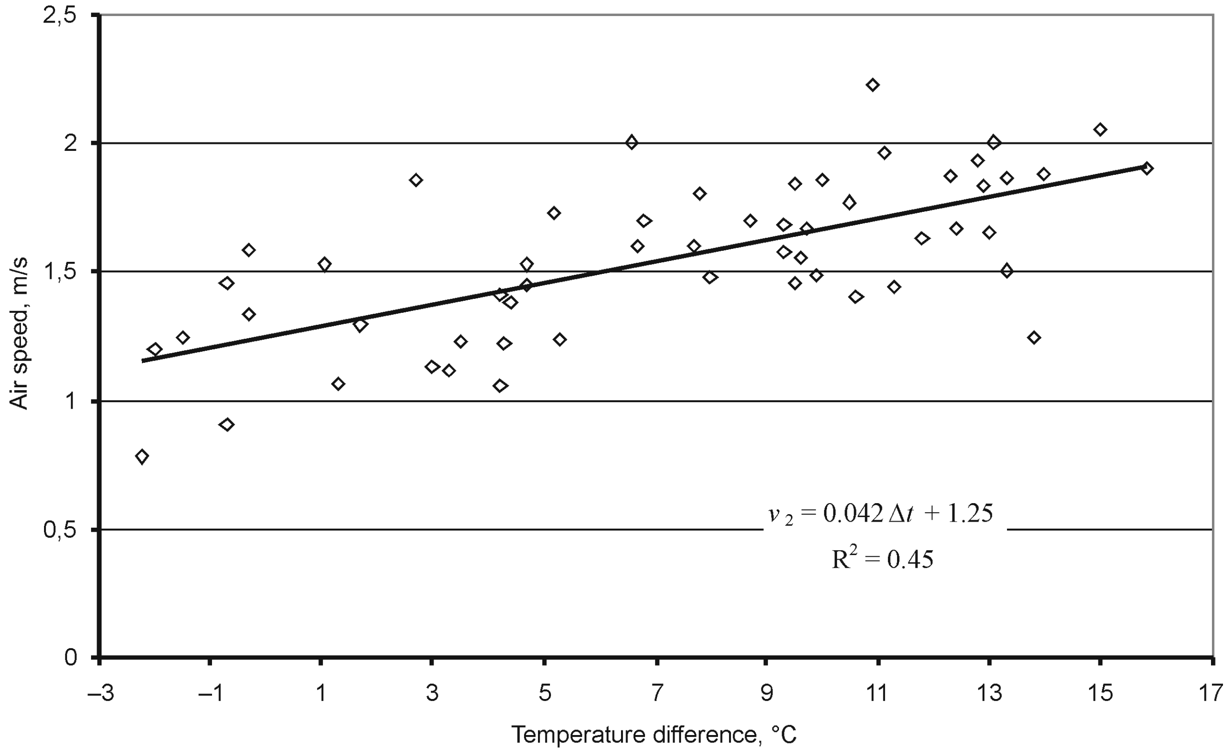

Fig. 3. Experimental dependence of air speed in outlet opening on indoor-outdoor temperature difference. Distance between air inlet and outlet openings is 6.5 m and wind speed – 3.6–4.2 m/s

When air speed in the outlet is known, the required total area of outlets (Kavolėlis and Bleizgys 2006) is:

$$A_2 = \frac{\Sigma Q_o}{c\rho_i v_2}\left(\frac{\varepsilon + x\eta I}{\Delta t} - x_o\right), \quad (17)$$

where ΣQ_o – total heat of animals flow rate, kW; c – specific heat capacity of air, kJ/(kg.K); ρ_i – indoor air density, kg/m^3; ε – ratio of sensible and total animal heat; x_o – total module of heat losses through building partitions (walls, roof, floor and foundations), 1/K; x – partial module of heat losses through building walls and roof, 1/K; η – coefficient that evaluates solar irradiance, m^2.K/W (about 0.022 m^2. K/W); I – average density of sun energy flow towards the building surface, W/ m^2; Δt – permissible difference between indoor and outdoor air temperature, °C.

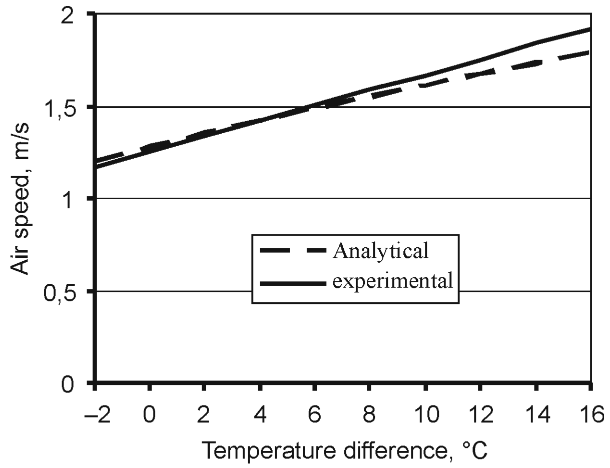

Fig. 4. Comparison of experimental (equation 16) and analytical (equation 13) data of air speed in outlet opening for various indoor-outdoor temperature differencies (difference between experimental and analytical values does not exceed 8%)

The total heat loss module (Kavolėlis and Bleizgys 2006)

$$x_o = \frac{\Sigma UA + \psi F}{\Sigma Q_o}, \quad (18)$$

the partial heat loss module

$$x = \frac{\Sigma UA}{\Sigma Q_o}, \quad (19)$$

where: U – heat transfer coefficient of partitions (walls and roof), W/(m^2.K); A – partition area, m^2; Ψ – specific heat losses through floor and foundations, W/(m.K); F – perimeter of foundations, m.

When foundations and floor are insulated, the specific heat losses Ψ = 0.9, and when they are uninsulated Ψ = 1.5 W/m.K (Albright 1990).

For an uninsulated animal shed, to avoid condensation of water vapor on the internal surface of the outer partitions, the permissible difference of indoor and outdoor temperatures is the following (Kavolėlis and Bleizgys 2006):

$$\Delta t = -\frac{260}{U}\lg\varphi_i, \quad (20)$$

where U – heat transfer coefficient of partition on which vapor condensation is not allowed, W/(m^2.K); φ_i – indoor air relative humidity, parts of unit.

Natural ventilation can efficiently provide year-round ventilation for animals. Air exchange is caused by thermal buoyancy- and wind-induced forces. Both of them are dependent on uncontrollable weather. The inlet and outlet openings are adjusted to control the air exchange rate. There are methods to calculate the parameters of air inlet and outlet openings of an animal shed ventilation system at the design stage (Albright 1990; Caenegem and Wechsler 2000; Stolpe 1985; Sallvik 1989). But the main problem is that these methods do not reveal the accuracy of predicted wind forces.

This work reveals the designers' practical knowledge on the behaviour of natural ventilation due to wind and thermal buoyancy forces acting in animal housing sheds during different seasons and under different animal housing conditions. By applying analytical and experimental investigation a methodology was prepared to establish ventilation intensity, caused by thermal buoyancy and wind, and data on thermal buoyancy and wind values and their relationship were achieved. The innovation of the methodology can be described by the fact, that a simple equation was formed to calculate the air speed in inlet and outlet openings, a mathematical expression of thermal buoyancy ventilation and wind ventilation ratio achieved and the required inlet opening area to let fresh air in compared with the outlet opening area to let polluted air out was substantiated to ensure all polluted air be removed through a rooftop outlet opening in winter.

4. Conclusions

The key data to design a natural ventilation system for an animal shed were substantiated. They can be summarized as follows:

1. The ratio of thermal buoyancy ventilation and wind ventilation was established and a methodology was prepared to forecast ventilation intensity based on the calculations of air speed in inlet and outlet openings. An equation was formed to calculate the ratio of the inlet and outlet areas.

2. The accuracy of the methodology was approved by experimental investigation under natural conditions of shed operation. The difference between the experimental and analytical air speed values in the outlet opening was insignificant and ranged 0–8%.

3. According to the analytical and experimental investigation results, it was calculated that in a typical animal shed air speed in the outlet of the rooftop was 1.3 m/s (when there is no wind, this speed decreases to 0.3 m/s), ratio ventilation rate of thermal buoyancy and wind forces – 0.27; and in order to have the entire air removed through the rooftop outlet opening in winter, the area of inlet openings in the walls does not have to exceed 40% of the rooftop outlet opening area.

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NATŪRALUS TVARTO VĖDINIMAS VEIKIANT GRAVITACINEI IR VĖJO TRAUKAI

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Santrauka

Projektuojant tvarto natūralaus vėdinimo sistemą, reikia įvertinti gravitacinės ir vėjo jėgos skirtingais metų laikais bei skirtingomis gyvulių laikymo sąlygomis sukeltą trauką. Taikant analizinius ir eksperimentinius tyrimus, sudaryta metodika gravitacinės ir vėjo traukos sukeltam vėdinimo intensyvumui nustatyti, gauti duomenys apie gravitacinės ir vėjo traukos reikšmes bei jų santykį. Metodika yra nauja. Sudaryta paprasta lygtis oro greičiui įėjimo ir šalinimo angose skaičiuoti, gauta gravitacinės ir vėjo traukos santykio matematinė išraiška, pagrįstas angų plotas, reikalingas šviežiam orui į patalpą įeiti. Šis plotas lyginamas su angų užterštam orui šalinti plotu, nes visas užterštas oras žiemą turi būti šalinamas per kraigo plyšį. Apskaičiuota, kad tipiškoje neapšiltintoje karvidėje vidutinis oro judėjimo greitis kraigo plyšyje – 1,3 m/s (kai vėjo nėra šis greitis sumažėja iki 0,3 m/s), gravitacinės ir vėjo traukos santykis – 0,27. Tam, kad žiemą visas šalinamas oras išeitų per kraigo plyšį, angų orui įeiti sienose plotas turi neviršyti 40 % kraigo plyšio ploto. Sudarytos metodikos tikslumas patikrintas tvarto natūralios eksploatacijos sąlygomis, kai atstumas tarp oro įėjimo angų ir šachtos viršaus – 6,5 m. Tiriant matuota patalpos ir lauko oro temperatūra, oro greitis šachtoje. Tyrimų metu patalpos ir lauko oro temperatūrų skirtumas kito nuo –2 °C iki +16 °C, o oro judėjimo greitis šachtoje – nuo 1,2 m/s iki 1,9 m/s. Analiziniai rezultatai pakankamai tiksliai atspindi eksperimentinių duomenų vidurkį tvarto natūralios eksploatacijos sąlygomis. Eksperimentinių ir apskaičiuotų oro judėjimo šalinimo angoje greičių reikšmių skirtumas buvo nežymus ir svyravo nuo 0 % iki 8 %.

Reikšminiai žodžiai: natūralus vėdinimas, gravitacija, vėjas, santykis, parametrai.

ВОЗДУХООБМЕН ЖИВОТНОВОДЧЕСКОГО ПОМЕЩЕНИЯ ПОД ДЕЙСТВИЕМ ТЕПЛОВЫХ ИЗБЫТКОВ И ВЕТРА

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Резюме

При проектировании системы воздухообмена здания необходимо учитывать воздухообмен, происходящий под действием тепловых избытков и ветра в разные сезоны года и при разных условиях содержания животных. В результате аналитических и зкспериментальных исследований разработана методика для расчета параметров системы воздухообмена животноводческого помещения, которая обеспечивает стабильную тягу. В методике приведены простые формулы для расчета скорости воздуха в приточных и вытяжных проемах, для определения рационального соотношения площади приточных и вытяжных проемов, для отдельного учета действия тепловых избытков и ветра на производительность системы воздухообмена. Согласно расчетам в типичном неутепленном коровнике шириной 21 м средняя скорость вытяжного воздуха в коньковой щели составляет 1,3 м/с (при отсутствии ветра – 0,3 м/с), соотношение гравитационной и ветровой тяги – 0,27. Для обеспечения стабильной тяги в холодный период года площадь приточных стеновых проемов должна составлять не более 40% площади конькового проема. Точность методики проверена в условиях натуральной эксплуатации животноводческого помещения, в котором расстояние между центрами приточных и вытяжных проемов равно 6,5 м. Во время исследований измерялась наружная и внутренняя температура воздуха, скорость движения воздуха в вытяжном проеме и скорость ветра. Температурный перепад между внутренним и наружным воздухом изменялся от –2 °C до +16 °C, а скорость движения воздуха в вытяжном проеме была 1,2–1,9 м/с. Разница между результатами расчета и экспериментальными данными не превышала 8%. Поэтому можно утверждать, что точность разработанной методики достаточна.

Ключевые слова: естественный воздухообмен, тепловые избытки, ветер, соотношение, параметры.

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