

Patterns of air temperature changes in air ducts of non-contact air heating or cooling systems

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Abstract. The article is devoted to the study of the patterns of changes in air temperature during its movement in a sheet steel air duct during the operation of non-contact air heating systems, when the temperature of the moving air through the duct is higher than the room air temperature, as well as in the process of non-contact air cooling, when the temperature of the moving air in the duct below the room temperature. Based on the mathematical model of the ongoing physical processes developed in the article, an analytical formula is derived that allows, taking into account the presence of a heat-insulating layer or without it, to determine the patterns of changes in the temperature of moving air through the air duct at a given distance, depending on the thickness of the wall of the air duct and the speed of its movement. A dependence is given that determines the value of the heat transfer coefficient of air moving inside the duct to the inner surface of the duct on the speed, which at a constant flow rate is presented as a dependence of heat transfer on the open cross-sectional area of the duct. An analysis was carried out of the change in temperature of the moving volume of air inside the duct as a result of heat exchange with the surrounding air. A comparative calculation of the change in temperature of air moving in a thermally insulated air duct and in an air duct without thermal insulation was performed. The necessity of carrying out a calculation to determine the patterns of changes in the temperature of moving air inside the air duct when designing air heating systems or air conditioning systems is substantiated. It is concluded that it is necessary to conduct a technical and economic analysis on the feasibility of installing thermal insulation of air ducts made of thin sheet steel.

1 Introduction

In some cases, the temperature of the air moving in the duct is very different from the temperature of the environment surrounding the duct outside. Often there is a need to evaluate the process of increasing or decreasing the temperature of the moving volume of air inside the air duct as a result of heat exchange between the latter and the environment. This environment can be the air of the premises in which the air duct is located, or a mass of soil if the air duct is underground. The greatest practical interest is in the case when it is necessary to determine the heating of air cooled in a central air conditioner and supplied with specified parameters to the serviced premises using a system of openly laid air ducts.

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If ducts are not thermally insulated, significantly more additional air cooling is required than with thermally insulated ducts. However, the installation of thermal insulation is associated with additional capital costs. In some cases, it is also necessary to take into account the cost of the additional volume of space occupied by thermal insulation. The decision on the advisability of installing thermal insulation or abandoning it should be made on the basis of a technical and economic analysis.

Review of the status of the issue. A number of works are devoted to the issue of using air heating systems for buildings and structures for various purposes [1-2].

The author of the work [3] notes that due to the growth of national production, the need for the construction of logistics centers and specialized warehouses is increasing. An important issue when designing such premises is to ensure the necessary air parameters for storing goods. The author concluded that the air heating system is the most rational from an economic point of view.

The features of air heating when supplying warm air with vertical jets from top to bottom are considered in works [4-5]. Article [5] analyzed an air heating system with heated air supplied through guide nozzles in the upper zone of rooms with large equipment. It is noted that the energy efficiency of such a heating scheme is ensured by a gradient-free temperature distribution along the height of the room due to the supply of supply air through nozzles at high speed with minimal air flow.

Air heating systems are often used in large premises: shopping centers, cinemas, exhibition spaces, industrial facilities, airports. This heating system allows you to heat the air in a large room much faster compared to traditional water heating. Works [6,7] consider various schemes for organizing air supply when installing air heating combined with ventilation. It is noted that in order to ensure standardized air parameters in the working area of the room, warm air must be supplied in the lower zone of the room, and removed from the upper zone.

The authors of [8] conducted a field study of the energy efficiency of using an air heating system with heat pumps in a two-story residential building in Japan during two heating periods (two winters). It was concluded that the use of this heating system leads to energy savings of an average of 11% while ensuring thermal comfort in the premises.

The issue of reducing electricity consumption with various layout solutions for air heating systems combined with forced ventilation is addressed in work [9]. The authors have developed a thermodynamic method for studying the thermodynamic characteristics of an air heating system. The calculation results indicate that the layout with a fan in front of the air heater makes it possible to reduce electricity consumption by a third.

Work [10] is devoted to the issue of ventilation and air conditioning of museum storage facilities. A method for calculating air exchange in rooms of this type is presented. A scheme for organizing air exchange according to the bottom-up scheme with recirculation is proposed.

Paper [11] presents an analysis of various types of air heaters for air heating systems. It was concluded that the most effective are air heaters in which the air is heated due to the combustion of the gas-air mixture. The advantages and disadvantages of an air heating system are discussed in [12].

The authors of [13] developed a mathematical model of a ducted air heating system.

The issue of heat recovery and the use of secondary heat from oil refining products in air heating of buildings is discussed in [14]. This method is one of the directions for increasing the efficiency of energy saving when designing heating and ventilation systems.

The work [15] presents the results of a study of the potential of a two-temperature warm air heating system in various climatic zones in comparison with a conventional single-temperature warm air heating system. A decrease in energy consumption was noted with

the above heating scheme. An assessment of the efficiency of using an air heating system in buildings for various purposes is presented in works [16, 17].

A team of authors carried out a comparative analysis of the efficiency of two heating systems in high (about 10 m) rooms of an industrial building: air and radiant heating systems. The experimental results indicate that the temperature gradient along the height of the room in both cases under consideration does not exceed 0.2 K. Temperature gradients remained fairly constant across all measured outdoor temperatures. Work examines various heating systems for industrial buildings, including air heating systems.

The analysis of the effectiveness of using an air heating system when designing heating and ventilation systems for buildings for various purposes allows us to conclude that this system has a number of advantages: uniform heating of large premises, economic efficiency, a small gradient in the height of the room, heating speed, ease of operation. Disadvantages include the need to install air purification filters, noise, etc.

2 Materials and methods

2.1. Formulation of the problem

To solve a problem related to the patterns of changes in the temperature of a moving air flow in an air duct of a given length, it is necessary to present a mathematical model that describes the physical process under study. As an object of study, the work considers the transit section of the air duct, which is laid in an air environment where the air temperature is constant and the same along its entire length: $t_c = \text{const}$. A constant value in the area under consideration, for a given speed of air movement in the air duct, is also the heat transfer coefficient ($k = \text{const}$) from the moving air to the air surrounding the air duct outside the room. The process takes place in a circular air duct, without a heat-insulating layer. Initially, the specified quantities for drawing up the model and solving the problem are: diameter of the live section – d_v , m; air flow rate – G , kg/hour; average value of air velocity along the open section of the air duct – w , m/hour.

It is assumed that the temperature of the air entering the considered section of the air duct in the initial section has a given time-constant temperature equal to t_0 . In this case, the temperature value in the initial section of the air duct under consideration differs from the air flow inside the air duct up or down $t_0 \neq t_c$. Having initially accepted that $t_0 < t_c$, we can assume that under the influence of heat exchange with the environment, the air temperature will change. Moreover, if we initially assume that $t_0 < t_c$, then as the air moves in the air duct, its temperature will increase.

2.2. Calculation methods

The problem comes down to determining the patterns of increase or decrease in air temperature in the air duct moving in a given direction of the function t_x .

It is also necessary to take into account the amount of heat released inside the duct as a result of friction against the walls. However, if we neglect the amount of heat that is released as a result of friction, assuming its insignificance in the section under consideration, the heat balance equation for an elementary section of an air duct with length dx can be expressed as:

$$k\pi[t_c - t(x)]dx = Gcdt(x) \quad (1)$$

where: k - heat transfer coefficient, $W/(m^2 \cdot ^\circ C)$; t_c - air temperature surrounding the duct, $^\circ C$; $t(x)$ - functional change in the temperature of the air flow moving in the air duct,

depending on the length of the air duct, °C; dx – elementary duct length, m; G – air flow in the duct, kg/hour; c – heat capacity of air, kJ/(kg °C).

The left side of the equation represents the amount of heat that enters the volume of moving air by heat exchange from the environment surrounding the air duct by heat transfer through the walls of the air duct. On the right side of the equation is the amount of heat that enters the moving air in the air duct and increases its temperature by the amount $dt(x)$.

As a result of the transformation, the heat balance equation can be represented as:

$$\frac{dt(x)}{t_c - t(x)} = \frac{k\pi d}{cG} dx \quad (2)$$

When integrating equality (2) within the length of the duct from the initial section, when $x = 0$, and the air temperature at the entrance to the section of the duct under consideration is t_o , to a section at a distance x from the initial section, where the temperature of the moving fluid depending on the distance x , will be equal to $t(x)$, we get:

$$-\ln \frac{t_c - t(x)}{t_c - t_o} = \frac{k\pi d}{cG} x \quad (3)$$

Then the required dependence for determining the air temperature at the outlet of the section under consideration $t(x)$ can be expressed as:

$$\frac{t_c - t(x)}{t_c - t_o} = e^{-\frac{k\pi d}{cG} x} \quad (4)$$

Or as an expression:

$$t(x) = t_c - e^{-\frac{k\pi d}{cG} x} (t_c - t_o) \quad (5)$$

In the case of air non-contact heating, air moves through the duct at a temperature much higher than the ambient air temperature $t(x) > t_c$. It must be taken into account that the air in the duct reduces its temperature as it moves. The functional dependence of air cooling can be represented by the dependence:

$$\frac{t_c - t(x)}{t_c - t_o} = \exp\left(-\frac{k\pi d}{cG} x\right) \quad (6)$$

Expressing the mass flow of air in the duct under consideration as the product of the open cross-sectional area and the speed of its movement, we obtain:

$$G = \rho \omega \frac{\pi d^2}{4} \quad (7)$$

where: ρ – density of the moving air in the considered section of the air duct, kg/m³; ω – speed of air movement in the duct, m/sec; d – duct diameter, m.

Substituting expression (7) into formula (6) it turns out that:

$$\frac{t_c - t(x)}{t_c - t_o} = \exp\left(-\frac{4k}{d\omega c\rho} x\right) \quad (8)$$

If in expression (8) the diameter is replaced by the radius, then we get:

$$\frac{t_c - t(x)}{t_c - t_o} = \exp\left(-\frac{2k}{r\omega c\rho} x\right) \quad (9)$$

In the case when the live cross-section of the air duct under consideration does not have a circular cross-section, but any other shape, then dependence (9) takes the form:

$$\frac{t_c - t(x)}{t_c - t_o} = \exp\left(-\frac{kS}{\rho c A \omega} x\right) \quad (10)$$

where S – perimeter of the cross-section of the air duct, m; A – cross-sectional area of the air duct, m².

The value of the heat transfer coefficient of the uninsulated wall of a steel air duct k , with some assumption, can be represented by the formula adopted for a flat wall. In this

case, the amount of heat supplied to the moving air inside a circular air duct one meter (1 m) long from the environment can be determined by the formula:

$$Q_{kr} = \frac{t_c - t_x}{\frac{1}{\alpha_1 \pi d_1} + \frac{1}{2\pi\lambda} \ln \frac{d_2}{d_1} + \frac{1}{\alpha_2 \pi d_2}} \quad (11)$$

where: d_1 - duct internal diameter, m; d_2 - duct outer diameter, m.

$$d_2 = d_1 + 2\delta \quad (12)$$

where: δ - duct wall thickness, m;

α_1 - coefficient of convective heat transfer to air moving in the duct from the inner surface of the wall of the duct in question, W/m² °C;

α_2 - the total value of the coefficients of convective and radiant heat transfer to the outer surface of the air duct under consideration from the external environment, W/m² °C;

δ - wall thickness of the air duct in question, m;

λ - thermal conductivity of the wall material of the air duct in question, W/m² °C.

If we assume that heat transfer conditions remain unchanged, only the shape of the living section changes (rectangular shape), then the amount of heat transferred through a flat wall will take the form:

$$Q_{пл} = \frac{\pi d_1 (t_c - t_x)}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2}} \quad (13)$$

If we take into account that the thermal conductivity of the metal wall of the air duct (λ) is high, and the thickness of the air duct (δ) is quite small, then the ratio (δ/λ) tends to a minimum and can be neglected.

In this case, we can assume with sufficient accuracy that the values of the coefficients α_1 and α_2 are comparable.

As noted, when determining the value of the coefficient α_2 , it is necessary to take into account simultaneously the convective heat exchange between the outer surface of the air duct and the environment and the radiant heat exchange between the internal surfaces of the enclosing structures and the outer surface of the air duct in question.

The amount of heat received by air that moves inside a circular duct per unit of time and the amount of heat received by air that moves inside a rectangular duct per unit of time are correlated:

$$Q_{kr} = \frac{t_c - t_x}{\frac{1}{\alpha_1 \pi d_1} + \frac{1}{2\pi\lambda} \ln \frac{d_2}{d_1} + \frac{1}{\alpha_2 \pi d_2}} \quad (14)$$

$$Q_{pl} = \frac{\pi d_1 (t_c - t_x)}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2}} \quad (15)$$

Dividing expression (14) by expression (15) and making some transformations, we obtain expression (16):

$$\frac{Q_{kr}}{Q_{pl}} = \frac{\alpha_1 + \alpha_2}{\frac{d_1}{d_2} \alpha_1 + \alpha_2} \quad (16)$$

Considering that the thickness of the metal air duct is quite small, it can be stated with sufficient accuracy that the ratio of the internal and external diameters tends to unity $d_1/d_2 = 1$. Taking this into account, expression (16) can be represented as $Q_{kr} = Q_{pl}$

Consequently, in the case where the air duct is not insulated, the heat transfer coefficient of the wall of a round air duct can be equal to that of an air duct with a flat wall.

Considering that the thermal resistance of the steel wall of the air duct is a fairly small value, the equation determining the value of the heat transfer coefficient can be presented in the following form:

$$k = \frac{1}{\frac{1}{\alpha_1} + \frac{1}{\alpha_2}} \quad (17)$$

To estimate the value of the heat transfer coefficient k , this article considers the case when the given ambient temperature varies in the range from 25 to 30 °C. The temperature of the cooled air moving inside the duct varies in the range from 10 to 15 °C.

The heat transfer coefficient from the air moving inside the duct to the inner surface of the duct is determined using the formulas of the similarity theory:

$$Nu = 0,023Re^{0,8}Pr^{0,4} \quad (18)$$

The process of heat transfer on the outer surface of the duct wall occurs by free convection in the air space of the external environment inside the room, as well as radiant heat exchange between the surfaces of the external environment and the outer surface of the air duct in question. In this case, the heat transfer coefficient is determined by the formula:

$$Nu = 0,135(GrPr)^{1/3} \quad (19)$$

When the air duct is not insulated, the temperature difference between the outer surface of the air duct and the outside air temperature in the room reaches from 10°C to 15°C. These values correspond to the value of the heat transfer coefficient:

$$\alpha_2 = 3,88 - 3,79 \text{ W/(m}^2 \cdot \text{°C)}$$

The average value of the heat transfer coefficient for calculation in the article is taken equal to: $\alpha_2 = 3,84 \text{ W/(m}^2 \cdot \text{°C)}$.

Assuming that the total area of the internal surfaces surrounding the air duct of the external environment of the room is equal to $A_n, \text{ m}^2$, with an average radiation temperature $T_n \text{ °C}$ and the area of the outer surface of the air duct $A_2, \text{ m}^2$, with a temperature $T_2 \text{ °C}$, then the value of radiant heat transfer can be determined by the formula of the Stefan equation - Boltzmann:

$$Q_{2,n} = \varepsilon' c_0 A_2 \left[\left(\frac{T_n}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \quad (20)$$

where: $\varepsilon' = \frac{1}{\frac{1}{\varepsilon_2} + \frac{A_2}{A_n} \left(\frac{1}{\varepsilon_n} - 1 \right)}$ reduced degree of emissivity of the presented system: fence surface - air duct surface;

c_0 - blackbody emissivity ($c_0 = 4,96$).

The degree of blackness of the outer surface of the air duct, taking into account the painting of its surface with oil paint early $\varepsilon_2 = 0,92$. The average size of the plastered surface of enclosing building structures is taken to be equal to: $\varepsilon_n = 0,91$. Taking the ratio $A_2/A_n = 0,25$, we determine the value $\varepsilon' = 0,9$.

Taking into account the above, the coefficient of radiant heat transfer between the surfaces under consideration will be equal to:

$$\alpha_{2,l} = \frac{0,9 \cdot 4,96 \left[\left(\frac{T_n}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]}{(T_n - T_2)} \quad (21)$$

The total value of the heat transfer coefficient to the outer surface of the non-insulated air duct under consideration will fluctuate within: $\alpha_2 = 9,3 \sim 9,9 \text{ Bt/(m}^2 \cdot \text{°C)}$.

For temperatures commonly encountered in practice, the temperature factor ranges from 1.07~1.08:

$$\frac{\left(\frac{T_n}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4}{(T_n - T_2)} = 1,07 \sim 1,08 \quad (22)$$

The value of the heat transfer coefficient per unit area of the internal surface of a round air duct having thermal insulation (thickness δ_{ins} , coefficient of thermal conductivity λ_{ins}), determined by the formula:

$$k_{kp} = \frac{1}{\left(\frac{1}{\alpha_1 \pi d_1} + \frac{1}{2\pi \lambda_{ins} \ln \frac{d_2}{d_1}} + \frac{1}{\alpha_2 \pi d_2} \right) \pi d_1} \quad (23)$$

(we neglect the thickness and thermal resistance of the steel wall).

The value of the heat transfer coefficient per unit area of the internal surface for a flat wall of an air duct having thermal insulation (thickness δ_{ins} , coefficient of thermal conductivity λ_{ins}), determined by the formula:

$$k_{пл} = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta_{ins}}{\lambda_{ins}} + \frac{1}{\alpha_2}} \quad (24)$$

3 Discussion of results

At a moving speed of 5 m/sec, with an initial temperature of 15 °C in an air duct with a diameter of 0.3 m, as a result of the temperature entering from the environment through the walls of the metal duct, the temperature increases by 0.077 °C/m (Figure 1).

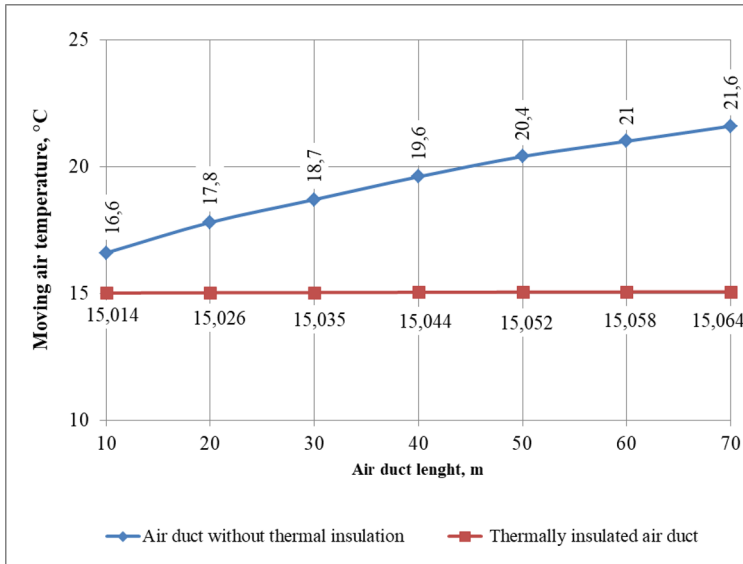


Fig. 1. Graph of changes in the temperature of moving air depending on the length of the air duct, taking into account the presence and absence of thermal insulation.

4 Conclusions

As a result of the study, the following results were obtained.

The coefficient of heat transfer through an uninsulated wall of an air duct made of thin sheet steel for air ducts with round or rectangular live sections has been derived.

An analytical formula has been derived that allows, taking into account the presence of a heat-insulating layer or without it, to determine the patterns of changes in the temperature of air moving inside the duct depending on the thickness of the duct wall and the speed of movement.

To prevent changes in the temperature of moving air flows, it is necessary to cover the air duct with a heat-insulating layer, the value of which is determined by calculation.

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