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Providing the Required Microclimate Parameters in Rooms with a Predominance of Heat Surpluses

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Annotation

This article discusses the provision of the required parameters of the microclimate in public and administrative premises with a predominant heat surplus. Detail the calculation of the heat gain according to the methodology [2] to the warm period, considered the scheme of supply air to the work area of public and administrative domestic premises by calculation, proved the possibility of using mixing valves, the calculation determined the minimum flow of supply air and the temperature of the cooled supply air. It is proved by the use of mixing air distributors that the flow of cooled supply air is comfortable. The air treatment process is constructed on the I-d diagram for the warm period for the city of Tashkent. The advantages of using air distributors of mixing type and economic efficiency are proved. To absorb the heat surpluses of the local-Central system, a traditional system is used to increase the productivity of the air conditioning system, which in turn leads to an increase in capital expenditures.

The greatest heat surpluses in the premises, calculated according to the norms of the sanitary rules and regulations [2], take place in the calculated conditions of the warm period of the year. This is typical for the formation of thermal conditions in residential, public and industrial buildings. Calculation of heat availability in buildings is carried out using well-known methods. The "top-up" scheme is the most traditional for ventilation and air conditioning systems in public, administrative and industrial buildings. The supply of supply air to the working area from above in the form of penetrating jets leads to mixing of air along the height of the room. This mode of organization of air exchange is called mixing ventilation, which causes the equalization of air temperatures along the height of the room (T_B=ty), and the indicator $K_L<1$. In field tests in modern premises of textile enterprises, where supply and exhaust devices are located in workshops up to 8 m high on the ceiling at a distance of up to 2 m from each other, lower temperatures ty of the removed air were noted compared to the air temperature t_B in the working area of the workshop. From these observations, an indicator $K_L<1$ was obtained, which indicates that part of the cooled supply air enters the exhaust ports, bypassing the working area, and this significantly reduces the efficiency of removing excess heat from the working area. In addition, mixing ventilation methods lead to mixing some of the harmful substances, such as water vapor and light gases, which are raised by convective flows to the ceiling, with the supply air from the upper area of the room. The return of some of the hazards from the upper zone to the working area by supply air significantly worsens the sanitary and energy performance of ventilation and air conditioning systems [1].

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When the supply air L_y is supplied directly to the work area, such hazards as heat and humidity, light gases and fine dust are displaced by convective flows to the ceiling and removed with the exhaust air L_y. The working area is filled with fresh supply air, and harmful substances are displaced to the ceiling. This is called the displacement ventilation method. It is necessary to strive for mass implementation in the design and construction of ventilation and air conditioning systems of methods for supplying supply air to the work area and hoods under the ceiling (displacement ventilation).

A restrictive condition for the use of displacement ventilation schemes is the mandatory fulfillment of the requirements for the comfort of supply air entering the work area. The norms of the sanitary rules and regulations [2] indicate that in the cold period of the year, the deviation of the temperature in the supply jet TP from the normalized air temperature in the serviced or working area TV should not exceed 3 °C. During the warm season, this temperature difference can be increased to 6°C.

The air temperature in the working area t_B in accordance with [2] is determined depending on the purpose of the room and the calculated outdoor temperature. For example, in the warm period of the year, the comfortable indoor air temperature in the climate of the Republic of Uzbekistan is 23÷26 °C with a relative humidity of φ_B=60÷40%. For residential premises, the sanitary norm of supply of supply outdoor air $l_{p.n}$ n is 3 m^3/m^2*h of the inhabited area [2]. In public and administrative buildings $l_{p.n}$=60 m^3/person, in the premises of hospital buildings $l_{p.n}$ =80 m^3/person*h.

In order to save energy and save heat and cold, it is recommended that the flow rate of the supply air L_P prepared in the Central air conditioner be taken equal to the minimum required but sanitary standards:

$$L_p = L_{p.n.\min} = Ll_{p.n}, m^3/h. \qquad (1)$$

For residential premises in the formula (1), instead of the number of people in the room L, enter the size of the area f vol., m^2. the Amount of excess heat removed by the minimum flow of supply air $L_{pn.min}$. min. is calculated using the formula

$$Q_{т.изб.п.н} = \frac{L_{п.н.мин} \cdot \rho_{п.н.} c_p \left(t_y - t_{п.н.}\right)}{3{,}6}, Вт. \qquad (2)$$

The temperature of the cooled supply outdoor air $t_{p.n}$ is selected from the conditions for ensuring the comfort of air distribution to the working area and the capabilities of the cooling means used in the Central outdoor air conditioner. To reduce it when the cooled external supply air is supplied to the working area, air distributors with a mixture of cooled external $t_{p.n.}$ and internal t_B air are successfully used [3].

The supply air temperature in mixing air distributors is determined as follows:

$$t_n = \frac{t_{п.н.} + K_э t_в}{1 + K_э}, °C \qquad (3)$$

coefficient in the internal air distributor L_V. e per unit of supply external air LP. n.

$K_e = \frac{L_v}{L_{p.n}}$ -where is the coefficient in the internal air distributor L_V. e per unit of supply external air $L_{p.n.}$

In modern air distribution devices, the Ke indicator=2.8 is achieved [1].

The calculation of the required performance in the Central air conditioner begins with the warm period of the year, when the calculated heat surpluses have the greatest value. For traditional mixing air distribution schemes in the formula (1), the temperature t_u = t_B and must

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meet the conditions of thermal comfort [2] or the requirements of the production technology. The temperature of the supply air TP is determined by rational methods of cooling the supply air and the conditions for the comfort of the cooled air entering the working area. In traditional mixing air distribution schemes, the operating temperature difference is

$$\Delta t_{wor} = t_v - t_p = 8 \div 10°C.$$

In displacement ventilation schemes, the operating temperature difference for the perception of heat surpluses can be significantly greater and is calculated by the formula

$$\Delta t_{раб} = t_в - t_п, °C. \qquad (4)$$

With the use of mixing air distributors, the comfort of the intake of cooled supply air can be ensured at significantly lower t_p. In the warm period of the year t_B = 25°C and the supply air temperature t_p but the air distribution comfort conditions can be assumed to be equal to 19°C. From the converted expression (3), it is possible to calculate the permissible minimum temperature of the cooled supply air outside when using a mixing air distributor:

$$\Delta t_{p.n} = t_p (1 + K_e) - K_e t_v, °C. \qquad (5)$$

Or for the accepted values t_B = 25 °C and t_p = 19 °C but (5) we find the possible minimum temperature of the supply outdoor air:

$$\Delta t_{p.n} = 19 \cdot (1 + 2{,}8) - 2{,}8 \cdot 25 = 2{,}2°C. \cdot$$

Cooling the outside air to the minimum permissible temperature $t_{pn.min}$ = 2°C is energy inefficient and is usually not used.

It is convenient to evaluate and select energy-efficient modes of supply air preparation using a graphical representation of processes on a wet air diagram. A detailed description of the construction and use of this diagram can be found in various literature sources (see, for example, [4]). We assume that the cooling of the supply outdoor air in the Central air conditioner is carried out from the initial state (point H): t_n = 37.5°C; d_n = 10.8 g/kg for the calculated parameters of the warm period of the year in Tashkent [1]. As a rule, moisture release in the premises of administrative buildings is not significant, they occur only from people, and it is rational to absorb them with the amount of sanitary norm of cooled supply air (the process of MON-Y in Fig. 1).

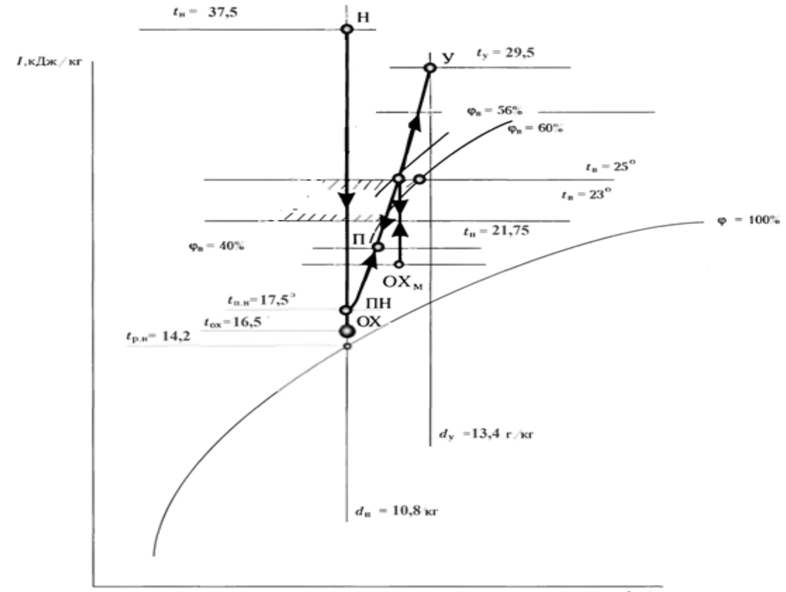

Figure: 1. The mode of cooling the sanitary norm of the supplied outside air and the removal of heat and moisture:

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-OX - "cooling in the supply unit; OX-ΠH - heating in the supply fan and air ducts; PN-V-U - absorption of heat and moisture emissions along the height of the room by cooled supply outside air supplied to the working area and removed under the ceiling of the room; PN -P-V - mixing in the air distributor of cooled and internal air; V-OXм - cooling of internal air in the local heat exchanger.

Figure 1 shows the area of comfortable air parameters in the working area of administrative and residential buildings during the warm season [2]. The absorption capacity of the sanitary norms of cooled air for the perception of moisture surpluses is calculated from the expression

$$\Delta d_{wor} = \frac{W_{ex.moi}}{L_{p.n}\rho_{p.n}}, gr/kg \qquad (6)$$

The amount of water surpluses $W_{ex.moi}$, g / h is calculated using the known method [2]. In relation to administrative buildings, at t = 25°C, the moisture release from one person during moderate work is 185 g/person-h. According to sannorms, $L_{p.n}$ = 60 m^3/h is supplied per person. From (6) we get

$$\Delta d_{wor} = \frac{185}{60 \cdot 1{,}2} = 2{,}6 gr/kg$$

Energetically, it is most rational to cool the supply air at a constant moisture content, when all the cold is spent only on lowering the supply air temperature. In the calculated conditions of the warm period of the year in Tashkent, t_n = 37.5°C; d_n = 10.8 g / kg; dew point temperature Tr. n = 14.2°C. This determines the maximum cooling at a constant moisture content of 10.8 g / kg to t_{ox}=16.5°C. In the supply fan and air ducts, the heating of the supply outside air is GS, and then $t_{p.n}$ = 17.5°C. To ensure a comfortable flow of cooled air into the working area, a mixing air distributor with a coefficient of K_e=1 is used [4]. Using the formula (3), we calculate the supply air temperature

$$t_n = \frac{17{,}5 + 1 \cdot 25}{1+1} = 21{,}25°C$$

We transform the expression (1) with respect to the unknown temperature of the removed air t_y:

$$t_y = K_L \cdot (t_в - t_n) + t_n, °C \qquad (7)$$

In [1], there is a graphical dependence of the K_L indicator on the ratio of the heat surpluses remaining in the working area to the total heat flows. In modern administrative buildings, a large number of office devices are used that consume energy, which is converted into heat carried away by convective flows to the ceiling. According to field observations, no more than 40°C of the calculated heat surpluses remain in the working area from the total heat flows in administrative premises. Under these conditions, according to the graph from [1], we get K_L = 2,2. using (7), we calculate the temperature of the removed air:

$$t_y = 2{,}2 \cdot (25 - 21{,}25) + 21{,}25 = 29{,}5 °C.$$

The cooled outdoor air enters the room at $d_{p.n}$ = 10 g / kg, and the previously calculated absorption capacity for removing excess moisture Δd_{rab}= 2.6 g/kg. Then the moisture content of the removed air will be as follows:

$$d_y = d_{p.n} + \Delta d_{wor} = 10{,}8 + 2{,}6 = 13{,}4 g/kg$$

In figure 1, at the intersection of the isotherms ty = 29.5°C and the moisture content d_y = 13.4 g/kg, we find the parameters of the removed air (point Y). Connecting the pointspn and y with a straight line, we get a ray of the process of absorption of constant heat and moisture in the room by cooled supply air.

The intersection of the straight MON-Y with the isotherm t_B = 25°C gives the point B-air parameters in the working area of the room at a comfortable humidity $\varphi_в$ = 56%. According to Fig. 1, when a sanitary norms of supply air of 60 m^3/h is supplied to a room for one person, the

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heat surpluses perceived by this amount of supply air are calculated by the expression (2):

$$Q_{t.ex.p.n} = \frac{60 \cdot 1{,}2 \cdot 1 \cdot (29{,}5 - 17{,}5)}{3{,}6} = 240 Vt.$$

In a modern administrative building, one employee usually uses a computer that consumes up to 200 watts of electricity. At t_b = 25°C, the apparent heat output from one person is 70 W when working with an average weight. The total constant heat output per person in a room is as follows: 200 + 70 = 270 W/h. It is energetically rational to remove constant heat by cooling $L_{p.n.min}$ [2]. The remaining heat surpluses in the room are determined by the formula

$$Q_{t.ex.m.ox} = Q_{t.ex} - Q_{t.ex.p.n}, Bm. \qquad (8)$$

Depending on the time of day, heat access to the room from solar radiation and transmission heat access through external fences change, which is rationally equated to the left side of the expression (8). This will allow the local air cooler to cool the internal air to a temperature (process B-Ohm in Fig. 1), which ensures the removal of heat surpluses $Q_{t.I.m.ox}$ variable by time of day. The cooling capacity of the local unit is easily automatically adjusted, which allows you to change the cold consumption in each room (or service area) depending on the thermal regime and save energy for generating cold at the Central cooling station.

Systems where Central air conditioners and exhaust units are used, and local air coolers are installed in the premises, are called local-Central. In [4], the energy and economic advantages of local-Central systems in comparison with traditional Central systems are considered in detail. for example, we calculate the cooling capacity of the supply air in the amount of 60 m^3 / h from the Central air conditioner, where the external supply air is cooled to t_p = 18°C and served on top:

$$Q_{t.ex.p.n} = \frac{60 \cdot 1{,}2 \cdot 1 \cdot (25 - 18)}{3{,}6} = 140 Vt.$$

To absorb the same heat surpluses with $Q_{t.I.p.n}$ = 228 W, as was calculated for the local-Central system, the use of a traditional system requires the following increase in the performance of the Central air conditioner:

$$L_n = \frac{240 \cdot 3{,}6}{1{,}2 \cdot 1 \cdot (25 - 18)} = 102{,}8 м^3/ч.$$

The resulting value is greater than the sanitary norm of external supply air in the following number of times:

$$\frac{L_p}{L_{p.n.\min}} = \frac{102{,}8}{60} = 1{,}7.$$

The time-of-day variable heat inputs account for at least half of the constant heat output, and their removal will require an additional two-fold increase in the performance of the air conditioner in a traditional Central system. The overall increase in the performance of the Central air conditioner and exhaust units in a traditional Central system will be at least four times compared to a local-Central system with the same ability to absorb calculated heat and moisture.

The obtained data allow us to draw a conclusion about the significant energy and economic advantages of locally-Central systems with the supply of cooled air to the working area.

It is rational to create Central air conditioners and exhaust units on the basis of technological blocks of air conditioning central frame panel developed and manufactured by Veza [5] , Zhihoz-vent [6].

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