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DETERMINATION OF DESIGN LOADS ON ROOM HEATING AND VENTILATION SYSTEMS USING THE METHOD OF ZONE-BY-ZONE BALANCES

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ABSTRACT

In the spaces of industrial buildings with sources of heat generation, there is a tendency for uneven vertical distribution of air temperatures. Most of the ventilation systems disturb the natural pattern in the development of convective flows above the heat sources and cause an intensive mixing of the room air with inflow jets, which results in rather even distribution of its parameters in the room.

Industrial spaces with heat sources should be ventilated such that the natural tendency of the air for temperature stratification would not be suppressed but stabilized, allowing an improved air exchange efficiency and reduced air exchange rate and providing the required parameters in the occupied zone.

The analytical method of evaluating the efficiency of air exchange for different methods of air supply is described in this paper. Air exchange efficiency coefficients for typical applications are also presented.

INTRODUCTION

In heated and ventilated shops, where the main concerns are excess summertime heat releases and wintertime heat losses, there is a tendency to create nonuniform air temperature distribution throughout the building height. Heated and less dense air tends to occupy the upper zone of the shop, and less heated and more dense air goes to the lower zone. In some cases, one can see nonuniform air temperature distribution in the outlay of a room. The air temperature rises behind heat release sources when they are arranged together in rooms with air heating, and the air temperature near walls and windows is lower than in the center of the room.

The nonuniform air temperature distribution throughout the height can be both a positive and a negative factor. Thus, for ventilated rooms with the working zone on the floor level, the existence of nonuniformity allows the air exchange to be reduced and, in heated rooms, overheating in the upper zone increases heat losses. Information about the nonuniformity of air temperature distribution in a room is needed to calculate loads on heating and ventilation systems and to choose the most suitable system design. In practice, besides experimental methods of evaluating air temperature distribution in heated and ventilated rooms, the analytical method of zone-by-zone heat balances is used. Its essence is in dividing the room into separate zones, where the temperature value is accepted as equal and is calculated as a result of a joint solution of thermal balance equations from each zone.

In general, the number of zones (and the number of heat balance equations) can be rather great. The method of zone-by-zone heat balances is used in designing radiant heating (Bogoslovsky 1982), natural ventilation (Shepelev 1962; Shilkrot 1976), and mechanical ventilation with different air supply schemes (Pozin 1983).

The process of dividing a room into zones and determining heat exchange coefficients between them can be formalized in using the method of zone-by-zone balances. To divide a room into zones, one should define a physical pattern of air- and heat-flow propagation, using common considerations from aerodynamics and thermal physics laws and experimental data.

Conductive, radiant, and convective heat exchanges between zones are accounted for. Convective heat exchange can take place both on the air-surface border and on the air-air border. Air-air heat exchange can be connected with mass transfer (jet transfer, air downflow to the jet) or without it (turbulent heat exchange, when the resulting mass flow is equal to zero). The reliability of calculation results greatly depends on the proper choice of zones and exchange coefficients.

THEORETICAL CONSIDERATIONS

Let us examine the application of the zone-by-zone heat balance method using calculation examples of the required capacity of the natural ventilation systems (aeration) and the warm air heating systems. In natural ventilation, air

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exchange of the room is arranged on a "from bottom to top" scheme. The supply outdoor air enters the room's working zone through openings in the walls and is removed through the skylight. The required air exchange rate, G_o, after the needed air temperature values in the working zone are achieved, can be defined from the room heat balance equation

$$G_o = \frac{mW}{C_p(t_{u,z} - t_{o,z})} \tag{1}$$

where

W = surplus heat releases in a room;
C_p = specific heat of air;
m = coefficient, which shows the part of heat releases affecting the air temperature in the occupied zone.

In the case of natural ventilation, it helps establish the relationship between the air temperature values in the occupied zone, $t_{o,z}$, the air removed from the upper zone, $t_{u,z}$, and outdoor air, t_o:

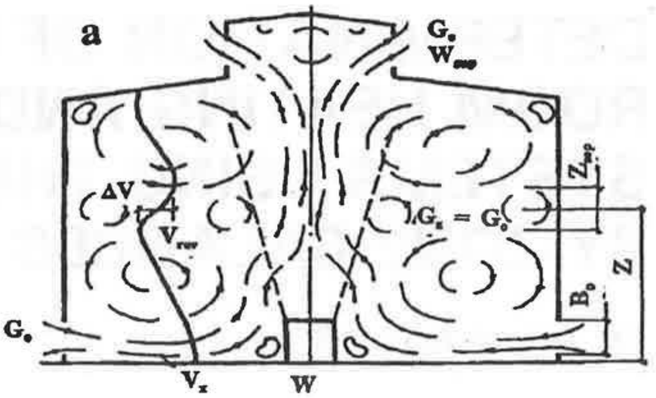
$$m = \frac{t_{o,z} - t_o}{t_{u,z} - t_o}. \tag{2}$$

To define m values by the method of zone-by-zone heat balances, let us examine the current conceptions about air- and heat-flow propagation in the naturally ventilated (aired) room. Shepelev (1962) stated that the air temperature separates into layers throughout the height, and two independent circulation zones are formed. The separation level, Z, is determined in terms of the equality between the airflow rate in the convective flows above the heat sources, G_{conv}, and the airflow, G_o, supplied into the occupied zone (Figure 1a). Other authors (Akinchev 1966; Ratter and Strizhenov 1968) think that no air temperature separation occurs, and there is just a single circulation zone in a room (Figure 1b).

Let us discuss various aspects of the circulation schemes concerned and the conditions for their formation. The scheme dividing the room into two zones seems to be more general. The through circulation can be considered as its partial case if the separation level height is equal to the room's height (the airflow rate in convective flows, G_{conv}, at the roof level does not exceed G_o) or if $G_{conv} > G_o$ at the occupied zone level.

The scheme with air temperature separation seems to be more natural, as in this case where the heated air is situated higher than the less heated one. At the same time, the cause of the through circulation can be both the ejection effect of supply air jets (Akinchev 1966) and the intensive turbulent exchange in the rooms, leading to the air temperature becoming equal throughout the height (Ratter and Strizhenov 1968).

There is a possibility of no temperature separation due to the supply jet's ejection. We assume that the supply jet is a two-dimensional jet attached to the floor and supplied through a wall opening of an infinite length and a finite width. Airflow outside the jet's boundaries is potential. The jet can destroy the air temperature separation in the case when the rarefaction, ΔP_{jet}, created by this jet on the level of separation, Z, is strong enough to move the heated air from the upper zone to the lower zone:



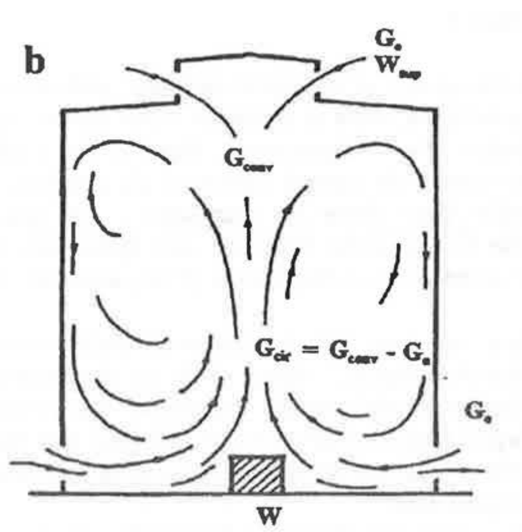


Figure 1a *Air circulation pattern in the naturally ventilated space with temperature separation throughout the height.*

Figure 1b *Air circulation pattern in the naturally ventilated space without temperature separation throughout the height.*

$$\Delta P_{jet} \geq g(\rho_{o,z} - \rho_{u,z})(Z - h_o) \tag{3}$$

where

g = acceleration of gravity;
$\rho_{o,z}$, $\rho_{u,z}$ = density of air in the occupied zone and the upper zone, respectively;
h_o = height of an air supply opening.

On the other hand,

$$\Delta P_{jet} = -\frac{\rho_{o,z} V_z^2}{2}. \tag{4}$$

The vertical air velocity component, V_z, in the potential downflow to the jet (maximum value near the wall) is determined under the supposition that the potential downflow rate is equal to airflow rate in the jet cross section.

Calculations from Equations 3 and 4 with Shepelev's relationships for linear jets have shown that when supply air velocity, V_o, in the slot is 2 to 3 m/s and the separation level, Z, is more than 1 m greater than the air supply opening, h_o ($Z - h_o > 1$ m), the temperature difference of 2° to 3°C between the air of the upper and lower zones ensures the separation stability with regard to the supply air ejection.

A stable temperature separation in rooms with mechanical ventilation is observed with a displacement air supply into the occupied zone. When needed, the stable temperature separation can be achieved by the artificial upper zone air heating of 1° to 3°C compared to the occupied zone air temperature.

Let us examine the influence of the turbulent exchange in the room on the temperature separation stability. The heat flux density due to turbulent exchange can be determined by the formula

$$q_{turb} = A_{turb}\, C_p\, g\rho\, \frac{\delta t}{\delta z} \tag{5}$$

where $\delta t/\delta z$ is the air temperature gradient in the separation zone.

To calculate the turbulent exchange coefficient, A_{turb}, for the separation zone, we use the V.H. Munk and E.R. Anderson relationship (Merrit and Redinger 1973), which is in good agreement with experimental data,

$$A_{turb} = A_{ex}(1 + 3.3\mathrm{Ri})^{-3/2}. \tag{6}$$

The exchange coefficient, A_{ex}, values have been evaluated in two ways. In the first way, it has been considered that A_{ex} is equal to the turbulent exchange coefficient in a convective flow above the heat release source at the separation level. In the second way, according to V.M. Alterman, it has been taken proportional to the total energy of incoming and convective jets attenuating in the room. The Richardson number is defined by the relationship

$$\mathrm{Ri} = \frac{g}{T}\,\frac{\delta t/\delta z}{(\delta V/\delta z)^2} \approx \frac{g}{T_{o,z}}\,\frac{\Delta t/\Delta z}{(\Delta V/\Delta z)^2} \tag{7}$$

where $\delta V/\delta z$ is the room's velocity gradient in the separation zone.

As a result of calculating the following ranges of parameters—the convective component of heat sources up to 11.6×10^3 kW, the heights of the temperature separation level up to 15 m, the room air exchange rates up 50 h_{-1}, the room's heat intensity up to 116 W/m^2, and the velocity of air supply up to 2 m/s—the following values were produced: $A_{ex} \approx 0.4$ m^2/s, Ri > 5. The heat flux density value due to the turbulent exchange is less than 10% of the heat release to the occupied zone and can be neglected.

Analysis of air circulation patterns in the ventilated rooms allowed four typical zones to be considered—the upper and the occupied zones of the room (under and above the temperature separation level, respectively), surfaces of the envelope (walls and ceiling), and equipment in those zones—and heat balance equations to be written for each of them.

The joint solution of the heat balance equations has defined the relationship for the coefficient m:

$$m = \frac{q_{rad\,o,z} + K\dfrac{\alpha_{rad}F_{o,z}}{C_p G_o}}{1 + K\dfrac{\alpha_{rad}F_{o,z}}{C_p G_o}}, \tag{8}$$

where

$F_{o,z}$	=	floor area of the occupied zone;
α_{rad}	=	coefficient of the radiant heat exchange between the envelope and equipment surfaces of the upper and lower zones;
$q_{rad\,o,z}$	=	radiant component of the heat releases into the occupied zone;
K	=	correlation of the surface temperature difference ($\tau_{u,z}$ and $\tau_{o,z}$) and the air temperature difference ($t_{u,z}$ and $t_{o,z}$) in the upper and occupied zones, respectively:

$$K = \frac{\tau_{u,z} - \tau_{o,z}}{t_{u,z} - t_{o,z}}. \tag{9}$$

As the numerical analysis has shown, the K value is close to 1 for the majority of ventilated rooms. It follows from Equation 8 that air temperature distribution throughout the height of the room depends on the relation of radiant and convective heat components, and it changes with the air exchange rate alteration. This conclusion corresponds well with the data received by V.M. Alterman, who showed that the ratio of the supply and convective jet specific energies in the rooms with natural ventilation is proportional to the ratio of radiant and convective heat. The results of the m values calculation according to Equation 8 are in agreement (Figure 2) with the experimental data (Ratter and Strizhenov 1968). Calculation of the required air exchange using Equations 1 and 8 is not difficult because data on the radiant component of the heat entering the occupied zone are available.

Ventilation systems providing warm air heating usually supply air (G_o) by horizontal or inclined air jets. The room has heat losses through the exterior walls (q_w), the roof (q_r), or the floor (q_{fl}), and by outdoor air infiltration (S_{inf}) through the exterior envelope slots (mostly in window frame ledges).

When spreading in the room, the supply air jet becomes wider, increasing airflow, G_{jet}, through its cross sections. As the result of confinement by the envelope, the reverse airflows in the occupied and the upper zones rise. Depending on the air jet trajectory, a part of supplied air, equal to θG_{jet}, gets into the occupied zone. Being cooled near the external wall, the indoor air forms near-wall, downward-

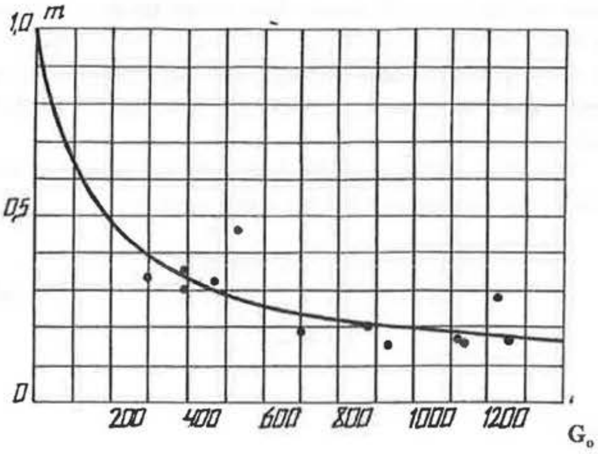


Figure 2 *Coefficient* m *versus air exchange rate:*
- *Strizhenov's experimental data.*
- *Equation 8.*

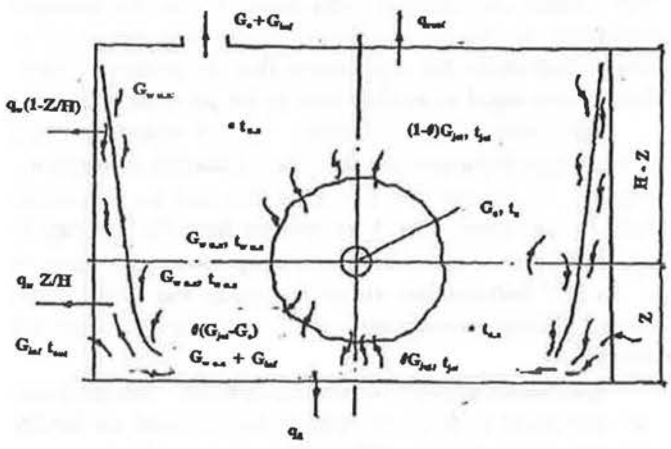


Figure 3 *Scheme of air- and heat-flow distribution in a space with warm air heating.*

projected, convective flows, ejecting air in the amount of $G_{w,u,z}$ and $G_{w,o,z}$ from the upper and occupied zones, respectively.

The outdoor infiltrated air enters the room mostly in the lower part of the vertical envelope and mixes with the cooled air of the near-wall convective flows. The part of the air that entered the occupied zone with the supply jet and the air of near-wall flows are injected partly by the supply jet and partly by the near-wall downflow. The remaining part moves to the upper zone. The air, which has entered the upper zone, feeds the supply jet, forms the near-wall flows, and is partly removed by exfiltration. Between the room's envelope structures, radiant heat exchange takes place. Heat is transferred through the envelope to an outdoor air.

According to the scheme in Figure 3, nine particular zones in a room with warm air heating can be chosen: the supply air jet; air in the lower (below the supply jet's axis) and upper zones; air in downward projected convective flows in the upper and lower zones; and surfaces of the floor, ceiling, and walls in the upper and lower zones. Heat balance equations can be written for each of these zones. In addition to the heat balance equation system, there is a limitation condition—that the buoyancy forces in the supply air jet should not have significant effect on its development:

$$\mathrm{Ar}_x = \frac{K_2}{K_1^2}\mathrm{Ar}_o\left[\frac{X}{\sqrt{A_o}}\right]^3 \leq \mathrm{Ar}_{x\,crit} \qquad (10)$$

where K_1 and K_2 are coefficients of velocity and temperature decay in the jet, and

$$\mathrm{Ar}_o = \frac{g\sqrt{A_o}}{V_o^2}\,\frac{\Delta t_o}{T_{o,z}} \qquad (11)$$

is the Archimedes number at the air supply cross section. Ar_x is the Archimedes number characterizing the ratio of buoyancy and inertial forces at the jet's cross section at the distance X from the air outlet.

The critical value, $\mathrm{Ar}_{x\,crit}$, for the local Archimedes number at the character cross section is 0.2 for horizontal and 0.5 for inclined air jets.

There is no sense in a general solution of the heat balance equation system, as it cannot be analyzed. As an example, the system of equations has been solved numerically for the case of warm air heating of a 72 m × 12 m × 9 m industrial building experiencing an outdoor air temperature of −25°C and an occupied zone air temperature of 16°C. Air is supplied horizontally (θ = 0.5). The coefficients of heat transfer are: walls (considering glazing), $K_w = 2.2 \times 10^{-3}$ kJ/(m²·s·°C), and roofing, $K_r = 1.1 \times 10^{-3}$ kJ/(m²·s·°C). The infiltration rate is from 0 to 2 h^{-1}.

Calculation results have shown that when the infiltration rate increases, the air temperature in the upper zone also increases from 16.4°C to 19.4°C. Air temperature in the occupied zone near the wall decreases from 14°C to 10°C. With a warm air heating system, infiltration reduces air temperature in the occupied zone more than in the upper zone and is the main cause of an uneven air temperature distribution throughout the height and width of the room.

Neglecting an uneven air temperature distribution throughout the room height can lead to underestimating the heating load up to 30%, which results in lowering of the air temperature in the occupied zone.

CONCLUSIONS

The cases discussed above show that the method of zone-by-zone heat balances allows one to more accurately calculate heating loads on heating and cooling systems. The reliability of providing the design temperatures in the occupied zone is enhanced, and in some cases the design heat and air consumptions for heating and ventilation can be reduced.

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