VARIABLE-AIR-VOLUME VENTILATION SYSTEMS FOR INDUSTRIAL BUILDINGS

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ABSTRACT

This paper discusses the application of variable-air-volume (VAV) heating and ventilating systems to industrial facilities with varying cooling loads. Cooling loads vary with gains and losses through building envelopes, ventilation loads, and internally generated heat. Applications for the methodologies discussed may include manufacturing plants, welding shops, and other types of industrial occupancies with similar characteristics.

The limitations of air distribution systems may impose restrictions that prevent operation of the heating and ventilating systems in effective ranges of operation. Conservative recommendations, based on experimental studies performed on physical models under field conditions, are presented for VAV air distribution techniques. Economically appropriate ranges of operation for the application of VAV heating and ventilating systems are defined.

INTRODUCTION

The heating and electrical energy consumption of ventilation systems depends on their operating conditions. Sotnikov, Steinacher, and Rickelton [1, 2, 3] have shown that significant energy savings are associated with the use of VAV systems. The savings achieved by using such systems are due to:

- Reductions in fan energy and, for some operating conditions, because of reduced air flow at part-load, and
- Reductions in heat energy, refrigerating effect, and water consumption resulting from application of logical criteria to the operation of the system.

The analysis in this paper demonstrates that using VAV systems is cost-effective in cases when:

1. Mechanical ventilation is required year-round to provide ventilation air for cooling during much of the year and to provide warm air during the heating season. Cooling loads tend to be variable due to the changing nature of the envelope and other loads, as illustrated in Figure 1.
2. Mechanical ventilation is required year-round to control moisture. Where wet processes are found, as in the wood-pulp, paper, woodworking, and tanning industries, opportunities exist for the use of VAV systems for moisture control. The amount of outside air required will vary with the absolute humidity of the outside air.
3. Noxious materials in the form of heat, vapors, or gases are released. Examples of this may include experimental production departments in chemical industries, welding facilities with local exhausts, and telephone exchanges.

The difficulties in selecting logical controlling parameters for the proper distribution of air are among the reasons that the application of VAV systems is limited today. The principles of identifying logical control parameters and methods of air distribution that support the use of VAV systems in industrial environments are discussed in this paper.

LOGICAL MODES OF OPERATION FOR VENTILATION SYSTEMS

A thermodynamic model was proposed by Rymkevich and Khalamaizer [4] to determine logical modes of operation for air-conditioning systems that meet the environmental requirements of an enclosed space. The principles of determining the main parameters of logical modes of operation for heating and ventilating systems are elaborated below for two modeled cases.

Case One

In the first case, the heating and ventilating system provides heated air in winter and adiabatically cooled air in summer. In winter the air temperature in the occupied zone, t_{oz}, should be maintained within the limits ($t_{oz,min}$ to $t_{oz,max}$) and relative humidity, ϕ_{oz}, should be less than $\phi_{oz,max}$. In summer t_{oz} should be lower than $t_{oz,max}$. The maximum volume flow rate of the inlet air, Q, supplied in summer for the assimilation of heat gains, exceeds the minimum air volume flow rate, Q_{out}, needed to assimilate harmful vapor and gases to meet sanitary and hygienic requirements. The flow rate, Q, must also compensate for the air extracted by local exhaust systems as well as creating a surplus pressure. In this case, the value of Q_{out} remains constant.

Case Two

In the second case, the heating and ventilating system is used to dilute or convey away harmful gases where the volume of exhaust is variable and may exceed in volume flow the maximum amount of air required to meet summer design cooling requirements.

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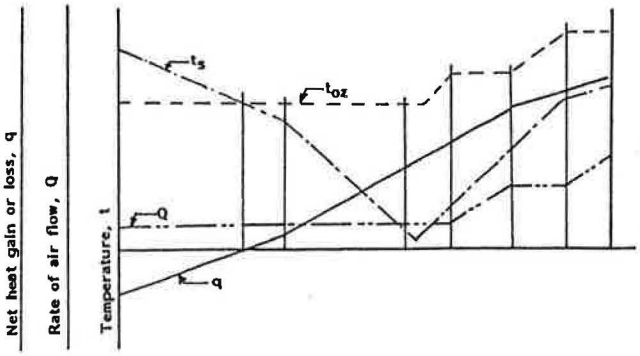

Figure 1 *Supply air temperature t_s, net heat gain or loss q, and rate of air flow Q as a function of outdoor air temperature*

Variations in the values of surplus heat release (q), outside airflow rate (Q), and the temperature (t_{oz}) depend upon the temperature of the outside air, t, as shown in Figure 1. This figure gives graphic information required for selecting and designing equipment for handling the inlet air, air distribution devices, and automatic controls for the system. Within the range of reference values for the outside air temperature, t_1 to t_7, the system operates in the following modes:

Mode 1: The first mode occurs when the temperature of the outside air is between t_1 and t_2. The supply air quantity, Q, is at a minimal value equal to the amount of outside air required to meet minimum ventilation requirements and is heated above the temperature of the occupied zone. When the supply air temperature (t_s) equals t_{oz}, internal gains offset envelope and ventilation losses.

Mode 2: The second mode occurs when the temperature of the outside air is between t_2 and t_3. The supply air quantity remains at the minimum level and is heated to a value less than the space temperature. When the outside air temperature reaches t_3, the supply air temperature, t_s, equals the outside air temperature.

Mode 3: When the outdoor temperature is between the values of t_3 and t_4, the supply air temperature is allowed to float up with the outside air temperature, and the supply air quantity remains at a minimum value. When the outside air temperature reaches t_4, the space temperature is at the upper range of the desired winter operating temperature.

Mode 4: When the outside air temperature is between t_4 and t_5, the air temperature is allowed to float with the outside air temperature, and the supply air quantity is proportionally increased to maintain the maximum winter operating temperature.

Mode 5: In this regime, t is between t_5 and t_6 and the outside air in the amount of Q_5, having the temperature $t_s = t$, is supplied into the space ($t_{oz,min} < t_{oz} < t_{oz,max}$) to absorb surplus heat.

Mode 6: In this regime, $t_6 < t < t_7 = t_{design}$ and outside air in the amount of $Q_{out,5} < Q_{out} < Q$ is supplied into the space to absorb surplus heat. It is adiabatically cooled if that is economically expedient.

The main obstacle for wide application of VAV systems is the difficulty of maintaining the required parameters of the atmosphere with variations in the supply airflow and the initial temperature differential, $\Delta t_s = t_s - t_{oz}$. Variable conditions at the point of discharge from the supply air terminal device during a year-round cycle of the system operation result in changing the ratio between gravitational and inertial forces in the air jets, which affects the trajectory of their movement as well as where they separate from the ceiling. This may cause the formation of areas with abnormal air motion or increase the area of poorly ventilated zones,

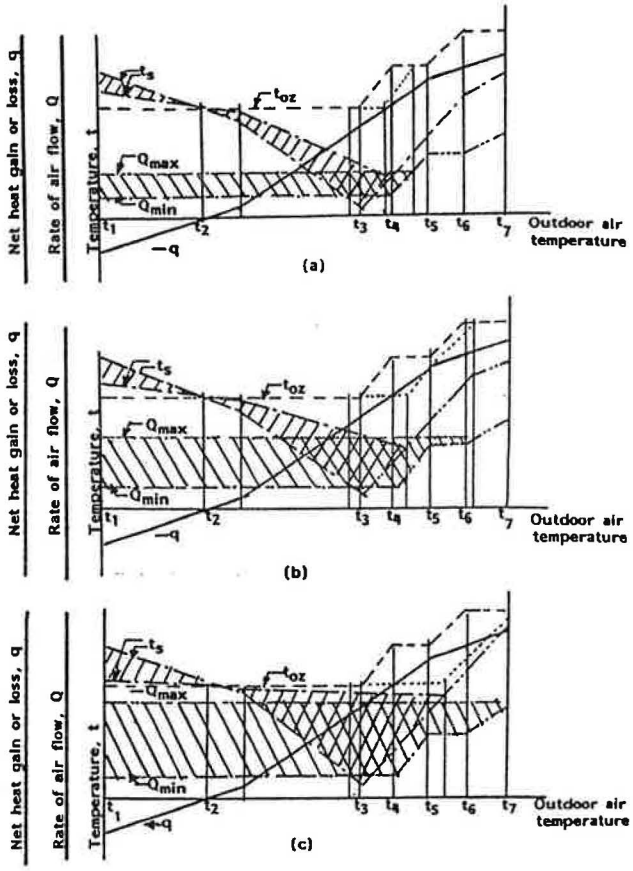

Figure 2 *Supply air temperature t_s, net heat gain or loss q, and rates of air flow Q for three different ranges of required Q's: (a) low maximum airflow rates Q_{max}, (b) moderate Q_{max}, and (c) high Q_{max}*

particularly in spaces with considerable machinery. The designing of air distribution (determination of the type, size, and number of supply air terminal devices, the height of their installation, and the position of controls) is performed at the design parameters of outside air in summertime ($t = t_{design}$). The possibility of providing standard atmospheric parameters in the occupied zone by the selected means of the air distribution is checked at reference values of outside air temperature (t_1 to t_6). If the required atmospheric parameters in the zone of occupation are not established at the values of Q and Δt_s corresponding to these regimes, it is necessary to increase the volume of supply air by mixing some recirculation air into it. If this recirculation is not allowed, the outside airflow should be increased, which reduces the initial temperature difference accordingly.

The characteristics of effective application ranges for the operation of heating and ventilating systems for spaces with variable gas emissions should be determined in the same way as in the first case for two extreme values of Q_{out}, namely, $Q_{out,min}$ and $Q_{out,max}$, calculated for minimum and maximum quantities of the released gases. With increasing Q_{out}, as can be seen from the graphs in Figure 2, all the reference points beginning with t_3 shift to the right along the abscissa.

The design of rational regimes for a number of heating and ventilating systems has shown that the degree of regulation of airflow rate in such systems,

$$D = (Q_{max} - Q_{min}) / Q_{max}$$

can reach significant values. For instance, D = 0.25 to 0.75

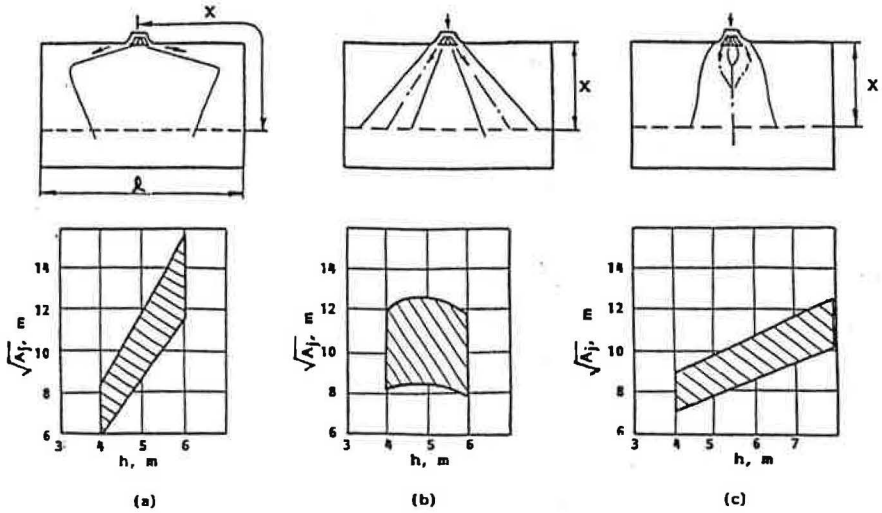

Figure 3 *(a) Radial jet, (b) conical jet, and (c) compact jet*

for machinery assembly and metalworking shops; $D = 0.3$ to 0.9 for welding shops.

AIR DISTRIBUTION

Let us consider the possibilities of varying the supply airflow rate for some conservative and special methods of air distribution. Air can be supplied through ceiling diffusers by radial, conical, or compact jets (Figure 3). Grimitlyn et al. [5] showed in experimental studies that the most uniform distribution of temperatures and velocities through the zone of occupation when ventilated by one jet is achieved when the ratio between the cross-sectional area of the jet at the point of its entering the occupation zone and the area in the zone $\bar{A}_j = A_j/A_{oz}$ is within certain limits. For air supplied through diffusers in the form of radial jets, choose $\bar{A}_j$ between 0.55 and 0.6; for conical jets, use $\bar{A}_j$ between 0.3 and 0.6; and when the air is supplied by compact jets, $\bar{A}_j$ should be between 0.3 and 0.5. Graphic interpretation of these conditions is shown in Figure 3. It can easily be seen when matching these graphs that transition from a radial jet to a compact one, resulting from reduced air volume at the air discharge height of 4 to 6 m, will not significantly change the uniform distribution of the parameters, with the area served by one diffuser being 50 to 100 m^2.

Computer-aided design for various heat loads in the space and supply temperature differentials leads to the conclusion that in the case of air supply by VAV heating, ventilating, and air-conditioning systems through the multi-diffuser ceilings, where only manual regulation of the jet shape in the process of mounting the system is possible, the degree of regulation does not usually exceed the value of 0.65.

When air is supplied through wall-mounted grilles with jets clinging to the ceiling, the length (L) of the room per one grille at which the most uniform distribution of the parameters across the zone of occupation is achieved can be found [5] from the relation:

$$\sqrt{m_2} + h - h_{oz} < L < 0.45\, m \sqrt{bh} \qquad (1)$$

where m is the aerodynamic characteristic of air diffuser. The width of the room, b, served by one grille should be within the limit of

$$b = (2.7 \text{ to } 4)\ (L + h - h_{oz}) / m. \qquad (2)$$

In the case of cooling, separation of the jet from the ceiling surface is allowed at a distance of $X > 0.5L$. Calculations have shown that the degree of regulation of the airflow rate in this case can reach the value of 0.5. There are special air distributors, with pneumatic electric drives for VAV ventilating and air-conditioning systems, used mainly in small offices, which allow changing the area of air discharge outlets as proposed by Connor and TROX [6, 7], as well as air distributors of the ejector type that change the ratio of the primary air (delivered to the air distributor) to the air ejected from the space into the mixing chamber of the air distributor, as shown by Carrier [8]. Their application makes it possible to increase the degree of regulation up to $D = 0.7$.

When air is supplied by inclined jets, the most uniform distribution of the parameters and the most effective utilization of the supplied air are achieved when the design patterns shown in Figure 4 are maintained [9]. Among the

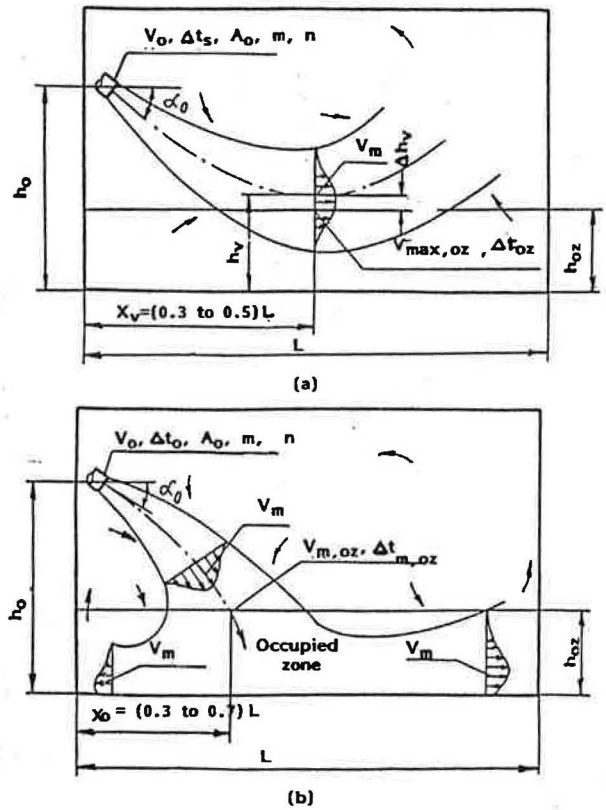

Figure 4 *Trajectory of: (a) a heated air jet and (b) a cooled air jet*

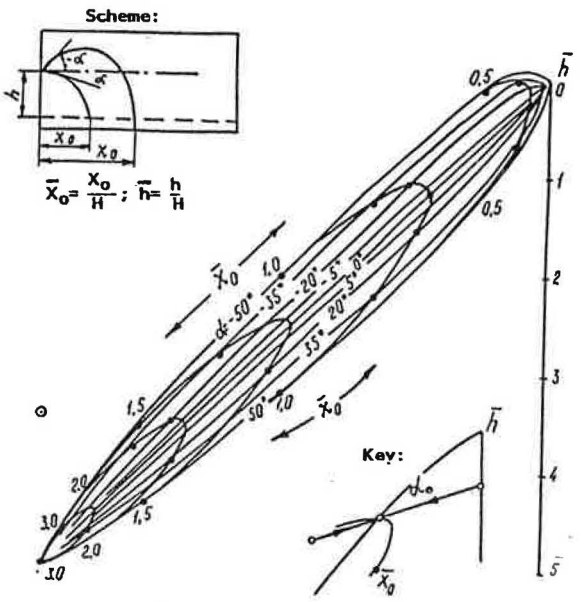

Figure 5 *Velocities and temperatures in the occupied zone with heated supply air*

most important characteristics of the method of supplying air with inclined jets, which determine to a great extent sanitary-hygienic and economic effectiveness of the system, are the coordinates of the point where the axis of a chilled air jet enters the occupied zone (X_o) and those of the vertex of the heated jet axis, (X_v, h_v). To solve these problems, an equation for the axis of a nonisothermal jet bent under the action of gravitational forces is used [10]:

$$h - ho = X \tan(\alpha_o) + \frac{0.47\ X^3}{H^2 \cos^3\alpha_o} \qquad (3)$$

where X and H are current coordinates of the jet axis (Figure 4). The geometric characteristic H is

$$H = \frac{m\ Vo\ \sqrt{t_{oz} + 273}\ \sqrt[4]{A_o}}{\sqrt{n\ \Delta t_s\ g}} \qquad (4)$$

where n = temperature characteristic.

Equation 3 is used to determine the trajectory of the air jet and, consequently, the place of intersection of the chilled air jet with the upper level of the occupied zone or the location of the lowest point of the trajectory of a heated air jet axis. In the case of chilled air supply, in order to achieve the most uniform distribution of the parameters through the occupied zone and to prevent the formation of stagnant (poorly ventilated) zones, the abscissa of the point of jet entering the occupied zone (X_o) should be located within 30% to 70% of the length of the ventilated area (Figure 4)

$$0.3\bar{L} < \overline{X_o} < 0.7\bar{L} \qquad (5)$$

where: $\overline{X_o} = X_o/h$, and $\bar{L} = L/h$.

To facilitate the calculations, a nomogram is suggested that allows locating X_o at preset values of the jet discharge angle α_o, the height h_o of mounting the air distributor, and its size A_o. In the case of heated air supply (Figure 5), in order to effectively use the heat of supplied air and to uniformly distribute the velocities and temperatures in the occupied zone, the abscissa of the jet vertex, X_v, should be within 30% to 50% of the length L of the zone ventilated by this jet and the ordinate should be above the upper level of the occupied zone at a distance not exceeding 0.2 S for air supply by a compact jet and 0.4 S for air supply by a jet with a coerced angle of expansion. The term S is the length of the jet trajectory to the vertex. This limitation can be represented by a system of inequalities,

$$0.3L < X_v < 0.5L; \qquad (6)$$

$$O < (h - h_v) < 0.2\ S \text{ for a compact jet} \qquad (7)$$

$$O < (h - h_v) < 0.2\ S \text{ for a jet with a forced angle of expansion} \qquad (8)$$

In the upper part of Figure 6 there is a shaded zone that meets inequality (6), and in the lower part of Figure 6, defined by curves 1 and 2, is a shaded zone that corresponds to inequality (7) and one defined by curves 1 and 3 that corresponds to inequality (8).

An analysis of the boundary curves in Figure 7 leads to the conclusion that it is possible to meet the system of inequalities 6 through 8 only under the condition

$$H > \max \text{ of } \frac{L}{2.07} \text{ or } \frac{h}{0.93}\ . \qquad (9)$$

The given nomogram indicates that the principal method of obtaining the specified range of variations for the values of X_o, X_v, and h_v at fixed values of H_o and A_o in VAV systems is to vary the discharge angle α_o of the air jet and its characteristic m. To supply air by inclined jets, air distribution of the type similar to ventilation grilles is used as a rule, which allows regulation of the aerodynamic characteristic (m = 2 to 6.3) in the process of mounting the system and gradual or seasonal regulation of the jet discharge angle (α = $\pm 30°$). The data given in [11] prove that with a fixed area of air discharge by inclined jets through a ventilation grille, the degree of regulation of the air volume can reach D = 0.5 to 0.65.

To increase the degree of regulation for this method of air distribution, a new column-mounted air distributor of VPRV type (Figure 8) has been developed [11], designed mainly for large industrial premises. This air distributor has two tiers of ventilation grilles mounted on branch pipe tees of round air ducts with an electric drive-equipped damper between them. The number of ventilation grilles that can be attached to each tier may be one, two, or four depending on their location: four at the columns, two at the wall, and one in the corner.

A bottom piece having stamped grilles is attached to the end face of the air duct in the lower tier through which

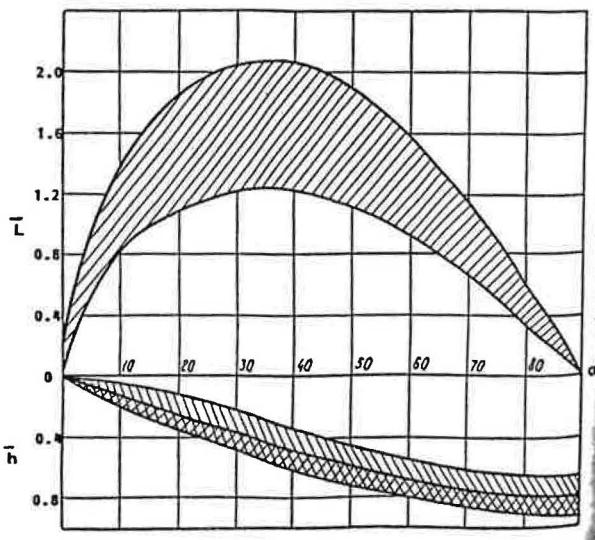

Figure 6 *Zones corresponding to the inequalities Equations 6, 7, and 8*

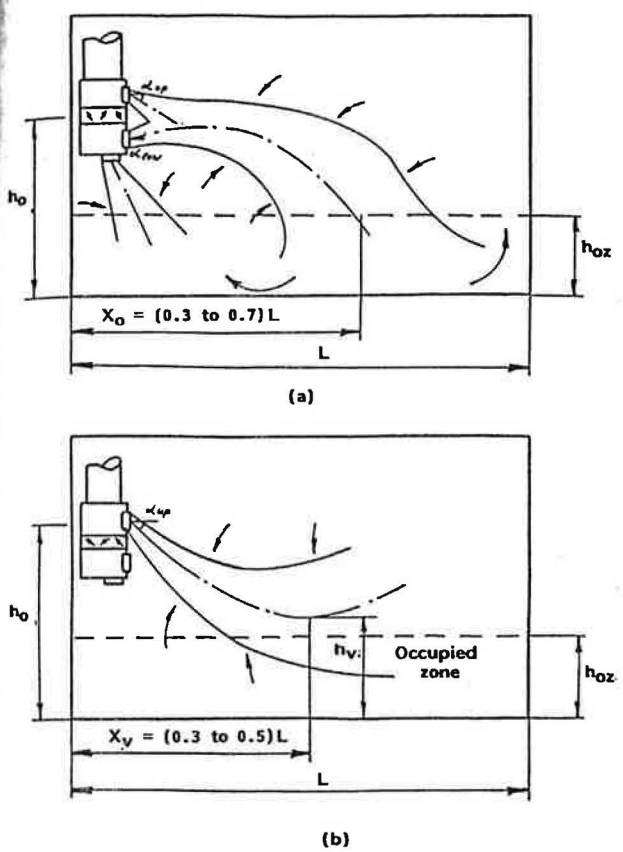

Figure 7 *Airflow patterns using the VPRV distributor with (a) a chilled jet and (b) a heated jet*

10% of the total air volume is supplied at an angle of 30° to the horizontal direction.

The vanes of the direction regulators for the grilles in the upper tier are placed at the angle of α_{up} and those for the grilles in the lower tier at the angle of α_{low} upward.

When the maximum air volume (in the warm season) of chilled air is discharged through the upper and lower tiers of grilles and also through the bottom of the air distributor blocks, the jets of heated or chilled air are only discharged through the upper tier of the grilles and their relative area is only 45% of the total area of discharge outlets. The angle of discharge of the upper jets is equal to the inclination angle of the direction regulator vanes α_{up} and is found from the condition of the air supply in the designed regime in the heating period. The angle of discharge of the lower jets, α_{low}, guarantees the required value of the total jet angle $a\Sigma$ for the designed regime in the cooling period ($t = t_7$) and is found from the expression

$$a\Sigma = arctg\,(\sin \alpha_{up} + \sin \alpha_{low})/(\cos \alpha_{up} + \cos \alpha_{low}).$$

The degree of regulation when using this type of air distributor can reach $D = 0.75$.

In industrial and office buildings of many European countries, systems of air distribution with directing jets that allow a significant depth of regulation are widely used. The principle of operation of the system, shown in Figures 9a and 9b [12, 13], is as follows: The main stream of fresh air (heated or chilled) is supplied through a small number of air outlets at a low initial velocity (2 to 4 m/s) and distributed within the space by horizontal and vertical directing jets that are discharged at a high velocity (20 to 35 m/s) from nozzles having a small diameter (10 to 100 mm). The air is delivered to these nozzles from a separate inlet unit. Studies have shown [14] that the air circulation within the space is caused mainly by the energy of the directing jets, which are

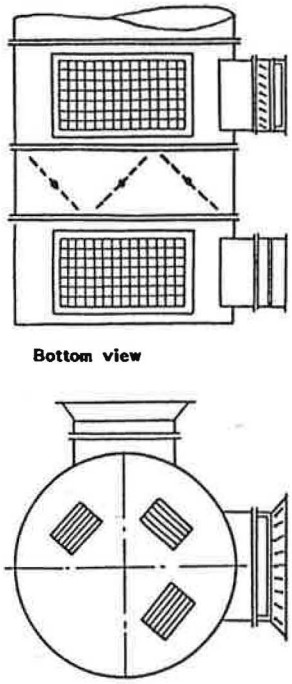

Figure 8 *Details of the VPRV distributor*

an order of magnitude greater than the energy of the main stream. That is why variations in the air volume supplied by the main stream do not affect the circulation pattern. Thus, with reduced emissions of harmful matter entering the premises, it is possible to reduce the air exchange rate to the volumes supplied through the nozzles that form the directing jets, that is, to 10% to 30% of the maximum air exchange rate. Directing jets are used in a number of other systems. For example, in the "air piston" system (Figure 9c) [15], axial-flow fans are aligned coaxially with the main

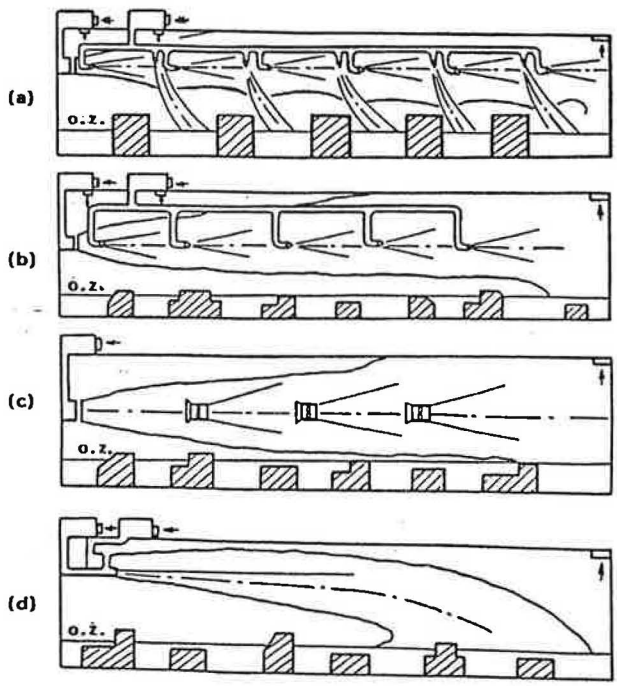

Figure 9 *Airflow patterns using a system with (a) a combination of horizontal and vertical jets, (b) horizontal jets only, (c) the air piston concept, and (d) unit with nozzles integrated into a single unit*

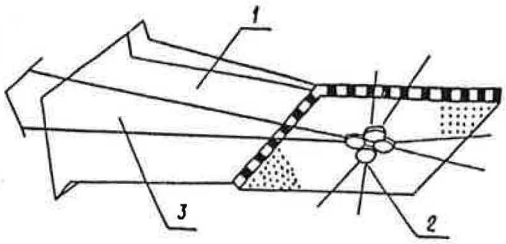

Figure 10 *Rectangular diffuser forming four jets, each of which consists of (1) the main stream, (2) a nozzle, and (3) the directing air jet*

streams instead of nozzles. The system shown in Figure 9d features an air diffusion device. Unlike the system of Figures 9a and 9b, the directing nozzles are integrated into a single unit with the inlet grille that forms the main stream. A similar concept was used in the system of Figure 10 [17]. The main streams in this system are shaped by the rectangular ceiling diffuser, which forms four jets spreading across the ceiling. Jets are discharged at high velocity through the nozzles located in the center of each ceiling when the supplied air volume is reduced. The depth of regulation for these methods of air distribution can reach D = 0.7 to 0.85.

CONCLUSION

The field of application for VAV ventilation systems in industrial buildings can be very wide. Rational strategies of operation for such systems are implemented using conservative ventilation equipment. In selecting a method of air distribution, the possibility of providing the required degree of regulation should be considered.

Supplied and extracted airflow rates can be changed gradually (with the help of guide vanes, a throttle valve, or a frequency inverter supplying power to the fan motor) or by steps, that is, by switching off the electric drives of part of the fans connected to a common header. For the degree of regulation up to 0.4, gradual regulation by means of guide vanes is preferable. For greater regulation, a combination of gradual and stepped regulation or a frequency converter should be used.

To regulate ventilation systems used in spaces with harmful gases or welding aerosol emissions, special gas analyzers or fine dust detectors are needed, which would permit their control in the occupation zone atmosphere within TLV limits.

NOMENCLATURE

A_j = cross-sectional area of jet entering occupied zone
A_{oz} = area of occupied zone
b = width of space
h = height that the diffusion is mounted
H = length of ventilated area
L = length of ventilated zone
S = length of jet trajectory to the vertex
t = temperature of outside air
t_{oz} = temperature in occupied zone
$t_{oz,max}$ = maximum target temperature in occupied zone
$t_{oz,min}$ = minimum target temperature in occupied zone
t_s = supply air temperature
q = net gain or loss of heat
Q = volume flow rate of supply air
Q_{out} = minimum airflow rate for contaminant control
x_o = distance from the air inlet to the occupied zone
ϕ_{oz} = relative humidity in the occupied zone
α_o = discharge angle

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