Helsinki University of Technology
Laboratory of Heating, Ventilating and Air Conditioning
Espoo 1999

CALCULATION METHODS FOR AIR SUPPLY DESIGN IN INDUSTRIAL FACILITIES

REPORT B60

Litterature review

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ABSTRACT

The objectives of air distribution systems for warm air heating, ventilating, and air-conditioning are to create the proper thermal environment conditions in the occupied zone (combination of temperature, humidity, and air movement), and to control vapor and air born particle concentration within the target levels set by the process requirements and/or threshold limit values based on health effects, fire and explosion prevention, or other considerations. HVAC systems designs are constrained by existing codes, standards, and guidelines, which specify some minimum requirements for the HVAC system elements, occupant's and process environmental quality and safety.

There is a variety of different methods consulting engineers use to design room air diffusion and to select and size air diffusers, such as assumption of perfect mixing , design methods employing the empirical relations determined through research, such as the air diffusion performance index (ADPI), air jet theory and computational fluid dynamics (CFD) codes.

Air supplied into the room through the various types of outlets (grills, ceiling mounted air diffusers, perforated panels etc.), is distributed by turbulent air jets. In mixing type air distribution systems, these air jets are the primary factor affecting room air motion. Numerous theoretical and experimental studies that developed a solid base for turbulent air jets theory were conducted concurrently in different countries (Germany, Sweden, Russia, U.K., USA) from the 1930's through the 1980's.

Design methods based on air jet theory allows for the prediction of extreme values of air velocities and air temperatures in the occupied zone of empty spaces. Current air jet theory techniques account for the effects of buoyancy, confinement, jets interaction. For many conditions of jet discharge, it is possible to analyze jet performance and determine:

- the angle of divergence of the jet boundary;
- the velocity patterns along heated or chilled the jet axis;
- the velocity and temperature profile at any cross section in the zone of maximum engineering importance;
- the entrainment ratios in the same zone;
- the jet trajectory;
- the vertically projected or inclined air jet throw;
- the separation point of the jet from the surface.
- influence of confinement on jet behavior;
- the multiple jet interaction.

The results of most analytical and experimental studies has been received in empty rooms and do not reflect the infuence of the obstructions on the air distribution and ventilation efficiency . Information on the influence of obstructions on room air distribution is limited.

Attempts has been made to utilize statistical data from the occupied zone conditions in order to extend predictions from extreme parameters to the rest of the occupied zone. The efficiency of the ventilation can be analyzed using zonal models.

PREWORD

This literature review is made within the Finnish research framework for industrial ventilation. The title of the research project to which the review belongs is "Development of a method for air distribution design in industrial environments". The goal of this project has been to develop an advanced but still simple calculation method for air distribution design and for predicting the conditions in the occupied zone of industrial buildings. The literature review is an important part of the foundation on which the development work is based. The review has been done in co-operation between Helsinki University of Technology and University of Illinois.

The project is funded by the Finnish Technology Development Center (TEKES) and the companies ABB, HALTON, JP-Building Engineering and VALMET. In the project management group have been Esko Tähti TAKE, Esko Virtanen TEKES, Dan Sarin ABB, Risto Kosonen HALTON, Markku Kaskimies JP-Building Engineering, Ilkka Jokioinen VALMET and Kai Sirén HUT.

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Espoo, 22 January, 1999.

Kai Siren

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1. TARGET LEVELS FOR AIR DISTRIBUTION DESIGN

1.1. Introduction

The objectives of air distribution systems for warm air heating, ventilating, and air-conditioning are to create the proper thermal environment conditions in the occupied zone (combination of temperature, humidity, and air movement), and to control vapor and air born particle concentration within the target levels set by the process requirements and/or threshold limit values based on health effects, fire and explosion prevention, or other considerations.

Compared to commercial and residential spaces, industrial environments may have other than health based constraints on indoor air quality and thermal conditions requirements: e.g., clean rooms in electronic industry allow limited number of particles in the room air; precision machinery shops put constraints on air temperature deviation throughout the space and within a time; textile shops have special requirements to room air humidity, welding and painting shops, and galvanic processes put limitation on maximum air velocities, etc.

The heating, ventilating, and air-conditioning system design process reflects the objective and subjective judgement of architects and HVAC designers. However, HVAC systems designs are constrained by existing codes, standards, and guidelines, which specify some minimum requirements for the HVAC system elements, occupant's and process environmental quality and safety.

Also, through the use of standards and codes, a compromise between energy efficiency and acceptable indoor environmental quality is attained. Standards most relevant to indoor environmental quality would include air quality standards, thermal comfort standards, ventilation standards, energy conservation standards, fire and explosion prevention codes.

Standards and guidelines typically are issued by national and international standard organizations and by professional organizations and groups of recognized experts.
Standards and guidelines may be mandatory or voluntary, but the distinction is not always clear. By distinction, codes are always mandatory and regulate activities which pose a threat to the public health and welfare. They are intended to serve as minimum requirements only and not as optimal design criteria. Because codes are legally enforceable, they generally contain provisions for noncompliance penalties. Standards are often incorporated into codes either directly or by reference.

Many codes, standards, and guidelines dealing with thermal comfort and indoor environmental quality exist both nationally and internationally. Technological requirements to thermal conditions or indoor air quality are typically set by technologists designing specific processes.

Codes generally are administered on a state or local level. Codes concerning indoor air quality and HVAC system design are subdivided into three categories: building codes, energy and fire and explosion prevention codes.

Standards and guidelines most relevant to indoor environmental quality include thermal comfort standards, ventilation standards, energy conservation standards. Relevant standard organizing bodies include: ISO (International Standard Organization), CEN (Center for Standards), EEC (European Communities), NKB (Nordic), DIN (German), ASHRAE, ANSI (U.S.A.), BSI (British), GOST and Ministry of Construction (Russian). While differences in these standards exist, most propose that a successful design consider thermal comfort, energy conservation, and IAQ requirements. International deficiencies in indoor environmental quality standards and guidelines are discussed by Zhivov et al. (1994) and Olesen et al. (1994).

1.2. Codes and standards

1.2.1. Thermal Environment in Industrial Work Spaces

1.2.1.1. Introduction

Both ASHRAE (American Society of Heating, Refrigerating and Ventilation Engineers) and ISO (International Organization for Standardization) have recently revised their standards for a comfortable thermal environment (ASHRAE 55-92,1992 and ISO 7730, 1993). The research, which forms the basis for these two standards are mainly performed under environmental conditions similar to commercial and residential buildings, with relatively low activity levels (mainly sedentary, 1.2 met), normal indoor clothing (0.5 to 1.0 clo) and a limited range of the environmental parameters. Also, there exist other standards and guidelines for evaluation of more severe and stressful thermal environments. There seems to be a gap between methods for evaluating moderate thermal environments typically in commercial and residential buildings and methods for evaluation of cold- and heat stress.

The requirements in the existing standards from ISO and ASHRAE were discussed by Olesen and Zhivov (1994) and comparison was made with similar Russian Code "Building Norms and Regulations (SNiP) -- Heating Ventilation and Air Conditioning", where the chapter on thermal comfort is based on numerous studies provided for different levels of human activity in industrial spaces. It was indicated, that the methods described in the ASHRAE and ISO standards for the acceptable overall temperature conditions, when taking into account clothing and activity may also be valid for industrial applications. However, the requirements for draft (air temperature, air speed, turbulence), vertical temperature differences at higher activity levels are different. At the higher activities in industrial spaces people are less sensitive to these non-uniformities. The acceptability of higher air speed at higher activity levels are reflected in ASHRAE and ISO standards, as well as in the reports on studies provided in Russia.

1.2.1.2. General thermal comfort

Existing methods for evaluation of the general thermal state of the body both in comfort and in heat- or cold stress, are based on an analysis of the heat balance for the human body:

$$\mathbf{S = M - W - C - R - E_{sk} - C_{res} - E_{res} - K} \tag{1.1}$$

where:
$\mathbf{S}$ = heat storage in body;
$\mathbf{M}$ = metabolic heat production;
$\mathbf{W}$ = external work;
$\mathbf{C}$ = heat loss by convection;
$\mathbf{R}$ = heat loss by radiation;
$\mathbf{E_{sk}}$ = evaporative heat loss from skin;
$\mathbf{C_{res}}$ = convective heat loss from respiration;
$\mathbf{E_{res}}$ = evaporative heat loss from respiration;
$\mathbf{K}$ = heat loss by conduction.

The factors influencing this heat balance are: activity level (metabolic rate, met or W/m^2); thermal resistance of clothing $\mathbf{I_{cl}}$ (clo or m^2 °C/W) evaporative resistance of clothing **Re**, clo (m^2 Pa/W); air temperature $\mathbf{t_a}$ (°C); mean radiant temperature $\mathbf{t_r}$ (°C); air speed $\mathbf{V_{ar}}$,(m/s); partial water vapor pressure $\mathbf{p_a}$ (Pa).

These parameters must be combined so that the thermal storage is 0 or else the working time has to be limited to avoid too much strain on the body. To provide comfort the mean skin temperature also has to be inside certain limits and the evaporative heat loss must be low. In existing standards, guidelines or handbooks, different methods are used to evaluate the general thermal state of the body in moderate environments, cold environments and hot environments; but all are based on the above heat balance and the listed factors.

Besides the general thermal state of the body, a person may find the thermal environment unacceptable or intolerable if local influences on the body from asymmetric radiation, air velocities, vertical air temperature differences or contact with hot or cold surfaces (floors, machinery, tools, etc.) are experienced.

1.2.1.3. Moderate Thermal Environments

ISO 7730 (1993) standardizes the PMV-PPD index (Fanger 1982) as the method for evaluation of moderate thermal environments. To quantify the degree of comfort the PMV-index gives a value on the 7-point ASHRAE thermal sensation scale: +3 hot, +2 warm, +1 slightly warm, 0 neutral, -1 slightly cool, -2 cool, -3 cold. An equation in the standard calculates the PMV-index based on the 6 factors (clothing, activity, air and mean radiant temperature, air speed, and humidity).

This method is also given in the ASHRAE Handbook of Fundamentals (ASHRAE 1993). Even if a PMV value of 0 is obtained, there will still be at least 5% of the occupants who will be dissatisfied with the thermal environment. The predicted percentage of dissatisfied PPD) index can be determined from the equation:

$$PPD = 100 - 95\, e^{-(0.03353\, PMV^4 + 0.2179\, PMV^2)} \tag{1.2}$$

The standard recommendation for an acceptable environment is: $-0.5 < PMV < 0.5$, $PPD < 10\%$. The use of the method is illustrated in Figure 1.1, where the optimal operative temperature (average of air and mean radiant temperatures, ASHRAE 1992, 1993, ISO 7730-1993) and the recommended temperature range are shown as a function of clothing and activity.

These data are reflected in the ASHRAE Standard 55-92 "Thermal Environment Conditions for Human Occupancy" (ASHRAE 1992).

ASHRAE Standard 55-92 also specifies a range for the operative temperature within which 90 % or more occupants find the environment thermally acceptable. For a sedentary activity (1.2 met), typical for commercial and residential buildings, and typical indoor clothing, the ASHRAE standard recommend the following operative temperature range: in winter (heating period, 0.9-1.0 clo) 20°C - 24°C, and in summer (cooling period, 0.5 clo) 23°C - 26°C.

Operative temperature for activities higher than 1.2 met (< 3 met), $\mathbf{t_a^{act}}$, can be found in Figure 1.1 (ISO 7730) or can be calculated from the operative temperature at sedentary conditions, $\mathbf{t_a^{sed}}$, °C (ASHRAE 55-92), using the following equation:

$$\mathbf{t_a^{act} = t_a^{sed} - 3(1 + I_{cl})(met - 1)} \tag{1.3}$$

Another method used to estimate the combined influence in moderate environments is the New Effective Temperature $\mathbf{ET^*}$ (ASHRAE 1993, Fobelets and Gagge 1988, Gagge et al 1971). In the comfort range it will give similar results as the PMV index. Because the PMV-index is assuming that ball evaporation from the skin is transported through the clothing to the environment, this method is not applicable for hot environments. It can be used within a range of PMV - index of $-2 < PMV < +2$, i.e. thermal environments where sweating is minimal.

1.2.1.4. Hot Environments

Tolerance limits to high temperature vary with the ability to sense temperature, lose heat by regulatory sweating, and move heat from the body core by blood flow to the skin surface, where cooling is the most effective. The maximum rate of sweating for an average man is about 0.5 g/s. Under conditions of low humidity and air movement, maximum cooling is about 675 W/m^2. However, this value does not normally occur because sweat rolls off the skin surface without evaporative cooling or is absorbed by or evaporated within clothing. A more typical cooling limit is 6 mets, 350 W/m^2, representing approximately 0.3 g/s of sweating for the average man (ASHRAE 1997).

The $\mathbf{ET^*}$ method can be used for hot environments. Another method Required Sweat Rate, $\mathbf{SW_{req}}$ (ISO 7933, 1989) is also based on the heat balance equation (1) and similar equations used in the $\mathbf{ET^*}$ method. Based on equation (1) and assuming that the heat storage is equal to 0, the necessary evaporation from the skin, $\mathbf{E_{req}}$, ensuring a heat balance, is calculated as follows:

$$\mathbf{E_{req} = M - W - C - R - E_{res} - C_{res}} \tag{1.4}$$

The maximum evaporation, $\mathbf{E^{max}}$, which can be absorbed by the environment, is estimated from the equation:

$$\mathbf{E^{max} = (psk_s - pa)/Re_T} \tag{1.5}$$

where: $\mathbf{psk_s}$ = saturated water vapor pressure at skin;
$\mathbf{pa}$ = water vapor pressure in the environment;
$\mathbf{Re_T}$ = total evaporative resistance of clothing and boundary layer.

Based on the required evaporation and the maximum evaporation it is then possible to estimate the following factors:

Required skin wetness,

$$\mathbf{w_{req} = E_{req}/E^{max}} \tag{1.6}$$

Sweating efficiency,

$$r = 1 - 0.5\, e^{-6.6\,(1 - w_{req})} \tag{1.7}$$

Required Sweat Rate,

$$\mathbf{SW_{req} = E_{req}/r} \tag{1.8}$$

These parameters are used to evaluate how stressful a given hot working environment is. Dependent on the physiological limitations for factors such as sweat rate, total sweat loss, heat storage and skin wetness, which are listed in the Table 1.1, it is possible to evaluate if a given environment is acceptable for continuous work. The method also allows to calculate an acceptable working time. Detailed equations for the calculations can be found in the standard (ISO 7933, 1989). Similar equations are given in the Chapter 8 of ASHRAE Fundamentals (ASHRAE 1993).

The relation between the operative temperature and $\mathbf{SW_{req}}$ for different combinations of activity, clothing, and $\mathbf{ET^*}$ is shown in Table 1.2.

1.2.1.5. Cold Environments

Many industrial work places are located in cold environments, like unheated cold storage, meat packing, work places located outdoors, etc. In cold environments, the clothing is the most important factor for obtaining an acceptable thermal environment. Based on the heat balance equation (1), an analytical method has been proposed by ISO, Required Clothing Insulation, $\mathbf{I_{req}}$ (ISO/TR 11079, 1993). The method calculates the insulation of the clothing necessary to keep a heat balance in a given cold environment by knowing the activity level, air- and mean radiant temperature, air speed and a relative humidity. The relationship between activity level, operative temperature and $\mathbf{I_{req}}$ is shown in Figure 1.2.

The index value may be used to select a clothing ensemble, which will provide the required insulation. The clothing insulation predicted is a minimum insulation, which does not necessarily provide comfort, but is enough for a person not to feel cold. If it is not possible to use or select a clothing ensemble with enough insulation, the method incudes a procedure for calculation of a recommended exposure time. The basis for the calculations are the physiological requirements listed in Table 1.3.

In all of the methods presented above, it is seen that the activity and clothing of the occupants are important parameters for estimation of an acceptable temperature level, influence of humidity or the cooling effect of an increased air velocity. Information on activity levels for different types of work and insulation values for work clothing is provided in Chapter 8 of ASHRAE Fundamentals (ASHRAE 1993), ASHRAE 55-92 (ASHRAE 1992), ISO 9920 (1993) and ISO 8996 (1990).

1.2.1.6. Local discomfort and individual parameters

In the previous sections several methods for evaluation of the combined influence of the thermal parameters were presented. Even if the combined influence provides a heat balance for the body as a whole, there may be influences on parts of the body from draft, radiant asymmetry, or vertical temperature differences which are not acceptable. In the following sections this will be discussed.

Air Speed

Air speed is often used in industry to provide cooling in warm environments. All of the above methods are capable of predicting the effect of an increased air speed. Depending on the conditions, too high of an air speed may cause a draft, which is an unwanted convective cooling on one or more parts of the body. A draft is probably the most common reason for complaints in air conditioned spaces with occupants at low activity levels (sedentary, standing).

For the sedentary activity (ISO 7730, ASHRAE 55-92) maximum value of mean air speed in the occupied zone to avoid draft depends upon air temperature, $\mathbf{t_a}$, and turbulence intensity, **Tu** (ratio of the standard deviation of the air speed to the mean air speed).

The impact of turbulence intensity (**Tu**) on sensation of draft has been investigated by Fanger P.O., Melikov A.K. and H. Hanzava (1988) for different methods of air distribution. The turbulence intensity shows the magnitude of the velocity fluctuations in comparison with the mean velocity.

$$Tu = \frac{\sqrt{\overline{V'^2}}}{\overline{V}} \qquad (1.9)$$

where

$$\overline{V} = \frac{1}{\tau}\int_{\tau_1}^{\tau_2} V d\tau \quad - \textit{mean velocity}$$

V'- velocity fluctuation.

Based on the studies of the different air distribution methods, they found that the turbulence intensity in ventilated spaces depends on the type of ventilation system and varies from 10 to 70 %. For air velocities ranging from 0.05 m/s to 0.40 m/s and air temperatures from 20°C to 26°C the tests were provided at three levels of the turbulent intensity: low (Tu < 12%), medium (20% < **Tu** < 35%) and high (**Tu** > 55%).

The researchers also found that periodical fluctuation of air flow is more uncomfortable than nonfluctuating air flow such as displacement ventilation systems at the same mean velocity and air temperature. The percentage of people dissatisfied due to draft as a function of air temperature, t_a ,°C, mean velocity V, m/s, and turbulence intensity, **Tu**, % was given by the equation:

$$PD = 3.143(34 - t_a)(\overline{V} - 0.05)^{0.6223} + 0.3696\overline{V}Tu(34 - t_a)(\overline{V} - 0.05)^{0.6223} \qquad (1.10)$$

ASHRAE Standard 55-1992 provides information on acceptable increase in air speed to compensate for the increase in temperature for sedentary, light and higher levels of human activity; however, precise relationships are not available from this standard. Air speeds as high as 1.5 m/s are acceptable if they do not cause problems unrelated to thermal comfort (e.g. blowing of papers or shield gas at semiautomatic welding).

Using the PMV equation (ISO 7730) the combinations of air temperature and air speed which will provide a thermal neutrality, i.e. PMV = 0 are presented in Figure 1.3 for different activity levels. The relationships are shown for heating (winter) period (I = 1.0 clo) and for cooling (summer) period (I = 0.5 clo).

Optimal and acceptable air temperature, relative humidity and air speed in the occupied zone of industrial buildings in Russia are restricted by the Building Code 2.04.05-91 (SNiP 1991) and are presented in Table 1.4. In column 7 and 8 of Table 1.4, temperatures are presented as fractions with numerator for regions with the design cooling season outdoor air temperature lower than 25°C, and the denominator for regions with the design cooling season outdoor air temperature higher than 25°C.

For regions with the design cooling season outdoor air temperature of 25°C and higher, the air temperatures in columns 7 and 8 can be within 4°C higher than the design outdoors cooling season air temperature, but less than those presented in columns 7 and 8 of the table. For regions with the design cooling season outdoor air temperature of 18°C and lower, the excessive temperature equal to 4°C in column 6 of the table can be replaced by 6°C. For regions with the design cooling season outdoor air temperature equal to **t**, °C, maximum air speed on the permanent and the temporarily work places stated in column 9 can be increased as follows by 0.1 m/s for each °C of the temperature difference (**t** - 28), but less, than by 0.3 m/s, when **t** > 28°C.

Teterevnikov, Kuksinskaya and Pavlukhin (1976) conducted study on thermal comfort in industrial spaces. Their data on combinations of air temperature and air speed for light activity - I (70 W/m²), medium activity - IIa (93 Wt/m²) and IIb (116 W/m²), and high activity - III (174 W/m²) are presented in Figure 1.4.

The above data show that with the activity level increase, people can accept higher air speed without a sensation of draft.

Radiant Temperature Asymmetry

Both in ASHRAE 55-92 and in ISO 7730 Standard, the limitations for the radiant asymmetry, < 10°C from cold walls and < 5°C from warm ceilings, are recommended. These data are based on testing with sedentary persons in 0.6 clo clothing (Fanger et al. 1985). Similar data are available for warm walls and cold ceilings. In industrial work places it is mainly the radiant asymmetry from overhead radiant heaters or a hot roof which may cause a problem. The values established for sedentary persons are too conservative for the higher activity levels and higher ceilings in industry.

A study by Langkilde et al. (1985) shows that significantly higher radiant asymmetry are acceptable. Based on criteria similar to the above requirements of less than 5% dissatisfied, the recommended asymmetry limit is 10 -14°C . In these studies there were no differences between seated and standing persons. Also the influence of wearing a helmet was insignificant. In the previous studies simulating the conditions in residential and commercial buildings, the distance between the heated ceiling and the head of the subjects was only 1 meter. While the distance between the infrared radiant heaters and the head was 4 meters in the experiments reported by Langkilde et al. For the same asymmetry level in the two types of experiments the difference in the radiant temperature level at feet and head height will be much larger when the distance from the person to the heaters is small (1 m). In industrial work places where the heaters normally are mounted more than 4 meters from the occupants a larger asymmetry can be accepted before the temperature difference felt between head and feet causes discomfort. That is also the reason why no difference was found between seated and standing persons.

In some work places very extreme radiation may occur from production processes (glass oven, hot metals etc.) and there may be a risk for burns on unprotected skin.In some cold work places there may also exist radiant asymmetry from cold ceilings. Normally these problems are limited because the temperature on the cold surfaces are limited by the dew point temperature. Also people are less sensitive to cold radiation from above than to warm radiation.

Vertical and Horizontal Air Temperature Differences

Another comfort parameter, which is described in ASHRAE 55-92 and ISO 7730, is the temperature difference between head and feet. The recommended limit is 3°C. The basis for this value are studies with sedentary persons (Olesen et al. 1979). This can not directly be used in industries with higher activities. Besides, it is not the air temperature difference alone, causing discomfort. It is really the combined effect of air temperature, air velocity and radiant temperature at head level compared to feet level, which are sensed by the occupants. These data for industrial environments is only available from SNiP (1991), where the vertical temperature difference in the occupied zone is limited by 4°C and horizontal temperature difference in the occupied zone can not exceed 4°C at light activity, 5°C at medium activity and 6°C at high activity.

1.2.1.7. Contact With Hot or Cold Surfaces

In ASHRAE 55-92 and ISO 7730 it is recommended that the floor temperature should be between 19°C and 29 °C. These values are also based on studies with sedentary or standing persons (Olesen 1977) using normal indoor footwear and may not be directly used in industrial environments. Discomfort and accidents may be caused by people touching too hot or too cold surfaces (machinery or hand tools etc.).

1.2.1.8. Spot Cooling

When the work places are located near the sources of radiant heat, which can not be entirely controlled by radiation shielding, ASHRAE Applications Handbook (ASHRAE 1995) recommends spot cooling. The air temperature at the work place is limited to the range of 20 to 30°C depending on the heat load and the air speed as in Table 1.5.

Combinations of air temperatures and air speed in the jet used for spot cooling are recommended also by SNiP (1991) and are presented in Table 1.6. When the air temperature in the occupied zone is different (lower/higher) from listed in Table 1.4, air temperature in the jet used for spot cooling can be increased/decreased, respectively, to compare with the one listed in Table 1.6 on 0.4°C for each degree of the temperature difference. It may not be lower than 16°C.

The temperature of the radiating surface should be averaged during the period of radiation. When the radiation effect lasts more than 15 min, air temperature in the jet can be 2°C higher or lower than one listed in Table 1.6.

1.2.1.9. Relation between air distribution and thermal comfort

Studies conducted at Kansas State University has led to the air distribution criteria based on effective draft temperature equation proposed by Rydberg and Norback (1949), subjective draft data of Houghten (1938), and modified by H.Straub in discussion of a paper by Koestel and Tuve (1955):

$$\theta = (t_x - \overline{t_{oz}}) - 0.07(V_x - 30) \tag{1.11}$$

Based on this data, the criteria representing conditions at which 80% of the occupants with a sedentary or light activity report themselves comfortable was developed and defined as the Air Diffusion Performance Index (ADPI). ADPI indicates the percentage of locations uniformly distributed throughout the occupied zone that meet specifications on effective draft temperature ($-3 < \theta < 2$F) and air velocity ($V < 0.35$ m/s). If the ADPI is maximum (approaching 100%), the most desirable conditions are achieved. ADPI is only a measure in cooling mode conditions and can not be applied for the heating conditions.

Analogous to ADPI criteria for evaluating Indoor Air Quality and Thermal Comfort - sanitary-hygienic efficiency index (SHEI), ϵ, suggested by Gunes (1974) is used in Russia;

$$\epsilon = A_{ac}/A_{o.z} \tag{1.12}$$

where, A_{ac} = area of the part of the occupied zone with acceptable IAQ and Thermal Comfort; $A_{o.z}$ = floor area of the occupied zone.

Current diffuser air jet theory makes it possible to predict maximum values of air temperature differential, t_x, and air velocity at the point of the jet entering the occupied zone. HVAC systems become bulky and expensive when they are designed to maintain maximum air velocities and temperature difference in the 100% of the occupied zone within a range, prescribed by the thermal comfort standards.

Taking in consideration the relatively small area of the jet cross-section with abnormal parameters, Building Norms and Regulations (SNiP 1991) suggests the following correlation between the maximum air velocity, V_x, the maximum/minimum air temperature, $\mathbf{t_x}$, in the jet entering the occupied zone (or in the reverse flow), normative values of air speed, $\mathbf{V_n}$, and temperatures, $\mathbf{t_n}$:

$$\mathbf{V_x = N\ V_n} \tag{1.13}$$

For heated air supply; $\mathbf{t_x = t_n + \Delta t_1}$; (1.14)

For chilled air supply; $\mathbf{t_x = t_n - \Delta t_2}$, (1.15)

where: $\mathbf{N}$, $\mathbf{\Delta t_1}$ and $\mathbf{\Delta t_2}$ are listed in the Tables 1.7 and 1.8.

To better show how these are used, assume you were designing a space for acceptable thermal conditions with a wall mounted grill air supply where the maximum air speed in the cross-section of the jet entering the occupied zone permitted can be 1.4 times higher (Table 1.7) for people located in reverse flow) compared with normative value 0.3 m/s (60 ft/min). Thus, according to jet theory, the air supply velocity (assuming same grille size) and the airflow rate can be increased 1.4 times. Such approach to the HVAC system design makes it possible to supply higher airflow rates in ventilated space with reduced costs, while maintaining air velocities and temperatures in the most occupied zone area within a required range.

1.2.2. Standards for Indoor Air Contaminant Control

The concentration of harmful gases, vapors and airborne particles in the occupied zone are limited by prescribed by National Occupational Safety and Health Administrations (OSHA) Permissible Exposure Levels (PELs) or by Threshold Limit Values (TLV) - the time-weighted average concentration for normal 8-hour workday and a 40-hour workweek, to which nearly all workers may be repeatedly exposed, day after day, without adverse effect, or by in-house TLV. These values (that may vary from country to country) are used for estimating quantity of outdoor air to be supplied by ventilation systems. It is desirable to set design objectives below statutory PELs and TLVs because of variation in sensitivity and because acceptable limits can be lowed. The sample data from the USA/Canada, Germany, Sweden, and Russia are presented in Table

1.9.

Uneven air distribution within occupied zone and specially in rooms with large obstructions can cause poorly ventilated zones. Contaminant distribution in the occupied zone also depends upon the method of air supplied used and the room size and configuration. Zhivov (1983) suggested that in buildings with sources of harmful contaminants, air velocity in the occupied zone should exceed 20 ft/min (0.1 m/s) to avoid poorly ventilated zones.

Uneven gas contaminant distribution in the occupied zone can be described by the following relation between the maximum permissible (target) contaminant concentration $C_{o.z.}^{max}$ and the average contaminant concentration in the occupied zone (Zhivov et al., 1996):

$$\overline{C_{o.z.}} = C_{o.z.}^{\max} - 2\,\sigma_c \tag{1.16}$$

where, σ_c = standard deviation of contaminant concentration distribution in the occupied zone.

When the airflow rate supplied into the occupied zone is designed to limit the average contaminant concentration in the occupied zone by TLV, the maximum contaminant concentration in the occupied zone will exceed the permissible concentration level by $2\sigma_c$. Thus, according to Zhivov et al. (1996), the supply airflow rate should be dimensioned based on the maximum contaminant concentration in the occupied zone rather, than its average value. However, the information on contaminant distribution within the occupied zone is limited and thus, further research is needed to study the distribution of contaminants in buildings with different methods of air distribution and the effects of obstructions.

Table 1.1. Reference Value for the different criteria of thermal stress and strain.

Criteria	Non-acclimatized subjects		Acclimatized subjects	
	Warning	Danger	Warning	Danger
Maximum skin wettedness w_{max}	0.85	0.85	1.0	1.0
Maximum sweat rate				
Rest:				
$M < 65$ W/m^2 SW_{max}, W/m^2	100	150	200	300
g/h	260	390	520	780
Work:				
$M > 65$ W/m^2 SW_{max}, W/m^2	200	250	300	400
g/h	520	650	780	1040
Maximum heat storage Q_{max}, W.h/m^2	50	60	50	60
Maximum water loss D_{max}, W.h/m^2	1000	1250	1500	2000
g	2600	3250	3900	5200

Table 1.2. Required Sweat Rate index $\mathbf{SW_{req}}$, W/m^2 and wettednes ($\mathbf{w_{req}}$) as a function of clothing, temperature, air speed and humidity at the activity level **M** equal to 70 W/m^2.

Clothing I_{cl}, clo	Relative humidity, %	Operative Temperature, t_o, °C	Air Velocity V, m/s				
			< 0.1	0.2	0.5	1.0	2.0
0.5	20	25	10(.03)	8(.02)	2(.01)		
		30	37(.13)	36(.12)	33(.09)	29(.06)	24(.04)
		35	65(.24)	64(.22)	64(.18)	63(.14)	62(.11)
		40	93(.37)	94(.34)	95(.29)	98(.24)	101(.20)
		50	169(.78)	165(.72)	165(.62)	171(.52)	182(.45)
		60	*(1.00)	*(1.00)	*(1.00)	*(1.00)	*(1.00)
	50	25	10(.04)	8(.03)	2(.01)		
		30	37(.17)	35(.15)	33(.12)	29(.08)	24(.05)
		35	65(.37)	65(.34)	64(.28)	63(.22)	62(.17)
		40	102(.74)	99(.69)	98(.57)	99(.47)	102(.39)
	80	30	37(.26)	36(.23)	33(.18)	29(.12)	24(.08)
		35	76(.82)	71(.75)	66(.61)	64(.48)	63(.38)

Table 1.3. Recommended physiological criteria for determination of $\mathbf{I_{req}}$, DLE and local cooling.

General cooling	Minimal $\mathbf{I_{req}}$	Neutral $\mathbf{I_{req}}$
$\mathbf{I_{req}}$ $\mathbf{t_{sk}}$, °C **w** (n.d.)	"high strain" $\mathbf{t_{sk}} = 30$ 0.06	"low strain" $\mathbf{t_{sk} = 35.7 - 0.285\ M}$ $w = \frac{(E_d + E_s\ )RT}{p_{sks} - p_a} \leq 0.25$
DLE $\mathbf{Q_{lim}}$ (W.h/m^2)	-4.0	-4.0

Table 1.4. Optimal and acceptable thermal comfort parameters for industrial spaces (SNiP 1991)

Season	Activity level	Optimum conditions			Acceptable conditions				
		Temperature, °C	Air speed, m/s	Relative humidity, %	Air temperature, °C			Air speed, m/s, less than	Relative humidity, %, less than
					At permanent and temporarily work places	At permanent work places[1]	At temporarily work places[1]	At permanent and temporarily work places	
Cooling	Light				4°C higher than the design outdoors temperature, but less, than those in columns 7 and 8 of this table				
	1a	23-25	0.1			28/31	30/32	0.2	
	1b	22	0.2			28/31	30/32	0.3	
	Moderate			40-60					75
	IIa	21-23	0.3			27/31	29/31	0,4	
	IIb	20-22	0.3			27/30	29/31	0,5	
	Heavy - III	18-20	0.4			26/29	28/30	0.6	
Heating and Mild Weather	Light								
	1a	22-24	0.1			21-25	18-26	0.1	
	1b	21-23	0.1			20-24	17-25	0.2	
	Moderate			40-60	-				75
	IIa	18-20	0.2			17-23	15-24	0.3	
	IIb	17-19	0.2			15-21	13-23	0.4	
	Heavy - III	16-18	0.3			13-19	12-20	0.5	

[1] Values in the numerator are for regions with the design cooling season outdoor air temperature lower than 25°C. Values in the denominator are for regions with the design cooling season outdoor air temperature higher than 25°C.

Table 1.5. Acceptable air speed at work place.

Activity level	Air speed, m/s
Continuous Exposure	
Air Conditioned space	0.25-0.4
Fixed work station, general ventilation or spot cooling:	
Sitting	0.4-0.6
Standing	0.5-1.0
Intermittent exposure, spot cooling or relief stations	
Light heat loads and activity	5 to 10
Moderate heat loads and activity	10 to 15
High heat loads and activity	15 to 20

Table 1.6. Recommended spot cooling air speed and air temperature.

Activity level	Air velocity in the jet averaged on 1 m^2 area of the work place, m/s	Average air temperature in the jet cross section, °C, on the work place, at the density of heat flux, W/m^2				
		140-350	700	1400	2100	2800
Light - I	1	28	24	21	16	-
	2	-	28	26	24	20
	3	-	-	28	26	24
	3.5	-	-	-	27	25
Moderate - II	1	27	22	-	-	-
	2	28	24	21	16	-
	3	-	27	24	21	18
	3.5	-	28	25	22	19
Heavy - III	2	25	19	16	-	-
	3	26	22	20	18	17
	3.5	-	23	22	20	19

Table 1.7. Coefficient of transition from normative air speed in the occupied zone to the maximum air velocity in the jet (SNiP 1991)

Comfort conditions	People location	N	
		light activity	medium and high activity
Acceptable	Within air jet:		
	- in the core zone	1	1
	- in the main zone	1.4	1.8
	Outside the jet	1.6	2
	In the reverse flow	1.4	1.8
Optimal	Within air jet:		
	- in the core zone	1	1
	- in the main zone	1.2	1.2
	Outside the jet or in the reverse flow	1.2	1.2

Table 1.8. Acceptable deviation of air temperature in the air jet from the normative air temperature in the occupied zone (SNiP 1991)

Comfort conditions	Room type	Acceptable temperature deviation			
		heating mode		cooling mode	
		People location			
		within air jet	outside air jet	within air jet	outside air jet
Acceptable	Residential and commercial:				
	Δt_1	3	3.5	-	-
	Δt_2	-	-	1.5	2
	Industrial:				
	Δt_1	5	6	-	-
	Δt_2	-	-	2.0	2.5
Optimal	Any, except those with a special process demands				
	Δt_1	1	1.5	-	-
	Δt_2	-	-	1	1

Table 1.9. Threshold Limit Value (TLV) for some typical fumes, particles and gases (as of January '91)

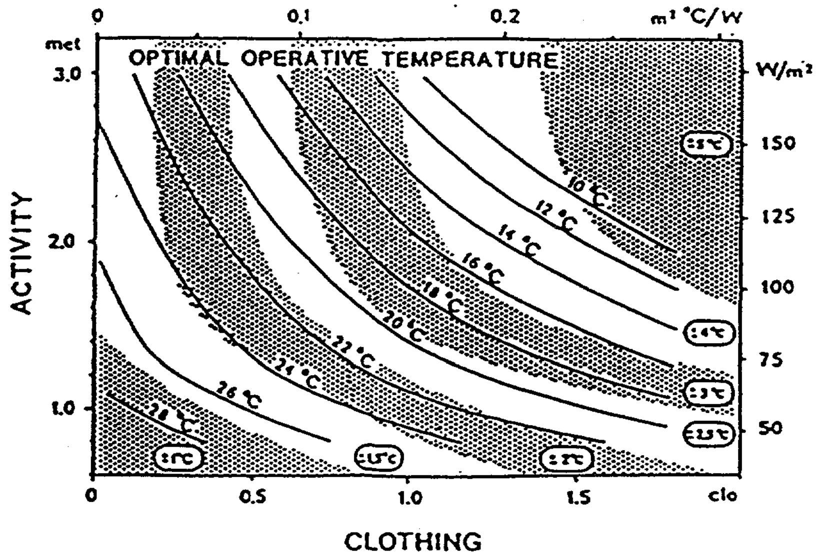


Figure 1.1. The optimal operative (corresponding to PMV = 0) as a function of activity and clothing. The shaded areas indicate the comfort range ± Δt around the optimal temperature inside which - 0.5 < PMV < + 0.5. The relative air velocity caused by body movement is estimated to be zero for M < 1 met and V_{ar} = 0.3 (M - 1) for M > 1 met. Relative humidity = 50 % (Reproduced from Olesen et al. 1994).

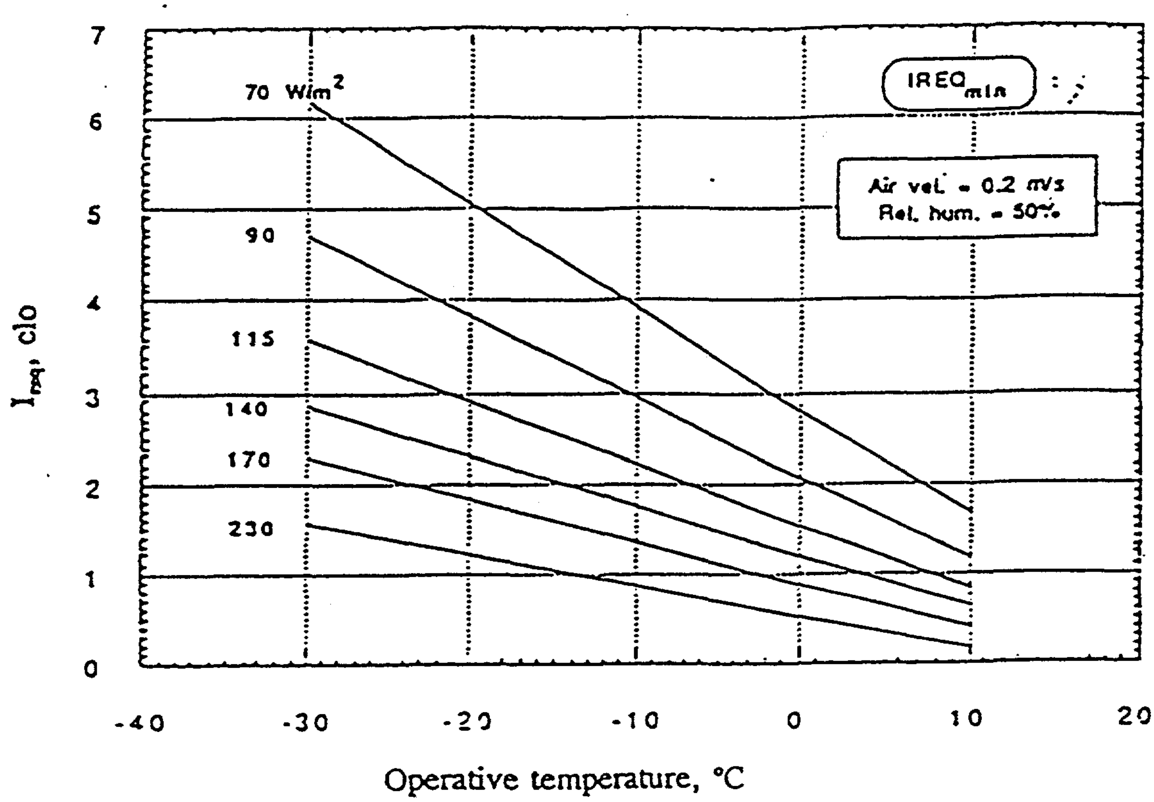


Figure 1.2. $I_{req.min}$ as a function of ambient operative temperature at six levels of metabolic heat production (Reproduced from Olesen et al. 1994).

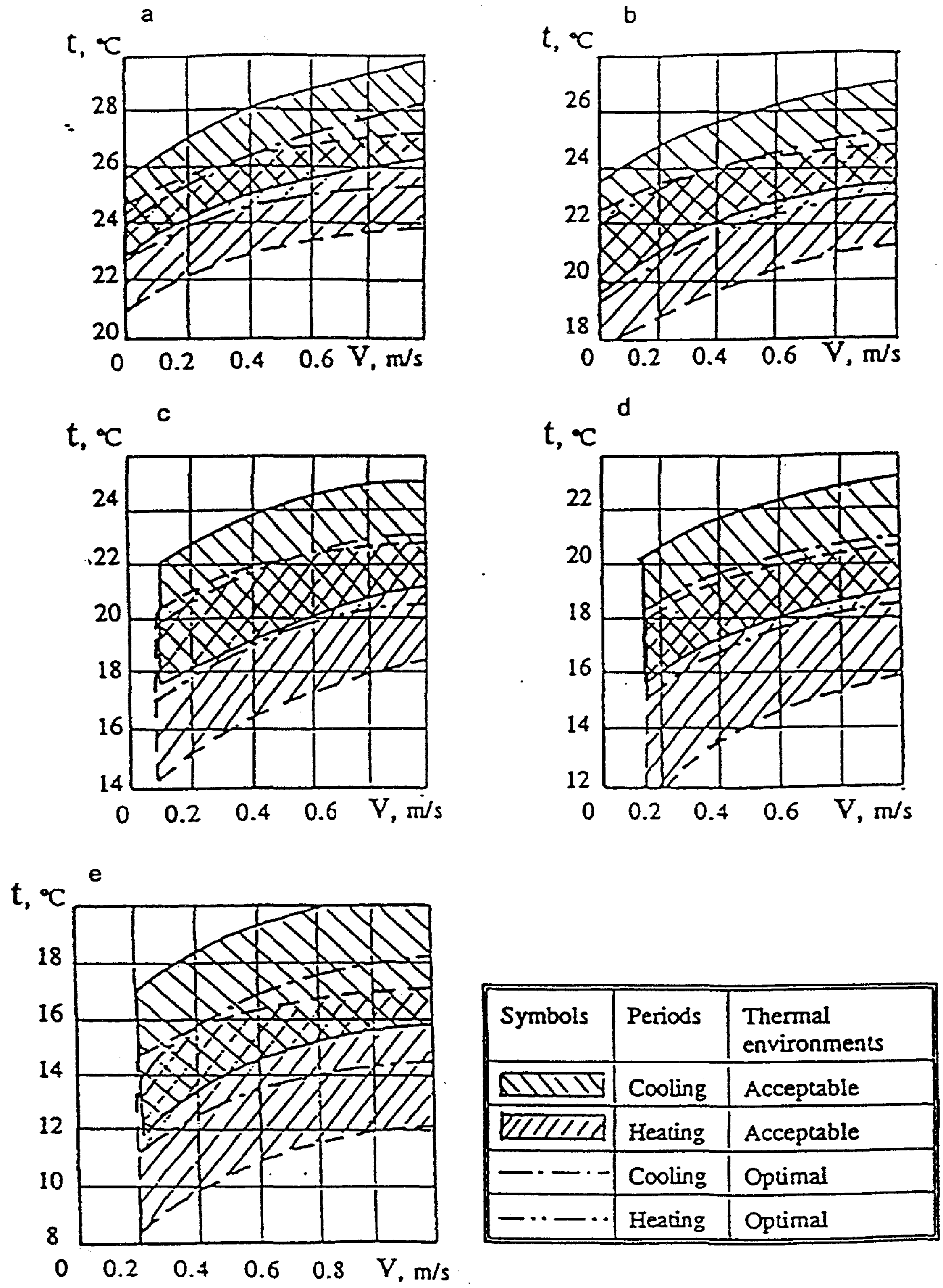


Figure 1.3. Optimal and acceptable ranges (ISO 1993) of air temperature and air speed in the occupied zone for different levels of human activity: a - sedentary (70 W/m^2); b - light (93 W/m^2); c - medium (116 W/m^2); d - medium (140 W/m^2); and high (174 W/m^2). Winter clothing = 1.0 clo. Summer clothing = 0.5 clo (Reproduced from Olesen et al. 1994).

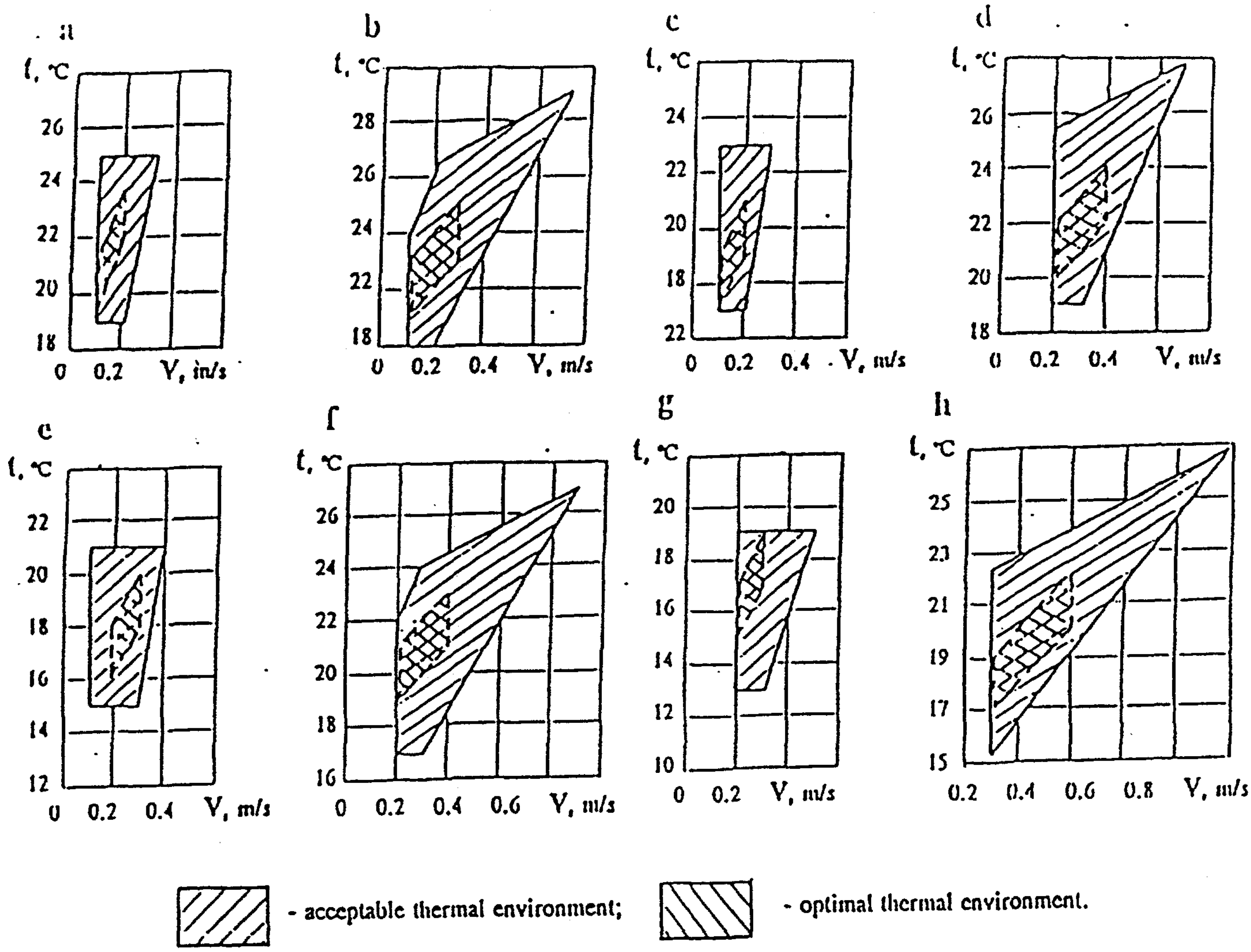


Figure 1.4. Optimal and acceptable ranges (Teterevnikov et al. 1974) of air temperature and air speed in the occupied zone for different levels of human activity:
a - light I (heating period);
b - light I (cooling period);
c - medium IIa (heating period);
d - medium IIa (cooling period);
e - medium IIb (heating period);
f - medium IIb (cooling period);
g - high III (heating period); and
h - high III (cooling period). (Reproduced from Olesen et al. 1994).

2. AIR DISTRIBUTION DESIGN METHODS IN LARGE INDUSTRIAL HALLS

2.1. Air Supply Methods

There are different methods of air supply into industrial spaces. To classify these methods we can initially separate air supply provided by Natural Ventilation and Mechanical Ventilation systems. Though Natural Ventilation systems driven by buoyancy forces and/or by wind effect, are still widely used in industrial spaces (especially in hot premises in cold and moderate climates), they are inefficient in large buildings, may cause drafts and can not solve air pollution problem. Thus, most of the ventilation systems in industrial spaces are either Mechanical or a combination of Mechanical supply with Natural exhaust systems.

The most commonly used methods for air supply into industrial spaces can be classified as:

- mixing
- displacement
- unidirectional airflow or "piston" flow
- spiral vortex
- localized

Mixing -type air distribution. In mixing systems air is normally supplied into the space at velocities much greater than those acceptable in the occupied zone. Supply air temperature can be above, below, or equal to the air temperature in the occupied zone, depending on the heating/cooling load. The supply air diffuser jet mixes with the room air by entrainment, which reduces air velocities and equalizes the air temperature. The occupied zone is either ventilated directly by the air jet or by the reverse flow created by the jet. Properly selected and designed mixing air distribution creates relatively uniform air velocity, temperature, humidity, and air quality conditions in the occupied zone.

In industrial spaces with mixing type air distribution, air can be supplied with:

- horizontal air jets (attached or not attached to the ceiling) or as termed "concentrated air supply" with the occupied zone ventilation by the reverse air flow (Figure 2.1);
- horizontal attached to the ceiling compact or linear jets supplied through the wall-mounted grills (Figure 2.2)
- horizontal ("concentrated") air jets assisted with an additional system of vertical and/or horizontal directing jets (Figure 2.3)
- inclined air jets through the grilles and nozzles installed on walls and/or columns at the height of 3m to 6m (Figure 2.4) ;
- radial, conical or compact air jets through the ceiling type air diffusers installed in or close to the ceiling (Figure 2.5a,b,c) or on the vertical duct drops (Figure 2.5d);
- radial or linear jets through perforated surfaces of horizontal round or rectangular ducts (Figure 2.6);

Displacement ventilation systems. Conditioned air with a temperature slightly lower than the desired room air temperature in the occupied is supplied from air outlets at low air velocities - 0.5 m/s (100 ft/min) or less (Jackman 1991, Skåret 1985, Skistad 1994, Shilkrot and Zhivov

1996). Under the influence of buoyancy forces cold air spreads along the floor, and floods the lower zone of the room. The air close to the heat source is heated and rises upward as a convective airstream. In the upper zone this stream spreads along the ceiling. The lower part of the convective stream induces the cold air of the lower zone of the room, and the upper part of the convective airstream induces the heated air of the upper zone of the room. The height of the lower zone depends on the air volume discharged through the panels into the occupied zone and on the amounts of convective heat discharged by the sources (Figure 2.7).

Typically the outlets are located at or near the floor level, and the supply air is directly introduced to the occupied zone. In some applications of displacement ventilation (in computer rooms or in hot industrial buildings) air can be supplied into the occupied zone through a false floor. In other applications supply air outlets can be located above the occupied zone. Returns are located at or close to the ceiling/roof through which air is exhausted from the room.

During the past twenty years, displacement ventilation has been common in Scandinavia and is becoming popular in the other European countries. Displacement ventilation is preferable when contaminants are released in combination with surplus heat and contaminated air is warmer and/or lighter than the surrounding air. Displacement ventilation design guidelines and limitations are described by Skistad (1994), Laurikainen (1995) and Zhivov et al. (1997).

<u>Unidirectional airflow ventilation</u>. Air is either supplied from the ceiling and exhausted through the floor or vice versa (Figure 2.8a), or supplied through the wall and exhausted through returns located on the opposite wall (Figure 2.8b). The outlets are uniformly distributed over the ceiling, floor, or wall to provide a low turbulent "plug" -type flow across the entire room. This type of system is mainly used for ventilating clean rooms, in which the main objective is to remove contaminant particles within the room, or in halls with high heat and/or contaminant loads with double flooring or floor pedestals.

<u>Spiral vortex flow</u> air distribution can be used to localize air contaminants in certain room areas and to evacuate the polluted air from those areas. A spiral vortex in a space can be formed by supplying air through the vertical supply ducts located along a closed contour (preferably along the walls) and thus generating a vertical vortex. An exhaust outlet can be located in the ceiling near the center of the rotational flow. Such combination of air supply and exhaust systems allows for concentrating contaminants in the vortex core and to transport them to the exhaust outlet along the core axis (Figure 2.9). Low pressure in the vortex core allows for collecting contaminants and for preventing their diffusion to the clean space (Kuz'mina et al., 1986; Nagasawa et al., 1990)

<u>Localized ventilation</u>. Air is supplied locally for occupied regions or a few permanent working places (Figure 2.10). Conditioned air is supplied towards the breathing zone of the occupants to create zones with comfortable conditions and/or to reduce the concentration of pollutants. These zones may have air 5 to 10 times cleaner than the surrounding air. In local ventilation systems air is supplied either (a) through the nozzles or grilles (e.g., for spot cooling); specially designed low velocity/low turbulence devices (Kristensson et al. 1993) or (b) through perforated panels suspended on the vertical duct drops and positioned close to the work place (VDI, 1994), or (c) through combination of vertical perforated supply air panels creating oasis around the working place and also acting as screens to separate the working place from the rest of the environment (Repus, ABB 1990, AIR-IX 1987).

2.2. Current Design Methods

There are only few documents and papers that guide consulting engineers in their selection of the method of air supply to be used for their specific applications (ASHRAE 1995, Designers Guide 1992, VDI 1994, AIR-IX 1987, ABB 1990, Zhivov 1992). Among the most important criteria that are used for selection are:

- room floor area and height;
- technological process and size of process equipment used, space obstruction with this equipment;
- number and type (permanent or temporarily) of working places, their location;
- type and amount of contaminants released into space, heating/cooling loads, air change rates;
- type of HVAC system used (Variable Air Volume or Constant Air Volume); and
- the data on ventilation effectiveness characterizing different air supply method.

Some guidelines on air distribution method selection can be found in ASHRAE Handbook, Applications (1995) and in VDI (1994). Table 2.1 reproduced from ASHRAE (1995) illustrates the air supply method selection principles.

In spite of variety of air supply methods, that are currently available to consulting engineers, more that 90% of air is supplied into industrial halls with mixing type ventilation systems. Almost all current ventilation systems in Russia and in other former USSR republics as well as in the USA are mixing type.

During the past twenty years, displacement ventilation has been common in Scandinavia. In 1989, it was estimated that displacement ventilation accounted for 50% of the Scandinavian market share in industrial applications. Displacement ventilation is becoming more popular in Germany, United Kingdom, Switzerland and Central European Countries. The analysis of internationally available technical information on displacement ventilation is available from (Zhivov et al. 1995). Limitations and design guidelines for displacement ventilation can be found from Stratos (1986), Skistad (1994), Laurikainen (1995), Zhivov et al. (1997). Other types of air supply systems have a very limited usage either due to their limited application (e.g., unidirectional flow, localized, or because of the limited information on their design procedure.

The following discussion will be limited only to design principles of mixing-type air distribution methods.

There is a variety of different methods consulting engineers use to design room air diffusion and to select and size air diffusers.

- Perfect mixing is commonly assumed meaning air distribution is uniform within a room. This approach can be acceptable in situations with the temporarily occupied places. However, with this approach there is no assurance, that the fresh air actually reaches the occupants nor there is a method of predicting thermal and air quality parameters vary throughout realistic rooms.

- Design methods employing the empirical relations determined through research. The data for these relations can be obtained in field experiments, reduced and full scale tests. The air diffusion performance index (ADPI) is one example of this technique. It utilizes only air velocity and effective draft temperature. Effectively, ADPI is a measure of cooling conditions with a sedentary or light activity. There exist many more of these types of indexes outlined for design and analysis of occupied regions; unfortunately, extrapolation of these techniques beyond their original data sets is questionable.

- Computational fluid dynamics (CFD) codes predict temperature, velocity and contaminant distributions. Boundary descriptions and computational requirements still limit general acceptance in the design community, although some specialty system designs have benefitted from CFD simulations. With CFD, it is theoretically possible to model any configuration assuming that time is not a constraint. However, it is difficult to efficiently and accurately prescribe the boundary conditions. At this time, CFD has been recognized as having a great potential but has not evolved into a design tool. Unique ventilation systems may currently utilize CFD when the design process involves gaseous or contaminate distributions of materials that have not been considered and where experimental data is absent. Computational fluid dynamic codes require expertise, computational availability and boundary conditions.

- Air jet theory is a well-practiced technique for analyzing and designing human environments. Air jet theory differs from the uniform mixing assumption used in the traditional design technique by dividing the room into a jet region and occupied or mixing zone as opposed to one region. Design methods based on air jet theory allows for the prediction of extreme values of air velocities and air temperatures in the occupied zone of empty spaces. Current air jet theory techniques account for the effects of buoyancy, confinement, jets interaction.

Air Jet Theory. Most national guidelines such as ASHRAE Fundamentals (1993) [USA], CIBSE Guide Volume B (1988) [UK], AICVF Aeraulique (1991) [France], Designers Guide (1992) [Russia], LVIS (1996) [Finland], Taschenbuch für Heizung and Klimatechnik (Schramek 1986) [Germany], Skistad (1995) [Norway] utilize air jet theory for room air distribution design. Numerous theoretical and experimental studies developed a solid base for turbulent air jets theory were conducted concurrently in different countries from the 1940's through the 1970's.

For many conditions of jet discharge, it is possible to analyze jet performance and determine:

- the angle of divergence of the jet boundary;
- the velocity patterns along the jet axis;
- the velocity profile at any cross section in the zone of maximum engineering importance;
- the entrainment ratios in the same zone;
- influence of confinement on jet behavior;
- the multiple jet interaction.

When the temperature of introduced air is different from the room air temperature, the behavior of the diffuser air jet is affected by the thermal buoyancy due to air density difference. A non-isothermal jet behavior is determined by the Archimedes number (Baturin, 1972):

$$Ar = \frac{gL_o(t_o - t_s)}{V_o^2 T_s}$$

where: **g** = the gravitational acceleration rate; $\mathbf{L_o}$ = the length scale of the diffuser outlet which is equal to the hydraulic diameter of the outlet, $\mathbf{t_o}$ = initial temperature of the jet; $\mathbf{t_s}$ = temperature of the surrounding air, $\mathbf{V_s}$ = initial air velocity of the jet, $\mathbf{T_s}$ = room air temperature, K.

If $Ar_0 \leq 0.1\, K_2^2/K_1(\sqrt{A_0}/X)^2$, the buoyancy forces do not significantly effect the jet behavior: its trajectory, point of jet's separation from the surface, velocity, etc. Such jet is called slightly nonisothermal jet (Grimitlyn, 1993).

In air jets with greater Ar_0 values buoyancy forces cause transformation of velocity and temperature profiles (Zhivov 1993) that can be neglected for the practical values of supply velocities and temperatures in mixing type air distribution systems. However, transformation of supply jet has significant effect on room air distribution at supply air velocities and temperatures with displacement ventilation systems ($V_0 < 0.25$ m/s, $\Delta t_0 = 0.5$ to 3°C).

The currently available theories for non-isothermal jets allow to predict:

- the velocity patterns along the heated or chilled air jet axis;
- the temperature profile at any cross section in the zone of maximum engineering importance;
- the jet trajectory;
- the vertically projected or inclined air jet throw;
- the separation point of the jet from the surface.

However, most of the information on ventilation jets is scattered throughout different sources. The only systematic design procedure based on air jet theory is practically available from Russian sources (Designers Guide 1992 and a number of guidelines for specific air diffusers selection, which are available only to Russian consulting companies in blue prints). This design algorithm for different methods of air supply was utilized (Pozin, 1993) in the computer program "PRIVOZ" (SUPPLY AIR).
Another computer model based on air jet equations is available from Halton (Livchak, 1993)

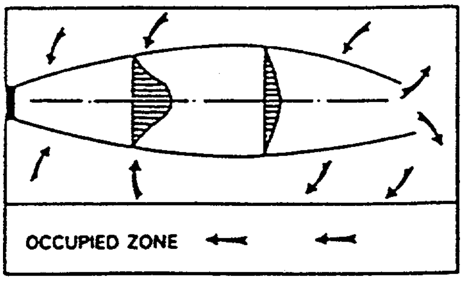
Computational Fluid Dynamic Codes. There are only few attempts to utilize computational fluid dynamic codes for air distribution design in industrial halls (Annex 26). In the demonstration project performed within IEA program, researchers studied mixing air distribution in fiber-glass reinforced polyester factory. They have concluded that the complexity of airflow and space geometry could not allow to make an accurate prediction.

Multizonal models. A number of multizonal models were developed by different researchers for analytical description of temperature and contaminant concentration distribution along the room height. The importance of such models is in capability to predict the efficiency of HVAC systems and to analyze the factors (i.e., method of air supply, location of air exhausts, location and type of heat and contaminant sources and sinks) that affect ventilation efficiency. For example, properly designed mixing type air distribution systems create comparatively even distribution of temperature and gas concentration along the room height. However, when the momentum of the heated air jet is not enough, warm air razes to the upper zone and the temperature in the occupied zone remains low, even when enough heat is supplied to the space. Also, information on ventilation effectiveness can be obtained from field studies, and from numerical and physical modeling. Discussion on the methods to predict ventilation efficiency in ventilated space is presented in Chapter 9.

Table 2.1. Guidelines for Selection of Air Distribution Method (Reproduced from ASHRAE 1995)

Characteristic of Ventilated Space						Recommended Air Supply Method
Height, m	Air Change Rate, ACH	Human Activity Type	Requirements to Parameters Uniformity in the Occupied Zone	Obstructions	HVAC System Type	
> 6-8m	< 5	Moderate and Heavy	None	Insignificant	CAV	Concentrated air supply; air supply by horizontal concentrated jets and vertical and/or horizontal directing jets, air supply with inclined jets, air supply with vertical compact jets, displacement air supply; a spiral vortex air supply.
					VAV	Air supply by horizontal concentrated jets and vertical and/or horizontal directing jets, air supply with inclined jets; downward air supply with compact jets.
				> 3 m	CAV	Air supply by horizontal concentrated jets and vertical and/or horizontal directing jets; air supply with inclined jets; displacement air supply.
					VAV	Air supply by horizontal concentrated jets and vertical and/or horizontal directing jets; air supply with inclined jets.
< 6-8m	>5	Sedentary and Light	High Degree of Uniformity	Insignificant	CAV	Air supply with vertical radial, conical or compact jets, air supply with horizontal attached or non-attached jets; displacement air supply.
					VAV	Air supply with vertical radial or compact jets, air supply with horizontal attached or non-attached jets.
				> 2 m	CAV	Air supply with vertical compact jets; air supply with horizontal attached or non-attached jets into the ails between process equipment displacement air supply;
					VAV	Air supply with vertical compact jets; air supply with horizontal attached or non-attached jets into the ails between process equipment.

a



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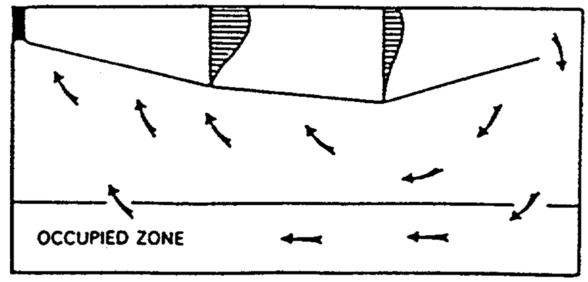

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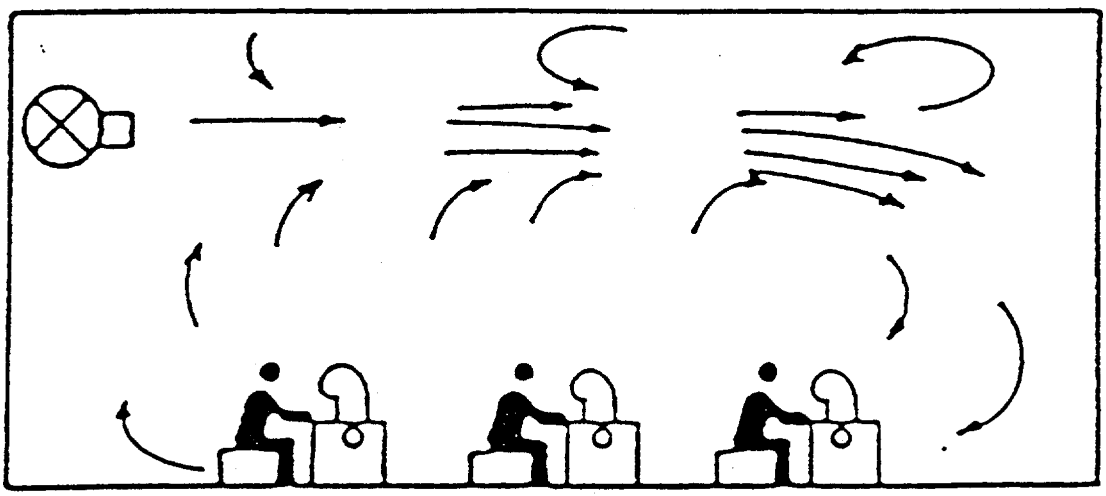


Figure 2.1. Concentrated air supply with occupied zone ventilation by reverse flow: a- schematic of air flow pattern with attached jet, b - schematic of air flow pattern with not attached jet, c - air supply with not attached jet in the shop (a,b - reproduced from Zhivov, 1992; c - reproduced from AIR-IX, 1987).

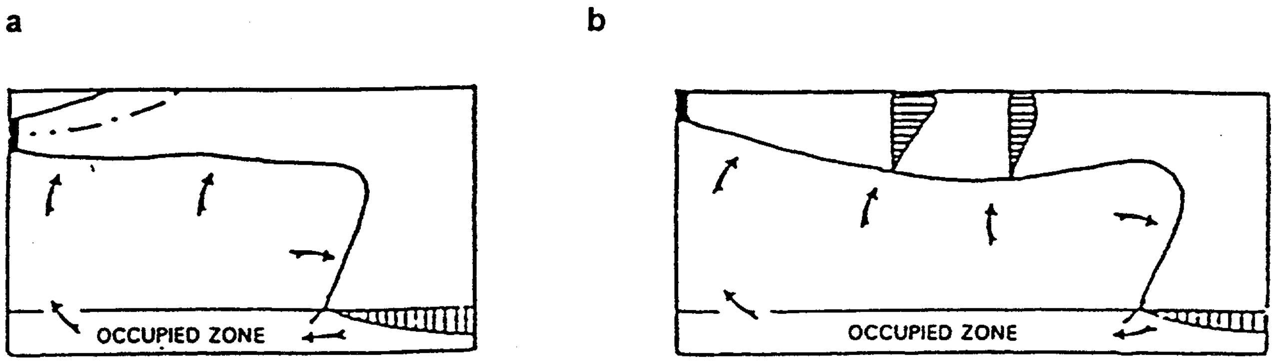


Figure 2.2. Air supply with horizontal jets: a - occupied zone ventilated directly by attached to the ceiling jet; b - occupied zone ventilated directly by attached to the ceiling jet and by reverse flow (Reproduced from Zhivov 1992).

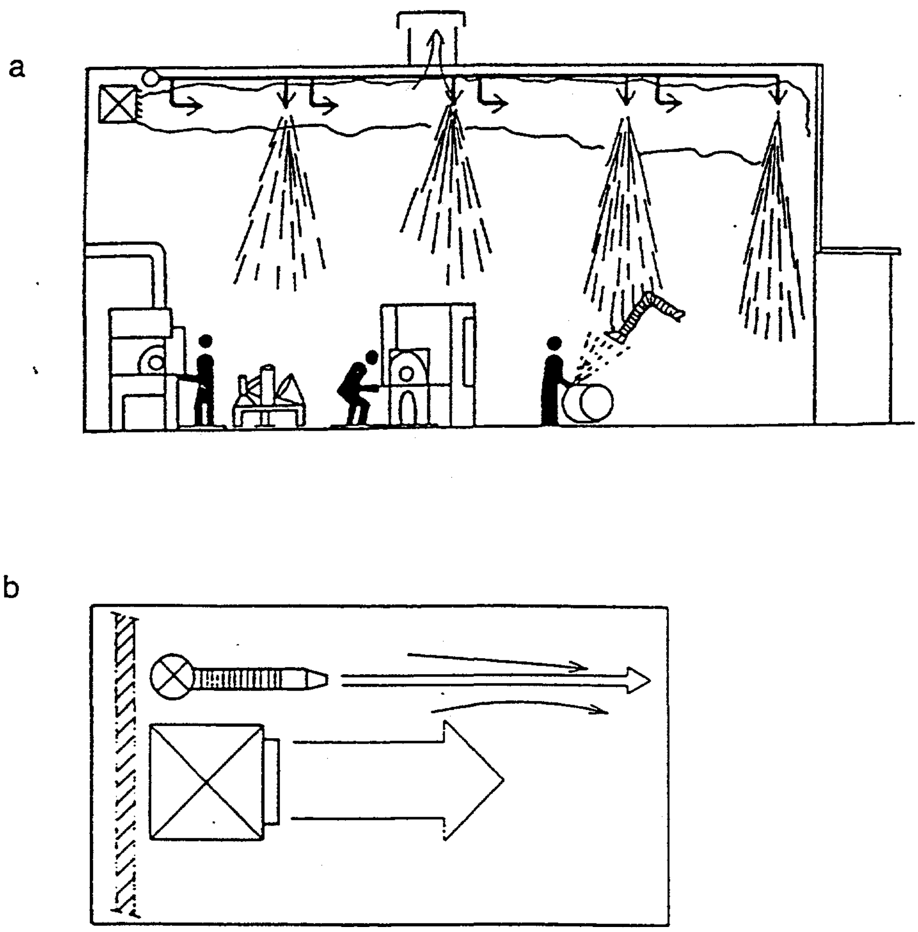


Figure 2.3. Concentrated air supply with (a) horizontal and vertical directing jets, (b) with only horizontal directing jets (Reproduced from AIR-IX 1987).

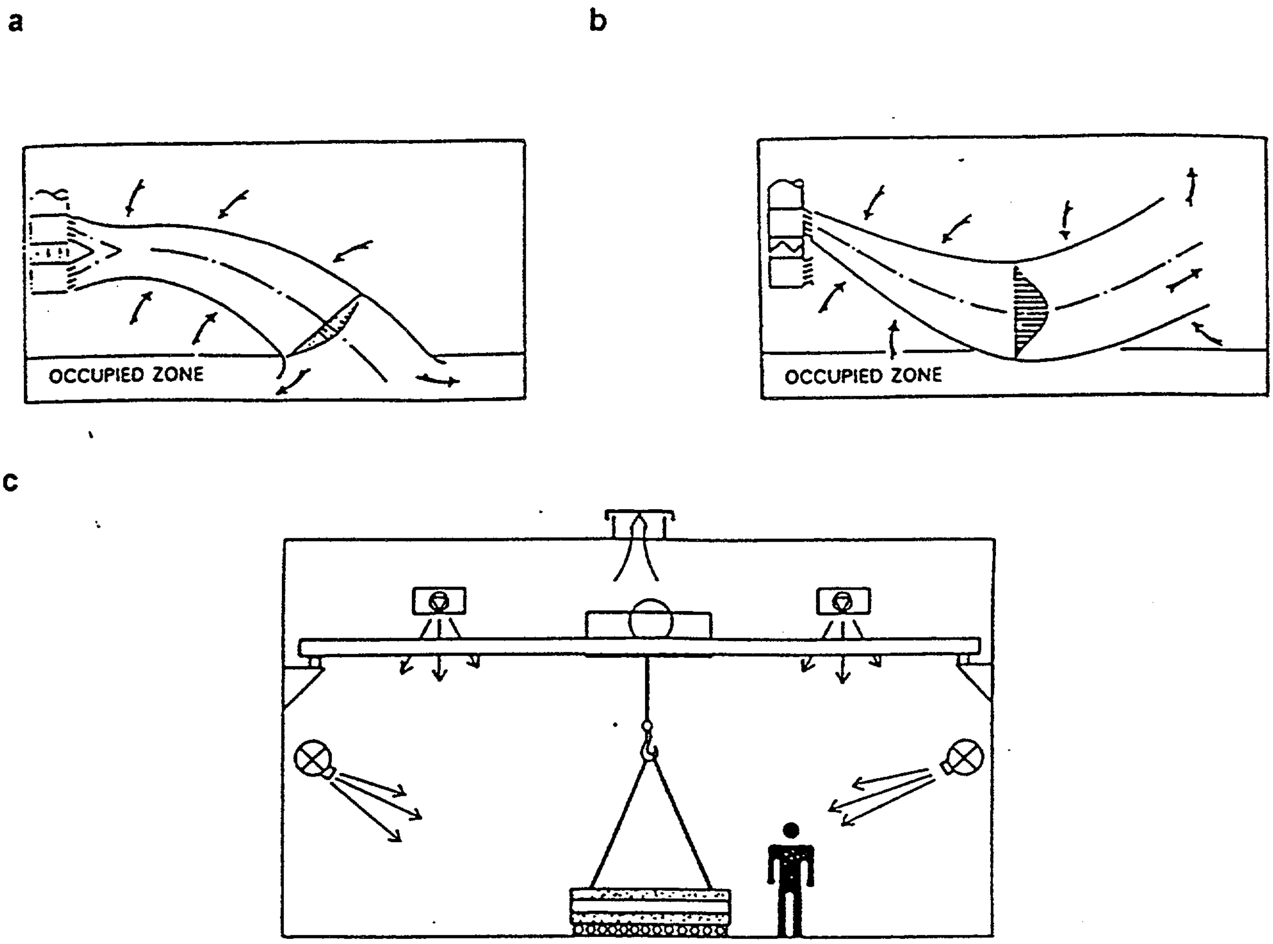


Figure 2.4. Air supply with inclined jets: a - schematic of cooled air supply; b - schematic of heated air supply; c - inclined air supply in mechanical shop (a and b - reproduced from Zhivov 1992, c - reproduced from AIR-IX, 1987)

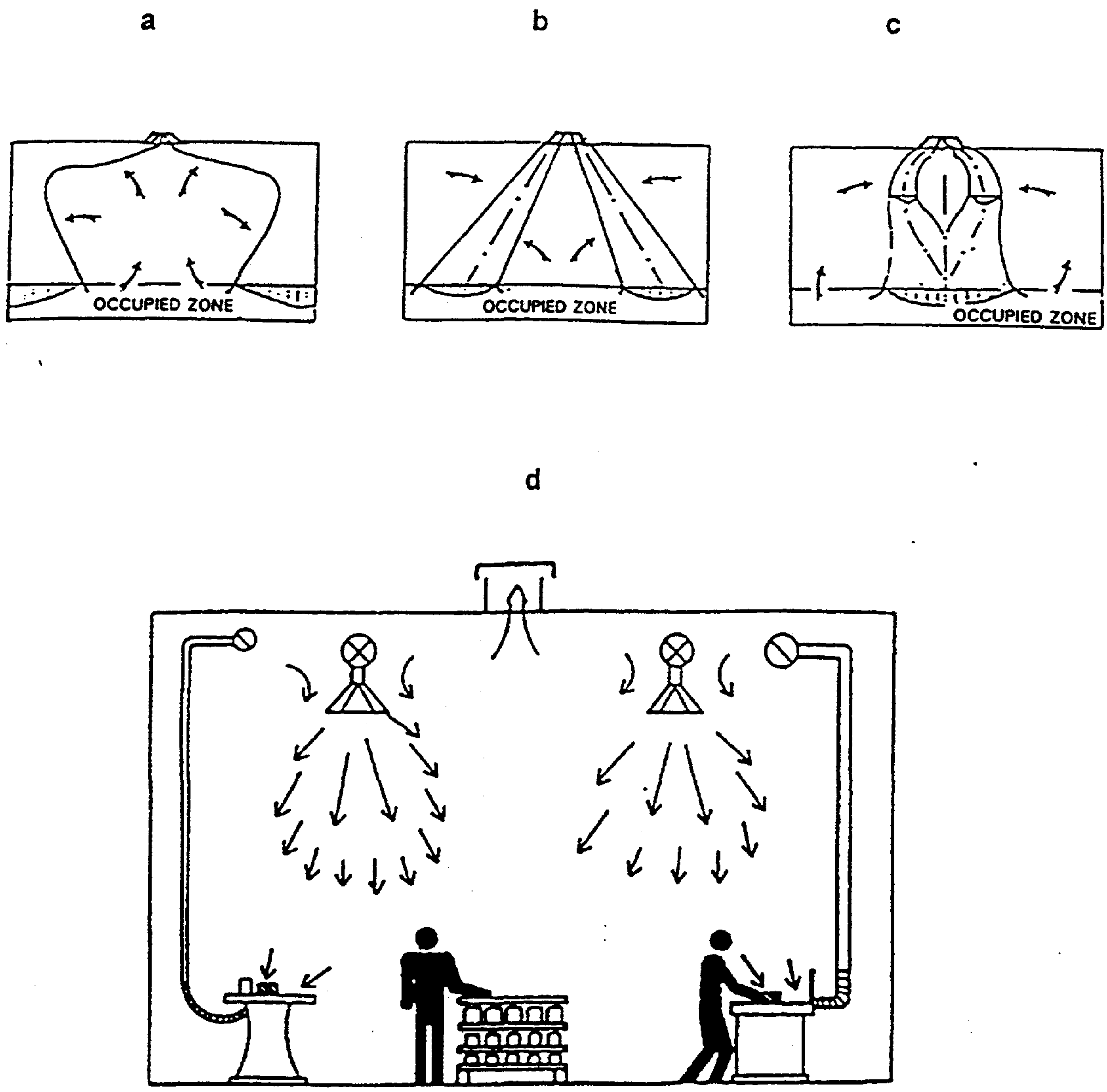


Figure 2.5. Air supply by (a) radial, (b) conical or (c) compact air jets through the ceiling type air diffusers installed in the ceiling, and (d) by conical jets through diffusers installed on vertical duct drops (a,b and c - reproduced from Zhivov 1992, d - reproduced from AIR-IX, 1987).

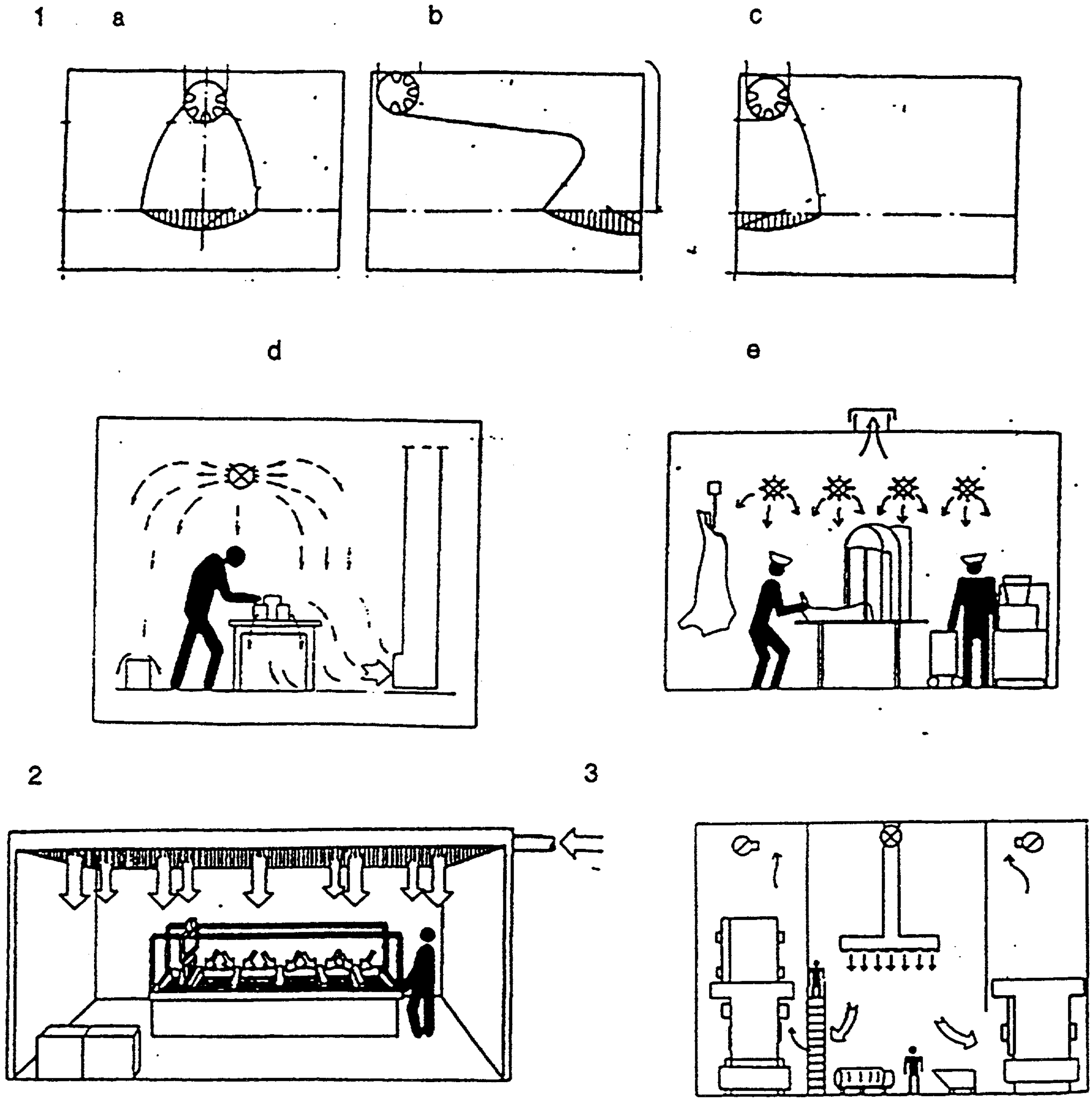


Figure 2.6. Air supply through (1) perforated ducts, (2) perforated ceilings and (3) perforated panels: 1 - vertical linear jet (a), horizontal attached linear jet (b), vertical attached linear jet (c); two unattached horizontal jets (d), radial jets (e) (1a,b,c -reproduced from Designer's Guide, 1992; 1d,e, 2 and 3 - reproduced from AIR-IX 1987).

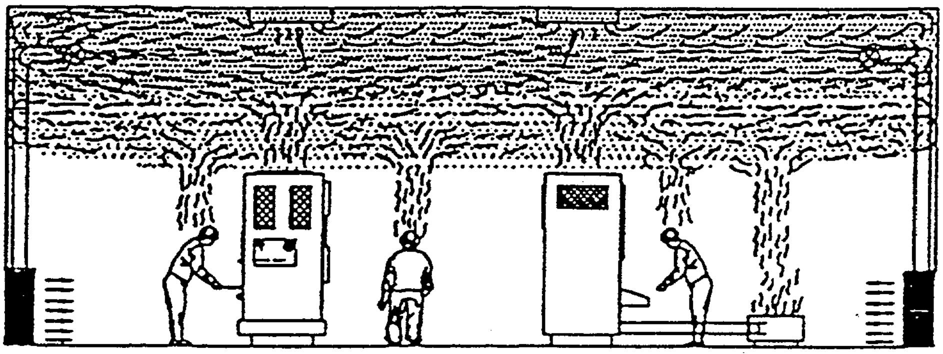
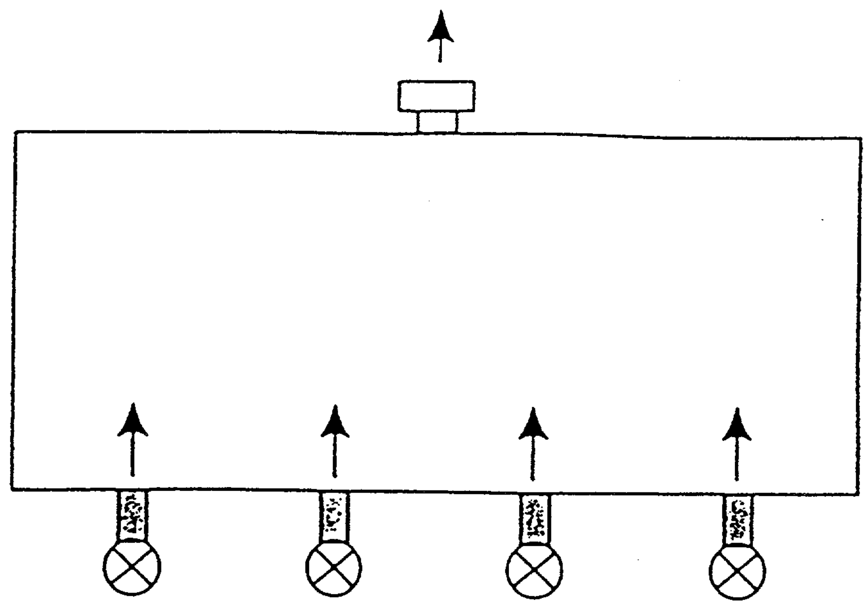


Figure 2.7. Displacement Ventilation (Reproduced from Kristensson, et al. 1993).

a



b

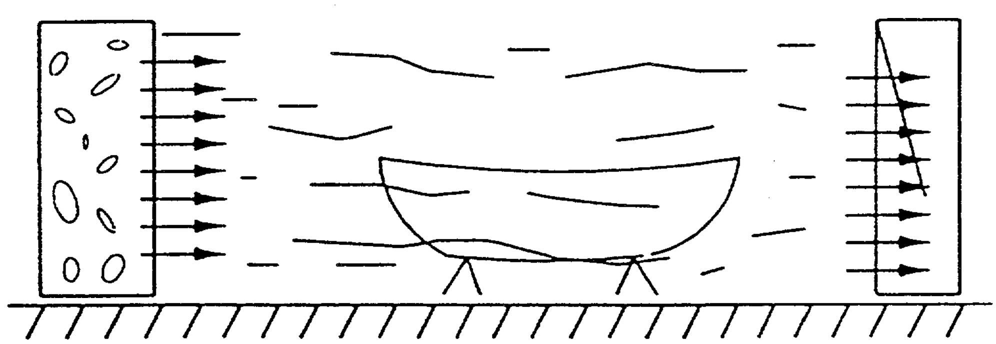


Figure 2.8. Unidirectional flow systems with (a) vertical air flow, (b) with horizontal air flow (a - reproduced from AIR-IX 1989, b - reproduced from LVIS 1996).

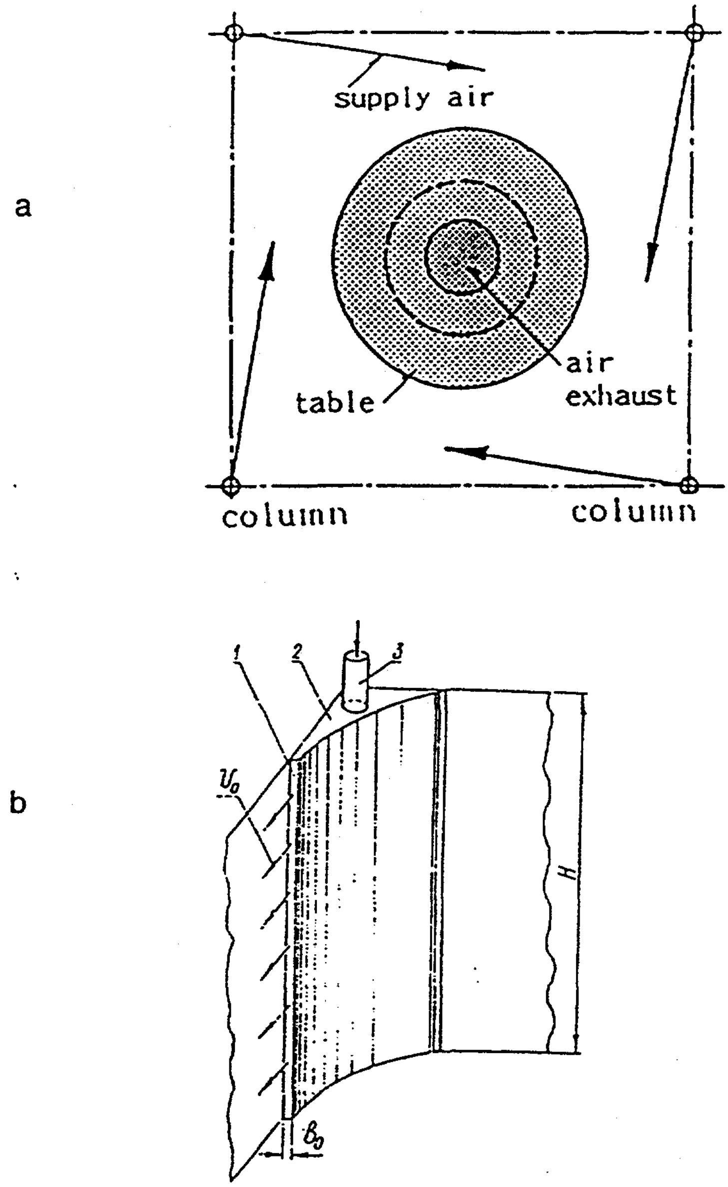


Figure 2.9. Spiral vortex ventilation system: a - schematic, b - air diffuser (a - reproduced from Nagasawa et al. 1990; b - reproduced from Kuz'mina et al. 1986).

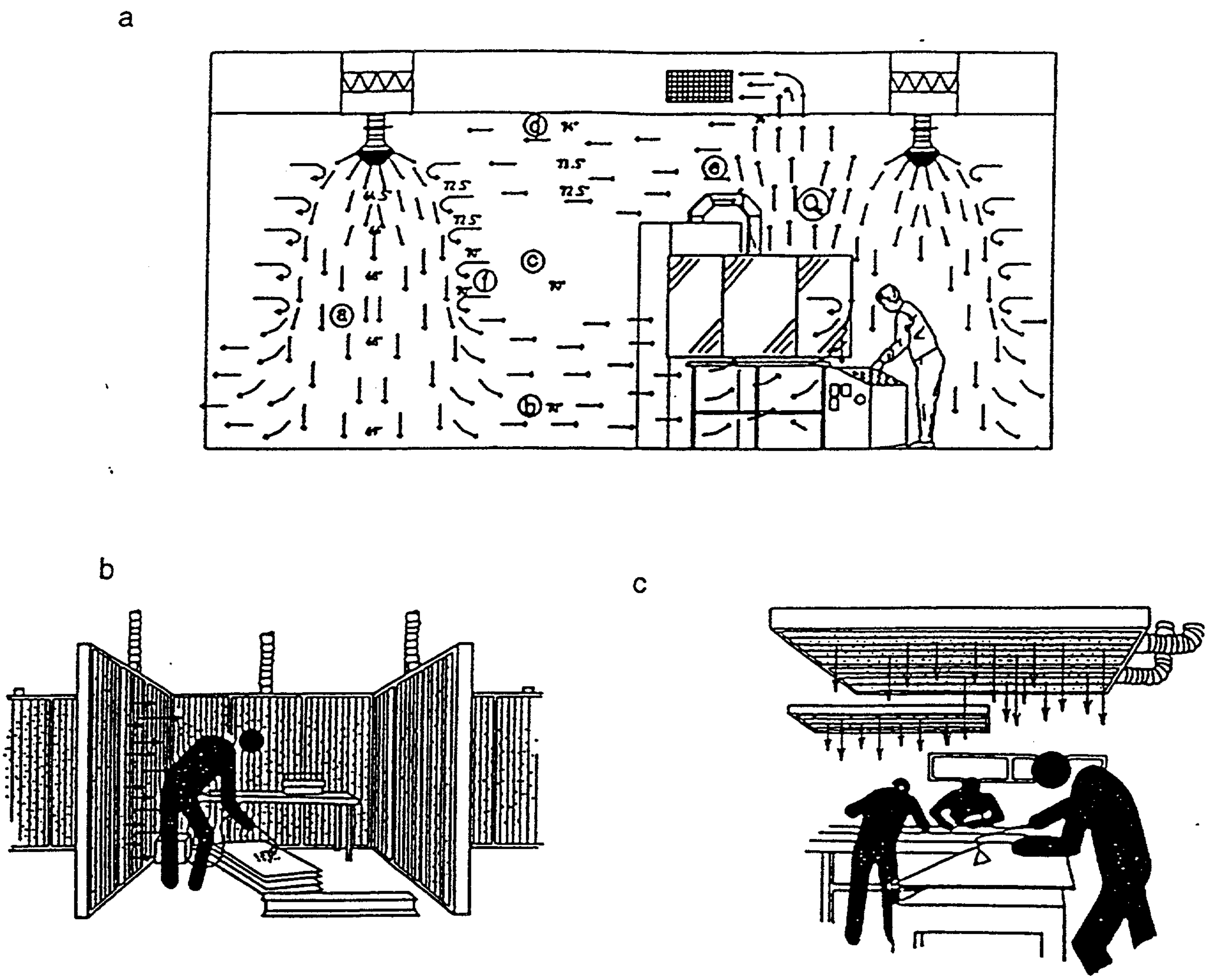


Figure 2.10. Localized ventilation systems: a - air showers, b - air oasis with horizontal air supply, c - air oasis with vertical air supply (a - reproduced from Kristensson, b and c - reproduced from AIR-IX 1987).

3. AIR DIFFUSERS AND THEIR PERFORMANCE

There is a wide range of air diffusion devices which can be used with different air distribution methods. The following is incomplete list of the most typical air diffusers.

Grilles are one of the most universal type of air diffusers (Figure 3.1). They can have one or two rows of vanes: vertical or horizontal and different aspect ratio and vane ratio. Vanes affect grill performance if their depth is at least equal to the distance between the vanes.

A grille discharging air uniformly forward (vanes in straight position) has a spread of 14 to 24 deg, depending on the type of diffuser, duct approach, and discharge velocity. Turning the vanes influences the direction and throw of the discharged airstream. The parallel horizontal vanes direct the air stream vertically within 45 deg. It should be noted that if the vane ratio is less than two, the jet inclination will be smaller than the angle of vanes.

Vertical vanes can be used to spread the air horizontally and horizontal vanes can be used to spread the air vertically. A grille with diverging vanes (vertical vanes with uniformly increasing angular deflection from the centerline to a maximum at each end of 45 deg) has a spread of about 60 deg and reduces the throw considerably. With increasing divergence, the quantity of air discharged by a grille for a given upstream total pressure decreases.

A grille with converging vanes (vertical vanes with uniformly decreasing angular deflection from the centerline) has a slightly higher throw than a grille with straight vanes, but the spread is approximately the same for both settings. The airstream converges slightly for a short distance in front of the outlet and then spreads more rapidly than air discharged from a grille with straight vanes.

Ceiling mounted air diffusers can be round, rectangular and linear and have outlets covered with grille, perforation, plaque, vanes forming multiple slots or a swirl insert (Figure 3.2). There are regulated and non-regulated ceiling mounted air diffusers. Depending on their design they can form attached radial, concentrated or linear jets as well as non-attached conical or concentrated air jets. Rectangular air diffusers with triangle or four sided grilles form non-uniform circular flow and can be considered as three or four separate jets.

For VAV application with considerable air volume and initial temperature differential range, air diffusers with regulated outlet area and/or direction of air supply as well as those with induction of room air create better performance within the year-round cycle of system operation. For application in industrial and commercial facilities with high ceilings, these diffusers can be mounted on duct drops (with installation height 3 m to 5 m) supplying compact conical or non-attached radial jets. Non-attached radial and conical jets typically collapse into conical or compact jets under the influence of buoyant forces.

Round, square and rectangular nozzles (Figure 3.3) with an outlet size from 10mm to 1 m are commonly used for different applications: from small residential rooms to large atriums, sport halls and industrial buildings. Converging nozzles form air jets with considerably higher throw and lower noise level compared to other air diffusers. Diverging nozzles are used to supply compact jets with increased angle of divergence and reduced throw. Typically, the latter type of jet can be achieved either by placing two or more concentric cones at the supply side of the air diffuser or by placing a swirl insert inside the straight nozzle.

Perforated panels (Figure 3.4) are used to supply air directly into the occupied zone for displacement ventilation systems or vertically downward. They discharge air with low velocities (0.2-0.5 m/s) and low turbulence. A diffuser panel is much more than a simple perforated plate or a filter mat. Simple perforated plates usually produce irregular flow of supply air and cannot be used as supply air devices. Another problem of filter mats is that they are blackened by particles in the supply air. Recently a new generations of perforate panels were introduced:

(1) with induction chambers which allows to mix supply air with room air inside air diffuser housing. This design allows to supply air with a greater air temperature difference without causing discomfort in the occupied zone, and

(2) with internal deflectors to adjust the flow direction. These panels are cable of decreasing the restricted zone (zone with abnormal velocities) in front of air diffuser.

Perforated ducts (Figure 3.5) round or rectangular with a partially or complete perforated/slotted walls. Primarily, they are used to supply air in spaces where a high air change rate is required and air velocities in the occupied zone are limited due to process limitations (e.g., to prevent contaminant spillage from local exhausts). The supply surface may be created either by perforating the duct wall, or by cutting the incomplete wholes in the wall and bending metal peaks inside/outside the duct (to deflect air jets in the right direction from the duct surface), or by stamping converging nozzles in the desired areas of the sheet metal bend that is used to form the spiro duct.

Typical applications for air supply diffusers are summarized in Table 3.1.

Table 3.1. Typical air diffusers and their applications

Air diffuser type	Air diffuser performance characteristics		Method of air distribution	Application
	K_1*	K_2*		
Large grills	2-6	1.8-5.1	Fig. 2.1, Fig. 2.3, Fig. 2.4	Large shops
Sidewall grills,	2-6	1.8-5.1	Fig. 2.2	Low rooms (< 20ft)
Grills mounted on duct drops	2-4	1.8-3.5	Fig. 2.5d	Large shops
Circular diffusers	1-3	0.9-3.2	Fig. 2.5a,b,c Fig. 2.9a	Low rooms (< 20ft) Large shops
Square diffusers	1-2.8	1.2-3.2	Fig. 2.5 a,b,c Fig. 2.9a	Low rooms (<20ft) Large shops
Linear diffusers	2.5	2	Fig. 2.2	Low, small rooms
Perforated panels (round, half/quater round, flat) mounted on or near the floor			Fig. 2.7, Fig. 2.8b Fig. 2.10b	Rooms higher than 12ft with surplus heat or combined heat and contaminant emissions
Perforated panels, mounted on duct drops	2.1	1.7	Fig. 2.6(3), Fig. 2.10c	Shops with surplus heat and a few work places or shops obstructed by precess equipment
Perforated ducts	0.5	1.2	Fig. 2.6(1)	Low industrial rooms with surplus heat, high air change rate and requirements for low velocities in the occupied zone
Perforated ceiling mounted panels or perforated panels	2.1	1.7	Fig. 2.6(2)	Same + special applications (e.g., clean rooms)
Nozzles				Large shops
converging	6 -6.8	4.2-4.8	Fig. 2.1, Fig. 2.3	Large shops; air supply into ails
diverging or with swirl inserts	1-2.5	0.8-2.0	Fig. 2.4, Fig. 2.10a	

*Approximate ranges are given for air diffuser characteristics: K_1 = coefficient of velocity decay along the jet; K_2 = coefficient of temperature decay along the jet. For actual values of these characteristics, consult manufacturers' guides.

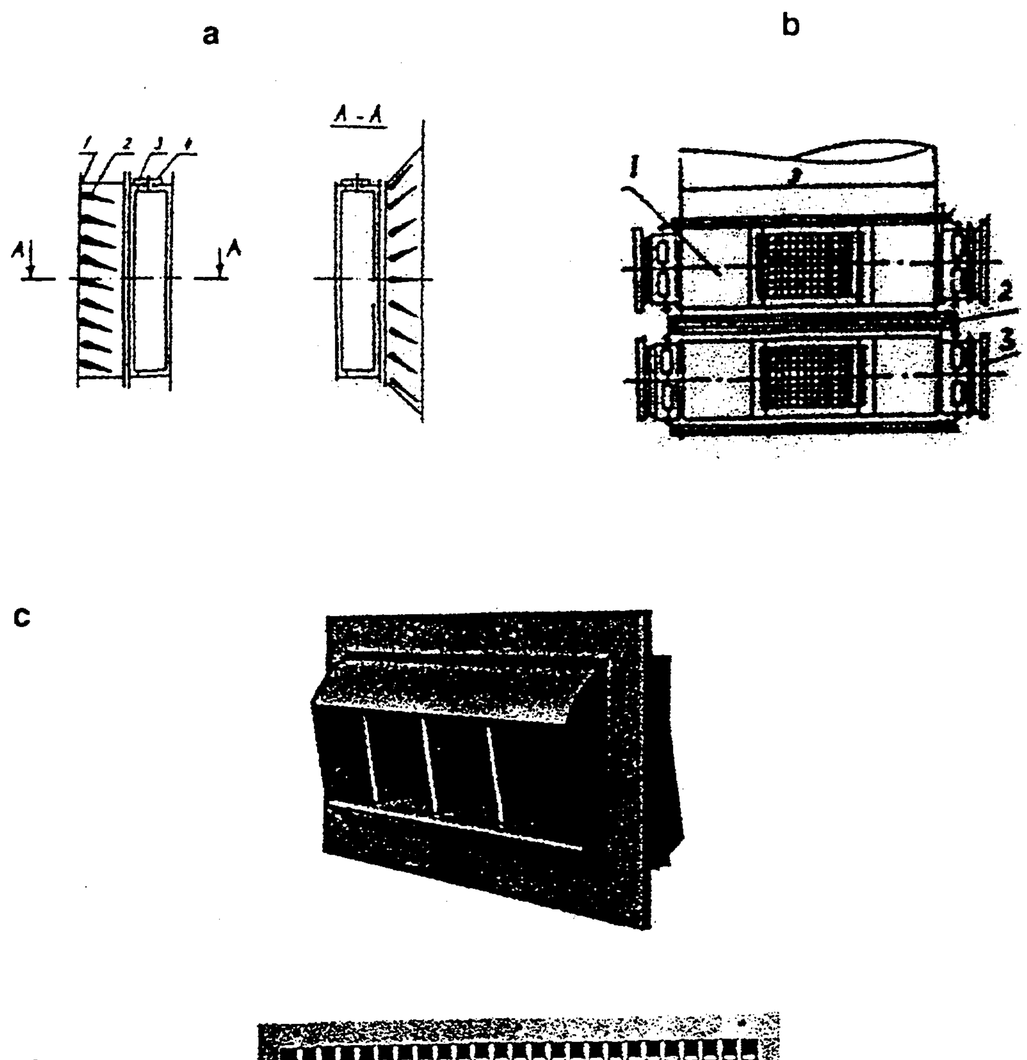


Figure 3.1 Air supply grilles: a - schematic of RV grille (Russia), 1 - grille frame, 2 - vanes, 3 - immovable axis, 4 - movable axis; b - VPRV air diffuser with RV grilles (Russia), 1 - round duct tees, 2 - damper, 3 - grilles; c - DL drum louver (TITUS, U.S.A.); d ventilation grille (TROX, Germany)

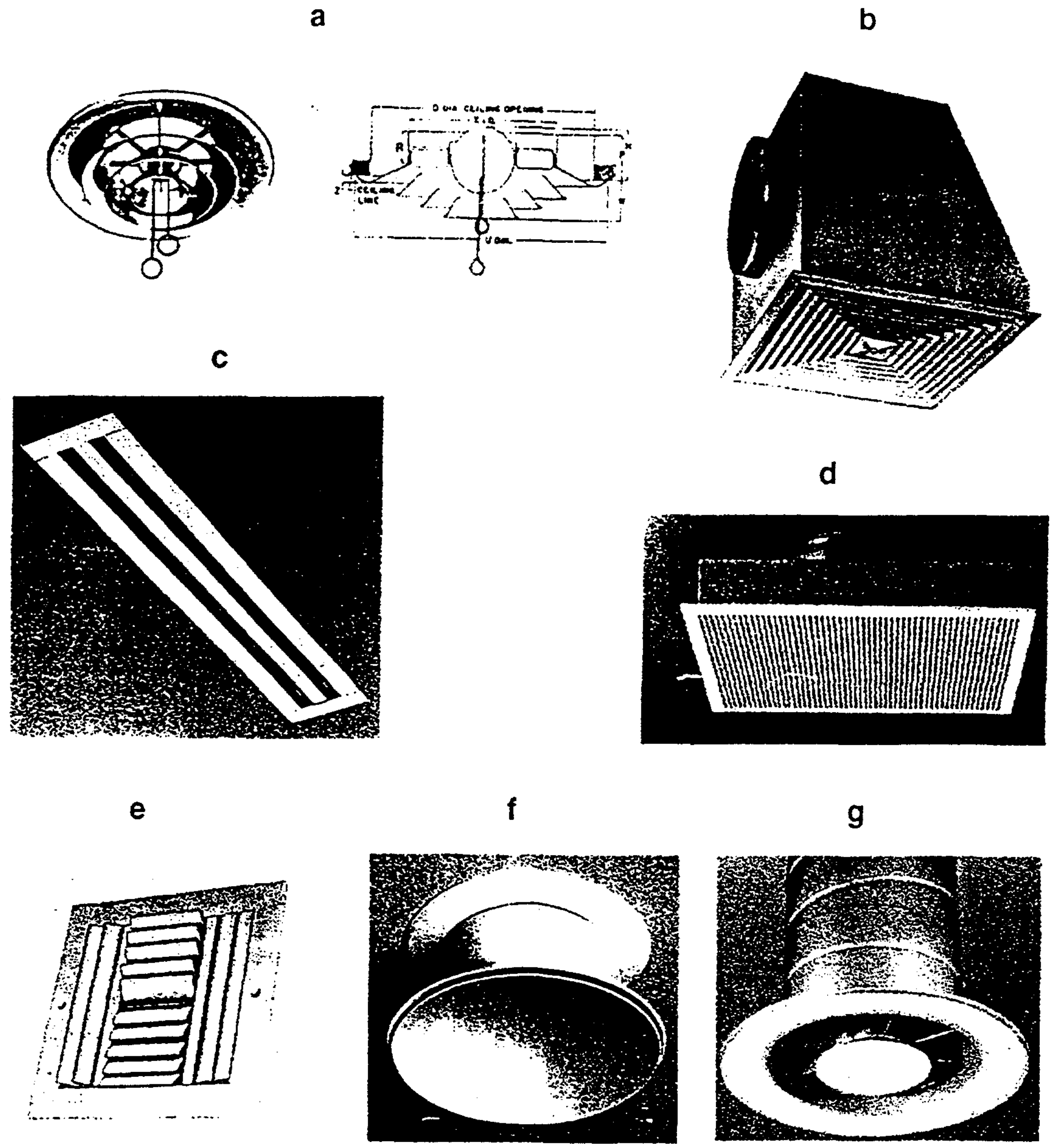
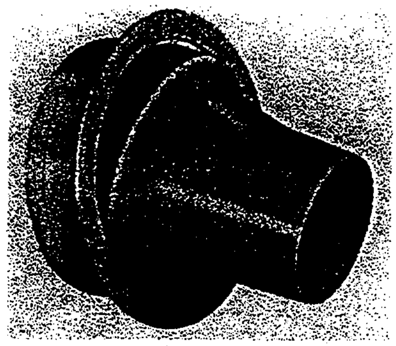


Figure 3.2 Ceiling mounted air diffusers: a - circular multidiffuser (ANEMOSTAT, U.S.A.), b - square/rectangular multidiffuser (TROX, Germany), c - linear (ABB, Sweden), d - rectangular with perforated outlet surface (Lindab, Sweden), e - rectangular with vanes forming multiple slots (KRUGER, U.S.A.), f - round with a plaque (ABB, Sweden), g - round with a swirl insert (KRANTZ-TKT, Germany)

a



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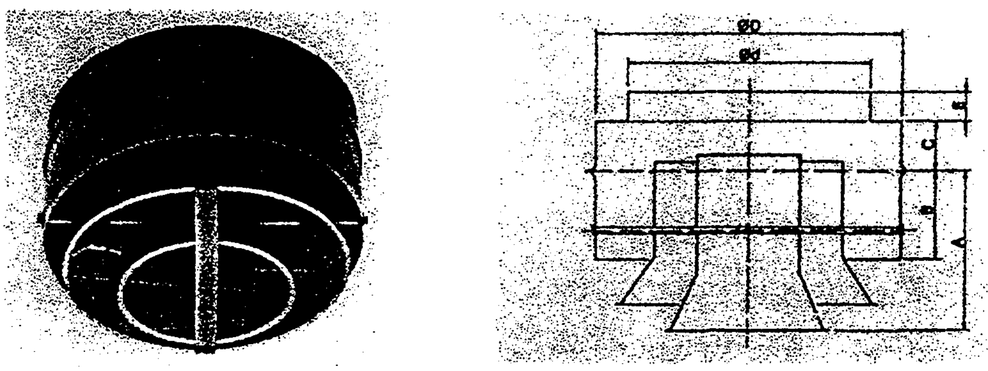

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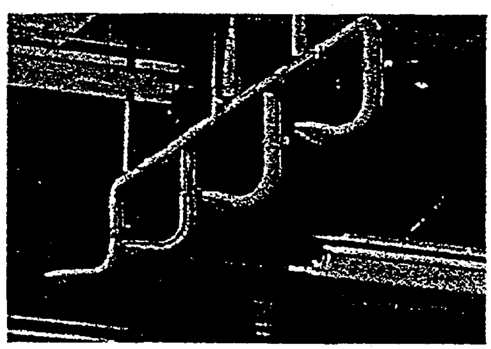


Figure 3.3 Nozles: a - with adjustable direction (TROX, Germany), b - nozzle with a concentric insert (Halton, Finland), c - small nozzles with flexible ducts for systems with directing jets (ABB, Sweden)

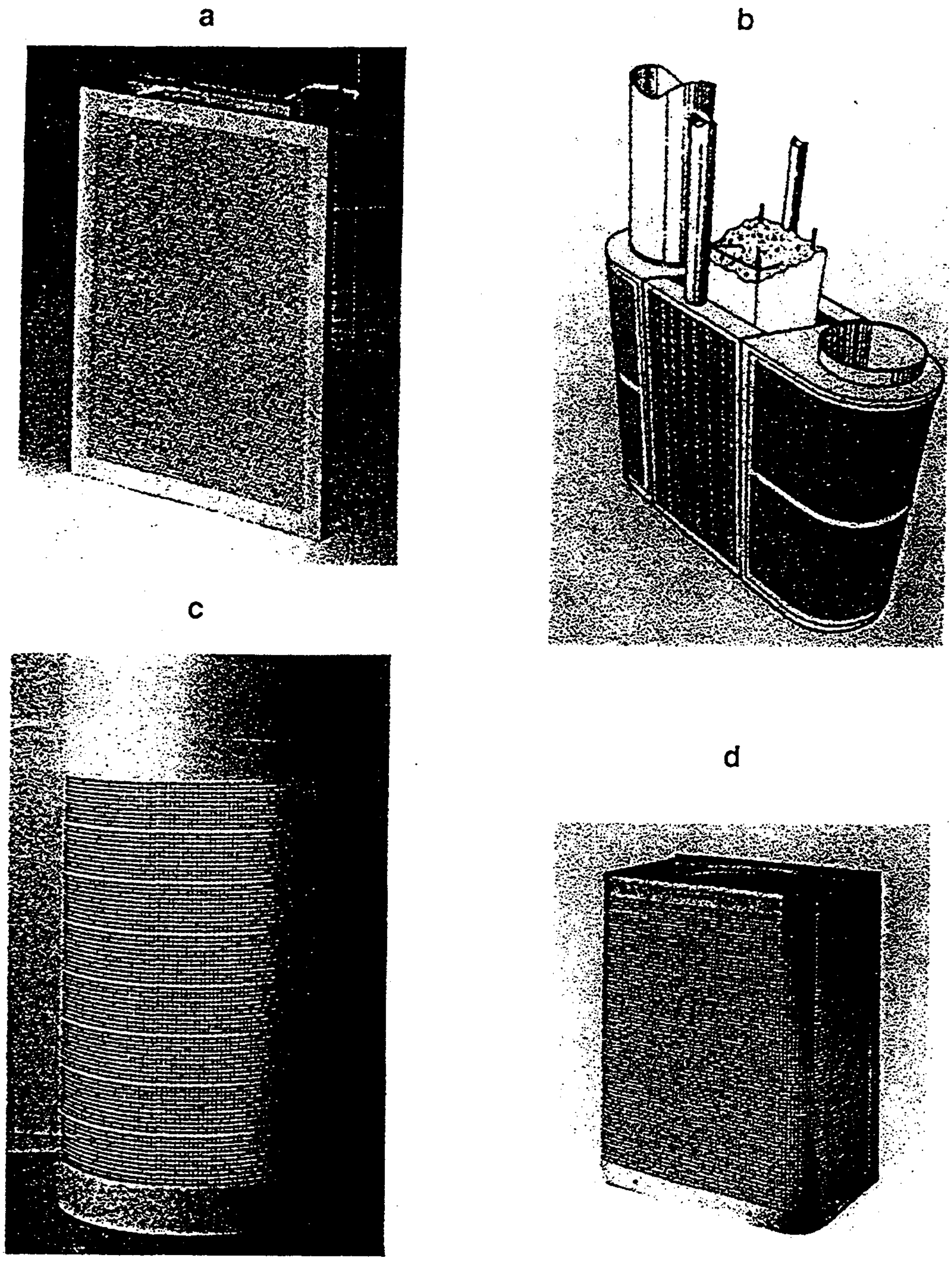


Figure 3.4 Perforated panels: a - flat rectangular (Halton, Finland), b - semicircular combined with rectangular (Repus, Sweden), c - with induction (ABB - Sweden), d - with movable air deflectors to adjust air spread pattern (Stifab, Sweden)

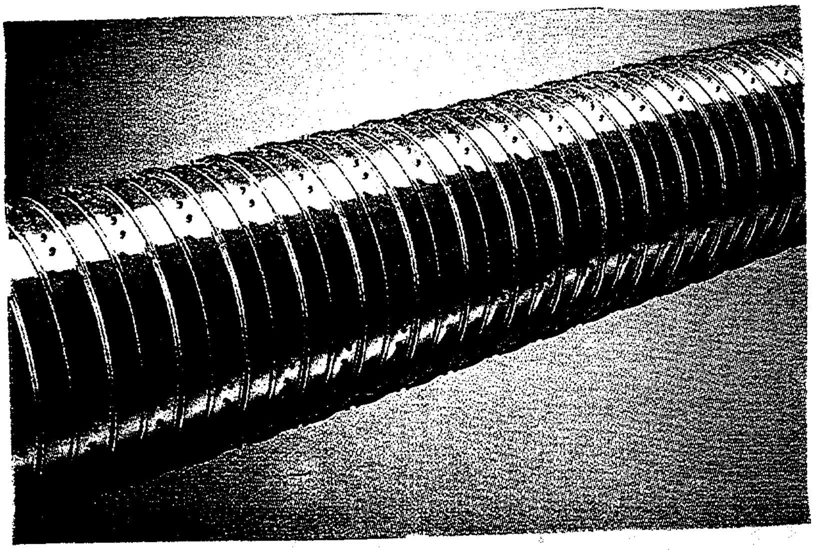


Figure 3.5 Perforated ducts (ABB, Finland)

4. AIR JETS THEORY

4.1. Introduction

Air supplied into the room through the various types of outlets (grills, ceiling mounted air diffusers, perforated panels etc.), is distributed by turbulent air jets. In mixing type air distribution systems, these air jets are the primary factor affecting room air motion. Numerous theoretical and experimental studies that developed a solid base for turbulent air jets theory were conducted concurrently in different countries (Germany, Sweden, Russia, U.K., USA) from the 1930's through the 1980's. Among the most important findings during that early stage are those made by L. Helander, H. Nottage, G. Tuve, A. Koestel, W. Linke, B. Regenscheit, G. Abramovich, I. Shepelev, P. Becher, H. Müllejans, V. Taliev, V. Baturin, M. Grimitlyn, P. Jackman, M. Holmes, P. Nielsen. Most of the current information is also available from ASHRAE/ASHVE Transactions, ASHRAE data base, proceedings of "ROOMVENT'87", "ROOMVENT'90" "ROOMVENT'92", "ROOMVENT'94" and "ROOMVENT'96" conferences. There are many valuable information on air jets, room air distribution and ventilation effectiveness published in English, German, Russian and Scandinavian languages. Most of air distribution design guidelines created on the national levels utilize air jet theory (e.g., ASHRAE 1997; AICVF 1991; CIBSE 1988; Designer's Guidebook 1992). Theory of air jets and air distribution design principles are discussed in this chapter.

4.2. Classification

If there is no influence of the walls, ceiling or obstructions on the air jet, it can be considered as a free jet. If the air jet is attached to a surface, it is called an attached air jet.

Characteristics of the air jet in the room might be influenced by the reverse flows, created by the same jet entraining the ambient air. This air jet is called a confined jet. If the temperature of the supplied air is equal to the temperature of the ambient room air, the jet is called isothermal jet.

A jet with an initial temperature different from the temperature of the ambient air is called non-isothermal jet. The air temperature differential between supplied and ambient room air generates buoyancy forces in the jet, affecting the trajectory of the jet, the location at which the jet attaches and separates from the ceiling/floor and the throw of the jet. The significance of such effect depends on the ration between the thermal buoyancy and inertial forces (characterized by Archimedes number).

Dependent upon the diffuser type, air jets can be classified as follows (Figure 4.1):

1. Compact air jets are formed by cylindric tubes, nozzles, square or rectangular openings with small aspect ratio, unshaded or shaded by perforated plates, grills etc. Compact air jets are three dimensional and axisymmetric at least on some distance from the diffuser opening. The maximum velocity in the cross-section of the compact jet is on the axis.
2. Linear air jets are jets are those formed by slots or rectangular openings with a large aspect ratio. The jet flows are approximately two dimensional. Air velocity are symmetric in the plane

at which air velocities in the cross-section are maximum. At some distance from the diffuser, linear air jet tend to transform into the compact one.
3. Radial air jet are formed by the ceiling cylindrical air diffusers with flat discs or multi diffusers to direct the air horizontally in all directions.
4. Conical air jets are formed by the cone-type or regulated multi diffusers ceiling air distribution devices, have an axis of symmetry. The vectors of air velocities are parallel to the conical surface (with an angle at the top of the cone equal to 120 deg.), which is also the place of maximum velocities in the cross-sections perpendicular to the axis.
5. Incomplete radial jets are supplied through the outlets with the grilles having diverging vanes and have a coerced angle of expansion. At some distance this kind of jet tends to transform into a compact one.
6. Swirling jets are supplied into the room through air diffusers with vortex-forming devices creating the rotation motion and have besides the axial component of velocity vectors tangential and radial ones. Depending upon the type of air diffuser swirling jets can be compact, conical or radial.

4.3. Isothermal free jet

4.3.1. Zones in a Jet

It was acknowledged, that for different types of free jets and air diffusers there is a similarity in a resulting flows. Four major zones are recognized along a free jet. These zones, as described by Tuve (1953), may be roughly defined in terms of the maximum or center core velocity, that exists at the jet cross-section being considered:
(Figure 4.2a). *Zone 1* is a short zone, extending about two to six diffuser diameters (for compact and radial jets) or slot widths (for linear jets) from the diffuser face. In this zone, the centerline velocity of the jet remains nearly equal to the original supply velocity throughout its length. *Zone 2* is a transition zone, and its length depends upon the diffuser type. For a compact jet a transition zone typically extends to 8 or 10 diameters. Within this zone, the maximum velocity may vary inversely as the square root of the distance from the outlet. Some researches (Abramovich, 1960; Shepelev, 1961; Grimitlyn, 1970)suggest for practical purpose to use simplified scheme of the jet (Figure 4.2b) with a transition cross-section.

Zone 3 is the zone of fully established turbulent flow. It has a major engineering importance, since it is usually this zone where the jet enters the occupied region. The length of this zone depends on the air jet shape, type and size of supply air diffuser, initial velocity and turbulence characteristics of the ambient air.

Zone 4 is a terminal zone in which the residual velocity decays quickly into large scale turbulence. Within a few diameters, the maximum velocity subsides to the range below 0.25 m/s. Though this zone was studied by several researchers (Madison et.al., 1946; Weinhold, 1969), its characteristics is still not well understood.

In some practical applications of air supply (e.g., multiples jet ceiling diffusers, annular jet collapsing into a compact, rectangular outlet jets transition into a round or elliptical, or multiple streams merge when air is supplied through perforated panels) measurements and accurate jet description in Zone 1 and Zone 2 may be difficult.

4.3.2. Velocity distribution in a jet cross-section within the Zone 3.

Velocity distribution profiles in Zone 3 of the jet were found to have similarity (Trüpel, 1915; Förtman, 1934; Albertson et al., 1948). They can be computed applying momentum-transfer theory (Prandl-Tollmein) and the vorticity-transfer theory (Taylor-Goldstein). Modification of these theories with different assumptions resulted in several equations for jet velocity profiles. These equation are presented in Table 4.1 and can be divided into two groups:

- profiles with a finite boundaries (Tollmein 1926, Schlichting 1930), with a zero velocity at the specified distance from the jet axis;
- profiles with an indefinite boundaries (Reichardt 1942, Görtler 1942), with air velocity decreasing with a distance from the axis asymptotical urging towards zero.

Table 4.1. Equations for velocity profiles in free jet

Author	V_i/V_x	
	round jet	linear jet
Schlichting	$(1 - r/\delta^{3/2})^2$	$(1 - Yy/\delta^{3/2})^2$
Tollmein*	$0 \le r/\delta \le 3.4$: $1 - 0.633\ r/\delta^{3/2} + ...$ $r/\delta \ge 3.4$: 0	$0 \le r/\delta \le 2.4$: $1 - 0.424\ r/\delta^{3/2} + ...$ $r/\delta \ge 2.4$: 0
Reichardt	$\exp\{-(r/\delta)^2\}$	$\exp\{-(r/\delta)^2\}$
Görtler	$\mathrm{Cosh}^{-2}(Cr/\delta)$	-

*Equations for velocity profiles in jets supplied from a round and slot nozzles derived by Tollmein have only numerical solutions. For practical use the resulting data is presented in tables (e.g., see Abramovich 1984). Table 4.1 presents approximation of Tollmein equations reproduced from Kraemer (1971).

Table 4.2. Jet angle of divergence.

Author	round			linear			radial	
	δ	$\beta/2$ Arctan(δ/X)	$\alpha/2$ Arctan($\delta_{0.5v}$/X)	δ	$\beta/2$ [Arctan(δ/X)]	$\alpha/2$	δ	$\alpha/2$
Tollmein, 1926	0.151X			0.272X				
Reichardt, 1942	0.085X		4.9		[0.112]	6.4		
Forthmann, 1934						5.8		
Miller, Comings 1957						5.5		
Corrsin,1943			4.8					
Ermshaus, 1963 (see Kraemer 1971)			5.2					
Kraemer, 1971			5.4					
Heskestad, 1966							0.288X	
Tuve, 1953							0.185X	10.5
Tuve and Priester 1944		10 to 11			11 to 11.5			
Becher (1949)		12	4.4		16.5	9.2		

In Equations listed in Table 4.1, **r** (**Y**) is the distance from the point of interest to the jet axis, **δ** is a distance to the jet boundary, that can be obtained from Equations summarized in Table 4.2.

It has been demonstrated by Ruden (1933), Albertson et al.(1948), Taylor et al. (1951), Keagy (1949), Pai (1949), Becher (1949), Forthmann (1934), Nottage et al. (1952), Shepelev (1961), Grimitlyn (1994) and other researchers, that the Gauss error-function profile by Reichard is comparable with a data taken in studies of both nozzle jets and manufactured air diffusers supplying similar jets. This profile is utilized the most by researchers using analytical (semi-empirical) approach in air jets studies. Table 4.3 lists some modifications of the Gauss error-function velocity profile equations as they are used for practical applications.

Schlichting finite boundaries profile is another one that is frequently used (Abramovich 1984). Utilization of this profile is specifically fruitful to describe velocity distribution on complex flows, e.g., jet in a cross flow (Gendrikson and Ivanov 1973), jets interaction under the right angle (Zhivov 1982). In such cases distance from the jet axis r_i to the point with an air velocity $\mathbf{V}_i$ is substituted by the parameter $\mathbf{r}_i = (\mathbf{S}_i/\pi)^{1/2}$, where $\mathbf{S}_i$ is the area within a contour with a constant velocity value $\mathbf{V}_i$.

Table 4.3 Practical modifications of the Gauss error-function velocity profile equations

Author	V/V_x		
	Round jet	Linear jet	Radial jet
Tuve, 1953	$e^{-0.7(y/y_{0.5V})^2}$ $Y_{0.5V} = X \tan\alpha_{0.5V}$	$e^{-0.7(y/y_{0.5V})^2}$ $Y_{0.5V} = X \tan\alpha_{0.5V}$	$e^{-0.7(y/y_{0.5V})^2}$ $Y_{0.5V} = X \tan\alpha_{0.5V}$
Regenscheight, 1971	$e^{-4\left(\frac{r}{mX}\right)^2}$ m = 0.1 to 0.4[1]	$e^{-\frac{\pi}{2}\left(\frac{Y}{mX}\right)^2}$ m = 0.1 to 0.4	$e^{-\frac{\pi}{2}\left(\frac{Y r_o A_F^2}{h_o r (1 - r_o/r)}\right)^2}$ $A_F = (1 + \alpha_F)\sqrt{1 - \frac{1}{1 + \alpha_F}}$ $\alpha_F = \frac{h_o}{m\, r_o}$ m = 0.1 to 0.4
Shepelev, 1961	$e^{-\frac{1}{2}\left(\frac{y}{0.082X}\right)^2}$	$e^{-\frac{1}{2}\left(\frac{y}{0.1X}\right)^2}$	$e^{-\frac{1}{2}\left(\frac{y}{0.1X}\right)^2}$
Grimitlyn, 1994	$e^{-0.7(y/y_{0.5V})^2}$ $Y_{0.5V} = X \tan\alpha_{0.5V}$	$e^{-0.7(y/y_{0.5V})^2}$ $Y_{0.5V} = X \tan\alpha_{0.5V}$	$e^{-0.7(y/y_{0.5V})^2}$ $Y_{0.5V} = X \tan\alpha_{0.5V}$

4.3.3. Centerline velocity

Compact jet. Centerline velocity in the Zone 3 of the jet supplied can be calculated from the equations based on the principe of momentum conservation along the jet (Abramovich, 1948;

[1]The parameter **m** value depands upon the type of nozzle used in experiments

Loitzansky, 1973):

$$M_o = 2\pi\varrho\int_o^{y^*} V^2 y dy \tag{4.1}$$

where:

$M_o = \rho V_o^2 A_c$ - initial jet momentum, y^* - distance from the axis to the jet boundary.

Application of the Gauss error-function equation for velocity profile in the form proposed by Shepelev (Table 4.3) in Equation (4.1) results in the following formula for the centerline velocity in Zone 3 of the compact jet:

$$V_x = \frac{1}{\sqrt{\pi}c}\sqrt{\frac{M_o}{\rho_\infty}\frac{1}{x}} \tag{4.2}$$

Equation (4.2) can be also presented as follows:

$$V_x = \frac{\theta\varphi}{\sqrt{\pi}c}\frac{V_o\sqrt{A_o}}{X} \tag{4.3}$$

where:

$$\theta = \sqrt{\rho_o/\rho_\infty} = \sqrt{T_\infty/T_o}$$

$$\varphi = \left[\int_0^1\left(\frac{V}{V_o}\right)^2 d\left(\frac{A}{A_o}\right)\right]^{\frac{1}{2}}$$

- coefficient of velocities distribution at the diffuser outlet.

When outlet velocity distribution is uniform $\phi = 1$, V_o - average air velocity at the diffuser outlet ($V_o = Q_o / A_o$). Complex of coefficients $\theta\phi/\sqrt{\pi}c$ reflecting conditions of air supply has a constant value for a given situation and is called (Shepelev 1961) a dynamic characteristic of diffuser jet. Dynamic characteristic describes the intensity of velocity decay along the air jet axis:

$$K_1 = \frac{\theta\varphi}{\sqrt{\pi}c} \tag{4.4}$$

The above approach for the centerline velocity computation was utilized by different researchers using other velocity profiles and resulted the following equation

$$V_x = K_1 V_o \frac{\sqrt{A_o}}{X} \tag{4.5}$$

Theoretical values of characteristic $\mathbf{K_1}$ depends upon what type of velocity profile equation and supply conditions were assumed. E.g., according to Shepelev, $\mathbf{K_1} = 6.88$; $\mathbf{K_1} = 6.7$ (Rydberg and Norbäck 1949). Shlichting profile results in $\mathbf{K_1} = 7.4$ and with Tollmein profile $\mathbf{K_1} = 7.76$ (Kraemer 1971). According to experimental studies reported by Tuve (1953) the range of K_1 characteristic for the compact jet discharged from round outlets varies between 5.7 and 7 depending upon supply air velocity and type of the outlet. Analysis of experimental data from different researchers by Rodi (1982) indicates that the $\mathbf{K_1}$ is close to 7.

Some researches (e.g. Abramovich 1948; Baturin 1972; Rajaratnam 1976; Nielsen et al. 1988) consider X to be a distance starting from some point located at some distance X_o upstream from the diffuser face. Equations for the jet boundaries and velocity profile used in the centerline velocity derivation assume jet is supplied from the point source. Addition of the distance X_o to the distance from the outlet corrects the influence of the outlet size on the jet geometry. For practical reasons some researchers neglect X_o.

Linear jet. The equations for centerline velocities in a linear diffuser jet can be derived using the same principles as are applied in case with a compact jet. For linear jet:

$$V_x = K_1 \; V_o \sqrt{\frac{H_o}{X}} \tag{4.6}$$

where: $\mathbf{H_o}$ = the height of the slot. Similar to the case with compact air jet supply, theoretical values of characteristic K_1 depend upon what type of velocity profile equation and supply conditions were assumed. E.g., according to Shepelev, $\mathbf{K_1} = 2.62$. Görtler profile results in $\mathbf{K_1} = 2.43$ and with Tollmein profile $\mathbf{K_1} = 2.51$ (Kraemer 1971). Becher (1950) reported K_1 characteristic for linear jet to be equal to 2.55. Experimental results by Knystautas (1964), Heskestad (1965), Miller and Comings (1960), van der Hegge Zijnen (1958), Gutmark and Wygnanski (1976), and Kotsovinos and List (1977) appear to satisfy $\mathbf{K_1} = 2.43$.

Radial jet. Principle of momentum conservation applied by Koestel (1957) to the radial jet result in the following equation for the centerline velocity (Figure 4.3):

$$\frac{V_R}{V_o} = \frac{\sqrt{K\,(H_o/R_o)\;Cos\theta\;[K\,(H_o/R_o)\;Cos\theta\;+\;1]}}{\dfrac{\sqrt{R\,(R\,-\,R_o)}}{R_o}} \tag{4.7}$$

The value of the numerator of the right-hand side of Equation (4.7) depends on the geometric configuration of the outlet (Ro, Ho, Cos θ). The denominator represents the dimentionless distance from the outlet. For a given diffuser or a plaque Equation (4.7) becomes:

$$\frac{V_R}{V_o} = \frac{C\,R_o}{\sqrt{R\,(R - R_o)}} \tag{4.8}$$

Experimental **C** values for radial slot and radial nozzle tested by Koestel equal to 1.13 and 1.19 correspondingly. Equation (4.8) is similar to Equation (4.5) if the distance from the outlet is large enough so that (R - R_o) is approximately equal to R. Theoretical value of the characteristic K_1 in Equation (4.5) applied to a radial jets is equal to 1.05, according to Shepelev (1978).

Similar derivations by Regenscheit (1971) resulted in the following Equation for the radial jet center line velocity:

$$\frac{V_r}{V_o} = \frac{R_o}{R}\,\frac{A_F}{\sqrt{(1 + R_o/R}} \tag{4.9}$$

where:

$$A_F = \alpha_F\sqrt{1 += 1/\alpha_F}$$

$$\alpha_F = \frac{h_o}{R_o\,m}$$

Referring to the data from Baturin (1959), Regenscheight evaluated α_F equal to 0.377. At significant distance from the supply outlet (**R_o/R**→ 0) Equation (4.9) can be transferred into (4.5).

From equations (4.5) and (4.6) one can see, that air velocity along compact, linear and radial jets is proportional to the value of K_1 coefficient. This parameter depends upon (1) jet type; (2) diffuser type and (3) initial air velocity (ASHRAE, 1997) or Reynolds number (Vulis et al.,1969; Hanel et al.,1979; Müllejans, 1966).

4.3.4. Universal equations for velocity computation along the jets supplied from outlets with finite dimensions.

A fruitful approach for velocity computation along first three zones of jet supplied from the outlet with finite size was developed based on the hypothesis that momentum diffuses with distance from the source in the same manner as heat energy (Carslaw and Jaeger 1947, Vulis 1960). This approach developed in the papers by Elrod (1954), Shepelev and Gelman (1966) and Regenscheit (1981) utilizes the method of superposition of jet momentum from the multiples jet system. These jets originate from the point sources with supply air velocity equal to the average air velocity at the outlet of the finite dimensions. This approach utilizes the following principles:

1. Momentum conservation along the jet;
2. Air velocity in each jet cross-section is described using Reichardt Gauss error-function profile;
3. Constant angle of divergence along the jet.

This method being applied by Shepelev et al. (1966) to air supply through rectangular outlet with dimensions **2L x 2B** (Figure 4.4) results in the following equation for air velocities U(y,z) in the jet cross-section located at the distance **X** from the outlet:

$$V_{x,y,z} = \frac{V_o}{2}\sqrt{\left(erf\frac{L-Y}{cX} + erf\frac{L+Y}{cX}\right)\left(erf\frac{B-z}{cX} + erf\frac{B+Z}{cX}\right)} \tag{4.10}$$

where: erf = probability integral:

$$erf\ t \cong \frac{2}{\sqrt{\pi}}\int_0^t e^{-t^2}dt$$

From Equation (4.10) the centerline velocity can be calculated by substituting of **Y** = 0 and **Z** = 0:

$$V_x = V_o\sqrt{erf\frac{L}{cX} \times erf\frac{B}{cX}} \tag{4.11}$$

Equations (4.10) and (4.11) describe air velocity in cross-section of the jet located in Zone 1 through Zone 3. The shape of the outlet can be from square (**2B x 2B**) to infinite slot with a width **2B** (L = ∞). In the case of linear jet supplied through the slot, Equations (4.10) and (4.11) become as follows:

$$V_{x,y,z} = \frac{V_o}{\sqrt{2}}\sqrt{erf\frac{B-z}{cX} + erf\frac{B+Z}{cX}} \tag{4.12}$$

$$V_x = V_o\sqrt{erf\frac{B}{cX}} \tag{4.13}$$

The solution for a circular jet as presented by Elrod (1954) and Regenscheit (1981) cannot be evaluated in closed form. The exact solution can be received only for the centerline velocity:

$$\frac{V_x}{V_o} = 1 - \exp\left(\frac{-D^2}{4\ c^2X^2}\right) \tag{4.14}$$

4.3.5. Jet throw

Diffuser jet **throw**, L - a parameter commonly used in air diffuser sizing, is defined as a distance from the diffuser face to the jet cross-section where the centerline velocity equals to a terminal velocity $\mathbf{V_x}$ ($\mathbf{V_x}$ is often assumed to be equal to 0.25 m/s). Therefore, the throw (**L**) can be determined by velocity decay equations with $\mathbf{V_x}$ equals the terminal velocity:

$$L = K_1\sqrt{A_o}\frac{V_o}{V_x} \tag{4.15}$$

4.3.6. Entrainment ratio.

Entrainment ratio is another jet characteristic commonly used in air distribution design practice. Specifically, it is used in analytical multizonal models (see Chapter 9.4) when one needs to evaluate the total air flow rate transported by the jet to some distance from a diffuser face. Airflow rate in the jet Q_x can be derived by taking integral of air velocity profile within jet boundaries:

$$Q_x = 2\pi\int_o^{y^*} Vydy \tag{4.16}$$

Equations for airflow rate computation in compact, linear and radial jets are presented in Table 4.4.

Table 4.4. Airflow rate through a jet cross-sectional area.

Author	Q_x/Q_o		
	Compact jet	Linear jet (per 1m slot length)	Radial jet
Baturin, 1965	$0.275\,\frac{X - X_o}{\sqrt{A_o}}$	$0.530\sqrt{\frac{X - X_o}{H_o}}$	$.530\sqrt{(1 - R/R_o)\frac{R - R}{H_o}}$
Regenscheight, 1971	$1.77\,m\,\frac{X}{\sqrt{A_o}}$	$\frac{1}{\sqrt{m}}\sqrt{\frac{X}{H_o}}$	$1.96\,\frac{R_o\sqrt{1 - R_o/R}}{R_o}$
Shepelev, 1961	$0.29\,\frac{X}{\sqrt{A_o}}$	$0.43\sqrt{m}\sqrt{\frac{X}{H_o}}$	$0.069\,\frac{R}{\sqrt{R_oH_o}}$
Grimitlyn, 1994	$\frac{2}{K_1}\frac{X}{\sqrt{A_o}}$	$\frac{\sqrt{2}}{K_1}\sqrt{\frac{X}{H_o}}$	$\frac{\sqrt{2}}{K_1}\frac{X}{\sqrt{A_{oX}}}$

For a given area of diffuser opening A_o, the entrainment ratio is proportional to the distance X (for compact, radial and conical diffuser jets) or proportional to the square root of the distance X (for linear jet). For the same type of jet, the entrainment ratio is less with a large K_1 than with a small K_1. Radial and conical diffuser jets have smaller entrainment ratio than compact (incomplete radial) jets with a same K_1 value. Linear diffuser jets have smaller entrainment ratio than radial and conical jets.

4.4. Non-isothermal free jets

4.4.1. Temperature profile in a jet

Along with a constant velocity zone (Zone 1) there is a constant temperature zone in the jet. Heat diffusion in a jet is more intense than momentum, therefore the core of constant temperatures fades away faster than that of constant velocities and the temperature difference profile is flatter than the velocity profile. Thus the length of the zone with constant temperatures (Figure 4.5) is shorter than the length of the constant velocity zone (Zone 1) (Abramovich 1940, Koestel, 1954, Grimitlyn, 1970).

From the Tolmin's theory and experimental data (e.g., Reichardt 1944) the relation between velocity profile and temperature profile in the jet cross-section can be expressed using an overall turbulent Prandl number: **Pr** $= \nu_t/\alpha_t$, where ν_t is a turbulent momentum exchange coefficient and α_t is a turbulent heat exchange coefficient:

$$\frac{V}{V_m} = \left(\frac{t - \overline{t_\infty}}{t_m - \overline{t_\infty}} \right)^{1/Pr} \tag{4.17}$$

where: t_o - temperature of supplied air, t_∞- average air temperature of the surrounding air, **t** - air temperature at the point of consideration.

A Prandl number **Pr** of 0.7 has been suggested for non-isothermal jets by Nottage (1951), Forstall and Shapiro (1950) and Corrsin and Uberoi (1949), Grimitlyn (1965). Abramovich (1960) suggested Prandl number for a compact jet equal to 0.75, and for a linear jet - equal to 0.5.

According to Abramovich (1940), Regenscheight (1959) and Shepelev (1961), the relation between velocity distribution and temperature distribution in the cross section of non-isothermal compact, linear or radial jets within Zone 3 can be expressed as follows:

$$\Delta t/\Delta t_x = (t - \overline{t_\infty})/(t_x - \overline{t_\infty}) = \sqrt{V/V_x}\ . \tag{4.18}$$

and thus, the Prandl number is equal to 0.5. Table 4.5 lists some temperature profile equations as they are used for practical applications.

Table 4.5. Temperature profile equations

Author	$\dfrac{t - \overline{t_\infty}}{t_{xm} - \overline{t_\infty}}$		
	Round jet	Linear jet	Radial jet
Regenscheight, 1959, 1971	$e^{-24\left(\frac{r}{mX}\right)^2}$ m = 0.1 to 0.4	$e^{-\frac{\pi}{42}\left(\frac{Y}{mX}\right)^2}$ m = 0.1 to 0.4	$e^{-\frac{\pi}{42}\left(\frac{Y r_o A_F^2}{h_o r (1 - r_o/r)}\right)^2}$ $A_F = (1 + \alpha_F)\sqrt{1 - \frac{1}{1 + \alpha_F}}$ $\alpha_F = \frac{h_o}{m r_o}$ m = 0.1 to 0.4
Shepelev, 1961	$e^{-\frac{1}{42}\left(\frac{y}{0.082X}\right)^2}$	$e^{-\frac{1}{42}\left(\frac{y}{0.1X}\right)^2}$	$e^{-\frac{1}{42}\left(\frac{y}{0.1X}\right)^2}$
Grimitlyn, 1994	$e^{-0.7(y/y_{0.5tv})^2}$ $Y_{0.5t} = X \tan\alpha_{0.5t} = X \frac{tg\alpha_{0.5v}}{\sqrt{Pr}}$	$e^{-0.7(y/y_{0.5tv})^2}$ $Y_{0.5t} = X \tan\alpha_{0.5t} = X \frac{tg\alpha_{0.5v}}{\sqrt{Pr}}$	$e^{-0.7(y/y_{0.5tv})^2}$ $Y_{0.5t} = X \tan\alpha_{0.5t} = X \frac{tg\alpha_{0.5v}}{\sqrt{Pr}}$
Abramovich, 1984	$(1 - r/\delta^{3/2})^{1.4}$	$(1 - Yr/\delta^{3/2})$	-

4.4.2. Centerline temperature differential in a horizontally supplied jet

Compact jet. Centerline temperature differential within a zone of fully established turbulent flow (Zone 3) of a non-isothermal jet can be derived using equations of momentum (Equation (4.1)] and excessive heat conservation along the jet (Koestel 1954; Abramovich 1960; Shepelev 1961, etc.):

$$W_o = 2\pi C_p \rho_\infty \int_o^{y*} V(t-t_\infty) y dy \tag{4.19}$$

where:

$W_o = C_p \rho_\infty V_o (t_o - t_\infty) A_o$ - excessive heat in supplied air, C_p - specific heat,

Equation for the centerline temperature differential in Zone 3 of the compact jet derived (Shepelev, 1961) from the Equation (4.19) using the Gauss-error function temperature profile (Table 4.5) is as follows:

$$\Delta t_x = \frac{1+\sigma}{2\sqrt{\pi c}} \frac{W_o}{C_p\ \rho_\infty} \frac{1}{\sqrt{M_o/\rho_\infty}} \frac{1}{X} \tag{4.20}$$

Equation (4.20) also can be presented as follows:

$$\Delta t_x = \frac{(1+\sigma)\theta}{2\sqrt{\pi c \varphi}} \frac{\Delta t_o \sqrt{A_o}}{X} \tag{4.21}$$

where

$$\Delta t_o = T_o - T_\infty = \frac{W_o}{C_p \rho_o Q_o} \tag{4.22}$$

Complex of coefficients having constant value $(1+\sigma)\theta/2\sqrt{\pi c\phi}$ is called (Shepelev, 1961) a thermal characteristic of diffuser jet, $\mathbf{K_2}$, and characterizes the temperature decay along the air jet. Assuming a perfect mixing in the room (i.e., $\mathbf{t_{o.z.}} \approx \mathbf{t_\infty}$), $\mathbf{t_\infty}$ can be substituted for $\mathbf{t_{o.z}}$, and Equation (4.21) can be presented as follows:

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \frac{\sqrt{A_o}}{X} \tag{4.23}$$

As in the case of equations for the velocity decay computation, in the equations for the temperature decay computation, some researches consider **X** to be a distance starting from some virtual source located at some distance X_o from the diffuser face, others for practical reason neglect $\mathbf{X_o}$.

As in the case of $\mathbf{K_1}$ characteristic, theoretical values of characteristic K_2 depends upon supply conditions. According to Shepelev (1961), in the case of air supply through the nozzle with a uniform outlet velocity profile $\mathbf{K_2 \times K_1 = (1 + Pr)/(2\pi\, 0.082^2)}$. Thus, when $\mathbf{K_1}$ = 6.88 and **Pr** = 0.7, $\mathbf{K_2}$ = 5.85. Grimitlyn (1994) suggest the following relation between $\mathbf{K_2}$ and $\mathbf{K_1}$

coefficients: $K_2 = \sqrt{\frac{1 + Pr}{2}} K_1$. According to Helander's unpublished data (progress Report, Downward Projection of Heated Air, January 6, 1951 referenced by Koestel 1954), $\mathbf{K_2} = 6$.

Linear jet. Derivation of equation for the centerline temperature differential in a linear jet is based on the same principles that are used in case of a compact jet. For the linear diffuser jet centerline temperature differential can be computed from the following equation:

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \sqrt{\frac{H_o}{X}} \tag{4.24}$$

Centerline temperature differential in the Zone 3 of diffuser jet is proportional to the value of $\mathbf{K_2}$ coefficient, which as the $\mathbf{K_1}$ coefficient depends upon jet and diffuser types.
and supply conditions. Theoretical value of $\mathbf{K_2}$ coefficient according to Shepelev (1961) is 2.49. Experimental data reported by Grimitlyn (1994) show $\mathbf{K_2}$ value to be 2.0

Radial jet. Equation for the centerline temperature differential in a radial and in a conical jets (Figure 4.3) is derived in a same way as for a compact and a linear jets (Shepelev 1961) and is similar to Equation (4.21):

$$\frac{t_x - t_\infty}{t_o - t_\infty} = \sqrt{\frac{1 + Pr}{4\sqrt{\pi c}\ Sin\alpha}\ \frac{\theta}{\varphi}\ \frac{1}{\sqrt{\beta}}}\ \frac{\sqrt{A_o}}{X} \tag{4.25}$$

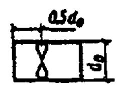
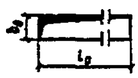
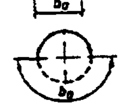
The complex of parameters $\sqrt{\frac{1 + Pr}{4\ \pi\ \sqrt{\pi c}\ Sin\alpha}\ \frac{\theta}{\varphi}\ \frac{1}{\sqrt{\beta}}}$ is a thermal characteristic, $\mathbf{K_2}$. In the case of a radial jet $\beta = 2\pi$, $\alpha = 90°C$. Assuming $\theta = 1$, $\varphi = 1$, $\mathbf{c} = 0.082$ and $\mathbf{Pr} = 0.5$, $\mathbf{K_2} = 1$. For conical jet with $\alpha = 60°C$ ($\beta = 2\pi$, $\theta = 1$, $\varphi = 1$, $\mathbf{c} = 0.082$ and $\mathbf{Pr} = 0.5$), $\mathbf{K_2} = 1.07$. Table 4.6 lists some equations used for centerline temperature differential computation in horizontal jets.

According to Shepelev (1978) the theoretical values of the $\mathbf{K_2/K_1}$ ratio for air supply through the nozzles with the uniform velocity distribution at the outlet cross-section is 0.9 for the compact jet and 0.95 for radial, conical and linear jets. Practically for different types of air diffusers this ratio can vary from 0.7 to 3.0 (see Table 4.7).

Table 4.6. Centerline temperature differential in horizontal jets

Author	$\frac{t_x - t_\infty}{t_o - t_\infty}$		
	Round jet	Linear jet	Radial jet
Regenscheight, 1959	$\frac{3}{4}\frac{D}{mX}$ m = 0.1 to 0.4	$\sqrt{\frac{3\,D}{4\,m\,X}}$ m = 0.1 to 0.4	-
Shepelev, 1961	$K_2\frac{\sqrt{A_o}}{X}$	$\frac{t_x-t_{o.z}}{t_o-t_{o.z}} = K_2\sqrt{\frac{H_c}{X}}$	$K_2\frac{1}{\sqrt{\beta\ Sin\ \alpha\ \beta}}\frac{\sqrt{A_o}}{X}$
Grimitlyn, 1994	$K_2\frac{\sqrt{A_o}}{X}$	$\frac{t_x-t_{o.z}}{t_o-t_{o.z}} = K_2\sqrt{\frac{H_c}{X}}$	$K_2\frac{\sqrt{A_o}}{X}$
Abramovich, 1940	$\frac{0.7}{\frac{aX}{R_o}+0.2941}$ a = 0.66 - 0.76	$\frac{1.04}{\sqrt{\frac{aX}{B_o}+0.41}}$ a = 0.09 - 0.12	-
Weidemann and Hanel, 1988	$4.2\frac{D}{X}$ @ $6 < X/D < 15$ $5.55\frac{D}{X} - 0.085$ @ $6<X/D<15$	-	-
Koestel, 1954	$5.35\frac{D}{X}$	-	-
Baturin, 1965	$\frac{9.24\,R_o}{X - X_o}\sqrt{\frac{T_\infty}{T_o}}$	$3.27\sqrt{\frac{B_o}{X}}\sqrt{\frac{T_\infty}{T_o}}$	

Table 4.7. Velocity and temperature decay coefficients for generic types of air diffusers manufactured in Russia (reproduced from Designer's Guide, 1992)[1]

Type of air diffuser	Schematic	K_1	K_2	ξ
Grilles PB type				
$\beta = 0$		**6.3**	**5.1**	**1.2-1.3**
$\beta = 90°$		**2**	**1.7**	**1.2-1.3**
Swirling type BEC or axial fan				
$X/\sqrt{A_o}$ = 8-14		**2.5**	**2.4**	**5.2**
$X/\sqrt{A_o}$ > 14		**2.8**	**2.7**	**5.2**
Rectungular opening (B_o/H_o > 10 ,				
$X > 6 B_o$		**2.5**	**2.0**	**1.8**
Ceiling mounted				
Multidiffuser type:				
PRM_k (round)				
radial attached jet		**1.2**	**1.0**	**1.4**
conical jet		**1.0**	**0.8**	**1.4**
conical-compact jet		**1.0-3.2**	**0.8-3.3**	**1.4**
PRM_p (square)				
attached jets		**1.8-2.8**	**2.0-3.6**	**1.7**
free jets		**1.0-1.7**	**1.3-2.6**	**1.3**
With a disk BDPM (h_o = 0.3 D_o)				
(attached radial jet)		**1.2**	**1.1**	**2.1**
With perforated disk BDPM				
(free radial jet)		**1.5**	**3.2**	**1.5**
Swirling type BTs		**0.7**	**1.0**	**2.5**
Perforated duct (downward jet):				
rectungular		**2.1**	**1.7**	**2.4**
round		**0.5**	**1.5**	**2.4**

[1] K_1 and K_2 are diffuser jet coefficients characterizing respectively velocity and temperature decay along the jet axis; ξ is the diffuser pressure loss coefficient.

4.4.3. Universal equations for temperature difference computation along the jets supplied from outlets with finite dimensions.

Shepelev and Gelman (1966) and Regenscheight (1981) computed air temperature along the first three zones of jet supplied from the outlet with finite size using the method of superposition of the multiples jet system. These jets originate from the point sources with supply air velocity equal to the average air velocity at the outlet of the finite dimensions.

Along with principles described in Section 4.3.4., this approach utilizes the following equations describing temperature distribution in a compact jet and a heat flux at a given point (**X,Y,Z**):

$$\frac{t - t_\infty}{t_o - t_\infty} = \frac{1 + Pr}{2 \times 0.082\sqrt{\pi}} \frac{\sqrt{A_o}}{X} e^{-\frac{Pr}{2}\left(\frac{r}{0.082X}\right)^2} \tag{4.26}$$

$$\frac{V(t - t_\infty)}{V_o(t_o - t_\infty)} = \frac{1 + Pr}{2\pi\, 0.082^2} \frac{A_o}{X^2} e^{-\frac{1+Pr}{2}\left(\frac{r}{0.082\,X}\right)^2} \tag{4.27}$$

The heat flux through the finite element **dA** of the jet cross-section at the distance **X** from the outlet can be calculated as:

$$\frac{d[V(t - t_\infty)]}{V_o(t_o - t_\infty)} = \frac{1 + Pr}{2\pi\, 0.082^2} \frac{dA_o}{X^2} e^{-\frac{1+Pr}{2}\left(\frac{r}{0.082\,X}\right)^2} \tag{4.28}$$

Double integral of Equation (4.28) across the outlet area **2A x 2B** results in the following Equation for a heat flux through a given point of a jet supplied through a rectangular outlet:

$$\frac{V_{XYZ}(t_{XYZ} - t_\infty)}{V_o(t_o - t_\infty)} = \frac{1}{4}\left(erf\sqrt{\frac{1+Pr}{2}}\frac{Y+A}{0.082\,X} - erf\sqrt{\frac{1+Pr}{2}}\frac{Y-A}{0.082\,X}\right) \times \left(erf\sqrt{\frac{1+Pr}{2}}\frac{Z+B}{0.082\,X} - erf\sqrt{\frac{1+Pr}{2}}\frac{Z-B}{0.082\,X}\right) \tag{4.29}$$

Joint solution of Equations (4.19) and (4.29) results in the following equation for temperature differential along the jet axis:

$$\frac{t_{XYZ} - t_\infty}{t_o - t_\infty} = \frac{erf\sqrt{\frac{1+Pr}{2}}\frac{A}{0.082\,X} \times erf\sqrt{\frac{1+Pr}{2}}\frac{B}{0.082\,X}}{\sqrt{erf\frac{A}{0.082\,X}\; erf\frac{\sqrt{B}}{0.082\,X}}} \tag{4.30}$$

In the case when air is supplied through the slot with a width **2B** (**A** = ∞), Equation (4.30) can be converted into the following:

$$\frac{t_{XYZ} - t_\infty}{t_o - t_\infty} = \frac{erf\sqrt{\frac{1 + Pr}{2}}\frac{B}{0.082\ X}}{\sqrt{erf\frac{B}{0.082\ X}}} \tag{4.31}$$

4.4.4. Velocities and temperatures in vertical non-isothermal jet

Studies by Helander et al. (1948,1953,1954,1957), Knaak (1957), Koestel (1954), Shepelev (1961), Regenscheit (1970), Grimitlyn (1970) resulted in equations for downward and upward projected diffuser jets.

For the circular jet Regenscheit (1970) obtained the following empirical equation for the maximum velocity in the downward and upward vertical jets of heated and cooled air:
For compact (round) jet:

$$\frac{V_x}{V_o} = \frac{m\sqrt{A_o}}{X} \pm \sqrt{\frac{Ar_o}{m}\left[1 + \ln\frac{2X}{m\sqrt{A_o}}\right]} \tag{4.32}$$

where **m** = parameter characterizing diffuser jet: **m** from 0.1 to 0.3.
For linear jet:

$$\frac{V_x}{V_o} = \sqrt{\frac{H_o}{X}} \pm \sqrt{\frac{Ar_o}{0.2}\left[2.83\sqrt{\frac{X}{H_o}} - 1\right]} \tag{4.33}$$

Based on the theoretical analyses Koestel (1954), Shepelev (1961) and Grimitlyn (1970) developed equations for velocities and temperatures in vertical heated and chilled air jets. The assumptions used by these authors are similar and the method used is described in (Koestel, 1954). The assumptions used in the analysis can be summarized as follows:

1. The jet of warm or cooled air is projected into an unbounded atmosphere of still air of uniform temperature;
2. The only force opposing the downward flow of the heated air or upward flow of the cooled air is a buoyancy force. In their analysis, Hellander et al. (1948) also suggested to account for inertial forces due to the entrainment of room air. However, this suggestion is not in an agreement with a principle of momentum conservation used in most of the existing models for isothermal jets;
3. The air entrained by the jet has a room air temperature.
4. A velocity profile and a temperature-difference profile have shapes that can be approximated by an error-function type curve.

For practical use the influence of buoyancy forces on temperature and velocity decay in vertical non-isothermal jet, as proposed by Grimitlyn (1982), can be considered by the coefficient K_n of non-isothermality.

For compact jets:

$$\frac{V_x}{V_o} = K_1 \frac{\sqrt{A_o}}{X} K_n \tag{4.34}$$

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \frac{\sqrt{A_o}}{X} \frac{1}{K_n} \tag{4.35}$$

where K_n for a compact jet can be computed using as:

$$K_n = \sqrt[3]{1 \pm 2.5 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o}} \right)^2} \tag{4.36}$$

For linear jets:

$$\frac{V_x}{V_o} = K_1 \sqrt{\frac{H_o}{X}} K_n \tag{4.37}$$

$$\frac{t_x - t_{o.z}}{t_o - t_{o.z}} = K_2 \sqrt{\frac{H_o}{X}} \frac{1}{K_n} \tag{4.38}$$

where:

$$K_n = \sqrt[3]{1 \pm 1.8 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{H_o} \right)^{1.5}} \tag{4.39}$$

The throw of downward projected heated jets or upward projected chilled air jets one can derive from the equations (4.34) and (4.37) considering $\mathbf{K_n}$ equal to some value, e.g. 0.1. Helander and his associates (1948), in their work on heated jets projected downward, have called attention to some of the differences between the actual conditions and those assumed for analysis. One of these is the radial escape of warm air in the terminal zone of a hot stream projected downward. This escaping warm air then rises and causes a change in ambient conditions for the upper part of the jet. The terminal zone and also the edge of the jet are zones of marked instability, with definite surges and fluctuations, so that the jet envelope is very difficult to define or to determine experimentally. In the closure to the paper presented by Knaak (1957), Dr. Helander suggested that from the point of view of practical application, the distance to the beginning of the unstable, terminal zone of the jet is about 80% of the jet throw (see summary in Table 4.8).

Table 4.8. Maximum downward/upward travel of heated/cooled jet

Author	Equation	Comments
Baturin and Shepelev, 1935	$\frac{Z^{max}}{D} = \frac{2}{3}\sqrt{\frac{1}{0.06Ar_o}\sqrt{\frac{T_{room}}{T_o}}}$	$Ar_o = \frac{g\ D}{V_o^2}\frac{T_o - T_{room}}{T_{room}}$
Helander, 1953	$\frac{Z^{max}}{D} > 8: \quad Z_{max} = 1.66 B_o^{1/2} \pm 10\%$ $\frac{Z^{max}}{D} \le 8: \quad Z_{max} = 0.36 B_o \pm 10\%$	$B_o = \frac{V_o^2}{g\ D}\frac{T_{room}}{T_o - T_{room}}$
Koestel, 1954	$\frac{Z^{max}}{D_o} = 3.4\left(\frac{V_o^2}{g\ D}\frac{T_{room}}{T_o - T_{room}}\right)^{1/2} - 2.85$	
Regenscheit, 1959	*for a compact jet:* $\frac{Z_{max}}{D} = 1.63\sqrt{\frac{1}{m}\frac{1}{Ar_o}}$ *for a linier jet:* $\frac{Z_{max}}{H_o} = 1.1\sqrt[3]{\frac{1}{m}\frac{1}{Ar_o^2}}$	$Ar_o = \frac{g\ D}{V_o^2}\frac{T_o - T_{room}}{T_{room}}$ $Ar_o = \frac{g\ H_o}{V_o^2}\frac{T_o - T_{room}}{T_{room}}$ m = 0.1 to 0.4
Shepelev, 1961	*for a compact jet:* $\frac{Z_{max}}{\sqrt{A_o}} = 0.53\frac{H}{\sqrt{A_o}}$ *for a linier jet:* $\frac{Z_{max}}{H_o} = 0.6\frac{H}{H_o}$	$H = \sqrt{\frac{K_1^2}{K_2}\frac{V_o^2\sqrt{A_o}}{g}\frac{T_{room}}{T_o - T_{room}}}$ $H = \left(\frac{K_1^2}{K_2}\frac{V_o^2\sqrt{H_o}}{g}\frac{T_{room}}{T_o - T_{room}}\right)^{2/3}$

Turner, 1973	*for a compact jet:* $\frac{Z_{max}}{D} = 1.74\frac{1}{\sqrt{Ar_o}}$	$Ar_o = \frac{g\ D}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$
Seban, et al., 1978	*for a compact jet:* $\frac{Z_{max}}{D} = 1.75\frac{1}{\sqrt{Ar_o}}$	$Ar_o = \frac{g\ D}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$
Sato et al., 1981	*for a compact jet:* $\frac{Z_{max}}{D} = 1.98\frac{1}{\sqrt{Ar_o}}$	$Ar_o = \frac{g\ D}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$
Grimitlyn, 1982	*for a compact jet:* $\frac{Z_{max}}{D} = \frac{0.63\ K_1}{\sqrt{K_2\ Ar_o}}$ *for a linier jet:* $\frac{Z_{max}}{H_o} = 0.67\ \frac{K_1^{4/3}}{(K_2\ Ar_o)^{2/3}}$	$Ar_o = \frac{g\ D}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$ $Ar_o = \frac{g\ H_o}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$
Mizuchina, 1982	*for a compact jet:* $\frac{Z_{max}}{D} = 1.66\frac{1}{\sqrt{\frac{1}{m}\ \frac{1}{Ar_o}}}$	$Ar_o = \frac{g\ D}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$
Weidemann and Hanel, 1988	*for a compact jet:* $\frac{Z_{max}}{D} = 1.59\frac{1}{\sqrt{Ar_o}}$	$Ar_o = \frac{g\ D}{V_o^2}\ \frac{T_o - T_{room}}{T_{room}}$

4.4.5. Trajectory of a horizontal and inclined jet

Buoyancy forces influence the trajectory of horizontally projected air jets or air jets supplied under some angle to the horizontal plane (Figure 4.6). Most non-isothermal air jets studies were devoted to horizon-tally projected compact air jets. Based on the analytical studies (Abramovich, 1960; Shepelev, 1978; Nosovitsky and Posokhin, 1966; Omelchuk, 1966, Filney and Nosovitsky, 1967; Fleischhacker and Schneider, 1980), the trajectory axis of inclined jets can be described by a polynomial function as follows:

$$\frac{Z}{\sqrt{A_o}} = \psi \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o}}\right)^3 . \tag{4.40}$$

The same type of equations were also suggested by experimental studies (Stein, 1953; Koestel, 1955; Grimitlyn, 1969). In some papers, authors suggest to determine the trajectory axis from equations of another kind, parabola for example (Lyakhovsky and Syrkin, 1939; Frean and Billington, 1955).

In some studies on inclined air jets, the equation for the trajectory differ from (4.40) by the additional term as follows:

$$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}} tg\alpha_o \pm \psi \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o}}\right)^3 \tag{4.41}$$

Taliev (1969) and Schneider (1975) received the equations for the trajectories by numerical methods. Experimental data for the trajectory of the inclined jet ($\alpha_o \neq 0$) were obtained only by Fleishbacker and Schneider (1975).

As it was shown by Zhivov (1993) the main difference in most of the equations for the jet trajectory is the value of the coefficient Ψ (Table 4.9). The difference in experimental data obtained by different authors are mainly due to the difficulties in the measurements of non-isothermal air jets supplied with low initial velocities (2 - 10 m/s). There is also a different understanding of the term "air jet trajectory". Some authors mean the geometrical place of the points with maximum velocities values, while others mean geometrical place of the centers of gravity of the cross sections of the jet.

Analytical method of jets trajectory study developed by Shepelev (1961) allows to receive several other useful features and is worth-while to be described. On the schematic of non-isothermal jet supplied under some angle α_o to the horizon (Figure 4.7), **S** is the jet's axis, **X** - horizontal axis and **Z** - vertical axis. Ordinate of the trajectory of this jet can be described as $\mathbf{Z = X\ tg\alpha + \Delta Z}$, where $\mathbf{\Delta Z}$ is jets raise due to buoyancy forces. To evaluate $\mathbf{\Delta Z}$ the elementary volume **dW** with a mass equal to $\mathbf{dm = \rho_s dV}$ on the jets trajectory was considered. The buoyancy force influencing this volume can be described as $\mathbf{dP = g\ (\varrho_\infty - \varrho_s)}$. Vertical acceleration of the volume under the consideration is $\mathbf{j = dP/dm = g(\varrho_\infty - \varrho_s)/\varrho_s \approx g\ (T_s - T_\infty)/T_s}$. Vertical acceleration can be presented with a help of the vertical velocity component $\mathbf{j = dV_z /d\tau}$, where time interval $\mathbf{d\tau}$ can be described as $\mathbf{dS/V_s}$. Based on these equations vertical component of air velocity can be presented as

$$V_z = \frac{g}{T_\infty} \int_o^s \frac{T_s - T_\infty}{V_s} dS \tag{4.42}$$

Ratio of local temperature difference and velocity on the jet's axis in (4.42) can be substituted by

$$\frac{T_s - T_\infty}{V_s} = \frac{K_2}{K_1}\frac{g}{T_\infty}\frac{T_o - T_\infty}{V_o}$$

resulting in the following:

$$V_z = \frac{K_2}{K_1}\frac{g}{T_\infty}\frac{T_o - T_\infty}{V_o} S \tag{4.43}$$

Considering that $\mathbf{V_z = dZ/d\tau}$, $\mathbf{V_z/V_s = dZ/dS}$ and $\mathbf{V_s = K_1V_o\sqrt{A_o}/S}$, the equation for calculating $\mathbf{\Delta Z}$ can be rewritten as

$$\Delta Z = \frac{K_2}{K_1}\frac{G}{T_\infty}\frac{T_o - T_\infty}{V_o}\int_o^s \frac{SdS}{V_S} \tag{4.44}$$

or

$$\Delta Z = \frac{K_2\, g\, (T_o - T_\infty)}{3\, K_1^2\, T_\infty\, V_o^2 \sqrt{A_o}} S^3 \tag{4.45}$$

Substituting $\mathbf{S}$ by $\mathbf{XCos\alpha_o}$ and complex of parameters by $\mathbf{Ar_o}$ the resulting equation for the trajectory can be as follows:

$$Z = Xtg\alpha_o \pm \frac{1}{3}\frac{K_2}{K_1^2}\frac{Ar_o}{A_o}\left(\frac{X}{Cos\alpha_o}\right)^3 \tag{4.46}$$

When chilled air jet is supplied at the angle α_o upwards it will cross the level of the supply outlet at the distance of $\mathbf{X_o}$. This distance can be calculated by substitution $\mathbf{Z = 0}$ in (4.46)

$$X_o = \frac{\sqrt{3}\, K_1 \sqrt{A_o}\, Cos\alpha_o \sqrt{Sin\alpha_o}}{\sqrt{K_2\, Ar_o}} \tag{4.47}$$

The abscissa and ordinate of the jet vortex in the case of inclined cold air jet supply upwards or inclined warm air supply downwards was derived from the equation (4.41) are described by equations (4.48) and (4.49), respectively:

$$\frac{X_v}{\sqrt{A_o}} = \frac{K_1\, Cos\alpha_o \sqrt{Sin\alpha_o}}{\sqrt{K_2\, Ar_o}} \tag{4.48}$$

$$\frac{Z_v}{\sqrt{A_o}} = \frac{2}{3}\frac{K_1\, (Sin\alpha_o)^{3/2}}{\sqrt{K_2\, Ar_o}} \tag{4.49}$$

The ratio ordinate $\mathbf{X_v}$ to $\mathbf{Z_v}$ depends only on $\mathbf{tg\alpha_o}$: $\mathbf{Z_v/X_v = 2/3\ tg\alpha_o}$ and the ratio of $\mathbf{X_v/X_o}$ has a constant value equal to 0.578.

To clarify the trajectory equation of inclined jets for the cases of air supply through different types of nozzles and grilles a series of experiments were conducted (Zhivov, 1993). The trajectory coordinates were defined as the geometrical places of the points where the mean values of the temperatures and velocities reached their maximum in the vertical cross sections of the jet. Values of coefficient Ψ received in these experiments, together with data of other authors, are presented in Table 4.10.

It is important to mention that, in such experiments, one meets with a number of problems such as deformation of temperature and velocity profiles and fluctuation of the air jet trajectory, which reduces the accuracy in the results of measurements (Zhivov, 1993). The mean value of the coefficient Ψ obtained from experimental data (Figure 4.7) is 0.47 ± 0.06. Thus the trajectory of the non-isothermal jet supplied through different types of the outlets can be calculated from the equation

$$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}} tg\alpha_o \pm 0.47 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\cos\alpha_o \sqrt{A_o}}\right)^3 \tag{4.50}$$

The accuracy of this value is sufficient (Figure 4.8) to be used in designing the trajectory of inclined ventilation jets at an angle $\boldsymbol{\alpha_o \leq \pm 45°}$. Considering experimental value of the coefficient the equation for the vortex abscissa $\mathbf{X_v}$ can presented as follows

$$X_v = \frac{\cos\alpha_o \sqrt{|\sin\alpha_o|}\ \sqrt{A_0}}{\sqrt{3\times 0.47\ (K_2/K_1^2)\ Ar_o}} \tag{4.51}$$

However, this clarification does not effect the ordinate to abscissa ratio, which remains equal to 0.578.

For the non-isothermal linear air jet trajectory equation derived by Shepelev (1961) is as follows:

$$\frac{Z}{H_o} = \frac{X}{H_o} tg\,\alpha_o \pm 0.4 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{H_o}\right)^{\frac{5}{2}} \tag{4.52}$$

Table 4.9. Compact Air Jet Trajectory Equations

Author	Equations for Axisymmetric Non-isothermal Jet		Comments
	Original	Transformed	
Baturin, 1965	$\frac{Z}{D_o} = 0.052 \frac{g D_o}{V_o^2} \frac{\Delta t_o}{T_\infty} \left(\frac{X}{D_o}\right)^3$	$\frac{Z}{\sqrt{A_o}} = 0.52 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o}}\right)^3$	$\alpha_o = 0; K_1 = 6.6;$ $K_2 = 4.9$;nozzle.
Shepelev, 1961	$Z = X\, tg\alpha_o \pm \frac{X^3}{3\, H^2\, Cos^3\alpha_o}$	$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}} tg\alpha_o \pm 0.336 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o} Cos\alpha_o}\right)^3$	$H = \frac{5.45 V_o K_1 \sqrt[4]{A_o}}{\sqrt{K_2\, \Delta t}}$, T_∞=293 K.
Nosovitsky and Posokhin, 1966	$Z = X\, tg\alpha_o \pm \frac{2}{3} \frac{X^3}{H^2 Cos^3\alpha_o}$	$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}} tg\alpha_o \pm 0.67\, Ar_o \left(\frac{X}{\sqrt{A_o}\, \cos\alpha_o}\right)^3$	$H = \frac{5.45 V_o\, K_1 \sqrt[4]{A_o}}{\sqrt{K_2\, \Delta t}}$ T_∞=293 K, nozzle.
Grimitlyn, 1969	$\frac{Z}{D_o} = 0.6 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{D_o}\right)^3$	$\frac{Z}{\sqrt{A_o}} = 0.47 \frac{K_2}{K_1^1} Ar_o \left(\frac{X}{\sqrt{A_o}}\right)^3$	$\alpha_o = 0$.
Baturin, 1965	$\frac{Z}{r_o} = \frac{X}{r_o} tg\alpha_o \pm 0.02\, Ar_o \sqrt{\frac{T_o}{T_\infty}} \left(\frac{X}{r_o Cos\alpha_o}\right)^3$	$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}} tg\alpha_o \pm 0.32 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o} Cos\alpha_o}\right)^3$	$\sqrt{\frac{T_o}{T_\infty}} \sim 1$, K_1=6.6, K_2=4.9, nozzle.
Lyakhovsky and Syrkin, 1939	$\frac{Z}{D_o} = 0.35\, Ar_o \left(\frac{X}{D_o}\right)^2$	$\frac{Z}{\sqrt{A_o}} = 2.76 \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o}}\right)^3$	$\alpha_o = 0; K_1 = 6.6;$ $K_2 = 4.9$;nozzle.

Omelchuk, 1966	$\frac{Z}{D_o}=\frac{X}{D_o}tg\alpha_o \pm 0.064\ Ar_o \sqrt{\frac{T_o}{T_\infty}}\left(\frac{X}{D_o Cos\alpha_o}\right)^3$	$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}}tg\alpha_o \pm 0.5\frac{K_2}{K_1^2}Ar_o\left(\frac{X}{\sqrt{A_o}\ Cos\alpha_o}\right)^3$	$\sqrt{\frac{T_o}{T_\infty}} \sim 1$, K_1=6.6, K_2=4.9, nozzle.
Filney and Nosovitsky, 1967	$\frac{Z}{\sqrt{A_o}} = 0.0565\ Ar_o \sqrt{\frac{T_o}{T_\infty}}\ \beta^{-1.5}\left(\frac{X}{\sqrt{A_o}}\right)^3$	$\frac{Z}{\sqrt{A_o}} = 0.46\ \frac{K_2}{K_1^2}\ Ar_o\left(\frac{X}{\sqrt{A_o}}\right)^3$	$\sqrt{\frac{T_o}{T_\infty}} \sim 1$, K_1=6.6, K_2=4.9, nozzle.
Stein, 1953	$\frac{Z}{D_o} = 0.056\ Ar_o\left(\frac{X}{D_o}\right)^3$	$\frac{Z}{\sqrt{A_o}} = 0.42\ \frac{K_2}{K_1^2}\ Ar_o\left(\frac{X}{\sqrt{A_o}}\right)^3$	$\alpha_o = 0; K_1 = 6.5; K_2 = 5.0$;nozzle.
Koestel, 1955	$\frac{Z}{\sqrt{A_o}} = 0.046\ Ar_o\left(\frac{X}{\sqrt{A_o}}\right)^3$	$\frac{Z}{\sqrt{A_o}} = 0.49\ \frac{K_2}{K_1^2}\ Ar_o\left(\frac{X}{\sqrt{A_o}}\right)^2$	$\alpha_o = 0; K_1 = 6.5$; $K_2 = 5.0$;nozzle.
Frean and Billington, 1955	$\frac{Z}{\sqrt{A_o}} = 0.238\ Ar_o\left(\frac{X}{\sqrt{A_o}}\right)^{2.5}$	$\frac{Z}{\sqrt{A_o}} = 2.2\ \frac{K_2}{K_1^2}\ Ar_o\left(\frac{X}{\sqrt{A_q}}\right)^{2.5}$	$\alpha_o = 0; K_1 = 7.5$; $K_2 = 6.1$; square opening in sheet metal.
Fleischhacker and Schneider, 1980	$\frac{Z}{\sqrt{A_o}} = \frac{X}{\sqrt{A_o}}tg\alpha_o \pm 0.0449Ar_o\left(\frac{X}{\sqrt{A_o}Cos\alpha_o}\right)^3$	$\frac{Z}{\sqrt{A_o}}=\frac{X}{\sqrt{A_o}}tg\alpha_o \pm 0.37\frac{K_2}{K_1^2}Ar_o\left(\frac{X}{\sqrt{A_o}Cos\alpha_o}\right)^3$	$K_1 = 6.5; K_2 = 5.1$; round tube.

Table 4.10. Data from jet trajectory experimental studies

Reference	Data	Outlet Type	D_o, $a_o \times b_o$, (m)	$\sqrt{A_o}$, (m)	K_1	K_2	α_o, (deg)	Δt_o, (ΨC)	V_o, (m/s)	Ar_o	Ψ
Grimitlyn,	1	grill	0.05 × 0.05	0.05	1.8	1.7	0	87.8	3.6	0.0113	0.46
Zhivov and	2	grill	0.05 × 0.05	0.05	1.8	1.7	0	63.5	4.9	0.0044	0.42
Kelina,	3	grill	0.05 × 0.05	0.05	1.8	1.7	0	63.3	6.0	0.0029	0.36
1988	4	grill	0.05 × 0.05	0.05	1.8	1.7	0	58.4	6.9	0.0021	0.40
	5	grill	0.05 × 0.05	0.05	1.8	1.7	-30	50.0	4.2	0.0047	0.45
	6	grill	0.05 × 0.05	0.05	1.8	1.7	-30	46.4	6.5	0.0018	0.46
	7	grill	0.05 × 0.05	0.05	1.8	1.7	-45	66.3	3.9	0.0073	0.57
	8	grill	0.05 × 0.05	0.05	1.8	1.7	-45	63.3	4.9	0.0044	0.39
	9	grill	0.05 × 0.05	0.05	1.8	1.7	-45	55.7	6.1	0.0025	0.44
	10	grill	0.05 × 0.05	0.05	6.3	1.7	-45	55.4	6.5	0.0022	0.47
	11	nozzle	0.02	0.018	6.3	5.0	0	69.8	7.9	0.0007	0.42
	12	nozzle	0.02	0.018	6.3	5.0	-30	57.2	9.1	0.0005	0.45
	13	nozzle	0.02	0.018	6.3	5.0	-30	60.8	11.6	0.0003	0.53
	14	nozzle	0.02	0.018	6.3	5.0	-45	77.9	10.3	0.0005	0.54
	15	nozzle	0.04	0.035	6.3	5.0	0	79.2	6.4	0.0026	0.56
	16	nozzle	0.04	0.035	6.3	5.0	-30	49.5	4.1	0.0039	0.39
	17	nozzle	0.04	0.035	6.3	5.0	-30	48.5	4.6	0.0031	0.49
	18	nozzle	0.04	0.035	6.3	5.0	-30	46.5	5.0	0.0025	0.54
	19	nozzle	0.04	0.035	6.3	5.0	-30	62.5	6.3	0.0021	0.57
	20	nozzle	0.04	0.035		5.0	-45	79.5	6.3	0.0027	0.51
Frean and Billington, 1955	21	rectang. tube	0.102 × 0.102	0.102	5.3	3.4	0	22.2	2.5	0.0118	0.58
Lyakhovsky	22	nozzle	0.05	0.044	6.6	4.9	0	136	6.4	0.0049	0.27
and Syrkin,	23	nozzle	0.1	0.089	6.6	4.9	0	133	5.4	0.0134	0.50
1939	24	nozzle	0.1	0.089	6.6	4.9	0	155	3.9	0.0293	0.53
	25	nozzle	0.2	0.177	6.6	4.9	0	119	2.3	0.1307	0.64
Fleischhack	26	nozzle	0.032	0.028	6.5	5.1	-30	75.6	3.6	0.0055	0.44
er	27	nozzle	0.032	0.028	6.5	5.1	-40	75.6	3.6	0.0055	0.43
and	28	nozzle	0.032	0.028	6.5	5.1	-20	75.6	3.6	0.0055	0.46
Schneider,	29	nozzle	0.032	0.028	6.5	5.1	0	75.6	3.6	0.0055	0.41
1980	30	nozzle	0.032	0.028	6.5	5.1	+30	75.6	3.6	0.0055	0.41
Stein, 1953	31	nozzle	0.047	0.042	6.5	5.1	0	45.5	5.7	0.0020	0.45

4.4.6. Jet attachment

Jet discharging close to the plane of the ceiling or the wall is a common case in ventilation practice. The presence of adjacent surface restricts air entrainment from the side of this surface. This results in a pressure difference across the jet, which curves towards the surface. The curvature of the jet increases until it attaches to the surface. This phenomenon is usually referred to as a "Coanda" effect. The attached jet or as it is commonly called a wall jet, can result from air supply through the outlet with one edge coincided with a plane of the wall or the ceiling (Figure 4.9). Jet supplied at some distance from the surface or at some angle to this surface also can become attached (Figure 4.10).

The results of studies of jets supplied through rectangular outlets by Kerka (Tuve 1953) with and without adjacent surface indicate the increase from 1.27 to 1.45 times of the velocity decay coefficient for wall jets compared with those obtained for the same outlets discharging into a free-open space. The angle of divergence of the wall jet in the direction perpendicular to wall was slightly less than one half of a free jet, while the angle of spread of the jet along the wall was greater, than the divergence of a free jet.

The results of experimental studies of compact wall jets by Mitkalinny (1961), Abdushev, Baharev and Fedorov, on linear wall jets by Kerka and Sakipov, and on radial wall jets by Gelman, are summarized by Grimitlyn (1994) and are presented in Figure 4.11). These data indicate, that the parameter $K^{wall} = K_1^{wall}/K_1$ reflecting the influence of the wall on the velocity decay along the jet increases from 1 to 1.4 with a distance from the outlet. E.g., for a compact jet $K^{wall} = 1$ when $X < 5d_o$; for a linear jet K^{wall} reaches its maximum value equal to 1.4 only at $X < 20b_o$, where b_o is an outlet width.

Studies of wall jets (e.g., Baturin 1965, Launder and Rodi 1983) show, that they have two layers: a turbulent boundary layer close to the wall and a outer shear layer. The thickness of the boundary wall layer can be neglected for the practical reason. Accordingly, to compute the maximum velocity in the wall jet researchers (Regenscheight 1959,Shepelev 1978, Etherindge and Sandberg 1996) apply the method of images by treating the wall jet as one half of a free jet. Application of this method gives a relationship between characteristics of a wall jet and a free jet which results in the value for a correction factor equal to $\sqrt{2}$. This approach has some inaccuracy even with a linear and radial jets. For a three dimensional wall jet the procedure is even more approximate. Discussion by Etherindge and Sandberg (1996) of some previous studies of attached jets indicates some loss of momentum in an attached jet due to the friction against the surface. The authors compiled an information from previous studies which with some edition is summarized in Table 4.11. According to Equations presented in Table 4.11, the maximum velocity in a wall jet is inverse proportional to the distance from the outlet in a different power compared to the case with a free jet.

Table 4.11. Maximum velocity deacy in a wall jets

Jet type	Decay of the maximum velocity	Reference
Linear wall jet	$\sim X^{-0.555}$ $\sim X^{-0.375}$	Schwartz and Cosart (1960) Regenscheight (1959)
Radial wall jet	$\sim X^{-1.12}$ $\sim X^{-1.15}$	Bakke (1957) Waschke (1974)
Rectangular wall jet $H_o/L=0.025$ $H_o/L = 0.05$ $H_o/L = 0.01$ $H_o/L = 1.0$	$\sim X^{-1.15}$ $\sim X^{-1.09}$ $\sim X^{-1.15}$ $\sim X^{-1.14}$	Sforza and Herbst (1970)
Compact wall jet	$\sim X^{-1}$	Nielsen (1987)

It is not uncommon to supply air into the room with jets attached both to the ceiling and to the wall surfaces (Nielsen 1981, Grimitlyn 1994). Air jets can be parallel to both surfaces or be directed at some angle to one or both surfaces (Figure 4.10). Studies of compact wall jets supplied parallel to both surfaces reported by Grimitlyn (1994) show that the correction factor value is in the range from 1.6 to 1.7, which means that restriction of entrainment from two sides reduces velocity decay by 20% to 30% compared to the case with a wall jet.

When a jet is supplied at some distance from the surface, the attachment occurs when the distance between the outlet and the surface is below a critical distance, otherwise the jet will propagate as a free jet (Awbi 1991). If the jet attaches to the surface the flow downstream of the attached point is similar to that of a wall jet. For a compact isothermal jet the critical distance for jet attachment to the surface is $L_{crit} = 6A_o^{1/2}$ (Farquharson 1952). For $L_{crit} < 6A_o^{1/2}$ the velocity decay coefficient K_1 becomes greater, than it would be in the case of free jet, and should be corrected using correction factor F (see Figure 4.12) to compensate for surface proximity.

The length of the recirculation zone, X_a, for a linear jet (the distance to the point of jet attachment to the surface) was studied by Sawyer (1963), Miller and Commings (1960), and Bourque and Newman (1960). The results of these studies summarized in (Awbi 1991) show that the length of recirculation zone (Figure 4.13) is proportional to the distance from the outlet to the surface and can be described as:

$$X_a / H_o = 0.73 \ (D / H_o) - 2.3 \tag{4.53}$$

where $\mathbf{H_o}$ is a width of the outlet.

Sandberg et al. (1992) conducted similar tests with a heated linear jet so, that the buoyancy forces opposed the forces due to the lower pressure in the circulation zone (bubble). Based on the results of these tests, it was concluded, that heating the jet does not change the location of the attachment point. Between $5 < D/H_o < 13$ the length of the circulation zone followed the relation:

$$X_a / H_{oD} = 1.175\ (D / H_o) + 6.25 \tag{4.54}$$

Calculation of X_a/H_o using Equations (4.53) and (4.54) results in significantly different results. For $D/H_o = 8$: $X_a/H_o = 3.54$ according to Equation (4.53), and $X_a/H_o = 15.65$ according to Equation (4.54).

The length of the circulation zone (bubble), $\mathbf{L_b}$, created when the linear jet supplied at the angle, α to the surface was studied experimentally by Bourque and Newman (1960) and theoretically by Sawyer (1963). The effect of the angle between the jet axis at the outlet and the surface on the length of the circulation bubble is shown in Figure 4.14, reproduced from Awbi (1991). The data presented in the Figure 4.14 show, that at sufficiently high Reynolds number the length of the circulation zone is independent of the Reynolds number.

Linear jet attachment to the plane not parallel to the supply direction was studied by Katz (1973). The critical angle, θ_c, of the plane to the jet supply direction as indicated in Figure 4.15, was found to be dependent on the supply velocity (Reynolds number). It also depends on the distance of the plane edge from the supply outlet (see Figure 4.16).

Baturin (1965) studied air jets supplied from the rectangular nozzle at some angle to the plane with an edge of the nozzle coincided with a plane. The results of his studies indicate, that the critical value of the angle of the jet supply direction to the plane is 45 deg. It was also shown, that the jet supplied through a rectangular outlet with a nozzle located at some distance from the plane does not attach to the surface.

4.4.7. Jet separation

When the temperature of the attached air jet is lower than the temperature of the ambient air, this jet will remain attached to the ceiling until the downward buoyancy force becomes greater than the upward static pressure ("Coanda" force). At this point the jet separates from the ceiling and begins downward curving trajectory (Miller, 1991). Studies of non-isothermal jets conducted by Grimitlyn (1970), Schwenke (1976), Nielsen and Moller (1987), Miller (1991), Anderson et al. (1991), Kirkpatrick et. al.(1991) showed that the distance to the point of jet's separation can be computed using the following equation:

$$\frac{X_{sep}}{\sqrt{A_o}} = \frac{a}{(Ar_o)^b} \tag{4.55}$$

For linear diffuser jets (Rodahl, 1977) $\mathbf{a} = 2.5H_o$ and $\mathbf{b} = 2/3$. For the compact diffuser jet $\mathbf{a} = 1.6\ \mathbf{A_o}^{0.5}$ and b = 0.5 (Anderson, 1991; Kirkpatrick, 1991). According to theoretical analysis and experimental data collected by Grimitlyn (1970), the separation distance of jets could be expressed by the equations similar to (4.55) considering diffuser characteristics $\mathbf{K_1}$ and $\mathbf{K_2}$.

For compact and incomplete radial jets:

$$X_s = \frac{0.55\ K_1\sqrt{A_o}}{\sqrt{K_2 Ar_o}} \tag{4.56}$$

For linear jets:

$$X_s = \frac{0.4\ K_1^{4/3}\ H_o}{(K_2 Ar_o)^{2/3}} \tag{4.57}$$

For radial jets:

$$X_s = \frac{0.45\ K_1\sqrt{A_o}}{\sqrt{K_2 Ar_o}} \tag{4.58}$$

4.4.8. Criteria for non-isothermal jets

In their review of experimental data on turbulent buoyant jets Chen and Rodi (1980) classify buoyant flows as:

- "buoyant jet" when the buoyancy force acts in direction of the jet supply velocity at the origin: upward projected heated air jet or downward projected cooled air jet;
- "negative buoyant jet" when buoyancy force acts in the opposite direction: downward projected heated air jet or upward projected cooled air jet;
- "non-buoyant jet' when the effect of buoyancy is negligible;
- plume when the buoyancy force completely dominate the flow: flow generated with a heat source.

In the general case, a buoyant jet has an initial momentum. In the region close to discharge, momentum forces dominate the flow so it behaves like a non-buoyant jet. there is an intermediate region where the influence of the initial momentum forces becomes smaller and smaller. In the final region, the buoyancy forces completely dominate the flow and it behaves like a plume. When the jet is supplied at an angle to the vertical direction, it is turned upwards by the buoyancy forces and behaves virtually like a vertical buoyant jet in a far field. A negative buoyant jet continuously loses momentum due the opposite direction of buoyancy forces to the supply air momentum and eventually tunes downward.

Between the non-buoyant jet region with its characteristic similarity behavior and the plume region with a different similarity behavior lies an intermediate region in which the flow changes from the former to the later (Chen and Rodi 1980). The axial location of the beginning of this intermediate region depends primarily on the exit Froude number:

$$F = \frac{V_o^2}{g\,D_o}\frac{T_\infty}{T_o - T_\infty} \tag{4.59}$$

The data from Harris (1967) and Kotsovinos (1977) for the linear jet that cover all three regions and from Rouse et al. (1952) are plotted in Figure 4.17 and Figure 4.18, reproduced from Chen and Rodi (1980). Figure 4.17 presents a dimensionless plot of centerline velocities and Figure 4.18 presents a dimentionless plot of centerline density (temperature difference). The data in Figures 4.17 and 4.18 also include as limiting curves the data of van der Hegge Zijnen (1958a and 1958b) for the non-buoyant jet ($F = \infty$) and of Rouse et al. (1952) for the plume ($F = 0$).

According to the data presented in Figures 4.17 and 4.18 the beginning of the transition zone in the linear jet is at approximately:

$$\frac{X_{beg}}{H_o} = 0.5\ F^{2/3}\left(\frac{T_\infty}{T_o}\right)^{-1/3} \tag{4.60}$$

and the end at approximately:

$$\frac{X_{end}}{H_o} = 5\ F^{2/3}\left(\frac{T_\infty}{T_o}\right)^{-1/3} \tag{4.61}$$

Using the relation between the Froude number and the Archimedes number: $\mathbf{Ar_o = 1/F}$, the length of the linear jet zone, **X**, where the buoyancy forces are negligibly small can be calculate as follows:

$$\frac{X}{H_o} = \frac{0.5}{Ar_o^{2/3}}\left(\frac{T_\infty}{T_o}\right)^{-1/3} \tag{4.62}$$

To characterize the relation between the buoyancy forces and momentum flux in different cross-sections of a non-isothermal jet at some distance **X** Grimitlyn (1982) proposed a local Archimedes number:

$$Ar_x = \frac{g\,X\,\Delta t_x}{V_x^2\,T_\infty} \tag{4.63}$$

where: **g** is acceleration due to buoyancy.

Equations for the local Archimedes number can be derived by substituting the expressions for axial velocity $\mathbf{V_x}$ and temperature differential $\mathbf{\Delta t_x}$ into (4.63):

for compact and radial jets

$$Ar_x = \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{\sqrt{A_o}} \right)^2 \tag{4.64}$$

for linear jet

$$Ar_x = \frac{K_2}{K_1^2} Ar_o \left(\frac{X}{H_o} \right)^{3/2} \tag{4.65}$$

where: $Ar_o = \frac{g\sqrt{A_o}}{V_o^2} \frac{\Delta t_o}{T_r}$ - Archimedes number at the outlet, characterizing the ratio of

buoyancy and inertial forces at the jet discharge from the outlet. When calculating $\mathbf{Ar_o}$ for linear jet, $\sqrt{\mathbf{A_o}}$ is substituted by the width of the slot $\mathbf{H_o}$.

Introduction of the local Archimedes criterion helped to clarify non-isothermal jet design procedure. Grimitlyn suggested critical local Archimedes number values, $\mathbf{Ar_x^{crit}}$, below which a jet can be considered unaffected by buoyancy forces <u>(moderate non-isothermal jet)</u>: $\mathbf{Ar_x} \leq 0.1$ for a compact jet; $\mathbf{Ar_x} \leq 0.15$ for a linear jet. Similar limitation for a linear jet resulted from Equation (4.62) @ $K_1 = 2.5$, $K_2 = 2.0$, $T_o = 293_0$C and $T_\infty = 313$°C is: $Ar_x < 0.14$. By applying the $\mathbf{Ar_x}$ criterion, equations (4.36) and (4.39) can be transformed into following:

$$K_n = \sqrt[3]{1 \pm a\, Ar_x} \tag{4.66}$$

where a = 2.5 for axially symmetric and incomplete radial jets and a = 1.8 for linear jet. The plus sign in Equation (4.66) corresponds to the situation when the directions of buoyancy and inertia forces coincide, whereas the minus sign corresponds to their counteraction. This equation can be used for vertical non-isothermal jets at $\mathbf{Ar_x} \leq 0.25$.

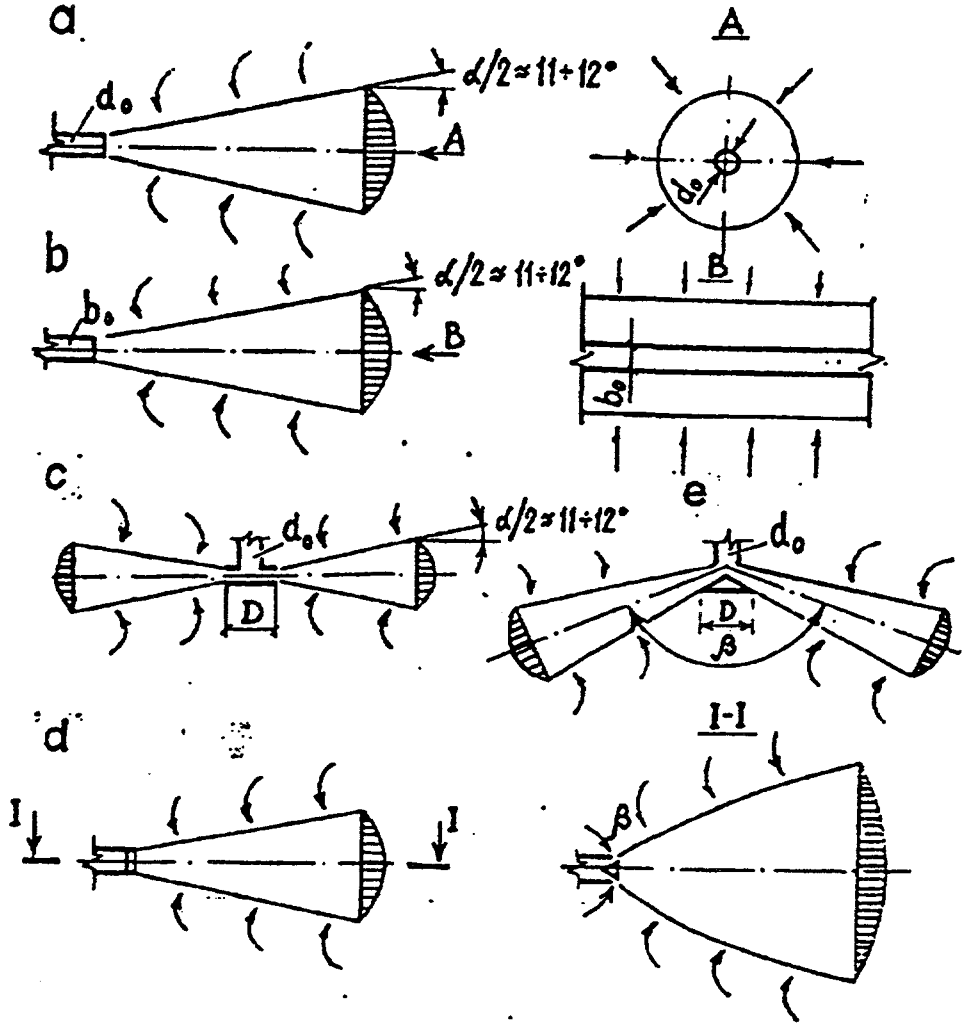


Figure 4.1. Types of diffuser jets: a - compact; b - linear; c - radial; d - incomplete radial; e - conical (Reproduced from Grimitlyn, 1994)

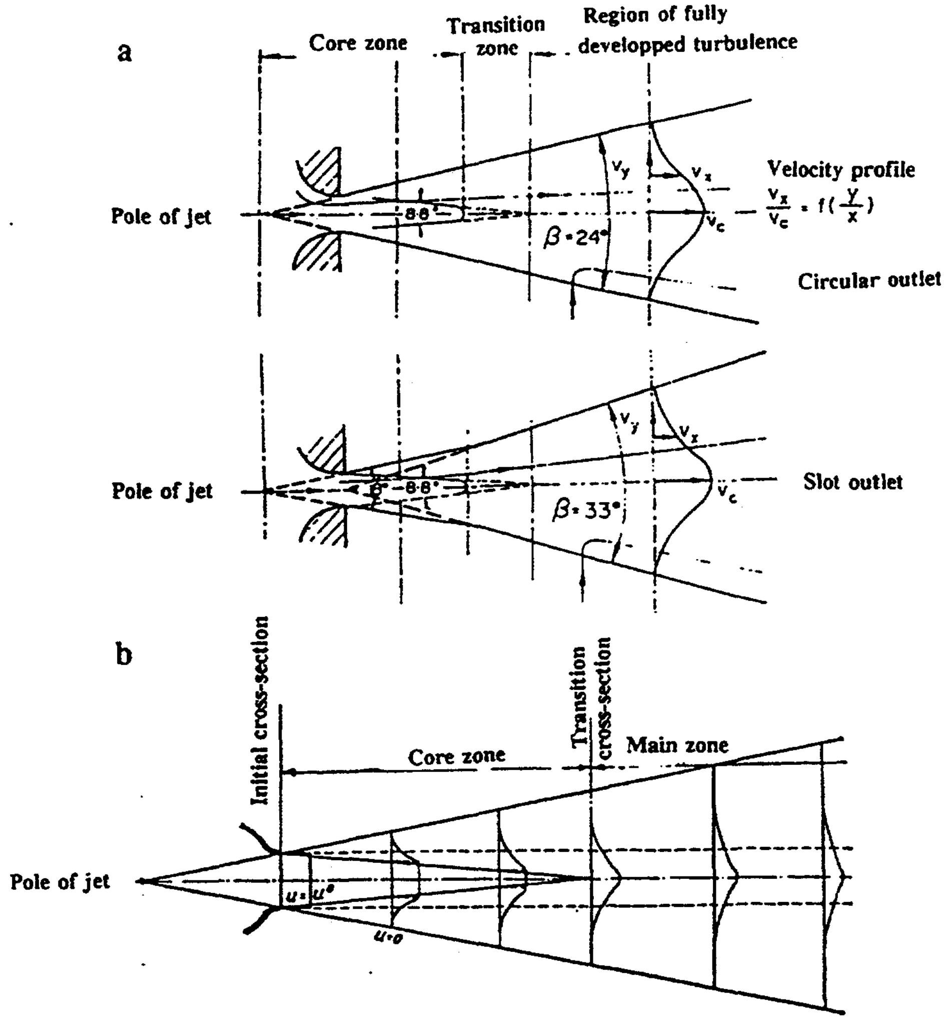


Figure 4.2. Turbulent jet: a - schematic with four zones; b - simplified jet schematic

a

b

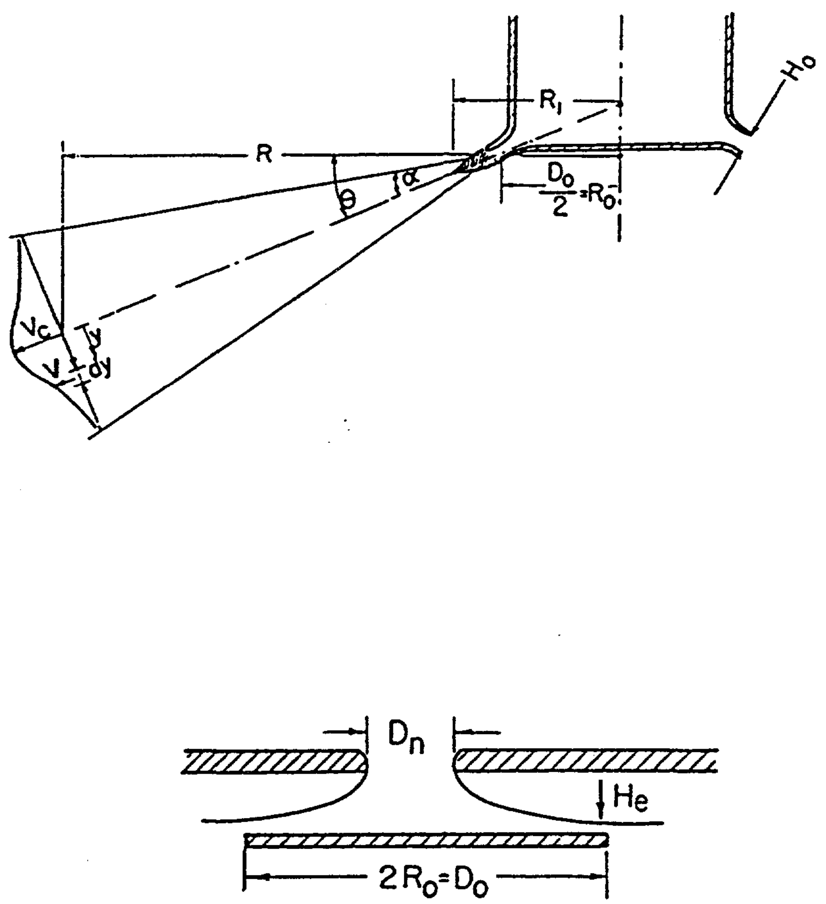


Figure 4.3. Schematic of radial jet: (a) general case: $\theta \geq 0$; (b) air supplied through diffuser with a plaque: $\theta = 0$. Reproduced from Koestel (1957).

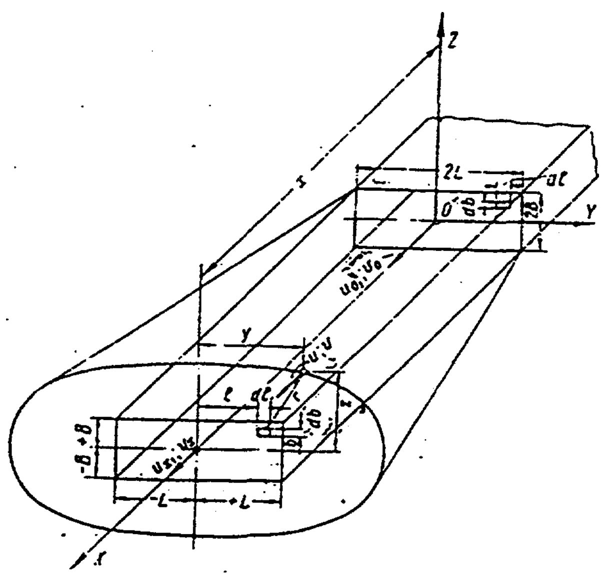


Figure 4.4. Schematic of the jet supplied through a rectangular outlet. Reproduced from Shepelev et al. (1966).

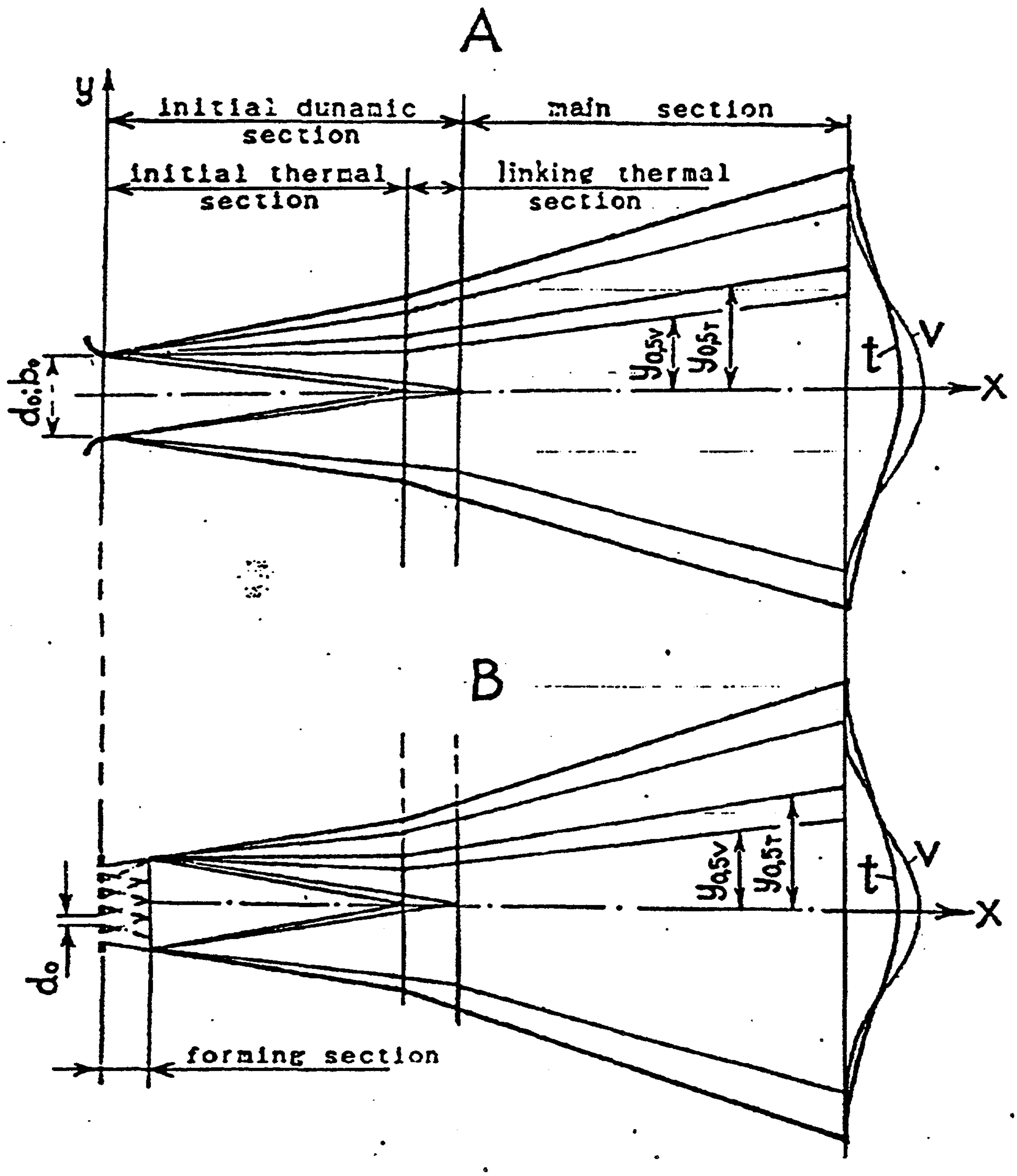


Figure 4.5. Scheme of a jet: A - from an open outlet; B - from a louvered (perforated) outlet. Reproduced from Grimitlyn (1982).

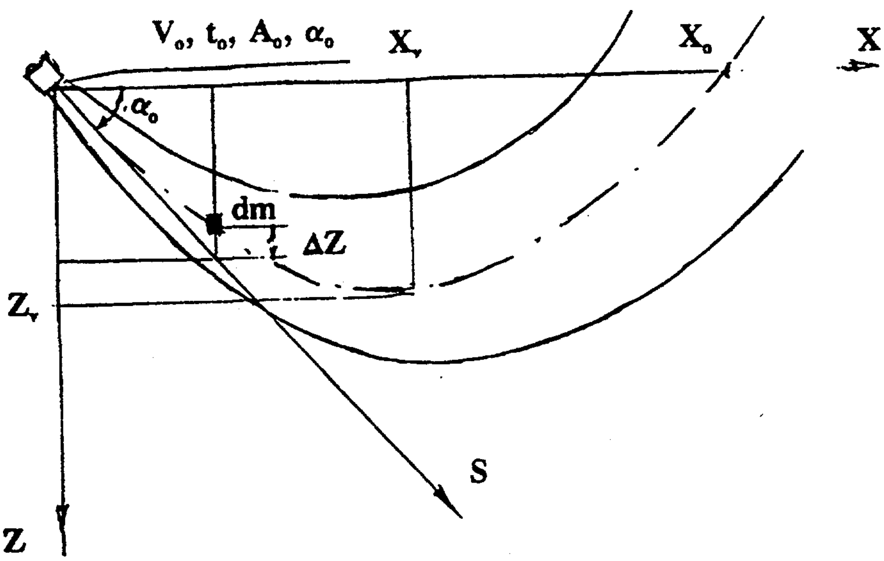


Figure 4.6. Schematic of inclined jet

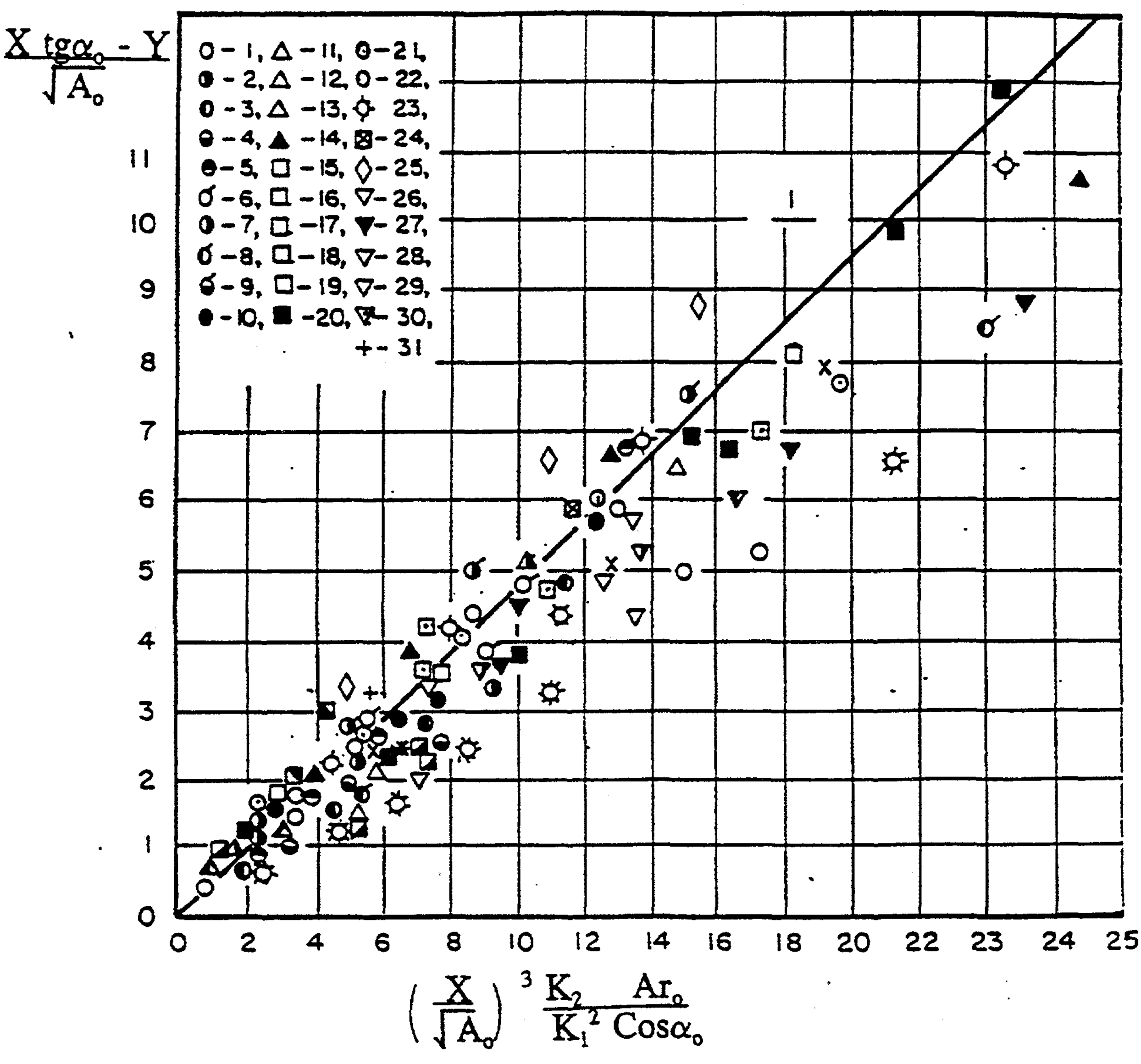


Figure 4.7. Experimental evaluation of the coefficient ψ. For experimental data references see Table 4.11. Reproduced from (Zhivov 1993).

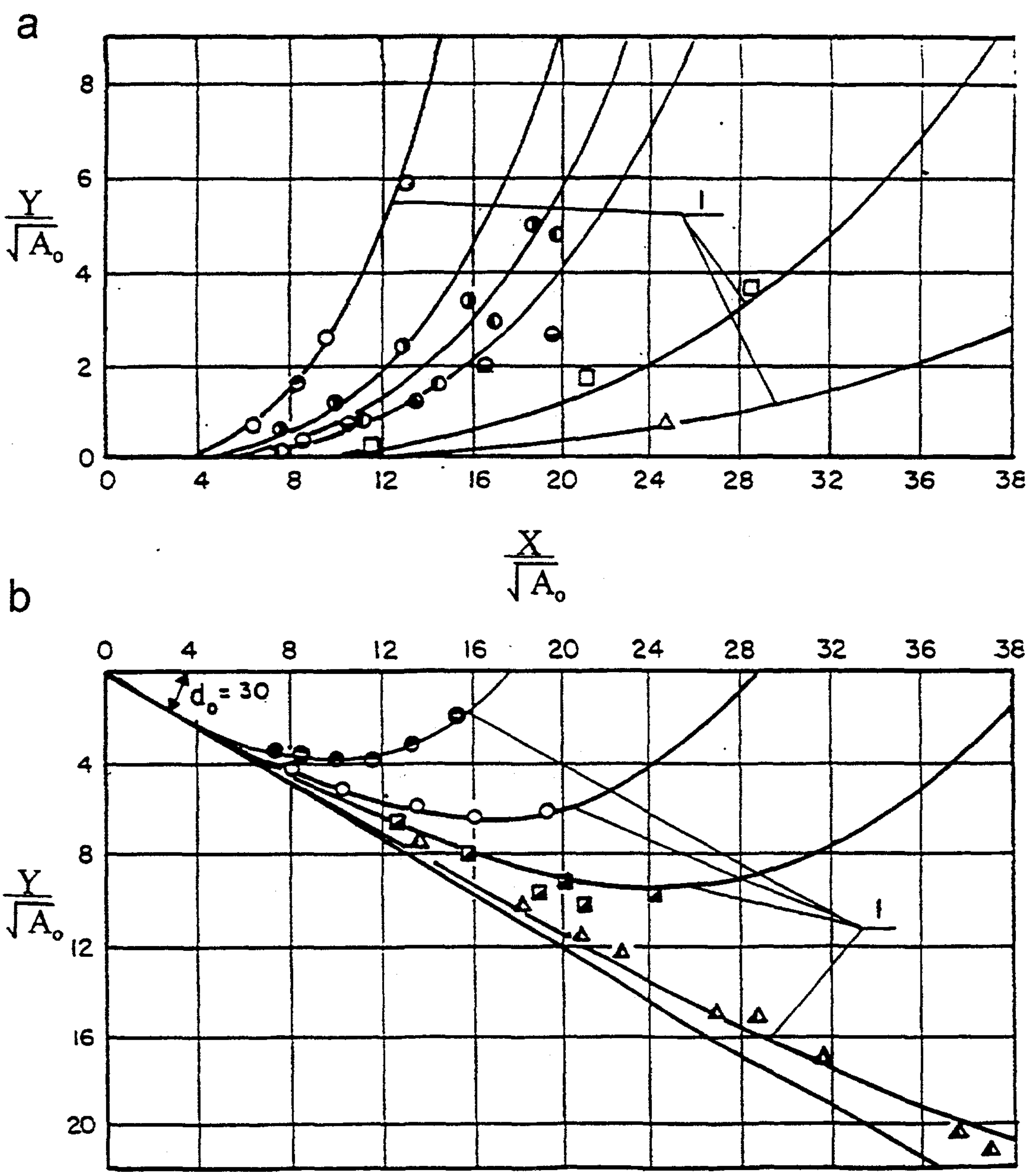


Figure 4.8. Trajectory of a non-isothermal jet supplied at an angle: (a) - α_o = O and (b) - α_o = 30 deg., 1 - equation (4.49). Reproduced from (Zhivov 1993).

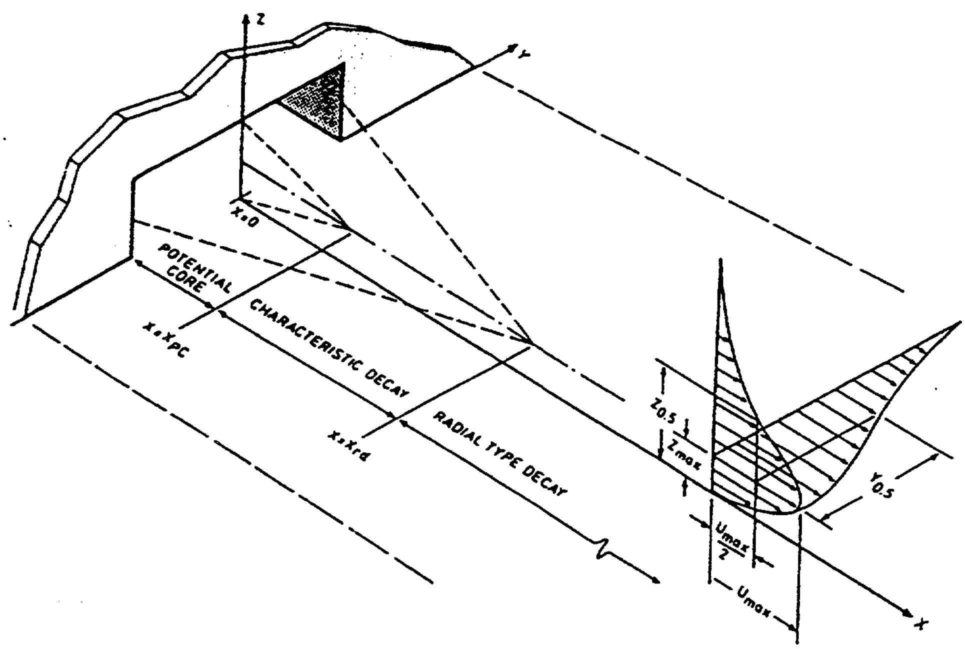


Figure 4.9. The three-dimentional wall jet. Reproduced from Etheridge and Sandberg (1996)

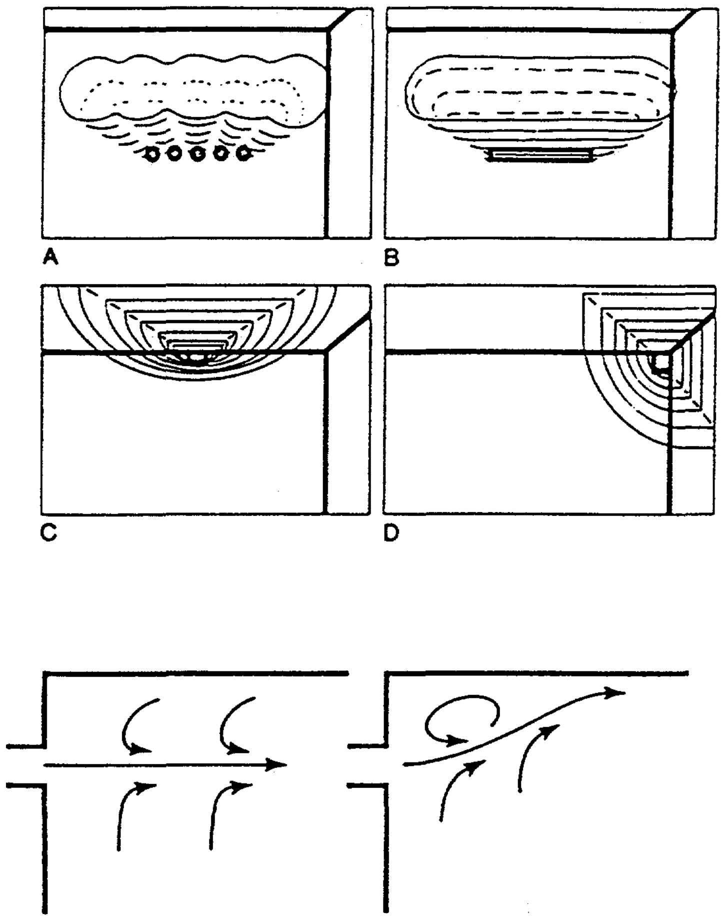


Figure 4.10. Jet attachement with air supply through outlets located at some distance from the surface: (A) multipal jets; (B) long slot jet; (C) rectangular jet; (D) rectangular conner jet; (E) general schematic. Reproduced from Nielsen (1981).

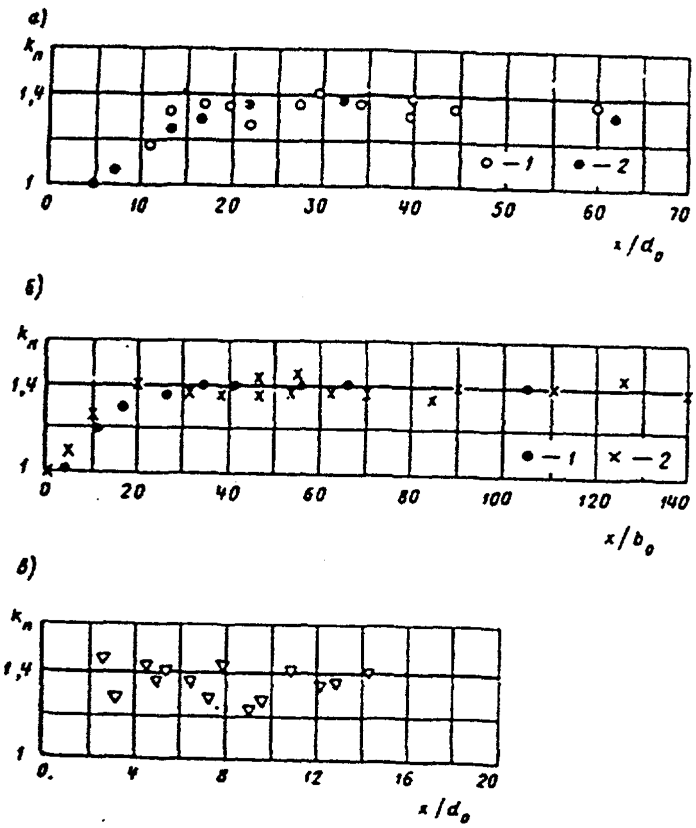


Figure 4.11. Correction parameter K^{wall} reflecting the influence of the wall: (a) compact jets - experimental data from V.Mitkalinny, A. Abdushev, V.Baharev and L.Fedorov; (b) linear jets - experimental data from W. Kerka and Z. Sakipov; (c) radial jets - experimental data from N. Gelman. Reporduced from Grimitlyn (1982).

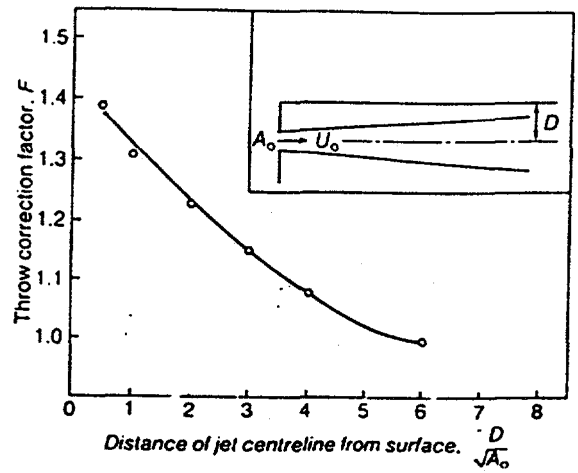


Figure 4.12. Correction factor F for surface proximity. Reproduced from Farquharson (1952).

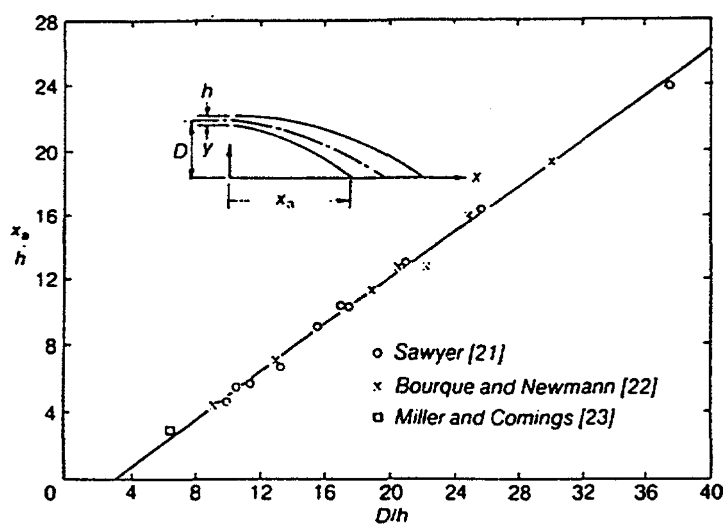


Figure 4.13. Effect of supply distance from surface on the attachment distance for a linear jet. Reproduced from Awbi (1991).

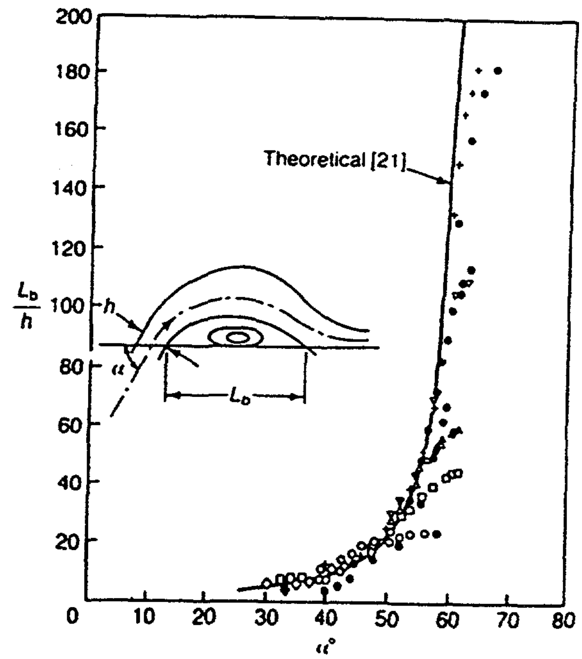


Figure 4.14. Effect of supply jet angle on recirculation buble length. Experimental data and theoretical curve from Sawyer (1960). Reproduced from Awbi (1991).

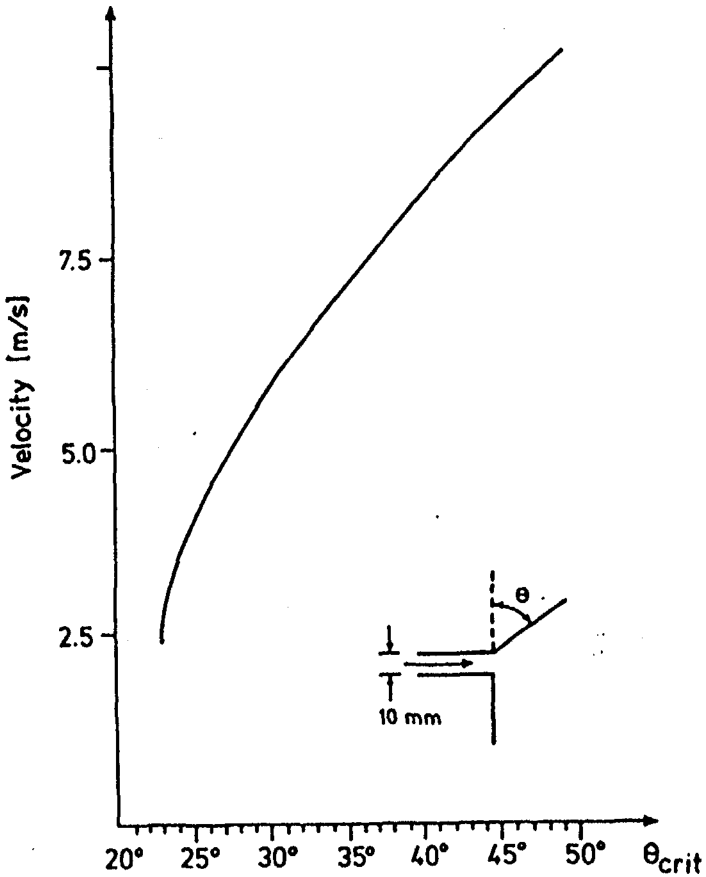


Figure 4.15. Critical angle. Wall is located close to supply outlet. Reproduced from Katz (1973).

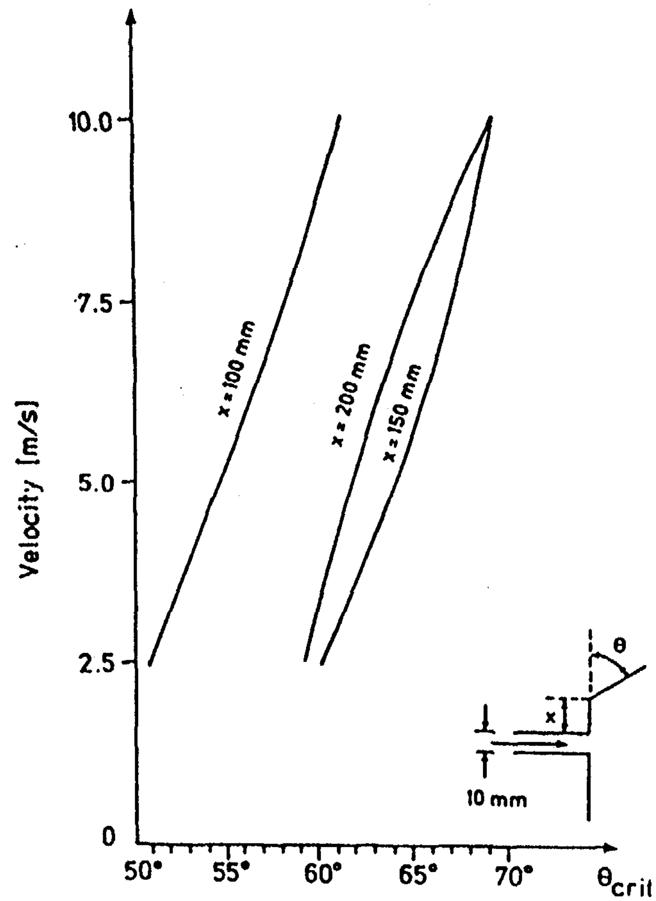


Figure 4.16. Critical angle. Wall is located at different distances from the air supply outlet. Reproduced from Katz (1973).

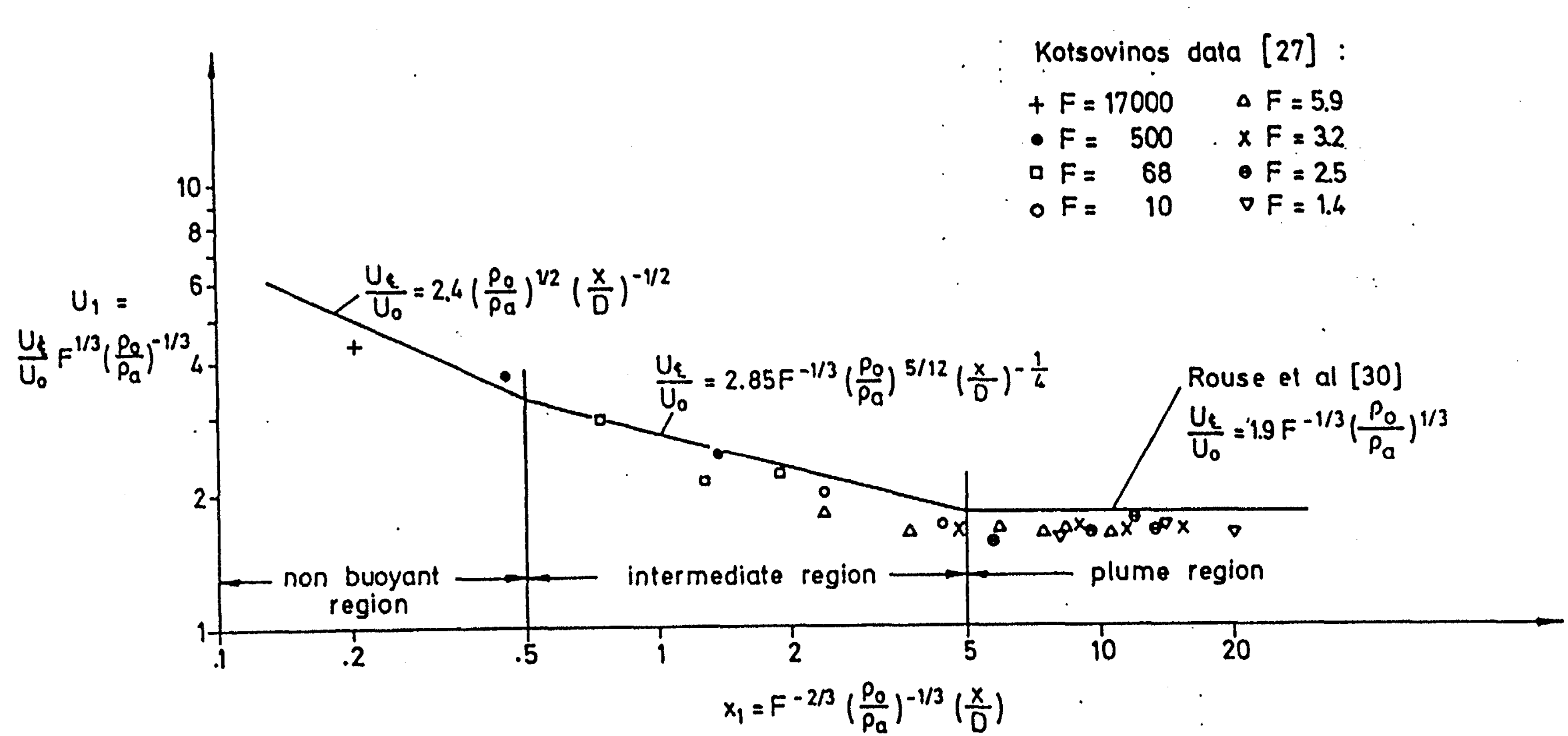


Figure 4.17. Decay of center-line velocity in a linear buoyant jet. Reporduced from Chen and Rodi (1980).

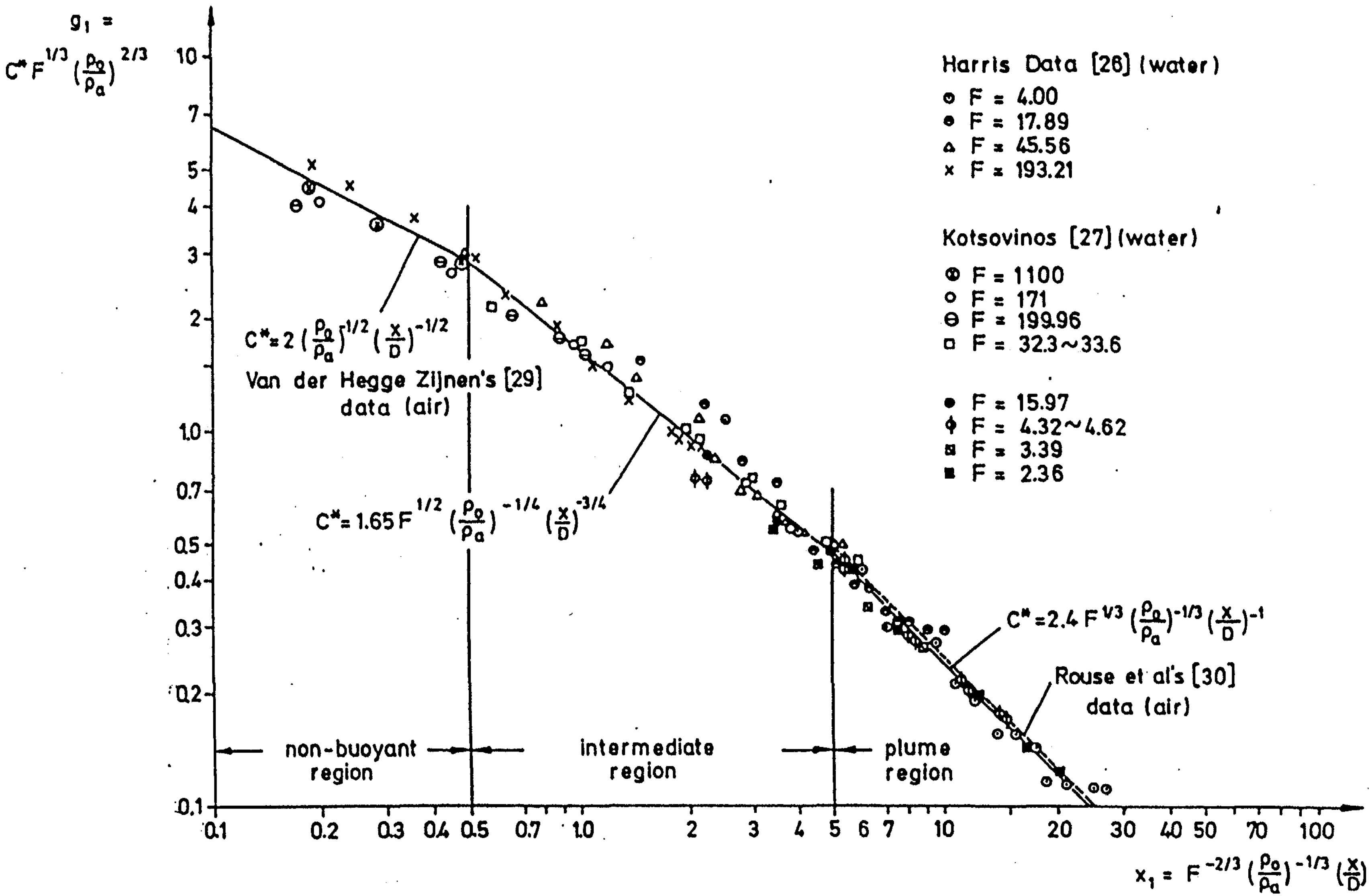


Figure 4.18. Decay of center-line density (temperature difference) in a plane jet. Reporduced from Chen and Rodi (1980).

5. JETS INTERACTION

Diffuser jets interaction is a common case when air is supplied into ventilated rooms through multiple air diffusers (e.g., side-wall mounted grilles or ceiling mounted air diffusers) or using speciality air distribution systems. Interacting jets can be supplied:

- parallel to each other;
- in the opposite direction toward each other (e.g., from the outlets located on the opposite walls);
- co-axial; and
- at an angle to each other.

The studies of some common cases of jet interaction are discussed in this section.

5.1. Interaction of parallel jets

Interaction of the parallel air jets is the most common case. It was thoroughly studied by Baturin (1951), Koestel et al.(1956), Grimitlyn (1960, 1982), Bashus and Kocheva (1963), Posokhin (1966), Nosovitskiy (1968), Kuzmina (1968), Vasilyeva (1969), Shepelev (1978). Researches developed equations describing velocities and temperatures in two or more interacting jets assuming that momentum and heat content of the flow through the elementary area in the cross-section of the resulting jet is equal to the sum of momentums and heat contents through the same area of the separate interacting jets. It was also assumed that separate jets do not influence each other. Derivation of velocities based on these assumptions described by Shepelev (1978) is presented below.

Air velocity $\mathbf{V_\Sigma}$ at any point (**X,Y,Z**) of the flow created by interaction of two parallel jets supplied from the outlets located at a distance **2a** from each other (Figure 5.1) can be described by equation:

$$V_\Sigma^2 = V_1^2 + V_2^2 \tag{5.1}$$

where

$$V_1 = \frac{K_1\ V_o \sqrt{A_o}}{X}\ e^{-\frac{1}{2}\frac{(Y-a)^2+Z^2}{(cX)^2}} \tag{5.2}$$

$$V_2 = \frac{K_1\ V_o \sqrt{A_o}}{X}\ e^{-\frac{1}{2}\frac{(Y+a)^2+Z^2}{(cX)^2}} \tag{5.3}$$

Air velocity in the resulting flow in the plane of interacting jets axis (**Z** = 0) derived from (5.1) -(5.3):

$$V_\Sigma = \frac{K_1\ V_o \sqrt{A_o}}{X}\left[e^{-\left(\frac{Y-a}{cX}\right)^2} + e^{-\left(\frac{Y+a}{cX}\right)^2}\right]^{\frac{1}{2}} \tag{5.4}$$

Air velocity on the axis of one of interacting jets (**Y = a** or **Y = -a**) is:

$$V_{x1} = \frac{K_1 \, V_o \sqrt{A_o}}{X} \left[1 + e^{-\left(\frac{2a}{cX}\right)^2}\right]^{\frac{1}{2}} \qquad (5.5)$$

If $\boldsymbol{\lambda} = (\mathbf{V_{x1}} - \mathbf{V_x})/\, \mathbf{V_x}$ is relative increase of axial air velocity in one the interacting jets due to the influence of the other, then the length of the jet $\mathbf{X_*}$ within which the influence of the interacting jet would be less than e.g. 10% ($\lambda = 0.1$) can be obtained from the equation:

$$X_* = \frac{2a}{c} \frac{1}{\sqrt{-\ln\lambda(2-\lambda)}} \approx \frac{24.4a}{\sqrt{-\lambda(2-\lambda)}} = 26.4a \qquad (5.6)$$

Which means that axial velocity of the interacting jet is influenced by another jet only beginning with $\mathbf{X_*}$ exceeding 13.2 of the distance between the outlets.

Velocity along the axis of symmetry between the interacting jets can be calculated assuming **Y** = 0 in (5.4)

$$V_{\Sigma x} = \frac{K_1 \, V_o \sqrt{2A_o}}{X} \, e^{-\left(\frac{a}{cX}\right)^2} \qquad (5.7)$$

Air velocity along the jet supplied from the outlet with opening area equal to $\mathbf{2A_o}$

$$V_x = \frac{K_1 \, V_o \sqrt{2A_o}}{X} \qquad (5.8)$$

If $\boldsymbol{\lambda} = (\mathbf{V_x} - \mathbf{V_{x1}})/\, \mathbf{V_x}$ is relative difference between axial air velocity in the jet with double opening area and air velocity along the axis of symmetry of the interacting jets, then the length of the jet $\mathbf{X_{**}}$ within which this ratio would be less than e.g. 10% ($\lambda = 0.1$) can be obtained from the equation:

$$X_{**} = \frac{a}{\sqrt{2}c} \frac{1}{\sqrt{-\ln(1-\lambda)}} \approx \frac{8.62}{\sqrt{-\ln(1-\lambda)}} = 26.5a \qquad (5.9)$$

This approach can be used for interacted jets supplied from the outlets with different area of discharge with different initial air velocities. In this case the equation for air velocity in the flow of interacting jets will be as follows:

$$V_\Sigma = \frac{K_1}{X}\left[V_{01}^2 \, A_{01} \, e^{-\left(\frac{Y-a}{cX}\right)^2} + V_{02} \, A_{02} \, e^{-\left(\frac{Y+a}{cX}\right)^2}\right] \qquad (5.10)$$

Comparison of the calculations according (5.4) with experimental data collected by Vasilyeva

(1969) at V_{01}=38.1 m/s, V_{02}=36.6 m/s, D_{01}=0.03 m, D_{02}=0.04 m, **a** = 0.05 m is presented on the Figure 5.1 from Shepelev (1978).

Jets interaction should not be taken in account when they are closely adjacent to each other, and are propagated in confined conditions and entrainment of the ambient air is restricted. This may be the case of concentrated air supply when air diffusers are uniformly positioned across the wall and the jets are replenished by the reverse flow, which does not increase but decrease the jet velocity. This effect should be taken into consideration using the confinement coefficient K_c discussed in Section 6. By the same reason jets interaction should not be taken into consideration when air is supplied through the ceiling mounted air diffusers and they are uniformly distributed across the ceiling (Grimitlyn 1982).

For the most common practical situation when air is supplied by parallel jets from several diffusers placed in one plane and having the same outlet area A_o and discharge velocity V_o the resulting velocity on the axis of the coalesced flow V_Σ can be found from the equations (Grimitlyn, 1982):

for compact and incomplete radial jets

$$\frac{V_\Sigma}{V_o} = K_1 \frac{\sqrt{A_o}}{X} K_{int} \tag{5.11}$$

for linear jet

$$\frac{V_\Sigma}{V_o} = K_1 \sqrt{\frac{H_o}{X}} K_{int} \tag{5.12}$$

The above relations can also be used as a first approximation to find temperature drop in interacting jets by substituting Δt_Σ ,Δt_o and K_2 for V_Σ, V_o and K_2 respectively. The values of interaction coefficient K_{int} for the even and odd number of outlets are given in Figure 5.2 reproduced from (Grimitlyn, 1982).

5.2. Interaction of jets supplied from opposite directions

There are few studies related to interaction of jets supplied from the opposite walls (Figure 5.3). For practical applications related to ventilation and air-conditioning interactions of similar jets with equal size and initial momentum were studied by Roeder (1967), Conrad (1973), Urbach (1971), Regenscheight (1974), Smirnova (Grimitlyn 1994).

Conrad (1973) compared impingement of two attached to the ceiling opposite linear jets with a linear jet changing its direction after impingement with a wall. For the attached jet maximum air velocity along the jet can be described by the following equation:

$$\frac{V_m}{V_o} = \left(\frac{h_o}{m\ XL} \right)^{0.375} \tag{5.13}$$

where, **m** is a supply outlet characteristic, which can be in the range from 0.1 through 0.4 (Regenscheight 1959), **X** is the distance from the slot to the point of interest. After changing the jet direction, velocity in the vertical jet can be obtained from the equation:

$$\frac{V_m}{V_o} = K \left(\frac{h_o}{m\ L} \right)^{0.375} \left(\frac{L}{L - Y} \right)^{q} \tag{5.14}$$

where, **Y** = vertical distance from the ceiling to the point of interest, **L** = length of jet travel along the ceiling; **K** = 1, **q** = 0.2 when jet travels vertically along the wall; **K** = 0.65, **q** = 1 in the case of two jet interaction. In discussion of the data obtained by Conrad and by Roeder, Regenscheit (1974) suggested, that the values K and q also depend upon the relative distances $\mathbf{L/h_o}$ and $\mathbf{Y/h_o}$ and the characteristic **m**. Based on the data by Urbach (1971), Regenscheight concluded, that **K** and **q** parameters also depend upon ratios **L/H** and $\mathbf{h_o/H}$. Research data also allow to conclude, that air velocities in the combined vertical jet are lower, than in the jet after its interaction with a wall.

The above data as well as studies of compact and radial jets interaction conducted by G.Smirnova, T. Avdeeeva and I.Gunes were summarized by Grimitlyn (1994). Grimitlyn suggested to describe air velocity in the jet resulting from impingement of two similar opposite directed jets with the following equation:

$$V_\Sigma = K_{int}^{op}\ V_x \tag{5.15}$$

where $\mathbf{K_{int}^{op}}$ can be evaluated from the graph in Figure 5.4. The graph shows, that smaller relative distance between jet supply outlets and the point of jets interaction, $\mathbf{a/b_o}$ (for linear jets) or $\mathbf{a/\sqrt{A_o}}$ (for radial and compact jets), result in smaller air velocities in the combined jet.

5.3. Interaction of coaxial jets

During the last two decades a new generation of HVAC systems with concentrated air supply assisted by directing jets was introduced in several European countries (Fläkt 1977, Wasiluk 1980). In one common modification, the main streams of ventilating air (heated or chilled), are supplied through a small number of air openings (grills) at low initial velocities and distributed within the space by horizontal (coaxial with main streams) and vertical (supplied perpendicular to the main streams) or only horizontal directing jets (Figure 5.5). These jets are discharged at high velocities from nozzles having small outlet diameters. The air is delivered to these nozzles from a separate air handling unit. The same principle is utilized by the "air piston system" (Wasiluk 1980), where horizontal directing jets are created by axial or radial fans located along the main streams.

Analytical and experimental studies on interaction of main streams of supplied air and horizontal directing jets in laboratories and in field conducted by Zhivov (1982, 1985, 1994) laid the ground for the design method of such systems (TsNIIPZ 1984).

Interaction of the free isothermal main stream and horizontal directing jets. The characteristic feature of the main stream and horizontal directing jets interaction is that the directing jets are supplied through the nozzles located at some distance from each other and from the outlet supplying the main stream (Figure 5.5).

Experimental studies with propane as a tracer gas, introduced into the main stream showed that directing jets make the main stream narrower (Zhivov 1985). Measurements of the velocity profiles $\mathbf{V_{i\Sigma}}$ in the cross section of the resulting stream (created by the main stream and directing jets) like that created by two linear coaxial jets studied by Stark (1950) and Landis and Shapiro (1951) can be described by the formula derived, assuming the resulting stream momentum is equal to the sum of interacting jets initial momentums. For maximum velocity in the resulting air flow in the cross section of the (**N - 1**) nozzle this equation is as follows:

$$V_{m\Sigma} = V_{02}\frac{d_{02}}{l_i}\sqrt{\frac{I_{01}}{I_{02}}\left(\frac{K_{11}}{K_{12}}\right)^2 + \sum_{k=1}^{N-1}\frac{1}{k^2}} \tag{5.16}$$

where, $\mathbf{K_{11}}$, $\mathbf{K_{12}}$ - coefficients of velocity decay in the main stream and directing jets, respectively, $\mathbf{V_{02}}$ - horizontal directing jet supply air velocity, $\mathbf{I_{01}}$ and $\mathbf{I_{02}}$ - main stream and horizontal directing jet momentum, respectively, $\mathbf{d_{02}}$ - horizontal directing jet nozzle diameter.

To derive the equation for the jet boundary resulted from the interaction of coaxial main flow and a directing jet supplied at the distance of $\mathbf{l_o}$ from the main outlet, this interaction was presented (Zhivov 1985) as the interaction of the main jet with a sink distributed along its axis (Figure 5.6). Considering the influence of the directing jet on the main flow boundary as $\mathbf{\Delta Y}$, the half width of the resulting flow can be presented as:

$$Y_b = \Lambda X - \Delta Y \tag{5.17}$$

where,

$$\Delta Y = \int_0^X \frac{d(\Delta Y)}{dX}\,dX \tag{5.18}$$

The Λ is the coefficient characterizing the angle γ of the main flow divergence (Figure 5.6) without directing jets influence. The following relationship was derived for the resulting flow boundary:

$$Y_b = \Lambda X - \frac{\theta}{K_{12}^2}\frac{I_{02}}{I_{01}}\, l_0\, \Phi(\overline{X}) \tag{5.19}$$

where, θ is an experimental coefficient and

$$\Phi(\bar{X}) = \int_0^{X}\left(1 + \frac{1}{\sqrt{1 + K_{11}^2\left(\frac{\bar{X}}{\bar{X}-1}\right)^2}}\right)^2 d\bar{X}; \qquad \bar{X} = \frac{X}{l_o} \tag{5.20}$$

In the case of several directing jets interacting with a main stream, the above approach was used assuming that each following directing jet interacts with a resulting flow created by the main flow and the previous directing jets. The equation for the resulting flow boundary differs from Equation (5.19) only by the expression for $\mathbf{\Phi(X)}$ function.

Interaction of the confined isothermal main and horizontal directing jets. The experimental studies (Zhivov 1985) show that the resulting jet length in the confined space can be divided into three zones (Figure 5.7). In the first zone, there is an expansion of the resulting jet boundaries. The length of this zone ($\mathbf{X_I}$) depends upon the relative momentum ($\mathbf{I_{02}/I_{01}}$) value and the relative distance ($\mathbf{l_i/\sqrt{A_r}}$) between the directing nozzles, where $\mathbf{A_r}$ is the room vertical cross section area ($\mathbf{b_r \times h_r}$). The distance ($\mathbf{X_I}$) increases when the relative momentum ($\mathbf{I_{02}/I_{01}}$) increases and the relative distance ($\mathbf{l_i/\sqrt{A_r}}$) decreases. In the second zone, the resulting jet width stays relatively constant. It expands up to the last nozzle, and its length is equal to ($\mathbf{X_{II} - X_I}$) and depends on the number of directing nozzles and the distance between them.

In the third zone, there is a significant decrease in resulting jet width. The cross section where the jet flow degradates is considered to be the end of the third zone. The length of the third zone ($\mathbf{X_{III} - X_{II}}$) is practically equal to the length of the jet's degradation zone developing in the confined space without directing jets, which is $\mathbf{2\sqrt{A_r}}$. Within the studied range of parameters, the resulting jet throw ($\mathbf{X_{III}}$) reached $\mathbf{10\sqrt{A_r}}$.

Beyond the third jet zone, there is a "stagnant" zone where the velocity values are relatively uniform and have an unstable direction. There is a reverse flow in zones I through III which is located between the jets boundaries and the cylinder walls. The maximum value of the velocity in the reverse flow is in the cross section at the end of zone I at the distance of $\mathbf{X_I}$. The following equation was derived to calculate the length of the first zone $\mathbf{X_I}$:

$$X_I = \frac{0.755}{\Lambda} Y_b + \frac{\theta}{\Lambda K_{11}^2}\frac{I_{02}}{I_{01}} l_i \Phi(\bar{X}) \tag{5.21}$$

The average experimental value of the coefficient $\mathbf{\theta}$ is 1.7 with a standard deviation (σ_θ) of 0.05. Equation (5.22) allows one to calculate the momentum ratio ($\mathbf{I_{02}/I_{01}}$) required to extend the length of the Zone I to the value equal to $\mathbf{X_I}$, given the distance between the directing nozzles equal to $\mathbf{l_i}$. Nomogram presented in the Figure 5.8 is plotted according to Equation (5.22) for $\mathbf{K_{11}}$ and $\mathbf{K_{12}}$ equal to 6.2.

The maximum value of reverse flow velocity ($\mathbf{V_{rev}}$) was found to be in the cross section at $\mathbf{X}$ equal to $\mathbf{X_I}$:

$$V_r^{max} = 0.73 V_{01} \sqrt{\frac{A_{01}}{A_r} (1 + S\, I_{02} / I_{01})} \tag{5.22}$$

where $\mathbf{A_0}$ is the main stream supply air diffuser area, **S** is the number of horizontal directing jets in the Zone 1.

Interaction of nonisothermal main stream and horizontal directing jets. Studies of nonisothermal main stream and horizontal directing jets interaction were conducted to evaluate maximum heat load which can be effectively supplied by such HVAC systems. To summarize experimental data received both in a free and confined conditions, it was suggested that the above limiting condition is achieved when the current Archimedes number $\mathbf{Ar_x}$ (ratio of the buoyancy forces over inertia forces along the resulting jet axis) does not exceed some value $\boldsymbol{\mu}$

$$Ar_x = \frac{gX}{V_{x\Sigma}^2} \frac{t_{m2} - t_r}{273 + t_r} \leq \mu \tag{5.23}$$

For the interaction of the main stream with N directing nozzles, the resulting expression for the $\mathbf{Ar_x}$ can be presented as follows:

$$Ar_x = \frac{K_{21}}{K_{11}^2} Ar_o \left(\frac{X}{d_{01}}\right)^2 \frac{1}{f} \tag{5.24}$$

where,

$$f = \frac{1}{N^2} + \frac{I_{02}}{I_{01}} \left(\frac{K_{12}}{K_{11}}\right) \left(\frac{X}{I_i}\right)^2 \sum_{i=1}^{N} \frac{1}{i^2} \tag{5.25}$$

The current Archimedes number for the resulting jet grows along the jet as it does in any nonisothermal jet. However, the consequent momentum additions by directing jets increases the inertial forces in the resulting jet and thus, at certain cross-section the current Archimedes number falls. The number of directing jets after which the $\mathbf{Ar_x}$ reaches the peak can be calculated using the following equation:

$$N_* = \frac{1}{\sqrt{\frac{I_{02}}{I_{01}} \left(\frac{K_{12}}{K_{11}}\right)^2 \left(1.202 - \sum_{i=1}^{N_*} \frac{1}{i^3}\right)}} \tag{5.26}$$

Based on the experimental data received for the free resulting jet, the $\boldsymbol{\mu}$ value in Equation (5.23) is equal to 0.075. Tests in field showed that the influence of the reverse flow and confining surfaces increase the value of $\boldsymbol{\mu}$ to 0.2. Based on Equation (5.23) and experimental $\boldsymbol{\mu}$ values the maximum initial air temperature supplied by the main stream is limited by:

$$t_o - t_r \leq a \frac{K_{11}^2}{K_{21}} \frac{V_{01}^2 d_{01}}{I_i^2} f \tag{5.27}$$

where, **a** is a coefficient equal to 2.65 for a free jet, and 7.07 for a confined jet.

5.4. Interaction of jets supplied at an angle to each other

There are only few studies of air jets supplied at some angle α ($0 < \alpha < 90$ deg) toward each other. To predict characteristics (trajectory, velocity decay, etc.) of the flow resulted from interaction of two jets supplied at some angle toward each other, Hudenko (1966) suggested to sum momentums of interacting jets as in the case with a parallel jets. He has estimated that the error of prediction will be smaller at a smaller:

- interaction angle;
- distance between the supply nozzles;
- difference in the nozzle sizes, and
- supply air velocity values.

Meshalin (1974) conducted experimental studies of two equal jets supplied at the angle of 15 deg, 30 deg and 45 deg toward each other. Based on the results of his studies, the author concluded that:

- The turbulent mass transfer in the flow resulted from the two jets interaction is more intensive, than in a single jet at the same supply conditions. The intensity of mass transfer in the flow in the plane of interacting jets axis is lower, than in the plane of symmetry;
- The intensity of mass transfer in the resulting flow increases with the angle of interaction increase.

Numerous studies of jet supplied into a uniform and nonuniform cross flow were conducted in application to such areas as air pollution control, burning processes, etc. Detailed discussion of these studies are beyond the current review. However, some results of these studies will be mentioned as needed in the following section.

Interaction of free isothermal main stream and directing jets supplied at the right angle to the main stream. As in the case with the interaction of coaxial directing jets, the interaction of main streams with directing jets supplied at a right angle was studied (Zhivov 1982, 1983) to develop a design method for air distribution with horizontal and vertical directing jets.

The discussion on interaction of air jets supplied an some angle toward each other shows, that application of the method that suggests to superimpose the interacting jets momentums and surplus heat to predict velocity and temperatures in the combined flow results in inaccuracy when two unequal jets are supplied under the right angle. Different approach was undertaken in the studies of interaction of the main stream with vertical directing jets (Zhivov 1982, 1983).

Visualization studies of the resulted flow showed that the directing jet due to its interaction with a main stream changes its initial direction. The interaction results in two separate flows; the first is a continuation of the main stream and the second is a continuation of the directing jet. The specific feature of this interaction is that vertical directing nozzles are located within a main stream (Figure 5.10). The median diameter of the directing jet is significantly (by several times) smaller than the main stream diameter within a zone of their interaction. Thus, the interaction of the main stream and the vertical directing jet can be seen as an interaction of the axisymmentric (directing) jet with a infinite cross draft with a nonuniform velocity profile.

An analytical solution of interaction in the case of isothermal main and directing jets, assume that the main stream (Figure 5.10) supplied with initial velocity (**V_{01}**) through the nozzle which has internal diameter (**d_{01}**) is developing within a zone (**$-l_0$, 0**) as a free jet. The momentum (**I_1**) of the jet within the zone (**$-l_0 + l_c$, 0**) remains equal to the initial momentum (**I_{01}**), and the velocity distribution in the cross section of interaction in the plane **XY** remains the same within the zone (**0, X_A**). The axisymmetric main stream within the zone (**0, X_A**) is substituted by the linear flow with velocity profile which can be described by the formula:

$$V_{i1} = V_{m1}\, e^{-\frac{1}{2}\left(\frac{Y-Y_o}{cl_o}\right)^2}, \quad X \in [0,\ X_A],\ c = 0.082 \tag{5.28}$$

The directing jet is supplied at a right angle to the main stream axis with an initial velocity of **V_{03}** from the nozzle with an inner diameter (**d_{03}**) located at the distance (**l_0**) from the plain of main stream supply and at the distance of **Y_0** from its geometrical axis. The momentum vector component along the **Y** axis remains constant and equal to the initial momentum (Figure 5.10):

$$I_3\ Cos\beta = I_{03} \tag{5.29}$$

where, **β** is angle between the **Y** axis and the tangent to the directing jet trajectory;

$$I_{03} = \rho_{03}\ V_{03}^2 \frac{\pi d_{03}^2}{4}. \tag{5.30}$$

The aerodynamic force (**P**) of the main stream and the momentum of the injected air (**I_{inj}**) change the **X** component of the directing jet. The **X** component of the directing jet can be calculated as follows:

$$I_3\ Sin\beta = P + I_{inj} \tag{5.31}$$

Experimental studies (Zhivov 1983) have shown that velocity distribution in the cross section of the directing jet can be described by the same equation as those in the axisymmetric jet in a cross draft

$$\frac{V_{i3}}{V_{m3}}\ Cos\ (\overline{V}_{i3},\ \overline{n}) = \left[1 - \left(\frac{r_i}{R_b}\right)^{1.5}\right]^2 \tag{5.32}$$

where, **$r_i = \sqrt{S_i/\pi}$, $R_b = b/2$; S_i** is a cross sectional area limited by the constant velocity line. The joint solution of Equations (5.29), (5.31) and (5.32) results in the following expression for the maximum velocity along the directing jet:

$$\frac{V_{m3}}{V_{03}} = X_v\ \frac{d_{03}}{Y} \tag{5.33}$$

where,

$$X_v = \frac{m_2}{1 + \frac{a}{q^k}} \frac{1}{\sqrt{Cos\beta}} \tag{5.34}$$

When $\beta < 25°$, $\sqrt{Cos\beta} \approx 1$.

The differential equation for the directing jet trajectory was received by the joint solution of Equations (19) and (21) assuming $\sqrt{Cos\beta} = 1$:

$$tg\beta = \frac{dX}{dY} = \frac{P + I_{inj}}{I_{03}} = \frac{c^2 k_o}{3} \frac{m_1}{X_v} \frac{I_{o1}}{I_{03}} A, \tag{5.35}$$

where,

$$A = e^{-\left(\frac{Y_o}{cl_o}\right)^2} - e^{-\left(\frac{Y-Y_o}{cl_o}\right)^2} + \frac{\sqrt{\pi}Y_o}{cl_o}\left[erf\left(\frac{Y-Y_o}{cl_o}\right) + erf\left(\frac{Y}{cl_o}\right)\right] \tag{5.36}$$

The equation for the directing jet trajectory was received by the integration of the Equation (25) at **X = 0**, **Y = 0**:

$$X = \frac{ck_o}{3} \frac{m_1^2}{X_v} \frac{I_{01}}{I_{03}} B \tag{5.37}$$

where,

$$B = (Y-Y_o)\left[e^{-\left(\frac{Y_o}{cl_o}\right)^2} + \frac{\sqrt{\pi}Y_o}{cl_o}\left(\frac{Y_o}{cl_o}\right)\right] - \frac{cl_o\sqrt{\pi}}{2}\left[erf\left(\frac{Y-Y_o}{cl_o}\right) + erf\left(\frac{Y_o}{cl_o}\right)\right] +$$

$$+ Y_o\left[e^{-\left(\frac{Y-Y_o}{cl_o}\right)^2} + \frac{\sqrt{\pi}(Y-Y_o)}{cl_o} erf\left(\frac{Y-Y_o}{cl_o}\right)\right] \tag{5.38}$$

Based on the Equations (5.36) and (5.38) one can conclude (Figure 5.10) that beyond the boundary of the main stream directing jet has a straight trajectory.

Visualization studies of the directing jet showed that after interacting with a main stream the directing jet has a straight trajectory when β is less than 50°. At a greater value of β ($tg\beta > 1.2$), the directing jet trajectory is significantly curved.

In the case of nonisothermal directing jet, the above assumptions are true except the momentum vector component along the **Y** axis changes due to the buoyancy force:

$$dI_{y3} = \pm\, dG \tag{5.39}$$

The amount of heat $\mathbf{W_3}$ along the directing jets remains constant.

Experimental studies have shown (Zhivov 1983) that temperature distribution in the cross section of the directing jets can be described as follows:

$$\frac{t_{i3}-t_r}{t_{m3}-t_r} = 1 - \left(\frac{r_i}{R_b}\right)^{1.5} \tag{5.40}$$

Based on the above assumptions the following equation was received to calculate the velocity an temperature decay along the nonisothermal directing jet:

$$\frac{V_{m3}}{V_{03}} = X_v \frac{d_{03}}{Y} \sqrt{1 \pm 1.8\, Ar_y \sqrt{Cos\beta} \sqrt{\frac{\rho_{03}}{\rho_r}}} \tag{5.41}$$

where,

$$Ar_y = \frac{1}{X_v} \frac{g\, d_{03}}{V_{03}^2} \frac{t_{03} - t_r}{273 + t_r} \left(\frac{Y}{d_{03}}\right)^2; \tag{5.42}$$

$$\frac{t_{m3} - t_r}{t_{03} - t_r} = X_t \frac{\sqrt{Cos\beta}\, \frac{\sqrt{\rho_{03}}}{\rho_r}}{\sqrt{1 \pm 1.8\, Ar_Y \sqrt{Cos\beta} \sqrt{\frac{\rho_{03}}{\rho_r}}}} \frac{d_{03}}{Y} \tag{5.43}$$

where,

$$X_t = \frac{n_3}{1 + \frac{a}{q^k}} \tag{5.44}$$

In the case when there is no main stream ($\mathbf{V_{01}} = 0$, $\mathbf{q} = \infty$ and $\boldsymbol{\beta} = 0$), Equations (5.41) to (5.44) transfer into those for free jets.

Joint solution of Equations (5.31) and (5.39) allows one to calculate the maximum heat amount supplied by directing jet with an assumption that the jet reaches the occupied zone ($\mathbf{V_{m3}} > 0.1$ m/s) and $\mathbf{(tg\beta_n - tg\beta)/tg\beta}$ less than 0.2 at the point where it enters the occupied zone. The maximum initial temperature difference of the air supplied by vertical directing jet is:

$$(t_{03} = t_r)^{max} = 32 \frac{V_{03}^2\, d_{03}}{(h_{03} - h_{o.z})^2} \frac{1}{1 + 23.7\sqrt{\frac{d_{01}}{l_o}}} \tag{5.45}$$

Based on the experimental results, the following values of coefficients were received: $\mathbf{a} = 9.5$, $\mathbf{k} = 0.25$, $\mathbf{k_0} = 3.3$ at $0° < \beta < 25°$ and $\mathbf{k_0} = 2.4$ at $25° < \beta < 50°$.

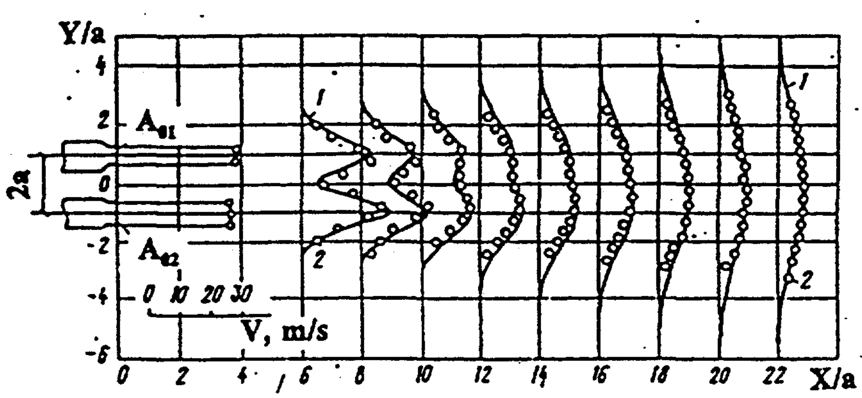


Figure 5.1. Interaction of two parallel compact jets.

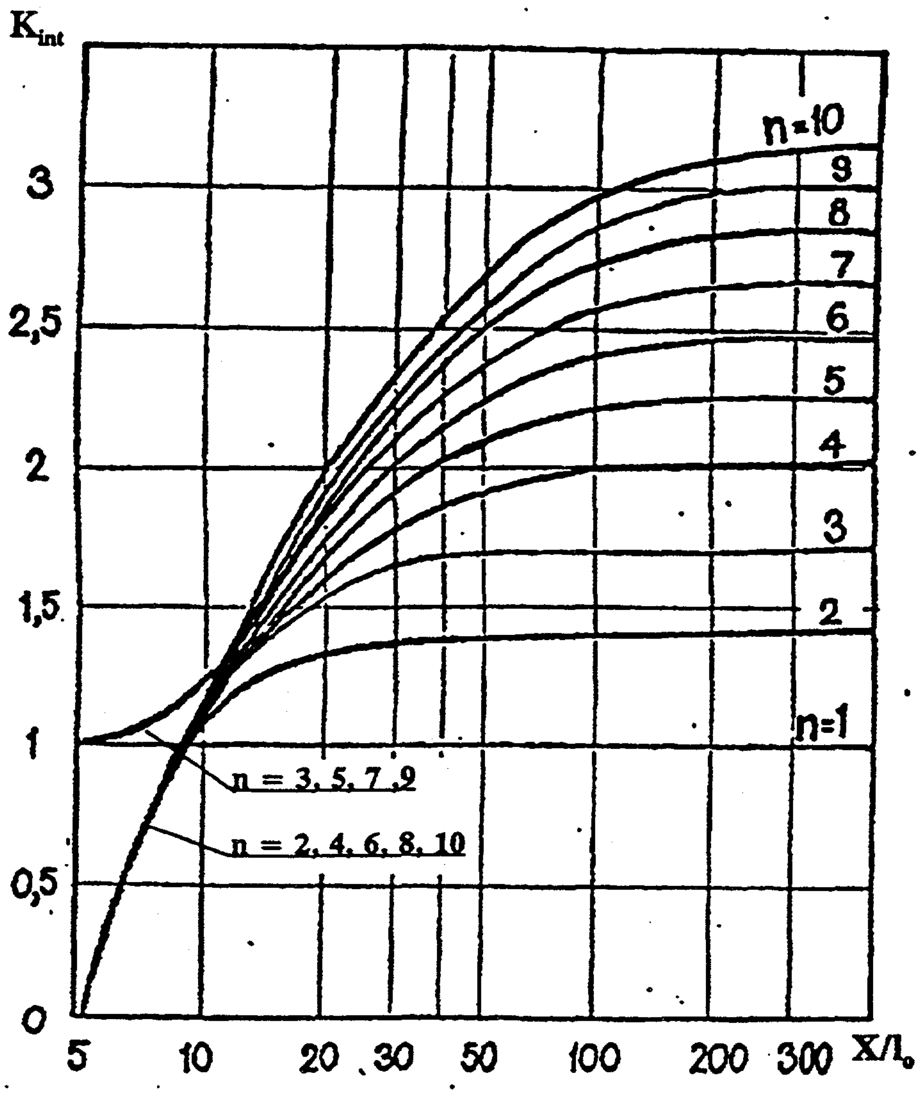


Figure 5.2. Coefficient of interaction for the jets discharging from the openings located in a single row. Reproduced from Grimitlyn (1982).

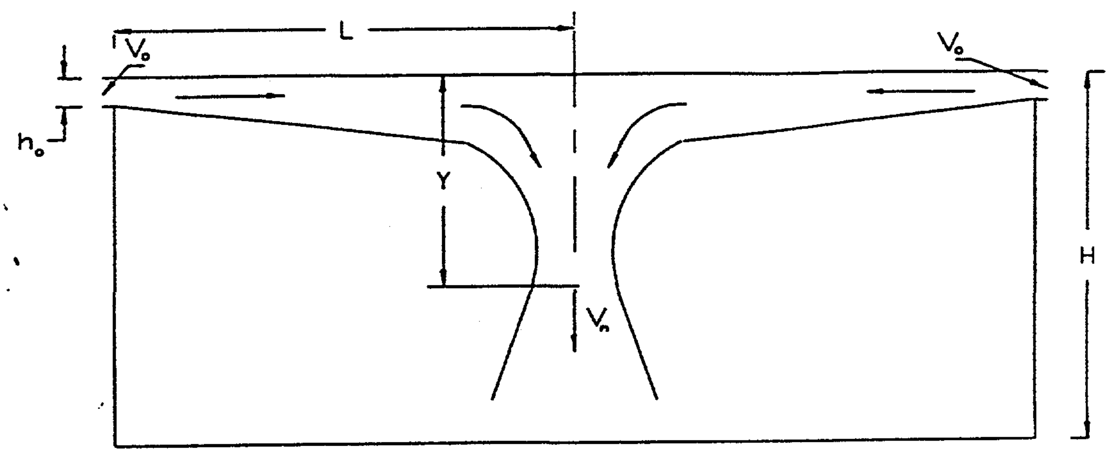


Figure 5.3. Interaction of two jets supplied from the opposite walls.

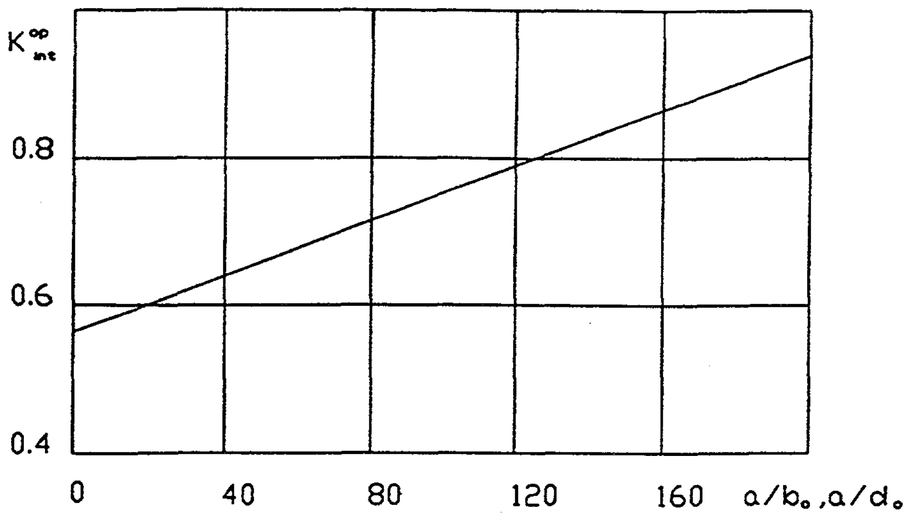


Figure 5.4. Coefficient K_{int}^{op} of opposite jets interaction. Reproduced from Grimitlyn, 1994

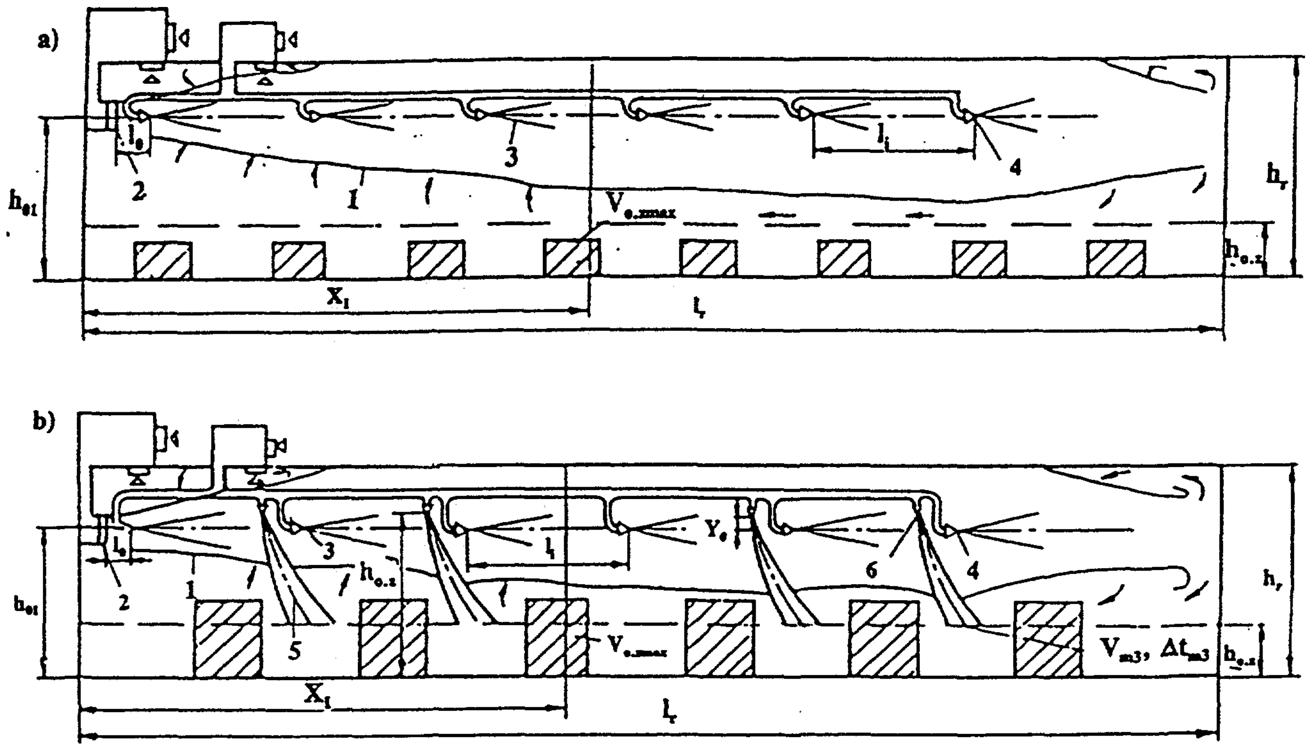


Figure 5.5. Concentrated air supply with directing jets: a - with horizontal directing jets; b - with vertical directing jets; 1 - main stream; 2 - main stream air diffuser; 3 - horizontal directing jet; 4 - horizontal directing jet nozzle; vertical directing jet; 6 - vertical directing jet nozzle.

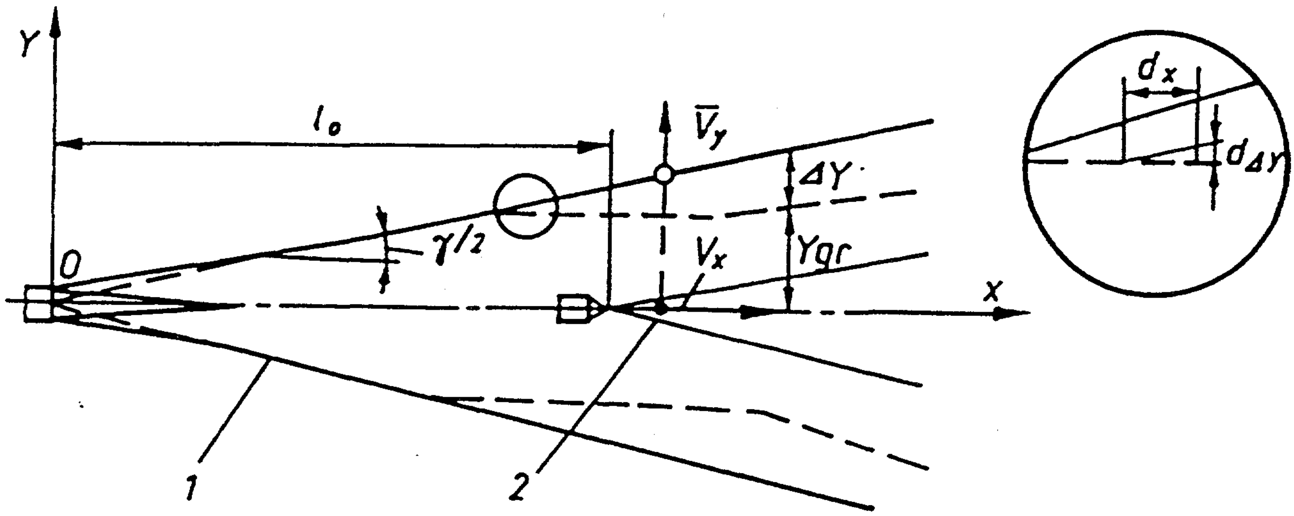


Figure 5.6. Schematic of free isothermal main stream and horizontal directing jet interaction: 1 - main stream (ρ_{01}, V_{01}, l_{01}, d_{01}, K_{11}); 2 - directing jet (ρ_{02}, V_{02}, l_{02}, d_{02}, K_{12}). Reproduced from Zhivov, 1983.

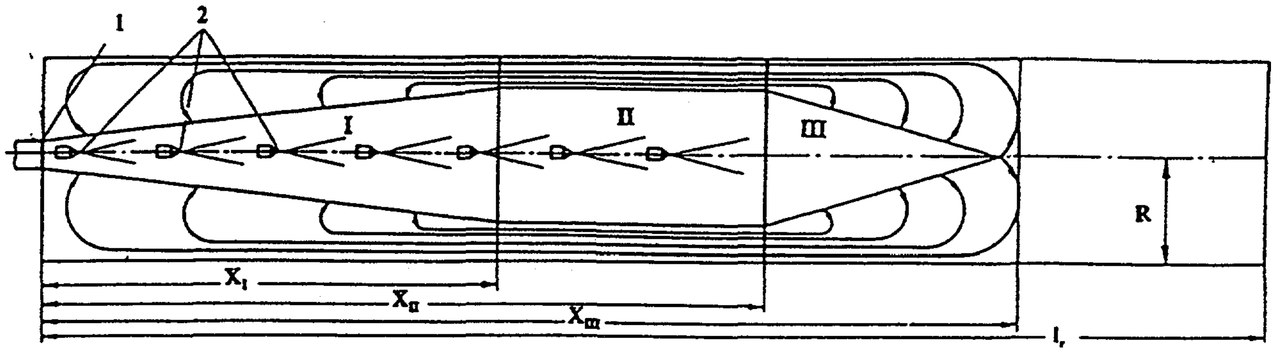


Figure 5.7. Schematic of confined main and horizontal directing jets interaction: 1 - main flow; 2 - directing jet. Reproduced from Zhivov (1994).

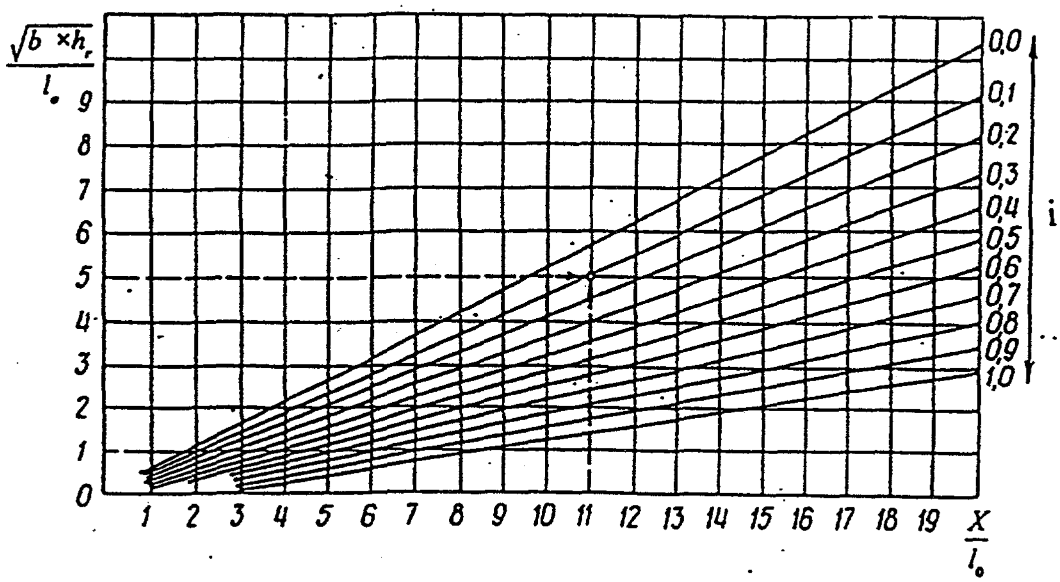


Figure 5.8. Nomogram for parameter i evaluation ($K_{11} = K_{12} = 6.2$). Reproduced from Zhivov 1983.

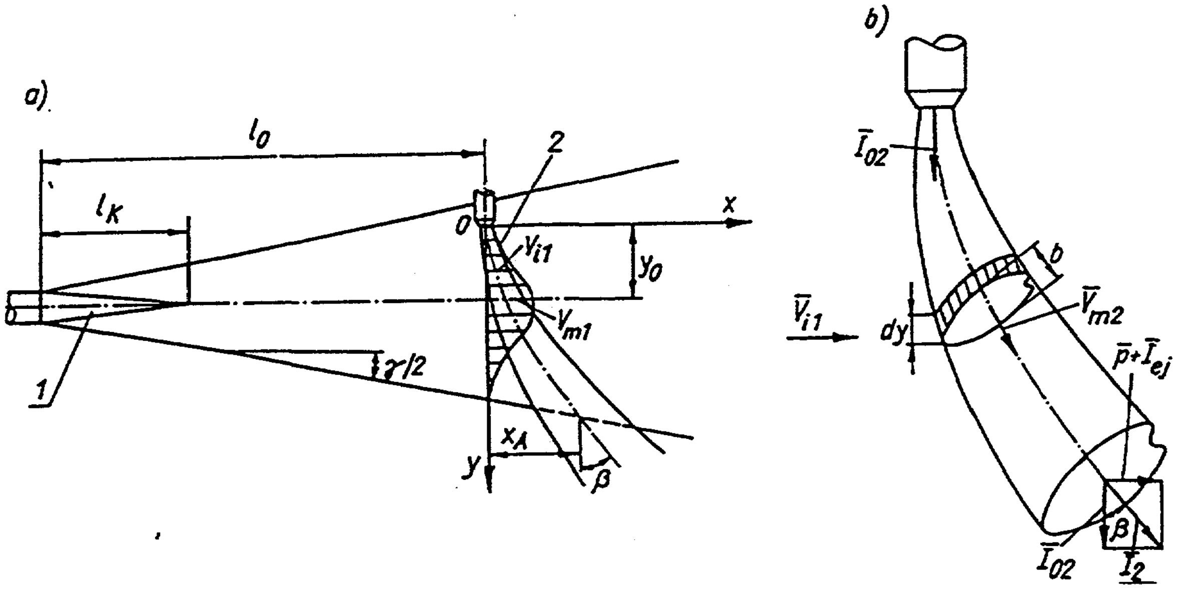
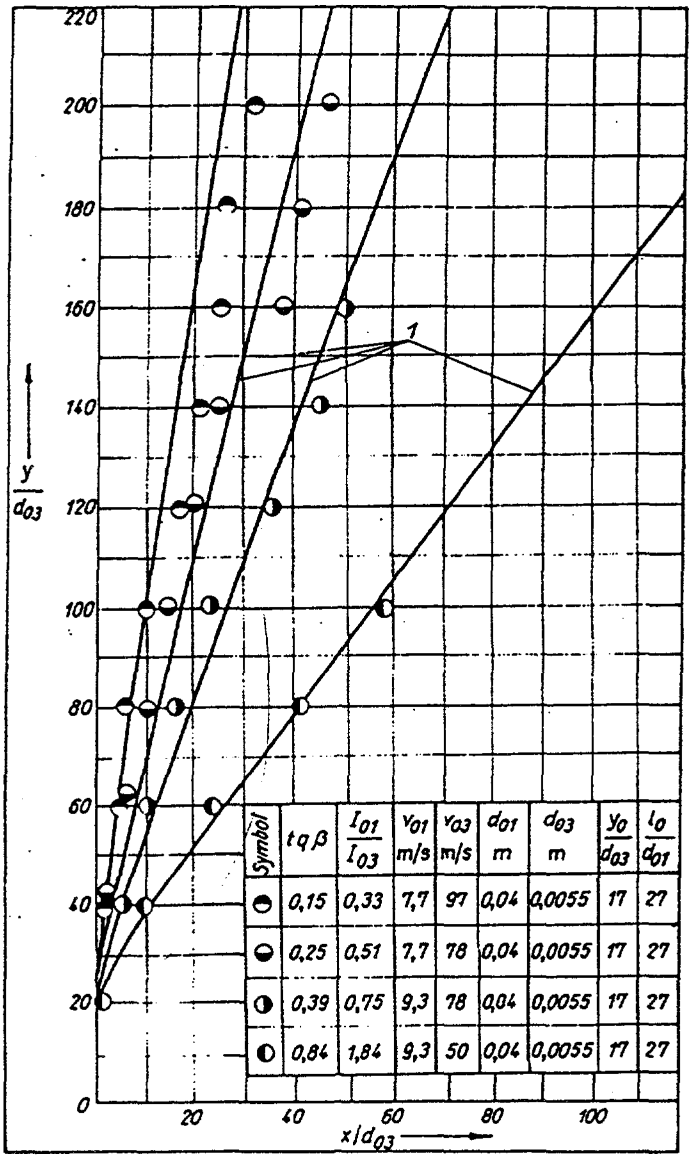


Figure 5.9. Schematic of free isothermal main stream and vertical directing jet interaction: 1 - main stream (ρ_{01}, V_{01}, l_{01}, d_{01}, K_{11}); 2 - directing jet (ρ_{03}, V_{03}, l_{03}, d_{03}, K_{13}). Reproduced from Zhivov, 1983.



Symbol	$tq\,\beta$	$\frac{I_{01}}{I_{03}}$	v_{01} m/s	v_{03} m/s	d_{01} m	d_{03} m	$\frac{y_0}{d_{03}}$	$\frac{l_0}{d_{01}}$
◓	0,15	0,33	7,7	97	0,04	0,0055	17	27
◒	0,25	0,51	7,7	78	0,04	0,0055	17	27
◑	0,39	0,75	9,3	78	0,04	0,0055	17	27
◐	0,84	1,84	9,3	50	0,04	0,0055	17	27

Figure 5.10. Trajectory of the vertical directing jet: 1 - Equation (5.37). Reproduced from Zhivov 1983.

6. JETS IN CONFINED SPACES

6.1. General description of confined flow

Current mixing type air distribution methods typically consider occupied zone ventilation with jets intercepting its upper boundary. These methods include air supply with vertical jet through ceiling mounted air duffusers, and air supply with inclined jets. They also include air supply with vertical upward directed jet or horizontal jet along one of the room surfaces. In the later case jet reaches the opposite wall/ceiling and follows room surfaces until it reaches the occupied zone (Figure 6.1). If the combination of room sizes (height, length and width) allows such airflow pattern, this room is considered to be "short" (Etheridge and Sandberg, 1996). The room where air jet dissolves before it reaches the opposite wall is considered to be "long". In such rooms occupied zone is ventilated by "reverse" flow. Initially studies of jets in confined spaces were carried out for mining, chemical and mechanical engineering applications (Abramovich, 1963, Rajaratnam 1976, Lyakhovskiy and Syrkin, 1939). In the current chapter three methods of air supply in confined spaces are discussed:

- horizontal jet supply;
- inclined jet supply;
- horizontal jet supply with directing jets;
- vertical jets.

6.2. Experimental studies of isothermal horizontal jet in confined spaces: airflow pattern, throw, velocities

First experimental data on confined air jets used for ventilation date back to 1939 when Baturin and Hanzhonkov studied air supply method with the occupied zone ventilation by "reverse" flow. Later, this method was called concentrated air jet supply. Baturin and Hanzhonkov concluded that the air flow pattern in the ventilated space depends on the location of air supply outlets and practically does not depend upon location of air exhausts.

Studies by Nelson and Stewart (1938), Bromley (1946), and Gunes (1948) allowed to receive experimental data on air velocities and temperatures distribution for this method of air supply at different room configurations, locations of air supply outlets, velocities and temperatures of air supply.

The effect of the room length, position and shape of the air supply outlets was studied by Linke (1957, 1966). These studies show there is a maximum room length, that can be effectively ventilated by the supply air jet (Figure 6.2a). For the linear (2-D) attached to the ceiling air jet supplied at the Reynolds number in the range from 1825 through 12000, the maximum room length does not exceed three room width. The rest of the room downstream is poorly ventilated. When the air supply slot is symmetrical (located at the 1/2H) the effectively ventilated room length increases to 4 room width. Air supply through a round nozzle with a non-attached jet allows to increase the effectively ventilated room length up to five transversal cross-section sizes, $(B \times H)^{1/2}$, of the room.

The airflow pattern in rooms ventilated by linear attached jets with **L/H** ratio greater than that for effectively ventilated rooms was studied by Schwenke (1976) and Müller (1977). The results of their air velocity measurements and visualization studies indicate that there are secondary vortexes, that are formed downstream the room and in the room corners. The number of the downstream vortexes and their size depend upon the room length (Figure 6.2b). Mass transfer between the primary vortex and the secondary vortex depends upon the difference in characteristic air velocities in the corresponding flows $U_1 - U_2$ and can be described using the Stanton number, St (Müller 1977):

$$St = \frac{U_1 + U_2}{U_1 - U_2} \; timds \; \frac{1}{4\,\sigma\,\sqrt{\pi}} \tag{6.1}$$

where

$$\sigma = \frac{1}{2\sqrt{\chi * c * \dfrac{U_1 - U_2}{U_1 + U_2}}} \tag{6.2}$$

χ and **c** are empirical coefficients. For the jet spreading along the wall ($\mathbf{U_2 = 0}$) Stanton number is equal to 0.01. This approach was used to predict mass transfer between the primary and secondary vortexes and the characteristic air velocities in the secondary vortexes. These predictions were compared with experimental data. Though experimental data deviates from predicted air velocities, the proposed model allows one to understand the mechanism of mass transfer between different room zone. Average rotation velocity and mass transfer decreases from the primary vortexes to the secondary and the subsequent vortexes.

Influence of room transversal cross-section configuration on airflow pattern created by air jet supplied through the round nozzle in proximity of the ceiling was studied by Baharev and Troyanovsky (1958) and P.Nielsen (1981), see Figure 6.3. Based on experimental data they concluded that when the room width **B** < 3.5**H** the jet attaches to the ceiling and spread filling the whole width of the room in a manner of a linear jet. The reverse flow develops under the jet. When **B** > 4**H**, the reverse flow also develops along the jet sides. Baharev and Troyanovsky (1958) indicated that air temperature and velocity distribution in the occupied zone is more uniform when the jet develops in the upper zone and the occupied zone is ventilated by the reverse flow. Thus, they proposed to limit room width to 3~3.5$\mathbf{H_r}$.

Detailed experimental data was received by Sadovskaya (1950, 1955) on physical model in isothermal conditions. She has found that the confined air jet has two critical cross-sections Figure 6.4). The first cross-section where the ratio of jet cross-sectional area to the area of ventilated space equals to 0.24. Till this cross-section jet develops as free one. Between the first and the second critical cross-section, where the jet occupies 40% of the room cross-sectional area is the zone of confined jet. Beyond the second critical cross-section is the zone of jet degradation. Sadovskaya has found that the length of all three zones depend upon the coefficient of turbulent structure **a** of the jet at the air supply and received empirical equations for the length of each zone and air velocities in the air jet and in the reverse flow:

$$X_1 = \frac{0.1\sqrt{BH}}{a} \tag{6.3}$$

$$X_2 = \frac{0.71\sqrt{BH}}{3.4\ a} \tag{6.4}$$

In these studies nozzles with **a** = 0.07 were used. For the values of parameter $(\mathbf{BH/A_o})^{1/2}$ used in the studies from 2.44 through 71.5, maximum jet throw is in the following range:

$$X_{max} = (from\ 4.07\ through\ 5.1)\sqrt{BH} \tag{6.5}$$

Based on experimental data received by Sadovskaya (1950,1955) and Rozenberg (1949) as well their own experimental results Baharev and Troyanovsky (1958) derived empirical equations to design air distribution with horizontally supplied confined jets.

Experimental studies conducted by Grimitlyn (1978) allowed to generalize equations (6.3) and (6.4) for air diffusers with different velocity decay characteristics K_1:

for compact jets

$$X_1 = 0.22\ K_1\sqrt{BH} \tag{6.6}$$

$$X_2 = 0.31\ K_1\sqrt{BH} \tag{6.7}$$

$$X_{max} = 0.62\ K_1\sqrt{BH} \tag{6.8}$$

for linear jets

$$X_1 = 0.1\ K_1^2\ H_r \tag{6.9}$$

$$X_2 = 0.15\ K_1^2\ H_r \tag{6.10}$$

$$X_{max} = 0.3\ K_1^2\sqrt{BH} \tag{6.11}$$

To avoid high velocities in the occupied zone due to direct effect from the supply air jet and to increase the zone length effectively ventilated zone by a single jet in rooms with a height, H_r, from 4m to 10m, Baharev and Troyanovsky (1958) proposed to supply air from the height $\mathbf{h_o}$ = 0.6~0.7$\mathbf{H_r}$.

Effect of jet proximity to the ceiling. Studies Sawyer (1960), Bourque and Newman (1960) and Regenscheit (1962) showed (Figure 6.5), that the two dimensional jet supplied from the slot with a width $\mathbf{h_o}$ at the distance $\mathbf{Z}$ from the ceiling surface will become attached to this surface at the distance $\mathbf{X_a}$ that can be evaluated from the following equation:

$$\frac{X_a}{h_o} = 0.2 + 2.7\left(\frac{Z}{h_o}\right)^{0.8} \tag{6.12}$$

Research reported by Jackman (1970) showed, that the effect of proximity of air supply to the ceiling is also important when air is supplied with compact jets. The non-uniform entrainment to either side of the jet resulted in a force deflecting the jet towards the ceiling. The attraction of the jet towards the surface is greater the closer the air supply is to the ceiling and the higher its aspect ratio (grill width over grill height).

Effect of ceiling beams/obstructions in the jet zone. Ceiling beam will not affects jet attachment to the ceiling if it is located further than $1.6\mathbf{X_a}$ from the air supply outlet (Regenscheight 1975). If it is located closer, the impingement of the jet with this beam will change the jet direction.

Effect of ceiling beams and light fittings on ventilation jets was also studied by Holmes and Sachariewicz (1973). Their studies were limited to two-dimensional case: air supply through linear slot and two dimensional barrier (Figure 6.6). The results of these studies show, that the ceiling jet can take one of three courses when encounters an obstruction:

1. Separate from the surface and take up a flow angle approximately equal to the angle between the upstream face of the obstruction and the surface;
2. Separate from the surface and re-attach some distance downstream from the barrier;
3. Almost ignore the existence of the barrier.

The jet will separate from the surface if the axial distance between the slot and the obstruction, $\mathbf{X_d}$, is less than a specified critical distance $\mathbf{X_c}$ (Figure 6.7). The values of $\mathbf{X_c}$ given in Figure 6.7 are only for an obstruction of transverse dimensions **w** equal to or less than the slot span **s**.

In the second course, the flow downstream of the barrier can be adequately represented by a determination of (a) the maximum separation of the line of maximum velocity from the surface (Figure 6.8) and (b) the velocity decay after the barrier:

$$\frac{V_{max}}{V_o} = 2.2\left(1 - 0.785\frac{w}{s}\right)^{1/2}\left(b_o\frac{\quad}{X - X_d}\right)^{1/2} \tag{6.13}$$

where: $\mathbf{X_d}$ is a distance of obstruction to the slot, $0.5 < w/s < 1$.

Obstruction does not affect the jet if the obstruction is further than about eight critical distances ($\mathbf{X_c}$) from the slot. In this case the velocity decay of the jet may be obtained from the following equation:

$$\frac{V_m}{V_o} = 2.2\left(\frac{b_o}{X - X_o}\right)^{1/2} \tag{6.14}$$

where $\mathbf{b_o}$ is an effective slot width and $\mathbf{X_o}$ the virtual origin of the jet.

Although the tests were conducted only for two-dimensional cases, authors suggest, that their results can be extended to three dimensional cases as follows:

- If the obstruction span is less than half the slot span the effect of obstructions can be ignored provided $\mathbf{X_d} > \mathbf{X_c}$;
- The value of $\mathbf{X_c}$ for a short barrier will be less than for a long one and it will be safe to use the values obtained in the studies;
- The velocity decay downstream of a short barrier may be represented by the equation:

$$\frac{V_m}{V_o} = 2.2\left(1 - 0.785\,\frac{w}{s}\right)^{1/2}\left(\frac{b_o}{X - X_d}\right)^{1/2} \tag{6.15}$$

  where $0.5 < \mathbf{s/w} < 1$.

- If the barrier is longer than the slot the flow will be deflected at right angles to the normal jet trajectory, causing a possible thinning of the jet at the barrier and increasing the critical barrier distance. It is not possible to estimate accurately the extend of such an increase from the studies on two dimensional situation. Authors consider that such an increase will not exceed 100%.

Graphical interpretation of the factors influencing the critical distance $\mathbf{X_c}$ from air supply to the linear obstacle with a height $\mathbf{d_c}$ for air supply through the slot diffuser with a height $\mathbf{h_o}$ and for air supply through a round nozzle with outlet diameter $\mathbf{d_o}$ (Nielsen 1995) are presented in Figure 6.9. Non-isothermal flow has an influence on the critical distance and Archimedes number Ar is an important parameter together with the geometrical relations (Nielsen 1995). Ventilation with cooled air increases the effect of obstacle and warm air supply decreases this effect (Söllner and Klinkenberg 1972, Nielsen 1980).

Air can be supplied in rooms by one or several jets. Air supply openings can be located along one wall - parallel air jet supply (Figure 6.10a) and/or on the opposite walls - contrary directed jets supply (Figure 6.10b). In special cases air can be supplied in a fan-type manner (Figure 6.10c).

Air circulation with a parallel jet supply is illustrated on the Figure 6.11. Jets are located at the distance **t** from each other and each jet forms return flow similar to that induced by a single jet in the room with a width $\mathbf{B} = \mathbf{t}$. Thus, in the case of **N** parallel jets supply, the room should be considered as divided into several zones with a width $\mathbf{B} = \mathbf{B_r/N}$ separated from each other by airtight walls.

6.3. Analytical studies

Abramovich (1960) was the first to study axisymmetric confined jet analytically. He suggested the method based on utilizing of the equations of continuity and momentum conservation. He also assumed that the width of the layer of jet mixing with a counterflow equals to the width of a free jet with a velocity distribution according to Shlihting formula:

$$\frac{V - V_{rev}}{V_x - V_{rev}} = \left[1 - \left(\frac{Y}{b}\right)^{3/2}\right]^2 \tag{6.16}$$

where **b = 0.22 X** - half of the free jet width.

Analytical methods suggested by Shepelev and Tarnopolsky (1965), Grimitlyn and Pozin (1974) and Sychev and Volov (1981) differ from the one described above only by the way the authors described velocity distribution in the mixing layer :

$$\frac{V - V_{rev}}{V_x} = e^{-\frac{1}{2}\left(\frac{Y}{cX}\right)^2}$$ - Shepelev and Tarnopolsky (6.17)

$$\frac{V - V_{rev}}{V_x} = e^{-0.7\left(\frac{K_1 Y}{0.66X}\right)^2}$$ - Grimitlym and Pozin (6.18)

$$\frac{V - V_{rev}}{V_x - V_{rev}} = 1 - 6\left(\frac{Y}{b}\right)^2 + 8\left(\frac{Y}{b}\right)^3 - 3\left(\frac{Y}{b}\right)^4$$ - Sychev and Volov (6.19)

It is assumed in above mentioned methods that the influence of confined space on the supplied jet can be described by the reduction of axial velocity component on the value $\mathbf{V_{rev}}$, like for jet development in the counterflow. The value of $\mathbf{V_{rev}}$ is assumed to be the same throughout each cross-section but variable along the jet length. The value of $\mathbf{V_{rev}}$ can be found from the continuity equation which in the case of jet distribution in a space of cylindrical shape can be presented as

$$2\int_o^{r_*} V\,rdr + V_{rev}(R^2 - r_*^2) = O \tag{6.20}$$

for air supply and air exhaust located in the same wall and for air supply and air exhaust located in the opposite walls.

According to Shepelev and Tarnopolsky (1965) air velocity on the axis of the jet at the distance **X** from the outlet for air supply and exhaust located on the same wall can be calculated from the equation

$$V_{xc} = V_x \left[1 - \frac{r_o}{R} \frac{1 - \exp\left[-\frac{1}{2}\left(\frac{R}{cX}\right)^2\right]}{\frac{1}{2}\left(\frac{R}{cX}\right)^2 K_1\sqrt{\pi}\ \frac{r_o}{R}} \right] \tag{6.21}$$

and the maximum (in the cross-section) velocity in the reverse flow-- from the equation

$$V_{rev} = V_o \left\{ K_1 \frac{\pi r_o^2}{X} \exp\left[-\frac{1}{2}\left(\frac{R}{cX}\right)^2\right] - \frac{r_o}{R} \frac{1 - \exp\left[-\frac{1}{2}\left(\frac{R}{cX}\right)^2\right]}{\frac{1}{2}\frac{R}{cX}} \right\} \tag{6.22}$$

Velocity in the reverse flow reaches maximum value at $\mathbf{X_*} = 4.88\ \mathbf{R}$ equal to

$$V_{rev}^{max} = 0.656\sqrt{\frac{M_o}{\rho\pi R^2}} \tag{6.23}$$

Equations presented above can be applied to spaces of rectangular shape by replacing $\pi\mathbf{R}^2$ for **BH**.

Similar approach was used by Zhivov (1983, 1994) in his studies of the system of coaxial jets in confined space. The distance **X** from the air diffuser to the cross-section with a maximum velocity in the reverse flow for the case without coaxial jets was found to be equal to 1.9 $(\mathbf{BH})^{0.5}$ at $\mathbf{K_1} = 6.2$ and 1.4 $(\mathbf{BH})^{0.5}$ at $\mathbf{K_1} = 4.5$. For a non-isothermal jet also it was found (Zhivov, 1994) that the reverse flow and confining surfaces increase the upper limit of the cold or heated supply air temperature $\mathbf{\Delta t_o}$, which insures a horizonal jet projection.

The results of different analytical and experimental studies of the confined horizontal jet described above are presented in the Table 6.1. The main reason in difference of analytical results is different approximations of reverse flow velocity profiles.

Influence of the reverse flow on the centerline velocities $\mathbf{V_{xc}}$ was proposed (Grimitlyn, 1994) to be expressed by the coefficient $\mathbf{K_c}$

$$V_{xc} = V_x K_c \tag{6.24}$$

which as it was shown above can be derived analytically. The value of $\mathbf{K_c}$ depends up on the ratio of the cross-section area of the free jet and the corresponding cross-section of the room. The graphs for evaluating $\mathbf{K_c}$ for compact, radial and linear air jets are presented in Figure 6.12.

6.3. Experimental studies of horizontal heated and cooled air supply in confined spaces

Gobza (1947, 1965) studied air supply with concentrated jets on physical models and in field and concluded that temperature stratification along the room height may occur if improper supply air temperature difference and air exchange rate are selected. Among the field cases reported by Gobza are industrial halls as long as 150 m (at width equal to 50 m and height - 12-15 m).

The effect of supply air temperature on jet behavior in confined spaces was studied by Müllejans (1966). Studies of cooled air jets were conducted in rooms with a size from 1.0 m x 1.0m x 1.6m to 2.27m x 3.33m x 5.31m and an air supply through the slot ($b = B_r$) or rectangular opening (**b << B_r**). Numerous smoke photographs were taken reflecting supply situations with different **Re** and **Ar** numbers. Archimedes number was defined by Müllejans as follows:

$$Ar = \frac{g\, D_h(t_w - t_o)}{V^2\, (T_w + T_o)/2} \qquad (6.25)$$

where: $V = Q_o/(B_r H_r)$; $D_h = \dfrac{4\, B_r\, H_r}{2(B_r + H_r)}$ - hydraulic diameter; $\mathbf{t_w}\,(\mathbf{T_w})$, $\mathbf{t_o}\,(\mathbf{T_o})$ - wall and supply air temperature,°C(K); **g** - acceleration due to gravity.

Müllejans has reported, that with air supply through rectangular openings, the jet behaves more or less as an isothermal flow when **Ar** $< 10^4$.

To establish a criterion for any size of room and outlet Ar number was adjusted using a geometrical factor. The modified **Ar*** was defined as:

$$Ar* = Ar\, \frac{b_o\, h_o}{D_h^2} \qquad (6.26)$$

With a common value of $\mathbf{b_o h_o/D_h^2} = 1/250$, the airflow pattern will be similar to isothermal with a modified **Ar*** number limited to 40.

In the case of room ventilated by a linear jet, this jet deflects towards the ceiling immediately after entering the room. Maximum Ar values depend upon the **L/H** ratio and are as follows:

L/H	4.7	3.0	2.0	1.0
Ar_{max}	2000	3000	10000	11000

Experimental studies conducted by Grimitlyn (1978) on heated and chilled confined jets allowed to determine that air flow pattern remains the same as for isothermal air supply when $\mathbf{Ar_x} < 0.2$ at $\mathbf{X = 0.22\, K_1\, \sqrt{(BH)}}$, in rooms with **H/B** ratio from 0.3 to 1.0, where

$$Ar_x = \frac{K_2}{K_1^2} \frac{g\sqrt{A_o}\,\Delta t_o}{V_o^2\, T_{o.z.}} \left(\frac{X}{\sqrt{A_o}}\right)^2 \tag{6.27}$$

The above limitation on the local Archimedes number results in the following equation for maximum temperature difference of supplied air:

$$\Delta t_o = 122\,\frac{V_o^2\sqrt{A_o}}{K_2 BH} \tag{6.28}$$

Similar studies were provided by Troyanovsky (1969), who concluded that to maintain air flow pattern in the room with warm or cooled air supply as in isothermal conditions, it is necessary that the raise of the horizontally supplied jet does not exceed $\mathbf{\Delta Y = 0.1BH}$ at the distance from the outlet $\mathbf{X = 0.15\, K_1 \sqrt{(BH)}}$. From this assumption the following equation for the maximum air temperature difference was derived

$$\Delta t_o = 1300\,\frac{V_o^2\sqrt{A_o}}{K_1 K_2 BH} \tag{6.29}$$

Comparing Equations (6.31) and (6.32) one can see that the value of the maximum temperature difference computed using Equation (6.32) is higher than that using Equation (6.31). Results of experimental studies on physical model (Grimitlyn, 1994) indicate that when **H/B** > 1, the limitation on supply air temperature difference should be even more constraint.

Table 6.1. Results of experimental and analytical studies of the compact confined jet

Reference	Throw	Throw definition	Zone 1 + Zone 2	Maximum velocity in the reverse flow
Baharev and Troyanovsky (1958)	$(4.7\text{-}5.4)\surd(BH)$	$V_{rev}=(0.05\text{-}0.1)V_o\surd(A_o/BH)$	$2.0\ \surd(BH)$	$0.78\ V_o\ \surd(A_o/BH)$
Abramovich (1960)	$3.5\ \surd(BH)$	$V_x = (0.05\text{-}0.1)V_o$	$2.4\surd(BH)$	$0.88\ V_o\ \surd(A_o/BH)$
Shepelev and Tarnopolsky (1965)	$5\ \surd(BH)$	$V_{rev}=0.07V_o\surd(A_o/BH)$	$2.75\surd(BH)$	$0.66\ V_o\ \surd(A_o/BH)$
Grimitlyn and Pozin (1974)	$0.7K_1\surd(BH)$	-	$0.31\ K_1\ \surd(BH)$	$0.78\ V_o\ \surd(A_o/BH)$
Gnatyuk et al.(1977)	$4.3\surd(BH)$ - $3.3\surd(A_o/BH)$	$V_x = (0.07 - 0.1)V_o$	-	-
Sychev and Volov (1981)	$5.5\surd(BH)$	$V_x = 0$	$3.2\surd(BH)$	$0.7V_o\surd(A_o/BH)$
Schwenke (1976)	$5.0\surd(BH)$ - compact jet 3H - linear non-attached jet	$V_{x\,ave}=0.2$ m/s or $V_{x\,max}=0.5$ m/s	-	-
Nielsen (1981,1987)	$5.0\surd(BH)$ - compact jet 4H - linear attached jet	-	$3\surd(BH)$	$0.95V_o\surd(A_o/BH)$ round attached jet
Zhivov (1983, 1994)	$3.9\surd(BH),\ K_1 = 6.2$ $3.4\surd(BH),\ K_1 = 4.5$	Cross-section velocities are uniform with an unstable direction	$1.9\surd(BH),\ K_1 = 6.2$ $1.4\surd(BH),\ K_1 = 4.5$	$0.73\ V_o\ \surd(A_o/BH)$

6.5. The effect of confinement on inclined air jet

There are only few publications where air supply with inclined jets in confined spaces is discussed. Numerous studies on horizontal and inclined air jet trajectory, velocity and temperature decay under buoyancy were carried out. However, these studies were conducted with free (non-confined) jets and were discussed in previous chapter. Though, there is no direct mentioning of inclined jet supply, discussion by Regenscheight (1970) can be related to this topic. This paper describes the results of studies of horizontal cooled air supply from linear and rectangular openings. Graphs in Figure 6.13. show how the relative distance $\mathbf{X_o/L}$ from the supply opening to the point of jet impingement with the floor surface is influenced by the modified Archimedes number Ar:

$$Ar = \frac{g\ \Delta T_o\ 4\ (BH)^3}{T_{o.z.}\ Q_o^2\ 2(B+H)} \tag{6.30}$$

Experimental and analytical studies of non-isothermal inclined jet in confined spaces were carried out by Zhivov (1993). Experimental studies were conducted on the physical model. The ratio of the model dimensions **L x B x H** was changed so that the value **H/B** was from 0.3 to 3.0 and $\mathbf{L/\surd(B\ x\ H)}$ = 2.4 - 4.9.

Visualization of airflow in room with a smoke and silk threads allowed to describe airflow pattern in room with inclined jet supply. Airflow created by inclined jet impinging the floor surface can be divided conditionally into three zones (Figure 6.14): free or confined jet (1), impingement zone (2) and flow along the floor (3). The width of the jet depends upon the supply characteristics, which can be primarily described the velocity decay characteristic $\mathbf{K_1}$. Some air diffusers (e.g., ventilation grilles) can create jets with coerced angle of deflection only in one direction. (Zhivov 1993).

The first zone of the jet can be described using equations for velocity and temperature decay as well as jet trajectory with a coefficient $\mathbf{K_c}$ accounting for jet confinement. The impingement zone can be characterized by a significant change of the static pressure and great curvature of the air current lines. After the impingement, the radial flow is formed as if it is supplied from the side surface of the truncated cylinder with a uniform initial velocity of $\mathbf{U_m^*}$. In the basement of the cylinder, there is particular line, that crosses the quasisorce of the radial flow. Equations provided in the paper allow to evaluate velocities along the branches with a maximum airflow, minimum airflow and along the particular line.

When the width of the jet (calculated for free conditions) is less than the width of the room, air flow after jet impingement with a floor is similar to that in non-confined conditions. When the side directed flow (along the particular line) reaches the wall, it is divided into two branches: one following direction of the branch with a maximum airflow and another flowing in the opposite direction.

When the air directed backwards reaches the back wall of the room, it flows upwards to be induced by the jet within its first zone (Figure 6.15).

Circulation zone is created above the branch with a maximum airflow spreading along the floor. The reverse flow also is induced by the inclined jet within its first zone.

If the width of the jet (calculated for free conditions) at the point of its intercept with the occupied zone exceeds the room width, side walls transform this jet to the flow as if it was formed by the linear jet impingement with a floor.

In design of air distribution with inclined cooled air jets, the following parameters should be considered: air velocity and temperature at the point of jet intercept with the occupied zone - for practical purpose this cross-section can be considered as the border between the first and the second zone of impinging jet; and velocities along jet branches - maximum airflow branch, minimum airflow branch and the branch along the particular line.

The latter information is important to evaluate the size of occupied zone, that can be effectively ventilated by inclined jet. It was proposed (Zhivov 1993), that the occupied zone of rooms is well ventilated by inclined jet (particularly in industrial rooms with contaminant release) if air velocity in the occupied zone exceeds 0.1 m/s.

The influence of confinement on air velocity $\mathbf{U_m}$ in the flow along the floor can be accounted for with the coefficient $\mathbf{K_c}$:

$$U_m^c = K_c \, U_m \tag{6.31}$$

The value of this coefficient depends on the relative height of the flow along the floor $\mathbf{h_f / H_r}$(calculated in the free conditions), where:

$$\mathbf{K_c = 1}, \text{ when } \mathbf{h_f \leq 1.1}, \; \mathbf{K_c = 1.79 - 0.72\, h_f / H_r}, \text{ when } \mathbf{h_f > 1.1} \tag{6.32}$$

6.6. Air supply with directing jets

Air supply with directing jets is one of modifications of concentrated air supply designed to increase the throw of primary flows in confined spaces. In one common modification of such system, the main streams of heated or cooled air are supplied through a small number of air outlets (grills) at low initial velocity V_{01} and distributed within the space by horizontal (coaxial with main streams) and vertical (supplied perpendicular to the main streams) or only horizontal directing jets (Figure 5.5). These jets are discharged at high velocities ($\mathbf{V_{02}}$ for horizontal directing jets and $\mathbf{V_{03}}$ for vertical directing jets) from nozzles having small outlet diameters ($\mathbf{d_{02}}$ and $\mathbf{d_{03}}$ for horizontal and vertical directing nozzles, respectively).

The momentum of horizontal directing jets increases the length of area ventilated by the main stream and allows the initial temperature differential of air supplied by the main streams to increase, relative to conventional systems. Vertical directing jets supplied at right angles to main streams inject fresh air (and excessive heat or cold contained in these streams) and deliver it to the workplaces, which may be located in the aisles between the process equipment.

Studies have shown (Zhivov 1994) that the air circulation within the space is caused primarily by the energy of the directing jets which are an order of magnitude greater than the energy of the main stream. This is why the variations in the air flow, supplied by the main stream, does not effect the circulation pattern. Interaction of the main stream with horizontal and vertical directing jets is discussed in Sections 5.3 and 5.4.

6.7. Air supply with vertical jets

Air supplied in confined space by downward vertical jets create similar flow pattern as in the case of air supply by horizontal non-attached jets. With vertical air supply occupied zone is ventilated directly by air jets. Grimitlyn (1993) suggests that the area of occupied zone ventilated by one jet is sized based on the jet's cross-section area at the point of its entering the occupied zone. The jet cross-section area and configuration depend upon the height of air supply, type of air jet and diffuser characteristics ($\mathbf{K_1}$ and $\mathbf{K_2}$). More discussion about the occupied zone size selection with air supply by vertical jets is in Section 8.

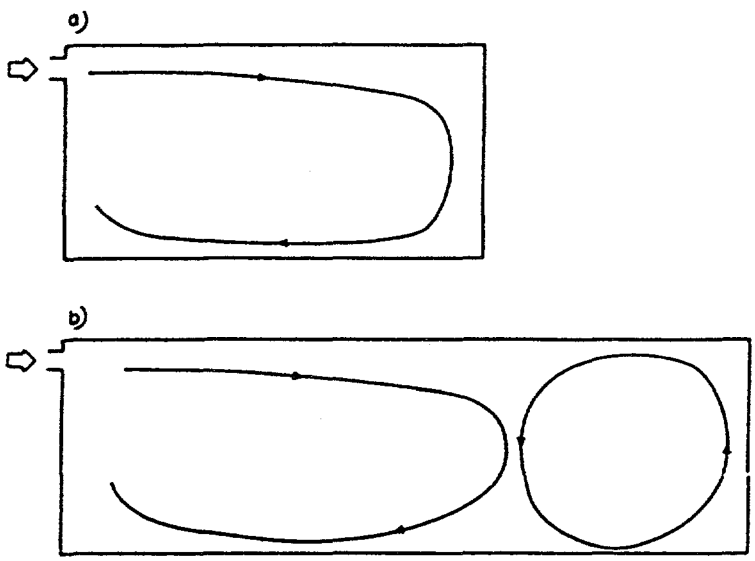


Figure 6.1. Jet flow in a room: a - "short" room, b - "long" room. Reproduced from Etherdge and Sandberg (1996).

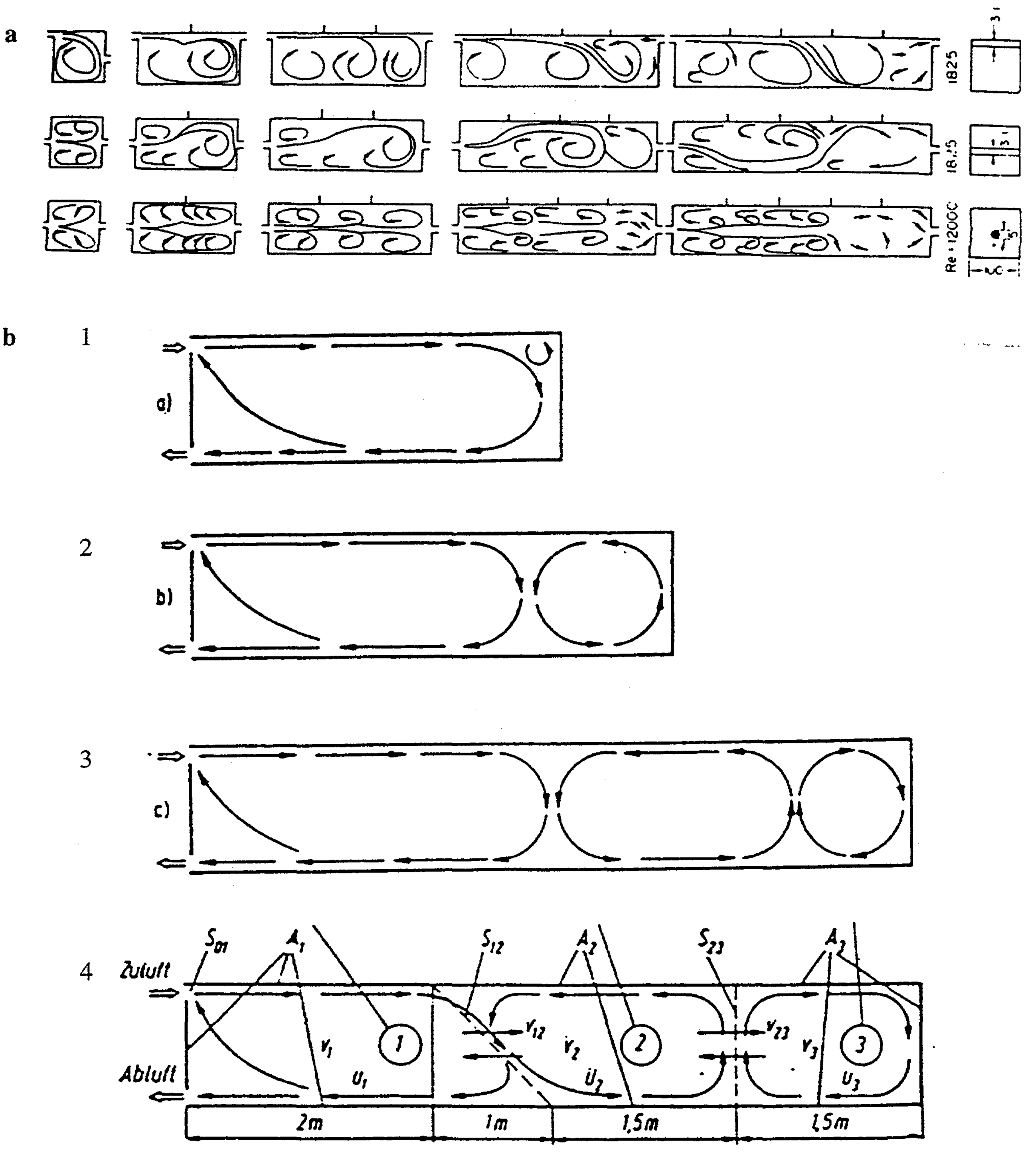
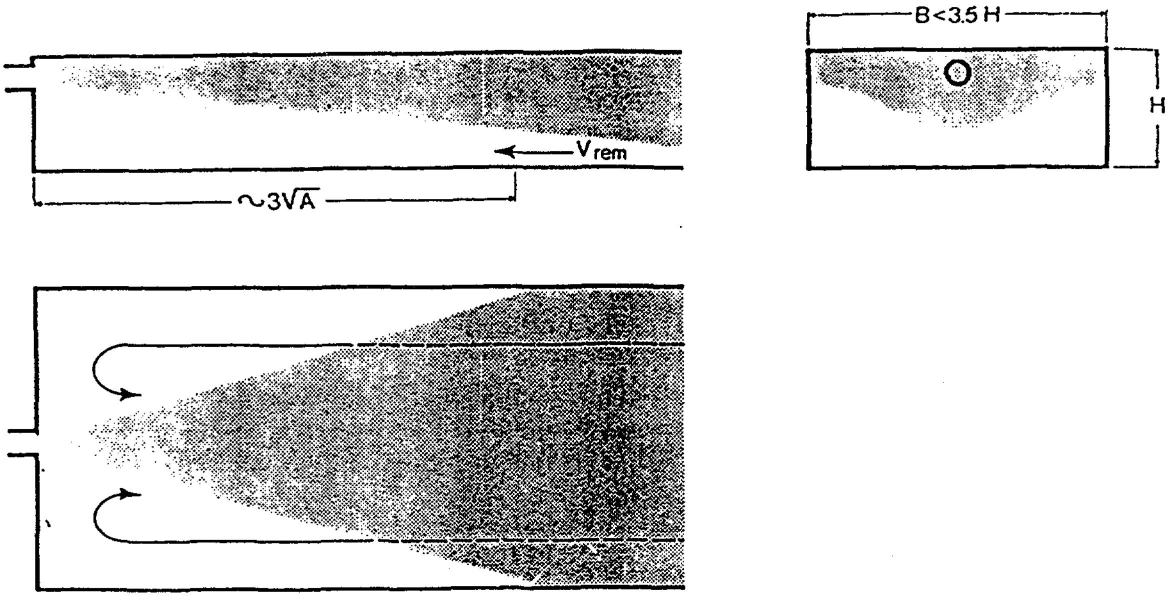


Figure 6.2. Flow patterns in rooms of different lengths with various types of air supply and exhaust: a - reproduced from Linke (1966), b - reproduced from Müller (1977), 1 - $\mathbf{L_r/H_r}$ = 3; 2 - $\mathbf{L_r/H_r}$ = 4; 3 - $\mathbf{L_r/H_r}$ = 6; 4 - schematic of primary, secondary and tertiary vortexes in the room with $\mathbf{L_r/H_r}$ = 6.

a



b

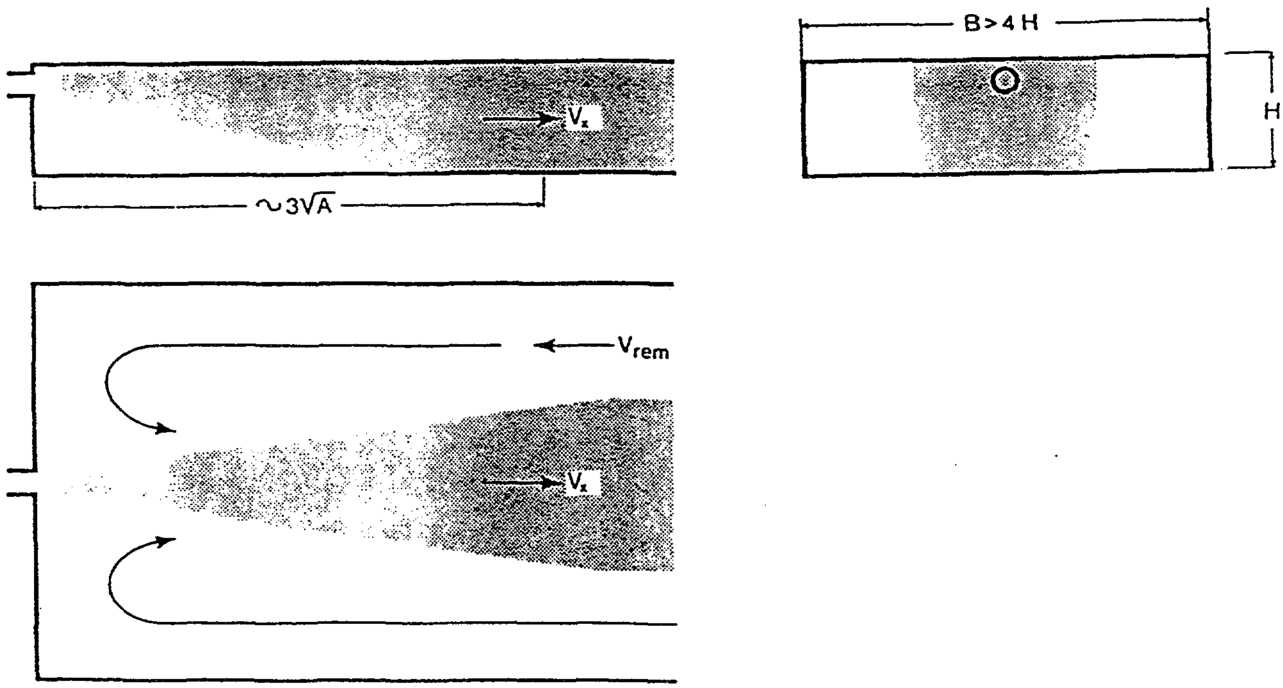


Figure 6.3. Influence of room configuration on airflow pattern: a - **B/H** < 3.5; b - **B/H** > 4. Reproduced from Nielsen (1981).

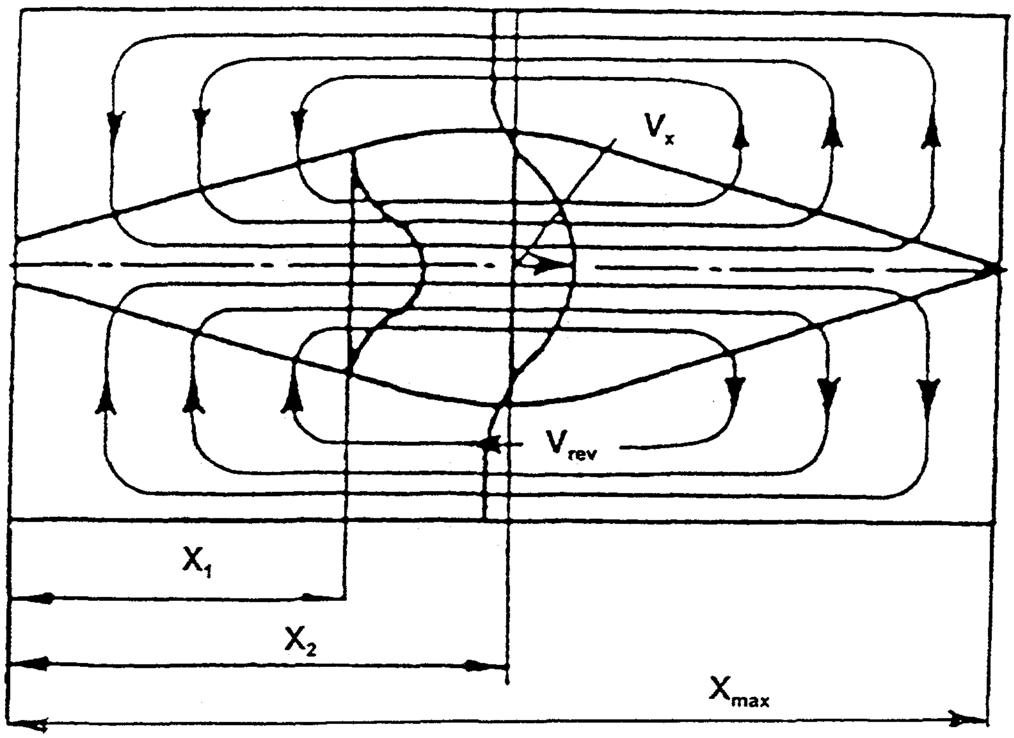


Figure 6.4. Schematic of air jet in confined space proposed by N.N. Sadovskaya. Reproduced from Grimitlyn (1970).

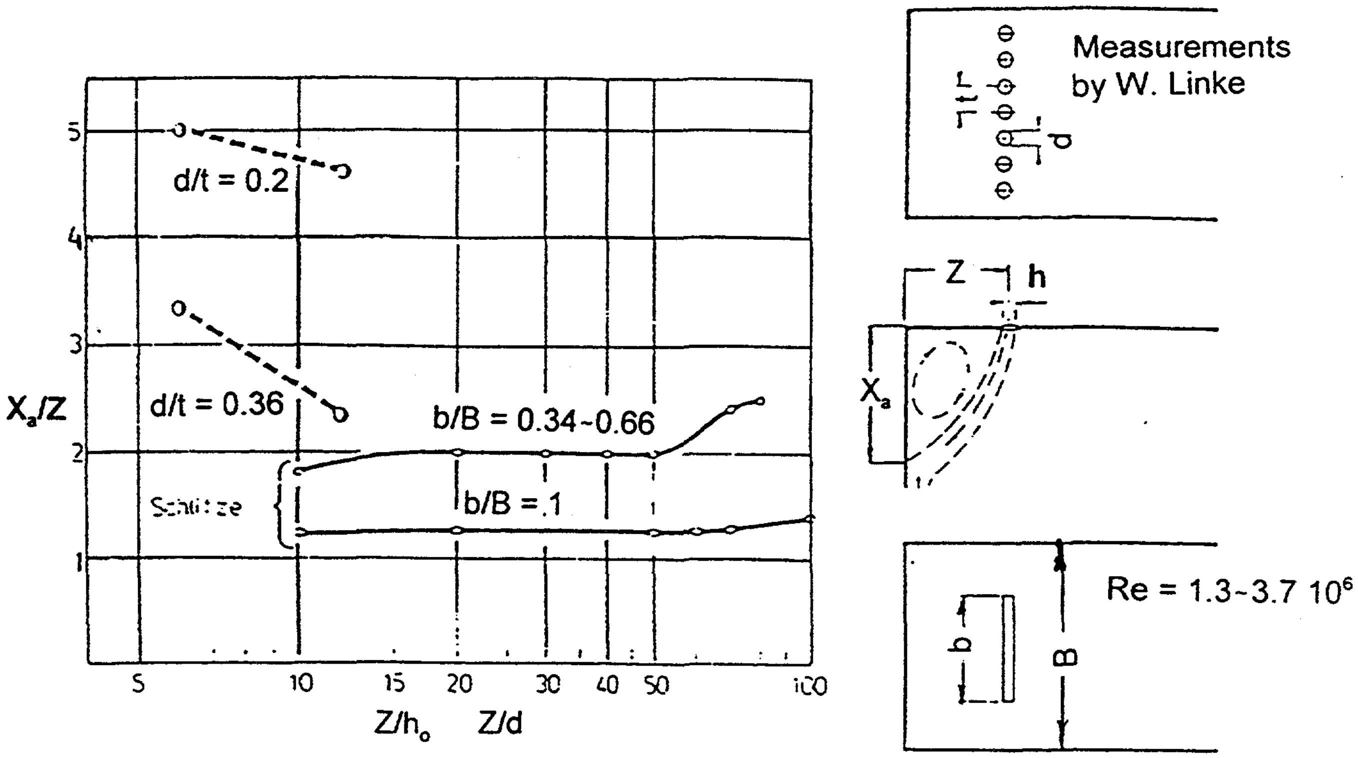


Figure 6.5. Reattachment length X_a vs. the distance **Z** from the ceiling surface to the supply outlet. reproduced from Regescheight (1962).

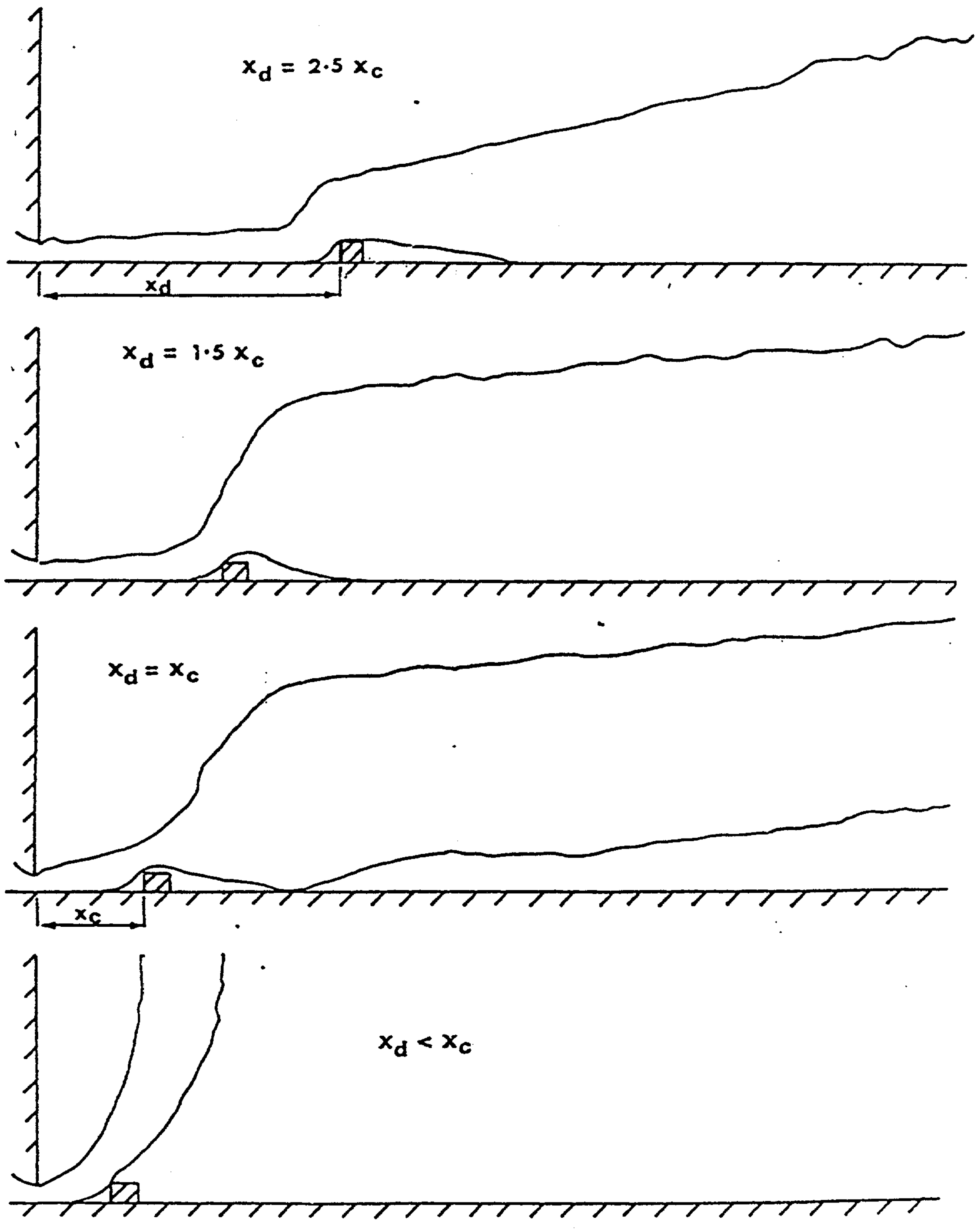


Figure 6.6. Beam influence on the airflow pattern along the ceiling. Reproduced from Holmes and Sachariewitz (1973).

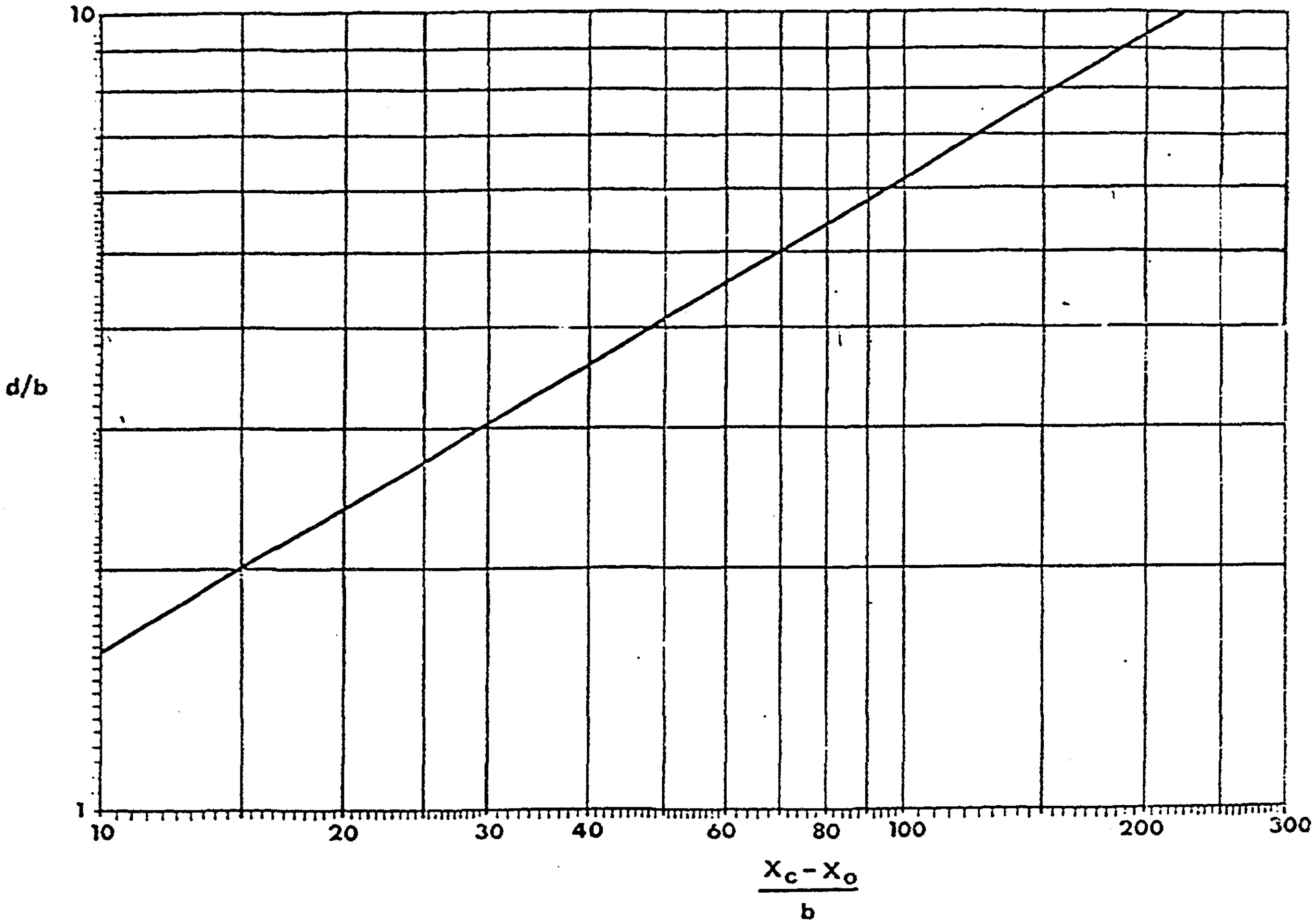


Figure 6.7. Critical distance $\mathbf{X_c}$ from the slot to the beam: **b** = width of two-dimensional slot, **d**= beam height, $\mathbf{X_o}$ = jet core zone length. Reproduced from Holmes and Sachariewitz (1973).

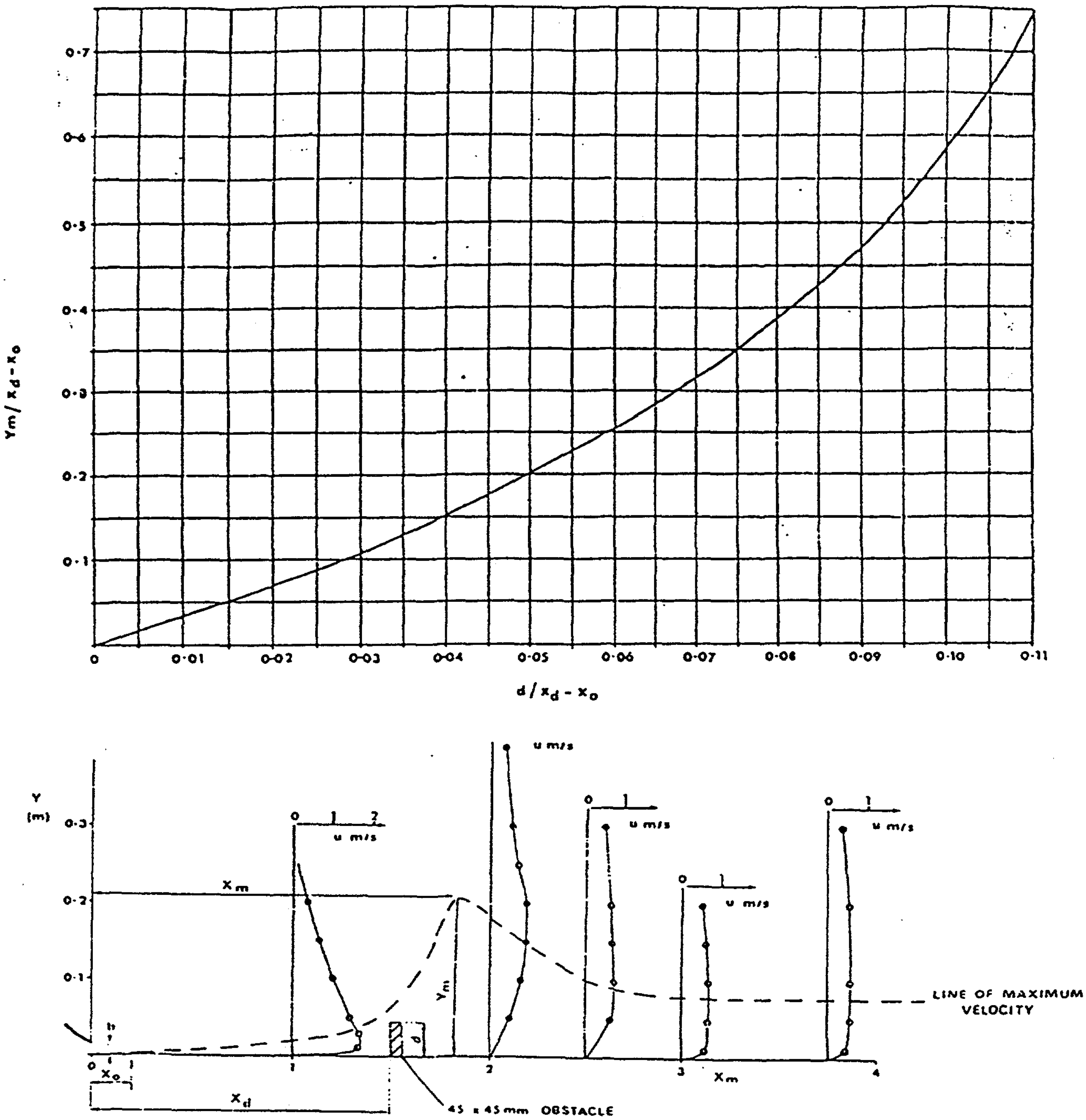


Figure 6.8. Maximum jet separation Y_m from ceiling: **b** = width of two-dimensional slot, **d**= beam height, X_o = jet core zone length. Reproduced from Holmes and Sachariewitz (1973).

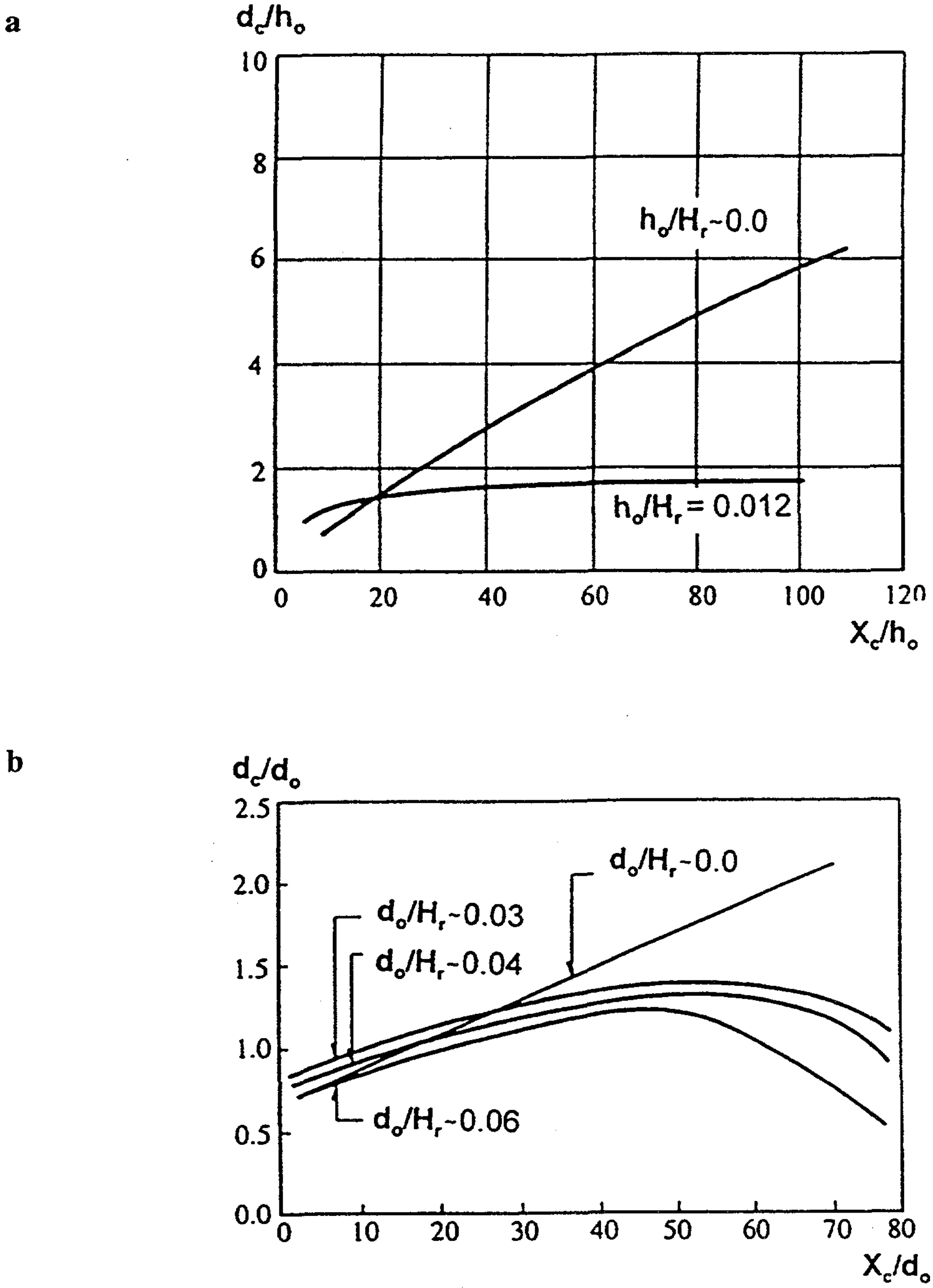


Figure 6.9. Critical height of an obstacle $\mathbf{d_c}$ vs. distance from supply slot with a height $\mathbf{h_o}$ (a), and supply nozzle with diameter $\mathbf{d_o}$ (b). Reproduced from Nielsen (1995).

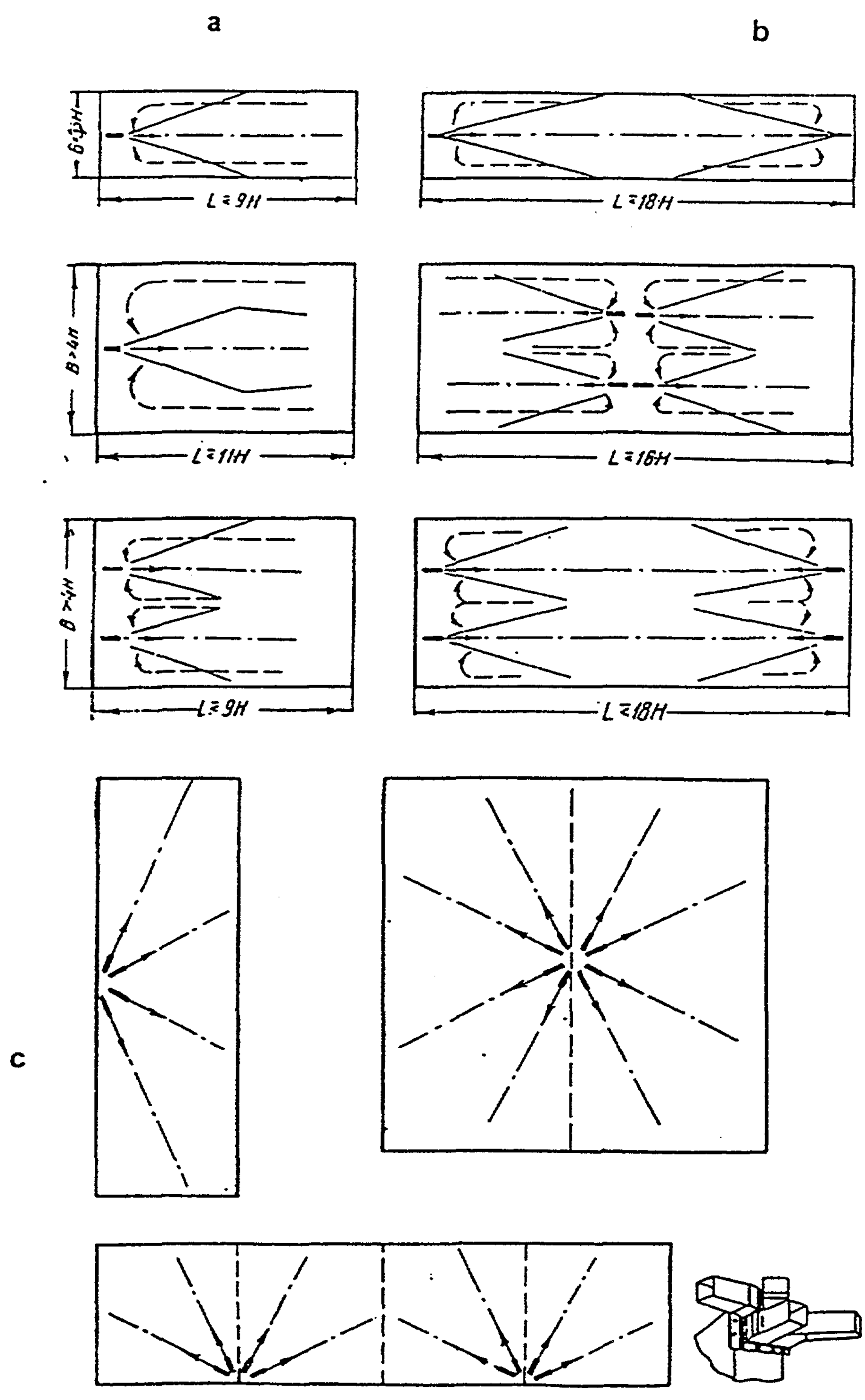


Figure 6.10. Schemes of room ventilation with parallel jet supplied from the same wall (a), from opposite walls (b), and in a fan-type manner (c) Reproduced from Baharev and Troyanovsky (1958).

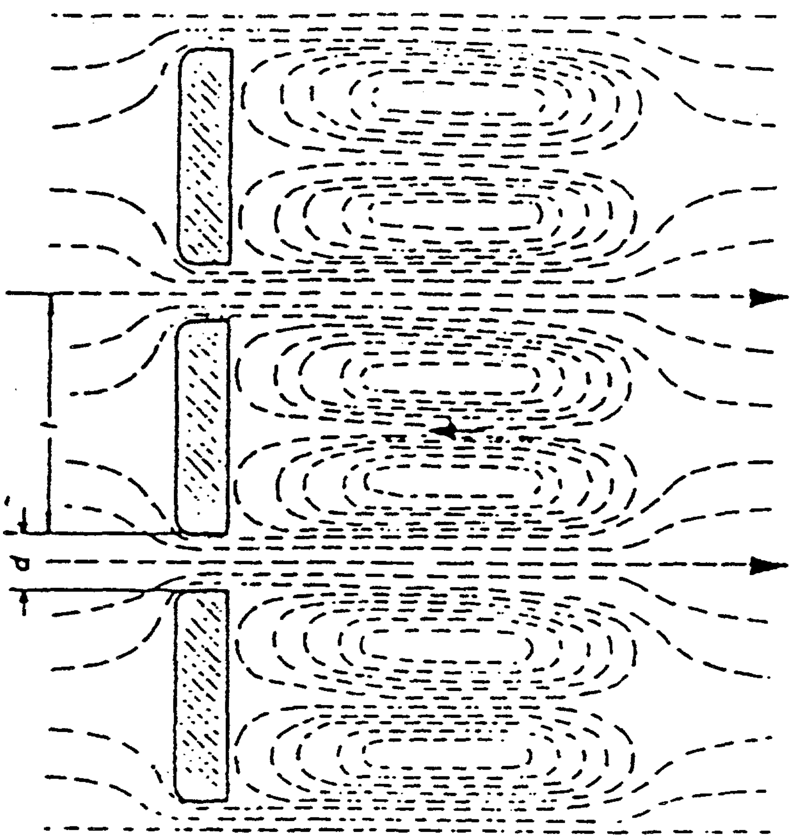


Figure 6.11. Room ventilation by parallel jets. Reproduced from Regenscheit (1975).

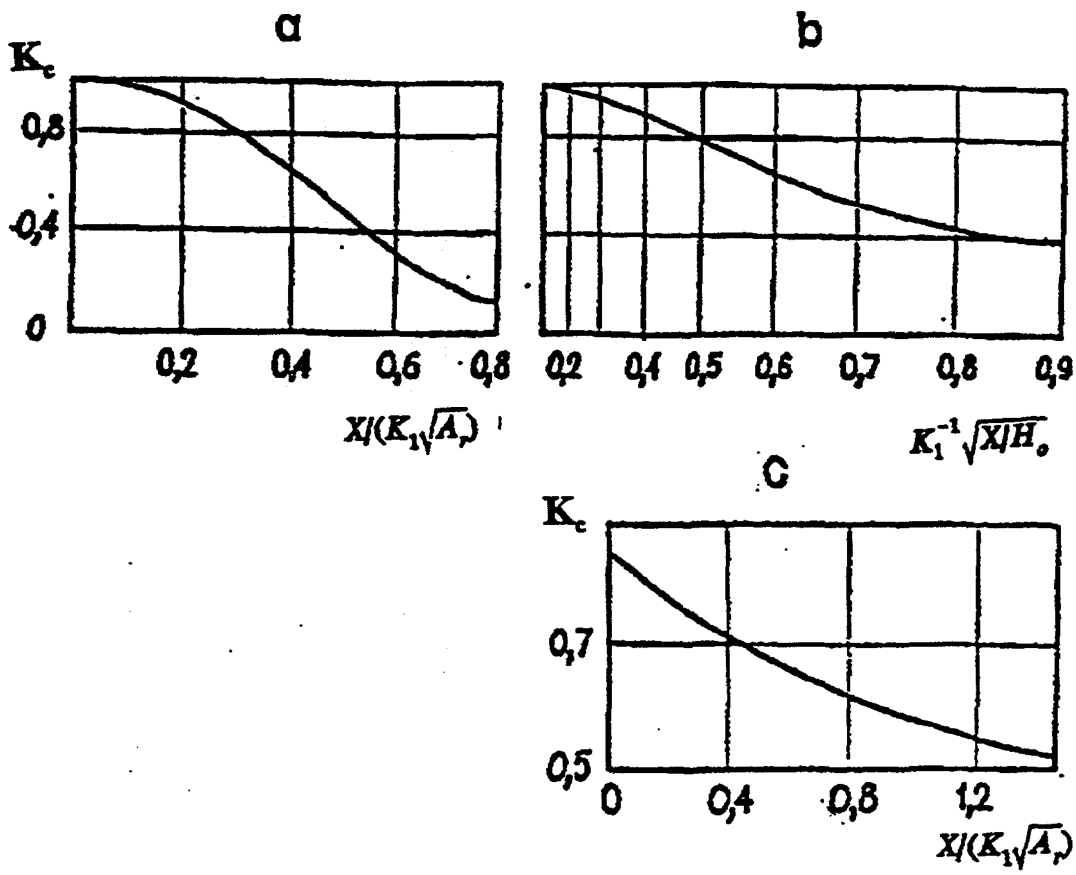


Figure 6.12. Coefficient of confinement: a - compact jet, $A_r = B * H$; b - linear jet; c - radial jet, $A_r = B * L$.

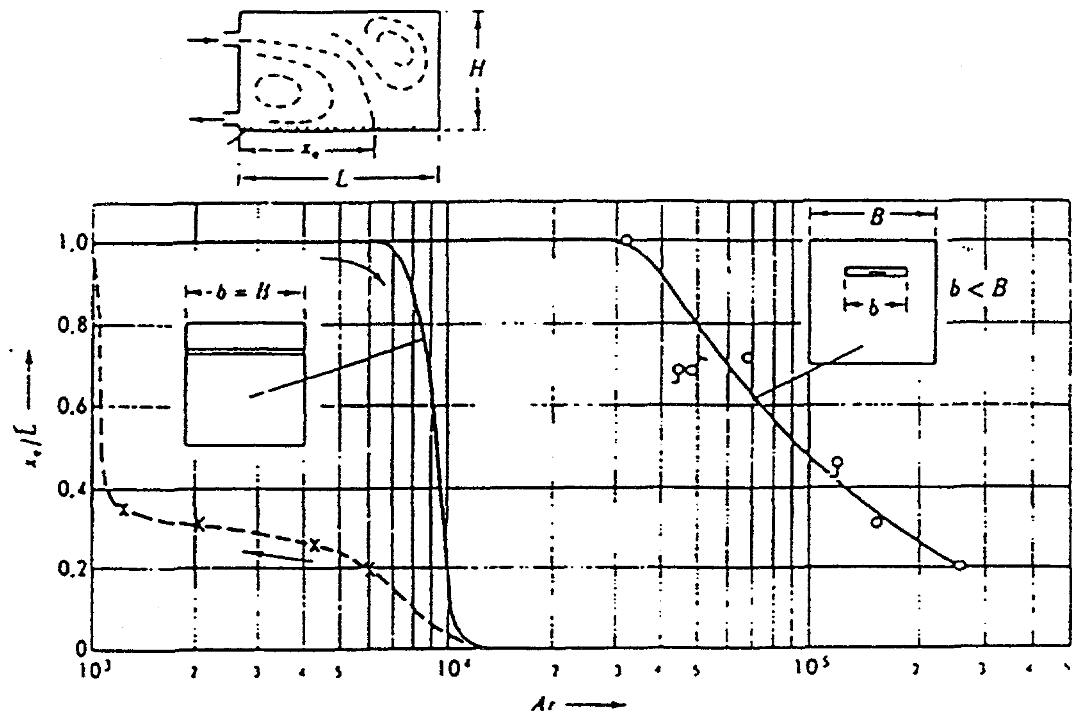


Figure 6.13. Non-isothermal jet trajectory in room. Reproduced from Regenscheight (1970)

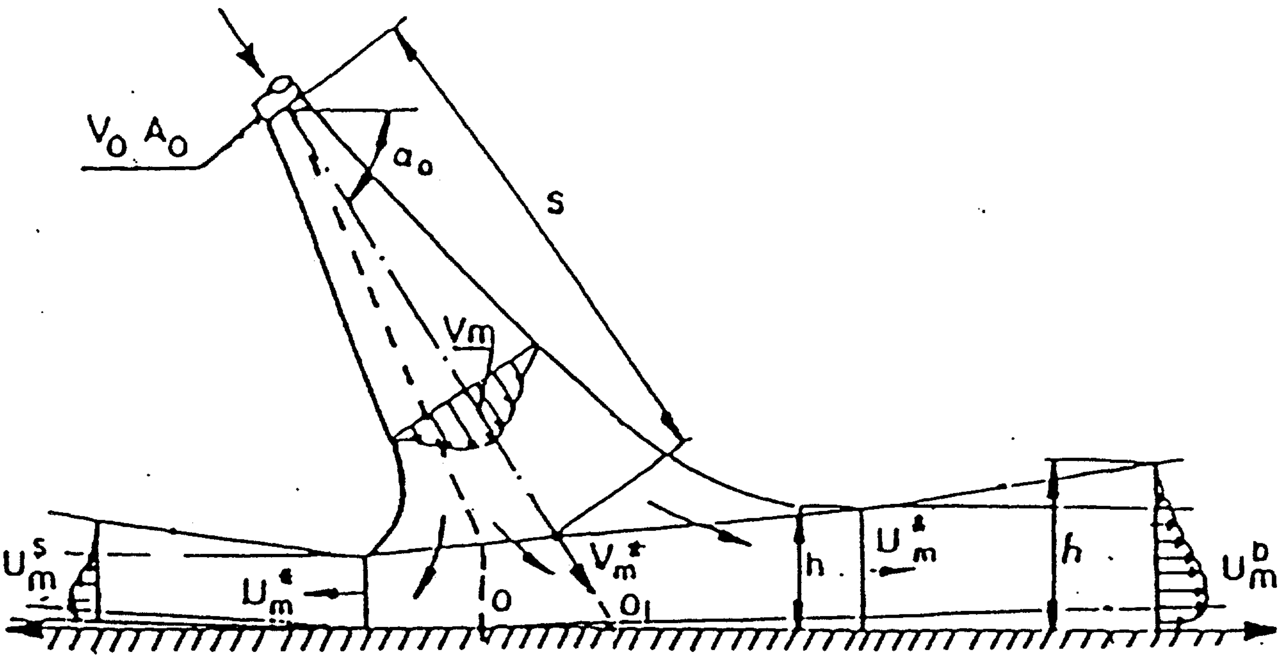


Figure 6.14. Scheme of inclined jet impingement with a floor surface. Reproduce from Zhivov (1993).

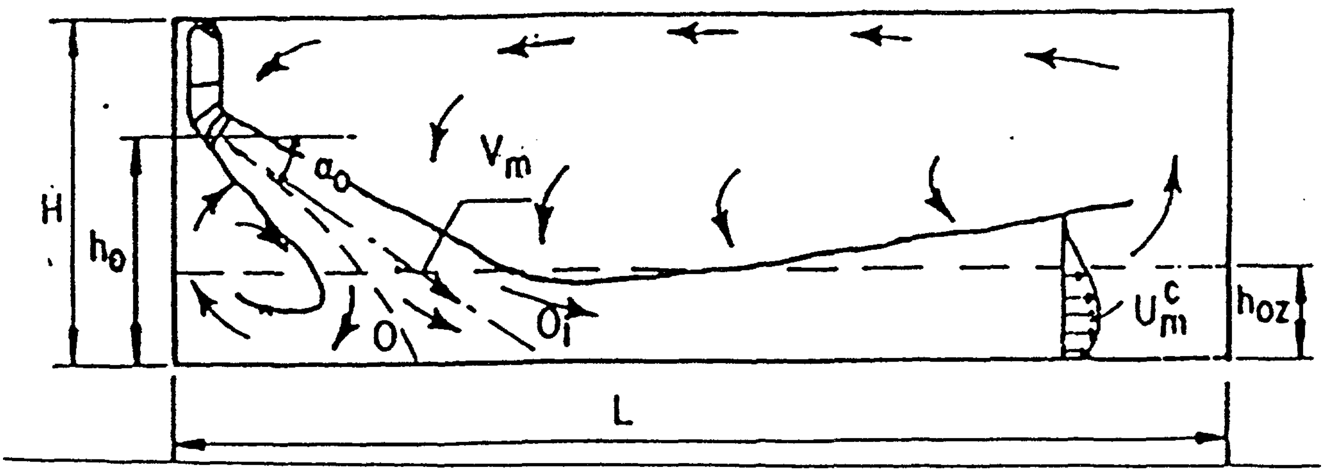


Figure 6.15. Airflow patter in room with inlined jet supply. Reproduced from Zhivov (1993).

7. INFLUENCE OF OBSTRUCTIONS ON ROOM AIR DISTRIBUTION

The results of most of analytical and experimental studies on different methods of air distribution were received from empty rooms and do not reflect the influence of the obstructions on the air distribution and ventilation (heat/contaminant removal) efficiency. Meanwhile in halls of some industrial buildings, process equipment may occupy a significant part of the floor area (Figure 7.1a), or the space height (Figure 7.1b and Figure 7.2). Workplaces can be located either within 2 m of the floor level or at different heights for operating and servicing of process equipment or to assemble workpieces (Figure 7.3). Thus, the requirements to the occupied zone thermal conditions and air quality should be extended also to those locations.

Information on the influence of obstructions on room air distribution is limited. The analysis of results of some field and laboratory studies based on the published technical information made by Zhivov (1983) show that big size obstructions effect air flow pattern and reduce ventilation efficiency. To evaluate the influence of process equipment Zhivov (1983) proposed to use the following parameters:

- ratio of the cross sectional area of the obstructions to the cross sectional area of the room σ_1 ;
- ratio of the floor area, occupied by obstructions to the area of occupied zone σ_2;
- ratio of volume of the technological equipment to the room volume σ_3.

Analysis of blueprints and field measurements in numerous industrial buildings indicates that the above ratios can reach correspondingly the following values: $\sigma_1 = 0.7$, $\sigma_2 = 0.6$ and $\sigma_3 = 0.4$.

Graphs in Figure 7.4 illustrate some conventional methods of air supply currently used in spaces with large size process equipment:

a. concentrated air supply with horizontal jets (attached or not attached to the ceiling) above obstructions: e.g., auto manufacturing plants, ship building yards, mechanical shops, welding shops, etc. (Shwenke 1975, Regenscheight 1964, Curilev and Pechatnikov 1966, 1967; Timofeeva and Veksler 1972, Cole, 1995);
b. concentrated air supply with horizontal jets into the corridors between obstructions: e.g., telephone exchange stations, warehouses with multistory racks, welding shops, assembly plants (Curilev and Pechatnikov 1966, 1967; Zhivov 1983) ;
c. with inclined jets supplied from the upper zone (above 6 m): auto manufacturing plants welding shops, machinery shops (VNIIOT 1981, Cole 1995);
d. through air supply panels with low velocity toward workplaces located at different heights: e.g., ship building/repair stocks, process tower stands in chemical plants (VNIIOT 1981);
e. vertical upward air supply through spiral, slotted or perforated air diffusers in the floor: in halls with double flooring: e.g., in chemical plants, non-ferrous metallurgy plants (VDI 1994, Angel and Rudman 1974);
f. by inclined jets supplied close to workplaces on different heights: e.g., hangars, ship building yards, paper mills, chemical plants (Zhivov 1983);
g. by inclined jets supplied close to workplaces from the height of 3 to 5m above the floor

level: e.g., mechanical shops, assembly shops (Gunes 1974, VDI 1994, Cole 1995);

h. vertical downward with compact jets into the corridors between obstructions: e.g., halls of textile plants, warehouses with multistory material racks (Sorokin 1965, Didenko 1972a,b; Uspenskaya 1975, Ströder 1991);

i. with horizontal jets into the occupied zone: e.g., plastic production plants, halls of textile plants (Grimitlyn et al. 1983).

Air distribution in rooms is influenced the most by obstruction of the occupied zone when concentrated air supply and air supply with inclined jets from a height greater, than 4 m above the floor level, are used (Zhivov 1983).

Curilev and Pechatnikov (1966, 1967) conducted experimental studies of concentrated air supply in significantly obstructed spaces. Their data shows, that obstruction of the room vertical cross-section area results in increased confinement of the supply air jets and thus reduces their throw. Curilev and Pechatnikov (1966) suggested modifying the equations for calculation of the jet throw (**X**) and the maximum air velocity in the reverse flow (V_{rev}), derived by Baharev and Troyanovsky (1958), to account for the influence of obstructions in the occupied zone:

$$X = (4.7 \text{ to } 5.4)\sqrt{BH(1 - \sigma_{12})} \tag{7.1}$$

$$V_{rev} = 0.78\ V_o \sqrt{\frac{A_o}{BH\,(1 - \sigma_{12})}} \tag{7.2}$$

The data by Curilev and Pechatnikov is in agreement with those published by Regescheit (1964) and Schwenke (1975).

Also, the results of field studies by Gunes (1970) indicate an increase in the reverse airflow velocity with concentrated air supply in obstructed spaces compared to the case of air supply into empty rooms.

Studies of air distribution with concentrated air jets at textile processing plants by Sorokin (1965) showed that when air is supplied into the corridors between the process equipment with a height greater than 3m, stagnant (poorly ventilated) zones occur behind this equipment. Air supply above the process equipment results in poor ventilation of the corridors between this equipment as well.

Timofeeva and Veksler (1972) studied air distribution with concentrated jets in the spaces of shipbuilding industry. The field tests of temperature gradient in the work areas on small parts of the ships ($\sigma_1 = 0.1$) and in the hall with an assembled ship ($\sigma_1 = 0.4$) showed, that in latter case temperature gradient increased 2.5 times compared with a former one.

Based on experimental studies of air supply with inclined jets into spaces with a significant confinement (e.g., corridors between obstructions), Zhivov (1993) considered confinement for the calculation of maximum velocities in the near floor air flow (see Chapter 6). Space obstruction by process equipment also decreases an inclined jet throw when air is supplied from the upper zone or into the corridors between the equipment.

Data received by Gunes (1974) on small scale physical models of workshops with a process equipment 3 m high that occupies 55% of the room floor area, show the influence of this equipment on the distribution of temperatures and velocities in the occupied zone. This data show, as well, the decrease of air distribution efficiency when air is supplied by inclined jets.

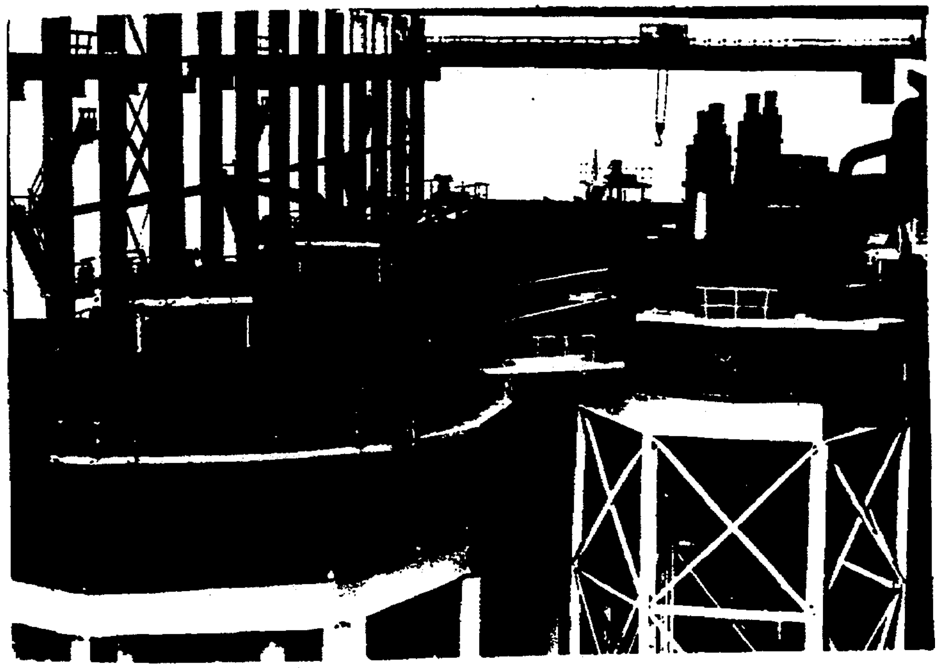
Review of technical literature and analysis of numerous HVAC systems designs (Zhivov 1983) show, that distributed methods of air supply (e.g., shown in Figures 7.4f,g,h, and i) are the most commonly used in obstructed industrial spaces.

However, according to the data received by Sorokin (1965), Didenko (1972) and Uspenskaya et al. (1975), obstructions with a height over 2m reduce air distribution efficiency when air is supplied with radial or conical jets through ceiling mounted diffusers. The effect of obstructions on jet throw, air velocity in the occupied zone and ventilation efficiency based on above cited research is illustrated by graphs in Figures 7.5 and 7.6.

The above discussion shows, that neglecting the influence of the obstructions on the room air distribution may cause a dramatic overestimation of air distribution efficiency.

In recent years, a new generation of HVAC systems supplying air into the halls of industrial buildings with directing jets and displacement ventilation systems were introduced in European countries. Studies have shown (Zhivov 1982, 1985, 1994) that in rooms with large size obstructions in the occupied zone horizontal directing jets increase the main streams throw. Vertical directing jets inject the clean and tempered air of the main streams and deliver it to workplaces, located in the aisles between the process equipment or distract the unwanted temperature gradient, when heated air is supplied. Air distribution system with directing jets is discussed in detail in Section 6. Also, numerous industrial applications of Displacement ventilation do not indicate there are any problems related to occupied zone obstruction by the process equipment. Unless, process equipment is installed within an air diffuser near zone, that can result in increased air velocities in the occupied zone.

a



b

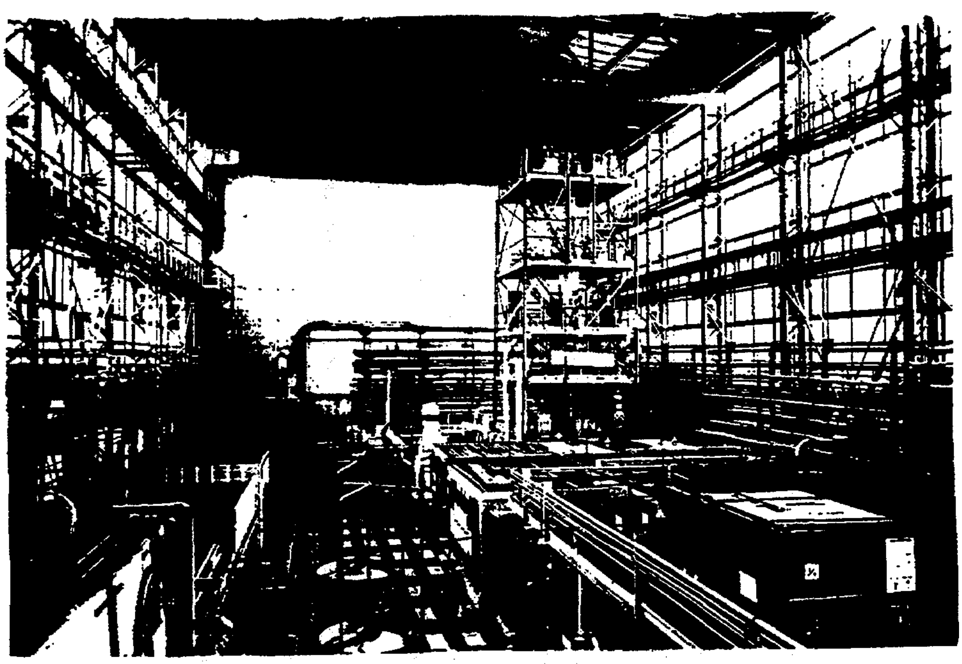


Figure 7.1. Chemical industry halls with a large size process equipment:
a - an occupied zone is significantly obstructed by process equipment,
b - workplaces for servicing process equipment are located at different heights. (Reproduced from Zhivov 1983)

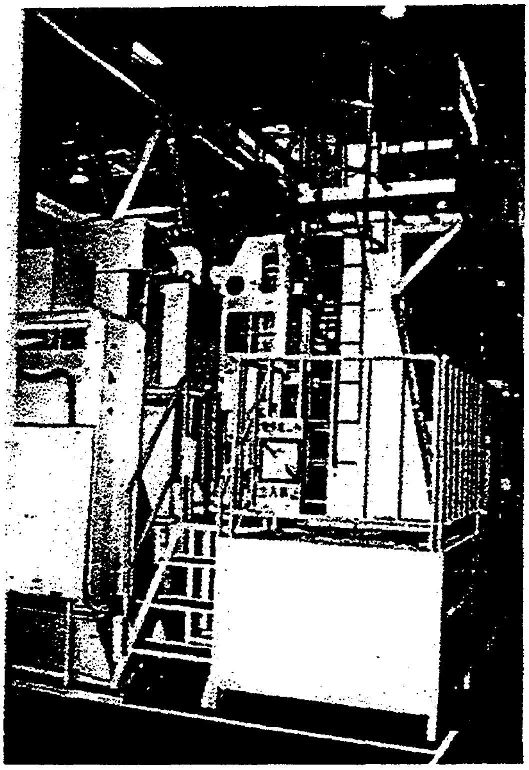


Figure 7.2. Machine shop with a significant occupied zone obstruction by process equipment (Courtesy of Caterpillar Corp.)

Figure 7.3. Assembly hall with workplaces located at different heights for process equipment servicing (Courtesy of Caterpillar Corp.).

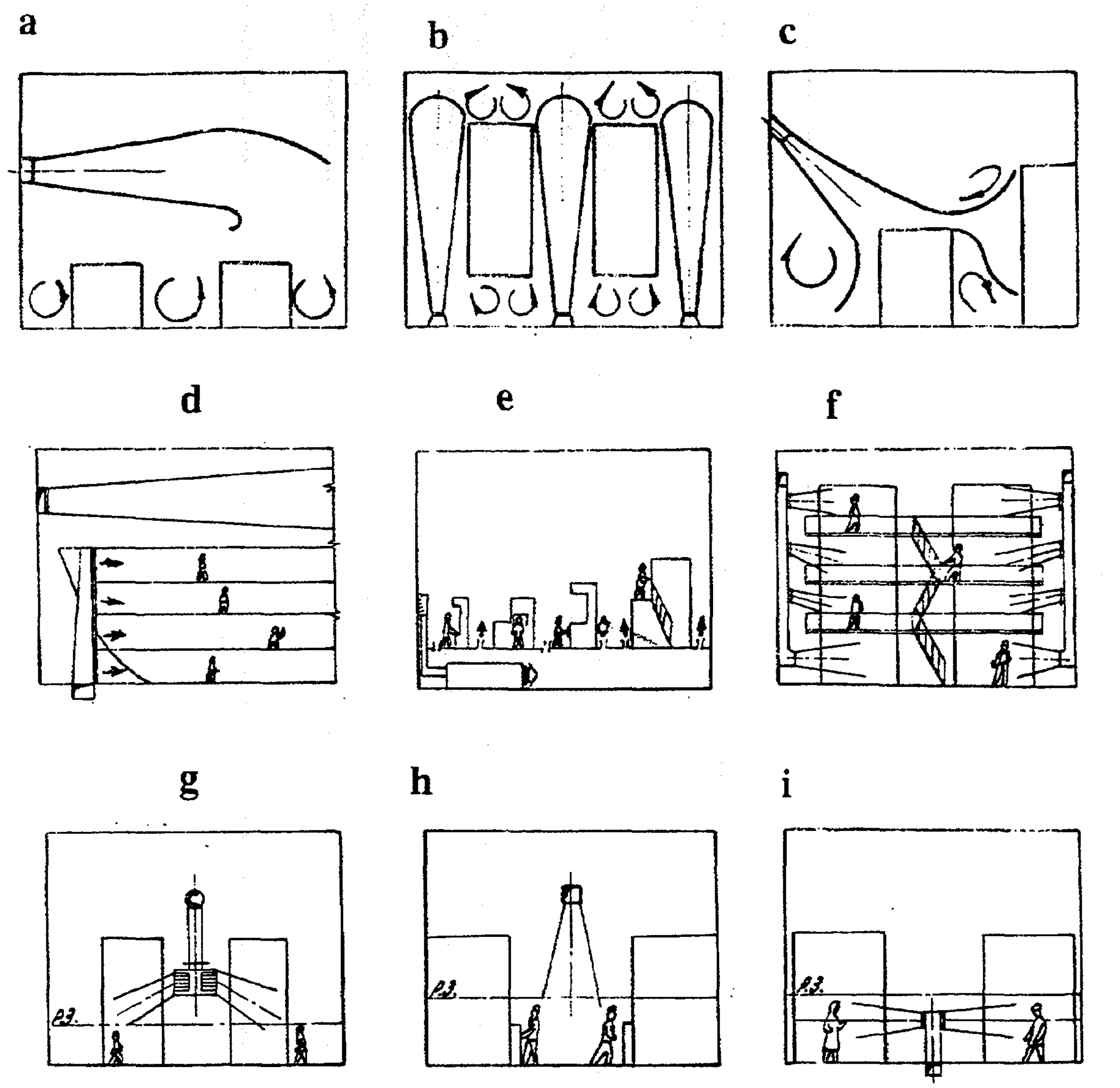


Figure 7.4. Examples of conventional air supply methods into obstructed spaces of industrial buildings (Reproduced from Zhivov 1983):

a - concentrated jets above obstructions;

b - concentrated jets into the corridors between obstructions;

c - inclined jets into the upper zone;

d - through panels on different levels of the occupied zone;

e - through the false floor;

f and g - by inclined jets close to the work places; h - vertical downward projected jets into the corridors; i - horizontal jets directly into the occupied zone.

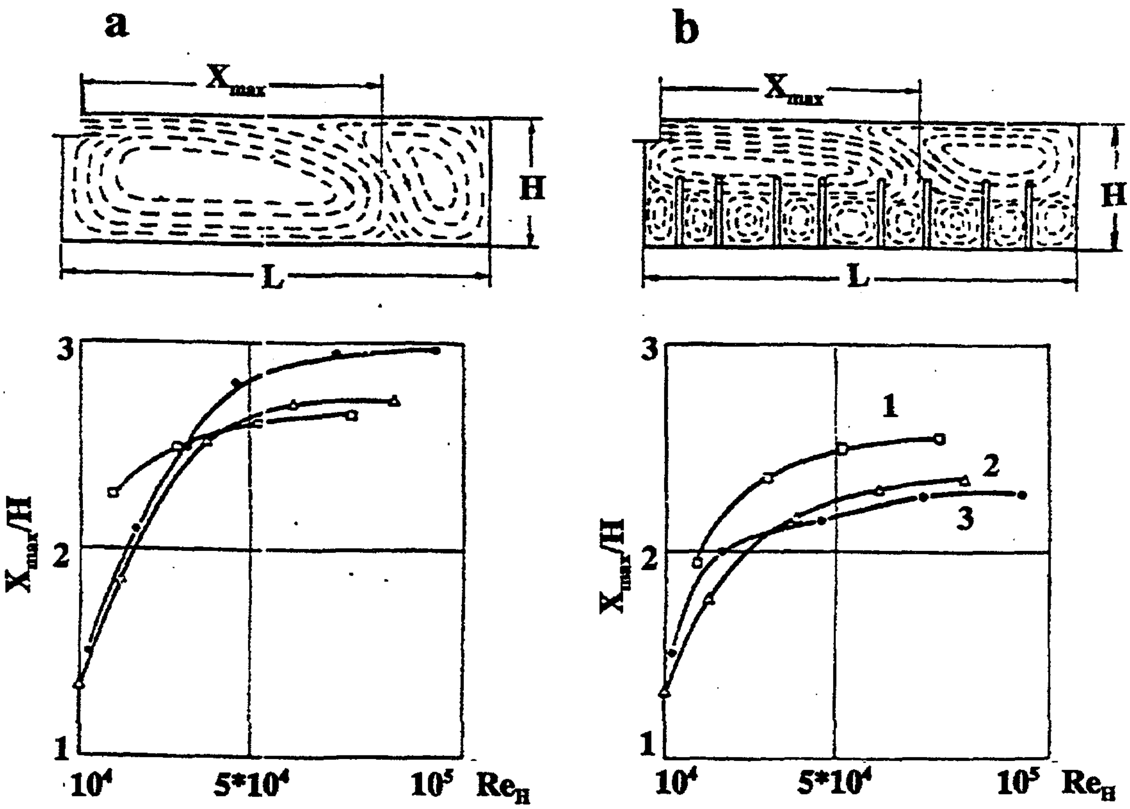


Figure 7.5. Influence of obstructions on jet throw:
a - jet throw in the empty room;
b - jet throw in the room with obstructions ($Re_H = VH/v$, where $V = Q_o /B*H$ - average supply air velocity, Q_o - supply airflow rate; 1 - H/L = 0.24; 2 - H/L = 0.28; H/L = 0.33 (Reproduced from Regenscheit 1964)

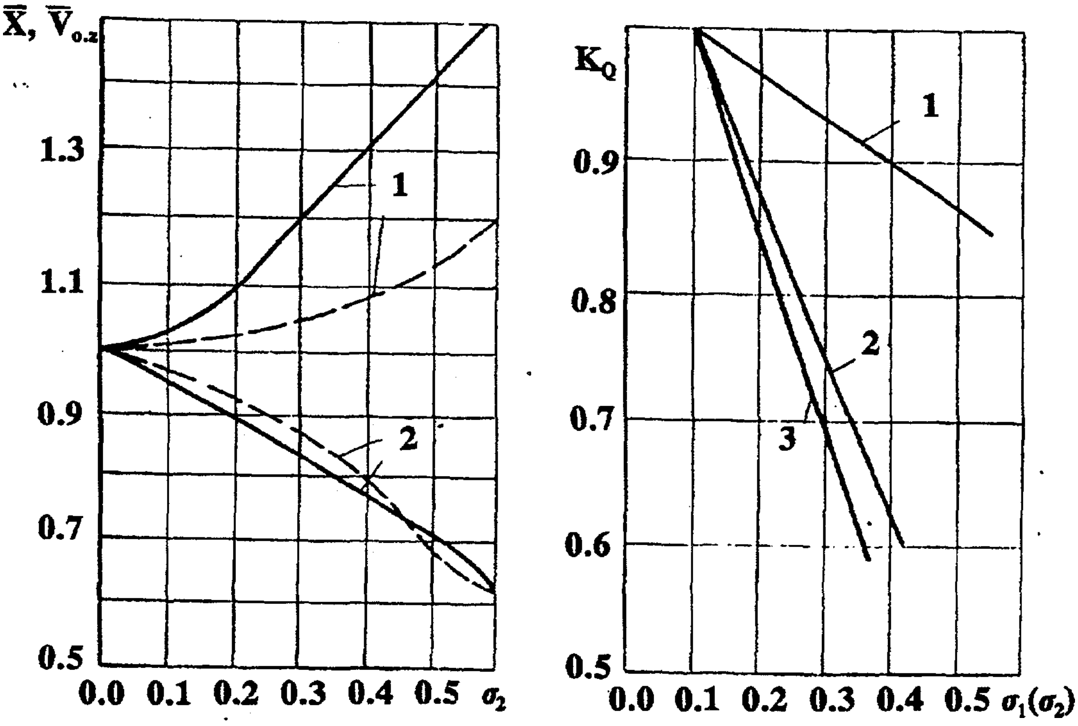


Figure 7.6. Influence of obstructions on air distribution in rooms (Reproduced from Zhivov 1983)

a: 1 - $V_{o.z}(\sigma_1)/V_{o.z}(\sigma_1 = 0.1)$; 2 - $X = X(\sigma_1)/X(\sigma_1=0.1)$;_____ - concentrated air jet supply (based on the data from Curilev and Pechatnikov (1966, 1967) and Baharev and Troyanovsky,1958); - - - - inclined air jet supply (based on the data by TsNIIPZ 1978); b: 1 - $K_w(\sigma_1)/K_w(\sigma_1=0.1)$ for concentrated air supply (based on the data by Timofeeva and Veksler (1972); 2 - $K_w(\sigma_2)/K_w(\sigma_2=0.1)$ - air supply by inclined jets from the height of 4m (based on the data by Gunes 1974); 3 - $K_w(\sigma_1)/K_w(\sigma_1=0.1)$ distributed air jet supply with horizontal jets into the occupied (based on the data from TsNIIPZ 1978).

8. PROBABILISTIC APPROACH TO THE OCCUPIED ZONE COMFORT AND CONTAMINANT DISTRIBUTION

Air velocity and temperature in the occupied zone have been measured by numerous researchers. Jackman (1970, 1971, 1973) studied air movement with side-wall mounted grilles, circular and linear diffusers. For all these methods of air supply, he found the correlation between momentum **Jo** of the supply air and the average velocity $\mathbf{V_{o.z.}}$ in the occupied zone. Fissore et al. (1991) studied air velocity distribution in a space ventilated by slot-type diffusers and found that the velocity distribution matched well with the Gaussian assumption:

$$\sigma_v = 0.3\ \overline{V_{o.z.}} \tag{8.1}$$

with the variation coefficient

$$K_v = \frac{\sigma_v}{\overline{V_{o.z.}}} = 0.0053 \tag{8.2}$$

where, $\overline{V_{o.z}}$ is an average air velocity in the occupied zone, σ_v is standard deviation of velocity.

Grimitlyn (1994) describes studies conducted on the physical model of industrial space with air jets supplied directly into the occupied zone, with inclined jets supplied through swirl type diffusers and grilles and with concentrated horizontal jets into the upper zone. One of the most variable in those experiments was the ratio of the occupied zone area ventilated by one air diffuser $\sqrt{A_r}$ to the size of the supply duct, d_o, as a characteristic size of the air diffuser.

The experimental data received for the case of air supply directly into the occupied zone (horizontal air jets supply through diffusers located within the occupied zone), show that the relative average air velocity in the occupied zone $\overline{V_{o.z.}}/V_o$ significantly increase (e.g., from 0.03 to 0.05) with a relative occupied zone area $\sqrt{A_r}/d_o$ change from 25 to 10. The velocity variance coefficient $K_v = \sigma_v / \overline{V_{o.z.}}$ decreases accordingly with the relative occupied zone area decrease. Studies were conducted with a small size equipment which was distributed uniformly across the floor area. The temperature variance coefficient in $K_t = \sigma_t / \overline{\Delta t_o}$, as defined by the author, was equal to 0.15 and thus, $\sigma_t = 0.15\ \Delta t_o$. Similar experiments with inclined jets air supply through swirl type air diffusers resulted in the values of the velocity variance coefficient $K_v = \sigma_v / \overline{V_{o.z.}}$ in the range from 0.3 through 0.4.

In the studies described by Grimitlyn (1994) of the concentrated air supply above the occupied zone, the average velocity in the occupied zone was found to be dependant upon the relative length of the ventilated room $L/\sqrt{A_r}$. According to Grimitlyn, the average velocity value can be calculated using the equation:

$$\overline{V_{o.z.}} = (\,from\ 0.5\ to\ 0.6)\ V_{\max}^{rev} \tag{8.3}$$

The highest uniformity of velocity distribution in the occupied is when the parameter $U/\sqrt{A_r}$ is from 3.5 to 4.

Experimental studies of the cold air supply through ceiling mounted air diffusers (Grimitlyn, 1994) showed, that the velocity and the temperature variance coefficients depend upon the ratio of the area of air jet cross-section A_{jet}, when it enters the occupied zone to the area of the occupied zone A_r ventilated by this jet, A_{jet}/A_r. The value of the velocity variance coefficient decreases with the increase of the A_{jet}/A_r ratio until $A_{jet}/A_r = 0.5$. (Figure 8.1). Further A_{jet}/A_r ration increase practically does not effect the velocity variance value. The opposite effect has the A_{jet}/A_r ratio on the temperature variance coefficient. The uniformity of temperature distribution in the occupied zone has the lowest value with $A_{jet}/A_r < 0.5$. The temperature variance coefficient significantly increases when $A_{jet}/A_r > 0.5$. The graph illustrating this phenomenon for compact, radial and linear jets is presented in Figure 8.2. The following equations to calculate A_{jet}/A_r were derived by Grimitlyn from the jet theory:

Compact air jet:

$$\frac{A_{jet}}{A_r} = 4.8\left(\frac{1}{K_1}\,\frac{X}{\sqrt{A_r}}\right)^2 \tag{8.4}$$

Radial jet:

$$\frac{A_{jet}}{A_r} = 1 - 0.15\left(2 - \frac{H_r - h_{o.z.}}{\sqrt{A_r}}\right)^2 \tag{8.5}$$

Linear jet

$$\frac{A_{jet}}{A_r} = 2.8\left(\frac{1}{K_1}\sqrt{\frac{X}{B_r}}\right)^2 \tag{8.6}$$

Grimitlyn (1994) suggests that the highest velocity and temperature uniformity in the occupied zone can be achieved at in the range from 0.3 to 0.5 for the compact and linear jets and in the range from 0.5 to 1.0 for radial jets. The ratio A_{jet}/A_r Is calculated using equations (8.4) through (8.6) **for free jets**. However, due to the effect of confinement, actual jet cross-section area will be smaller.

Extensive experimental studies of room air distribution in room ventilation simulator were conducted in All-Union Research Institute for Hydro technique and Sanitary Technique (Gunes,1977; Uspenskaya et al. 1970,1975). Researchers studied the influences of different factors (heating/cooling loads, room configuration and size, equipment size and number of pieces, air change rate, air diffuser installation height, etc. on velocity and temperature distribution uniformity in the occupied zone and on ventilation effectiveness. They used the approach when special experiment matrixes were developed to cover practical ranges of factors that they considered important. The collected data was used for the regression analysis allowing to select significant influences and disregard other. These studies resulted in equations that can be used by consulting engineers to specify air duiffusers.

The results of air distribution study with ceiling mounted air diffusers supplying compact and radial jets are reported by Gunes et al. (1977). The data presented in Figure 8.3 for compact vertical jets show, that the height of air diffuser installation above the occupied zone significantly effect the uniformity of temperature distribution: with $X/\sqrt{A_r}$ increase from 0.5 to 2.0, the complex $\frac{\sigma_t}{\Delta t_o} \times \frac{\sqrt{A_r}}{d_o}$ decreases from 1.2 to 0.3.

The conclusions made based on the data presented in Figure 8.3 are in conflict with those made by Grimitlyn (1994) (see Figure 8.2). Figure 8.4 illustrates the influence of Δt_o on temperature uniformity in the occupied zone characterized by σ_t for different $X/\sqrt{A_r}$ ratios based on experimental data from Gunes et al. (1977).

From this data it is clear, that the increase in the temperature difference Δt_o results in the increase of the standard deviation σ_t, and the value of σ_t decreases with the increase of the ratio $X/\sqrt{A_r}$. Thus, the uniformity of temperature distribution is a function of Δt_o and $X/\sqrt{A_r}$. According to the data received by Gunes (1977), for air supply with radial jets, $\sigma_t / \Delta t_o = 0.12$. The data on velocity distribution, received by Gunes for air distribution with conical jets reflects the decrease in the velocity variance number (increase in velocity uniformity) with an increase of the relative jet length $X/\sqrt{A_r}$ (Figure 8.5).

Experimental studies of air distribution in the occupied zone of the machinery building plants (concentrated air supply into the upper zone) and through ceiling air diffusers into the upper zone of the textile industry spaces were conducted by Uspenskaya et al. (1970, 1975). These experimental studies resulted in the following equation for the maximum temperature difference in the occupied zone:

$$\Delta t_{o.z.}^{\max} = t_{o.z.}^{\max} - \overline{t_{o.z.}} = A \ \sigma_t, \tag{8.7}$$

where A is an experimental coefficient, and σ_t is a standard deviation, that can be calculated from the following equation:

$$\sigma_t = B \ \frac{Wq}{ACH^{\,m}} \ K \tag{8.8}$$

ACH is an air change rate; B, m and K are experimental coefficients reflecting the influence of process equipment location, relative area of the occupied zone covered by the process equipment, uniformity of heat sources distribution across the occupied zone, etc.

The influence of cooling load **W** and **ACH** on Δt_o and σ_t values based on the experimental data from Uspenskaya et al. (1970, 1975) is presented in Figure 8.6. Based on these data the graphs reflecting the relationships of σ_t and K_t are plotted in Figure 8.7. These graphs show that the increase in Δt_o due to decrease in ACH at **W** = constant, result in the increase of σ_t.

Studies of air distribution in rooms with air supply through ceiling mounted diffusers and sidewall grilles were conducted at the All-Union Research Institute for Labor Protection in Leningrad (Grimitlyn et al. 1982, 1986). Experimental data was collected on the physical model and from 10 field tests of air distribution in rooms of commercial and industrial buildings.

The laboratory studies of air distribution by radial jets in room with heat sources distributed throughout the room showed that the extreme velocities and temperature in the occupied zone can be described by the following relationships:

$$t_{o.z.}^{\max} - \overline{t_{o.z.}} = 2.2\ \sigma_t\ , \qquad \overline{t_{o.z.}} - t_{o.z.}^{\min} = 1.4\ \sigma_t, \qquad t_{o.z.}^{\max} - t_{o.z.}^{\min} = 3.6\ \sigma_t \tag{8.9}$$

$$V_{o.z.}^{\max} - \overline{V_{o.z.}} = 1.8\ \sigma_v\ , \qquad \overline{V_{o.z.}} - V_{o.z.}^{\min} = 1.7\ \sigma_v, \qquad V_{o.z.}^{\max} - V_{o.z.}^{\min} = 3.5\ \sigma_v \tag{8.10}$$

where $t_{o.z.}^{max}$ and $t_{o.z.}^{min}$ are maximum and minimum air temperatures in the occupied zone; $V_{o.z.}^{max}$ and $V_{o.z.}^{min}$ are maximum and minimum air velocities in the occupied zone.

Similar equation were received in studies of air distribution by air jets supplied into the room through side-wall mounted grilles:

$$t_{o.z.}^{\max} - \overline{t_{o.z.}} = 1.8\ \sigma_t\ , \qquad \overline{t_{o.z.}} - t_{o.z.}^{\min} = 2.2\ \sigma_t, \qquad t_{o.z.}^{\max} - t_{o.z.}^{\min} = 4.0\ \sigma_t \tag{8.11}$$

$$V_{o.z.}^{\max} - \overline{V_{o.z.}} = 2.8\ \sigma_v\ , \qquad \overline{V_{o.z.}} - V_{o.z.}^{\min} = 1.3\ \sigma_v, \qquad V_{o.z.}^{\max} - V_{o.z.}^{\min} = 4.1\ \sigma_v \tag{8.12}$$

Based on the data resulted from the literature review, and numerous experiments conducted on scaled physical model and in full-scale field facilities, Grimitlyn, et al., (1986) developed guidelines for air distribution design in spaces with special requirements to thermal parameters in the occupied zone of room when air is supplied through grilles and ceiling mounted diffusers.

When air is supplied through the wall mounted grilles with jets attached to the ceiling, to achieve the most uniform temperatures and velocities in the occupied zone, the length **L** of the room per one grille is recommended (Grimitlyn et al. 1986) to be calculated from the following expression:

$$H - h_{o.z.} \le L \le 0.45\ K_1\sqrt{B\ H} \quad . \tag{8.13}$$

The width of the room, **B**, served by one grill should be within the limit of

$$B = (2.7 \sim 4)\ \frac{L + H - h_{o.z}}{K_1} \quad . \tag{8.14}$$

In the case of cooling the separation of the jet from the ceiling surface is allowed at a distance exceeding **0.5 L**.

The most uniform distribution of temperatures and velocities in the occupied zone when ventilated by one jet supplied through the ceiling mounted diffuser is achieved, when the ratio A_{jet}/A_r between **0.55** and **0.6** for radial jets, between **0.3** and **0.6** for conical jets, and between **0.3** and **0.5** for compact jets. Graphic interpretation of these conditions is shown in Figure 8.8. It was suggested (Grimitlyn et al. 1986) that the velocities and temperatures in the occupied zone can be described by evaluating the air jet maximum velocity $\mathbf{V_x}$ and temperature difference $\mathbf{\Delta t_x}$ at the point within the room where the jet enters the occupied zone. In the case of velocities, the deviations created from the influence of obstructions are incorporated into the jet analysis as follows:

$$\overline{V_{o.z.}} = V_x - 2\,\sigma_v \quad (a), \qquad or \qquad V_{o.z.}^{min} = V_x - 4\,\sigma_v \quad (b) \tag{8.15}$$

Thus, the resulting range of velocities in the occupied zone is based on the maximum velocity in the jet entering the occupied zone determined from the air jet theory. Using two (a) or four (b) times the experimental deviation will provide a confidence that 95% of the averaged occupied zone velocities will be within the predicted range.

A temperature equation for the occupied zone is organized in the same way. Grimitlyn et al. (1986) recommended using four times the experimental deviation of the temperature σ_t. This results in a 95% confidence interval for both the maximum $\mathbf{t_{o.z.}^{max}}$ and minimum $\mathbf{t_{o.z.}^{min}}$ temperature. The occupied zone temperature difference is calculated as follows:

$$t_{o.z.}^{max} - t_{o.z.}^{min} = 4\,\sigma_t \tag{8.16}$$

By introducing an artificial temperature deviation variable, the equation for the temperature range in the occupied zone can be reorganized as the function of the diffuser conditions as follows:

$$t_{o.z.}^{max} - t_{o.z.}^{min} = 4\,K_t\,K_2\,\Delta t_o\,\frac{\sqrt{A_o}}{X} \tag{8.17}$$

where, $\mathbf{K_t} = \sigma_t / \Delta t_x$ is a temperature variance related to the point where the jet enters the occupied zone, $\mathbf{K_2}$ is an air jet temperature decay coefficient, $\mathbf{A_o}$ is a characteristic size of the air diffuser, and **X** is a distance along the jet to the point of its entering the occupied zone.

Maximum initial temperature differential $\mathbf{\Delta t_o}$ which doesn't cause temperature differences in the occupied zone more than desired can be found from the equation

$$\Delta t_o = \frac{t_{o.z}^{max} - t_{o.z}^{min}}{4\ K_t\ K_2}\ \frac{K_n\ K_c}{K_{int}}\ \frac{L + H_r - h_{o.z.}}{\sqrt{A_o}}. \tag{8.18}$$

where, $\mathbf{K_t = \sigma_t / \Delta t_x}$ is a temperature variance related to the point where the jet enters the occupied zone, K_2 is an air jet temperature decay coefficient, $\mathbf{A_o}$ is a characteristic size of the air diffuser, and **X** is a distance along the jet to the point of its entering the occupied zone. The experimental data that laid the groundwork for $\mathbf{K_t}$ coefficient evaluation are presented in Figures 8.9, 8.10 and 8.11. The values of the coefficients $\mathbf{K_n}$, $\mathbf{K_{int}}$, $\mathbf{K_c}$ and the variation coefficient $\mathbf{K_t}$ of air temperature distribution within occupied zone are presented in the Table 8.1.

Based on Equation (8.18), one can evaluate the maximum initial temperature differential $\mathbf{\Delta t_o}$ which results in the occupied zone air temperature within the required range as follows:

$$\Delta t_o = \frac{t_{o.z.}^{max} - t_{o.z.}^{min}}{4\ K_t\ K_2}\ \frac{K_n\ K_c}{K_{int}}\ \frac{X}{\sqrt{A_o}} \tag{8.19}$$

Table 8.1. Coefficients for Side-Wall and Ceiling Mounted Diffuser Air Jets

Type of jet	K_{int}	K_c	K_t	K_n	**X**
Compact (ceiling diffuser)	1	0.9	0.8	$\sqrt[3]{1 \pm 2{,}5\ \frac{K_2}{K_1^2}\ Ar_o \left(\frac{h - h_{o.z.}}{\sqrt{A_o}}\right)^2}$	$H - h_{o.z}$
Conical (ceiling diffuser)	1	0.8	0.8	$\sqrt[3]{1 \pm 2{,}5\ \frac{K_2}{K_1^2}\ Ar_o \left(\frac{h - h_{o.z.}}{\sqrt{A_o}}\right)^2}$	$H - h_{o.z}$
Radial (ceiling diffuser)	1	0.8	1.5	1	
Compact or incomplete radial (side-wall grilles)	1	0.8	1.5	1	

SUMMARY

Generalized statistical data obtained from the previous laboratory, field and numerical experiments, should be complimented with additional studies in obstructed rooms different configurations and with different air supply methods. These data can extend predictions from extreme parameters to the rest of the occupied zone. Through this technique, the designer can determine if velocity, temperature (contaminant concentration) distribution are within an acceptable range. If they are not, a different diffuser velocity and temperature and/or diffuser type will need to be selected. Obstructions influence the uniformity of velocity, temperature and contaminant concentrations distributions. The influence of obstructions height and the floor area occupied as well as the impact of different methods of air supply and room configurations should be accounted for by corresponding values of deviation variables.

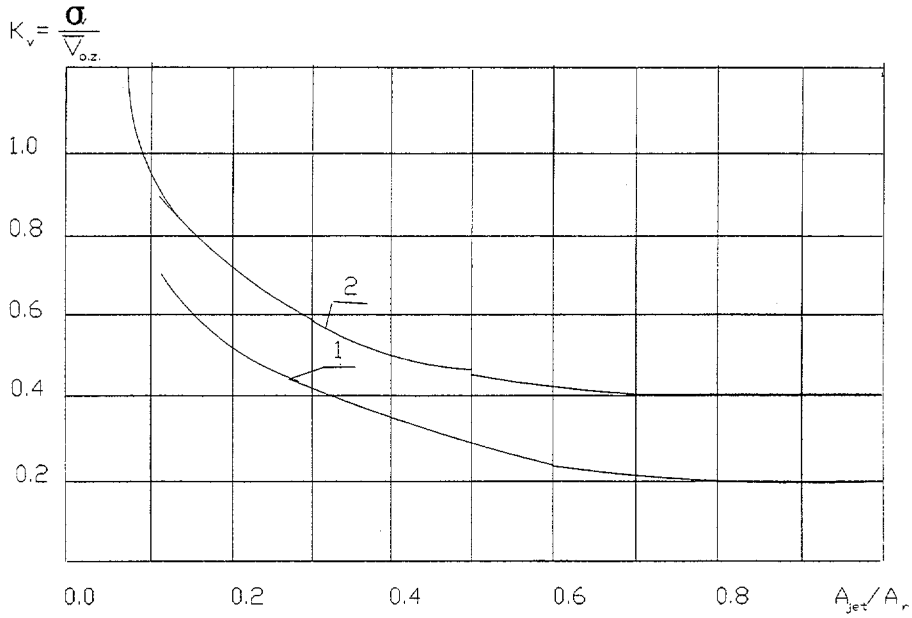


Figure 8.1. Relationship between the velocity variance coefficient K_v and the ratio of the air jet cross-section area, A_{jet}, over the occupied zone area, A_r, ventilated by this jet. Air was supplied vertically through ceiling mounted air diffusers: 1 - compact jet, 2 - radial jet (Grimitlyn, 1973)

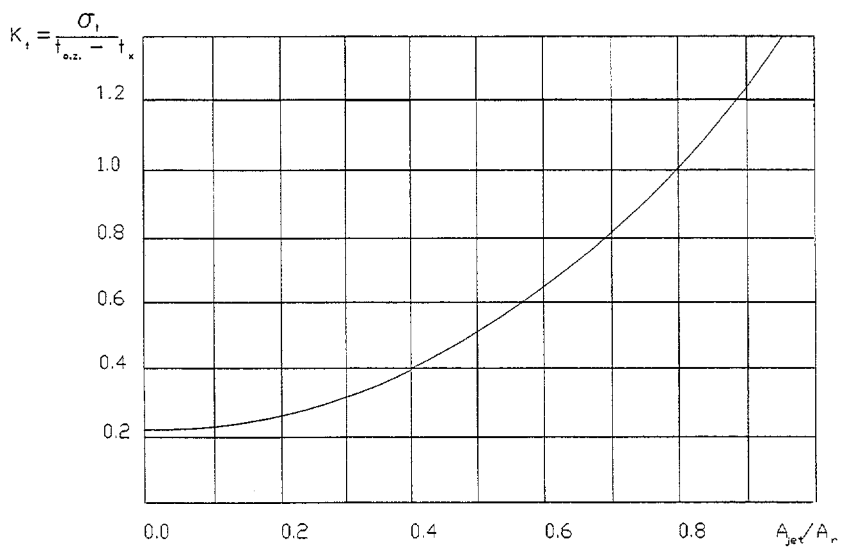


Figure 8.2. Relationship between the temperature variance coefficient K_t and the ratio of the jet cross-section area A_{jet} over the occupied zone area A_r ventilated by this jet. Air was supplied vertically with a compact jet through ceiling mounted air diffusers (Grimitlyn, 1973)

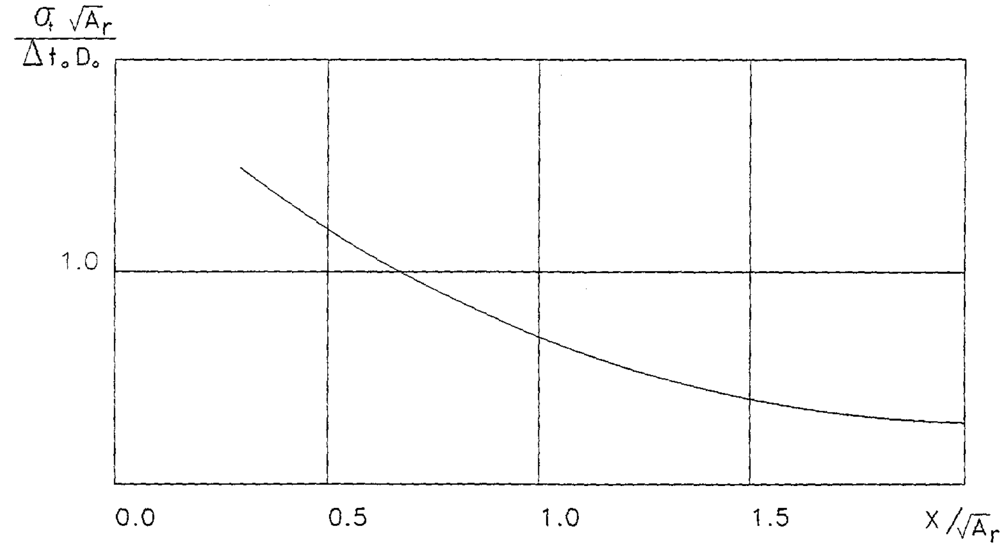


Figure 8.3. Relationship between the ratio $\sigma_t / \Delta t_o \times \sqrt{A_r} / D_o$ and the relative distance of the jet travel $X / \sqrt{A_o}$ to the occupied zone (Gunes, 1977).

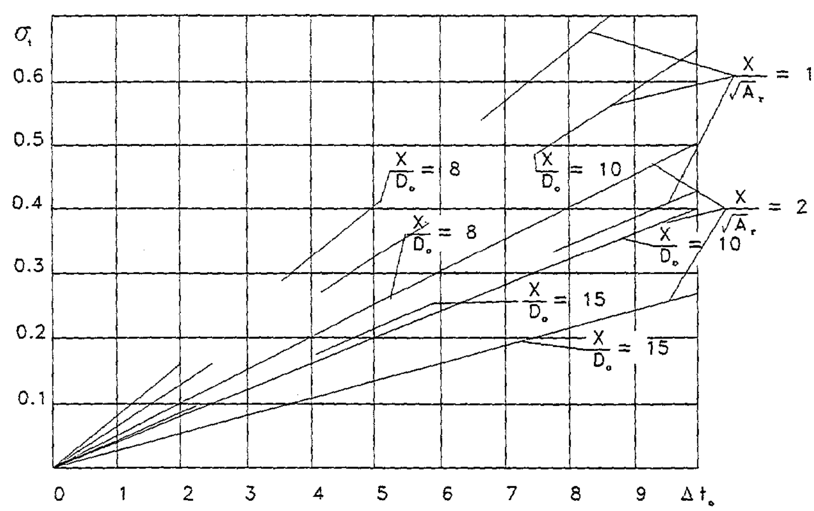


Figure 8.4. Relationship between the standard deviation of air temperature in the occupied zone σ_t and supply air temperature difference Δt_o with air supply through ceiling mounted air diffusers (Gunes et al., 1977).

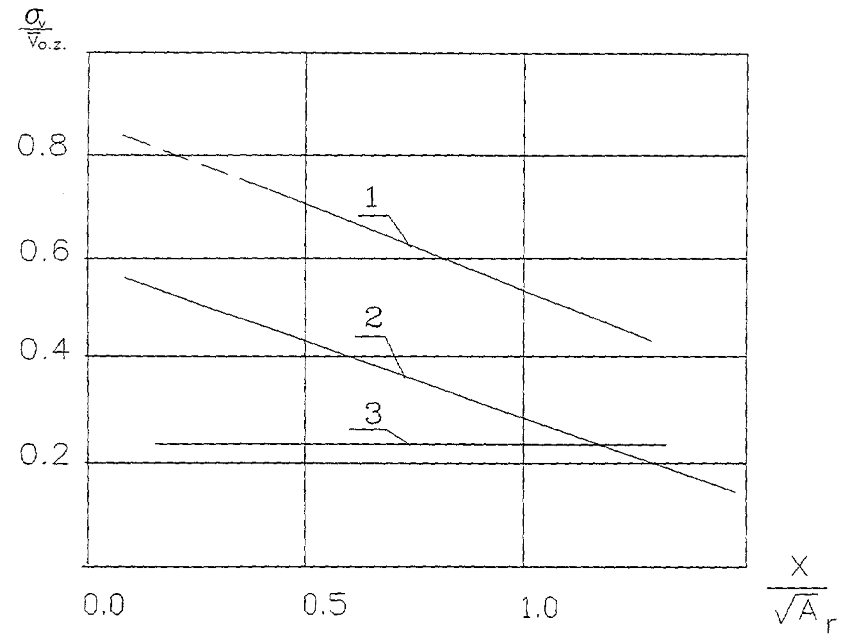


Figure 8.5. Relationship between the velocity variance coefficient $K_v = \frac{\sigma_v}{V_{o.z.}}$ and the relative distance $X/\sqrt{A_o}$ of the jet travel to the occupied zone: 1 - conical jets, 2 - compact jets, 3 - radial jets (Gunes, 1977).

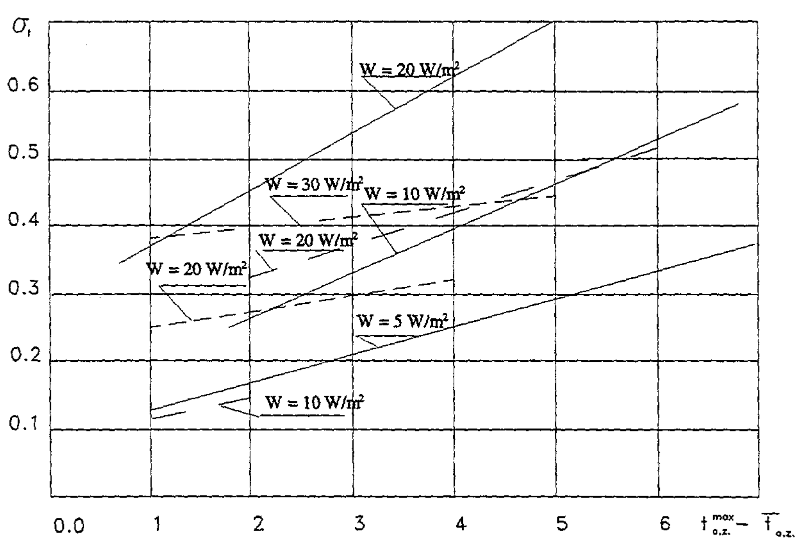


Figure 8.6. Relationship between the standard deviation of air temperature in the occupied zone σ_t and the temperature difference $t^{max}_{o.z.} - \overline{t_{o.z.}}$ (Uspenskaya, 1980):

_______ - air supply directly into occupied zone,

- - - - - air supply through ceiling mounted air diffusers with radial attached jets,

__ . ___ - air supply through ceiling mounted diffusers with swirl conical jets.

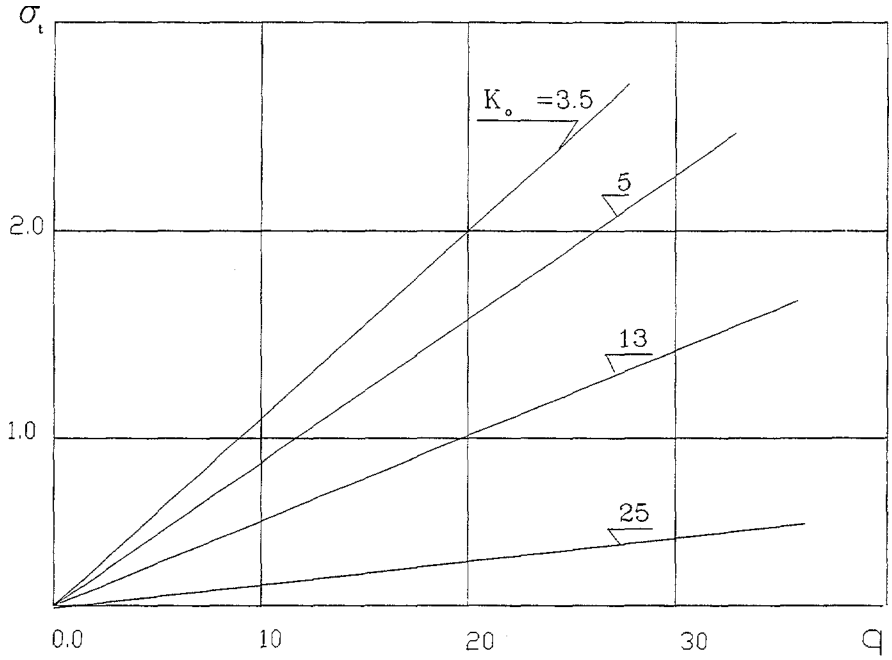


Figure 8.7. Influence of air change rate K_o, 1/h, and cooling load q, W/m^3 on standard deviation σ_t of temperature distribution in the occupied zone (Uspenskaya, 1980).

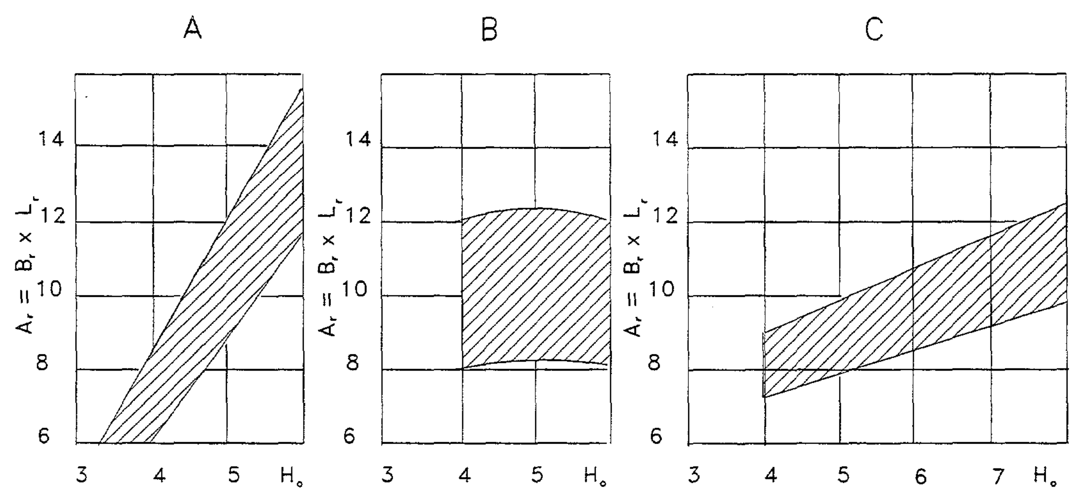


Figure 8.8. Graphs for evaluation of the occupied zone area A_r, m^2, ventilated by air jet supplied from the height H_o, m through a ceiling mounted air diffuser: A - radial jet. B - conical jet, C - compact jet.

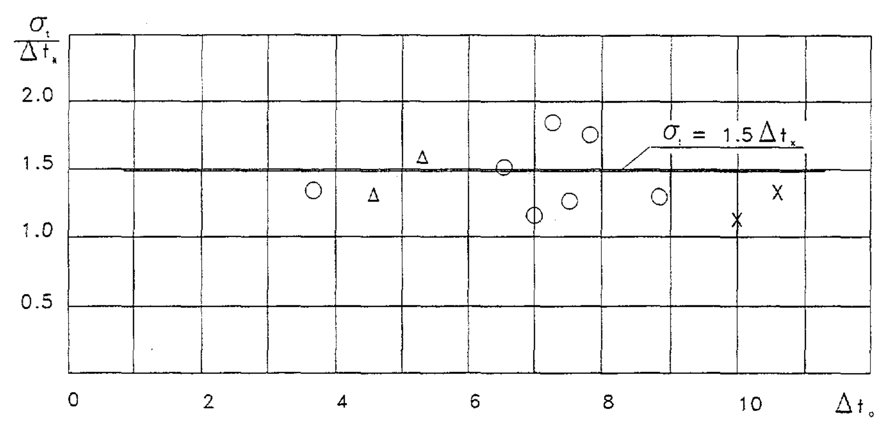


Figure 8.9. Relationship between the temperature variance coefficient $K_t = \sigma_t /(t_x - \overline{t_{o.z.}})$ and the supply air temperature difference Δt_o for air supply through wall mounted grilles:

o - laboratory studies (VNIIOT, 1982);

Δ - field studies (VNIIOT, 1982);

x - laboratory studies (VNIIOT, 1970).

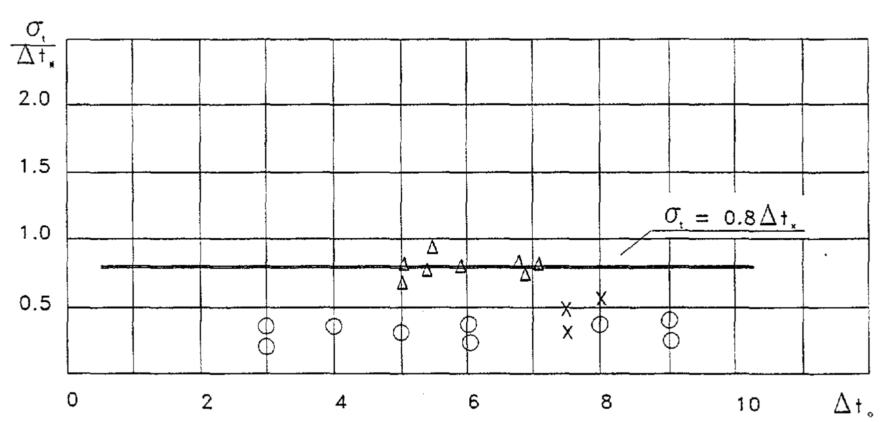


Figure 8.10. Relationship between the temperature variance coefficient $K_t = \sigma_t /(t_x - \overline{t_{o.z.}})$ and the supply air temperature difference Δt_o for air supply through ceiling mounted air diffusers with compact jets:

o - laboratory studies (VNIIOT, 1982);

Δ - field studies (VNIIOT, 1982);

x - laboratory studies (VNIIOT, 1970).

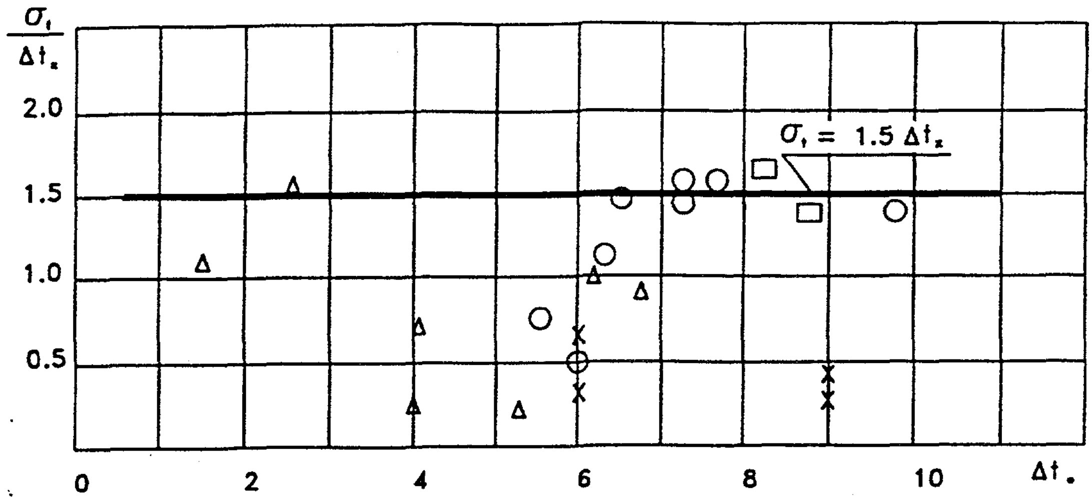


Figure 8.11. Relationship between the temperature variance coefficient $K_t = \sigma_t / (t_x - \overline{t_{o.z.}})$ and the supply air temperature difference Δt_o for air supply through ceiling mounted air diffusers with radial jets:

○ - laboratory studies (VNIIOT, 1982);

△ - field studies (VNIIOT, 1982);

X - (Gunes et al. 1977),

□ - laboratory studies (VNIIOT, 1970).

9. VENTILATION EFFICIENCY

9.1. INTRODUCTION

Effective air distribution in ventilated rooms and proper quantity of conditioned air is essential for creating comfortable conditions, removing contaminants and reducing initial and operating costs of air conditioning and ventilation systems. The degree to which a ventilation system fulfills ventilation requirements is described in the literature in terms of "ventilation effectiveness", "ventilation efficiency", " ventilation performance", etc. Liddament (1991) in his review of technical information related to ventilation effectiveness stated, that "the subject of ventilation effectiveness is made unnecessarily complex by the lack of uniformity in terminology. Frequently, terms are interchanged or different terms are used to describe the same concepts".

Sandberg and Skåret (1985) differentiate between the terms "air change efficiency" and "contaminant removal effectiveness". Air change efficiency is "a measure of how effectively the air present in a room is replaced by fresh air from ventilation system" whereas "**contaminant removal effectiveness, ε^c**" is "a measure of how quickly an air-borne contaminant is removed from the room". Another similar criteria, that is used is a **"contaminant removal efficiency, η^c"**. The value of this criteria can be derived from the "contaminant removal effectiveness":

$$\eta^c = \left(\frac{\varepsilon^c}{1 + \varepsilon^c}\right) \tag{9.1}$$

In the current review, the term **"effectiveness"** of air distribution will be used to describe the ratio of the occupied zone area (where thermal comfort and contaminant concentration are within ranges required by standards and codes) to the total occupied zone area. This a **hygienic criteria**, that allows one to judge how well the HVAC system fulfills its main task - creating thermal comfort conditions and controlling contaminants in the occupied zone.

The level of uniformity of temperatures, velocities and contaminant concentrations depends upon the method of room air distribution used, room sizes, geometry and sizes of obstructions, etc. Different criteria are used to describe air distribution/ ventilation **"effectiveness"**. Among the commonly used criteria is the *Air Distribution Performance Index (ADPI)* (ASHRAE 1997), that is defined as the percentage of locations where a combination of air temperature and air velocity meets comfort requirements. This criteria is based on experimental results of air diffuser performance for specifically tested room configurations. The data on the ADPI is available only for sedentary activity.

The outdoor airflow rate and amount of heating and cooling capacity used to ventilate and conditioned the space depends upon the ratio of temperatures and contaminant concentration in the exhaust air to the occupied zone air, that is primarily influenced by the method of air supply used. Temperature, relative humidity and contaminant concentration stratification in mechanically and naturally ventilated and air conditioned spaces can be characterized by the *coefficients of ventilation* ***efficiency:*** *K_w, K_{vap} and K_g* for heat removal, water vapor removal, and gaseous/particulates contaminant removal, respectively:

$$K_w = \frac{t_{exh} - t_o}{t_{o.z.} - t_o}, \qquad K_{vap} = \frac{W_{exh} - W_o}{W_{o.z.} - W_o}, \qquad K_g = \frac{C_{exh} - C_o}{C_{o.z.} - C_o} \tag{9.2}$$

where: $\mathbf{t_{exh}}$, $\mathbf{t_o}$ and $\mathbf{t_{o.z.}}$ = temperatures of exhaust, supplied and occupied zone air;
$\mathbf{W_{exh}}$, $\mathbf{W_o}$ and $\mathbf{W_{o.z.}}$ = humidity ratio of exhaust, supplied and occupied zone air;
$\mathbf{C_{exh}}$, $\mathbf{C_o}$ and $\mathbf{C_{o.z.}}$ = contaminant concentration in exhaust, supplied and occupied zone air. $\mathbf{K_w}$, $\mathbf{K_{vap}}$ and $\mathbf{K_g}$ values are used to calculate loads on HVAC systems and to select the best system design.

These definitions of heat/moisture/contaminant removal efficiency correspond with the definition of "contaminant removal effectiveness", given in "A Guide to contaminant removal effectiveness" (Brouns and Waters, 1991), considering that the later is also taken as the level above the value in the supply duct. In some instances it is useful to apply the parameters reflecting the heat/moisture/contaminant removal efficiency to the average conditions in the heated/cooled and ventilated space [i.e., $\mathbf{C_{o.z.}} = \mathbf{C_{room}}^{ave}$ in Equation (9.2)] or to the average parameters in the occupied zone air (i.e., $\mathbf{C_{o.z.}} = \mathbf{C_{o.z.}}^{ave}$) or to a specific locality within the occupied zone (e.g., the operator breathing zone). In the industrial shop the size of operator(s) location could be confined to the relatively small part of the shop volume. Sandberg (1981) emphasizes, the importance to consider ventilation effectiveness in both the "steady state" and in the "transient" phases. In industrial buildings where a sudden release of toxic gas or smoke is possible, special emergency ventilation systems should be used and ventilation efficiency in the short period is an important design parameter.

The knowledge of parameters $\mathbf{K_w}$, $\mathbf{K_{vap}}$ and $\mathbf{K_g}$ values allows the calculation of the amount of air to be supplied in the ventilated space to remove excessive heat, water vapor and impurities. Than the maximum from those values is compared with the amount of air evacuated by local exhausts and process equipment as well as that the amount of air supply based on other considerations (e.g., to keep a positive pressure in the building) to be used for HVAC system and ventilation equipment sizing.

According to Designers Guide (1992) and ASHRAE Handbook (ASHRAE 1995), air flow rates to be supplied into the space for heat/water vapor and contaminant removal can be calculated from the following equations:

$$Q_o = Q_{o.z} + \frac{W - C_p\, \rho\, Q_{o.z}\, (t_{o.z} - t_o)}{C_p\, \rho\, K_w\, (t_{o.z} - t_o)} \tag{9.3}$$

$$Q_o = Q_{o.z} + \frac{M_w - \rho\, Q_{o.z.}\, (W_{o.z} - W_o)}{\rho\, K_{vap}\, (W_{o.z} - W_o)} \tag{9.4}$$

$$Q_o = Q_{o.z} + \frac{G - \rho\, Q_{o.z.}\, (C_{o.z} - C_o)}{\rho\, K_t\, (C_{o.z} - C_o)} \tag{9.5}$$

where: $\mathbf{Q_o}$ - amount of air supplied in the room, m^3/h

$\mathbf{Q_{o.z}}$ - amount of air exhausted from the occupied zone, m^3/h

ρ - air density, kg/m^3

W - excess heat, produced by internal sources (heat gains minus heat losses),W

M - amount of water vapor released into the space, kg/h

G - rate of contaminant release into the space, kg/h

$\mathbf{t_o}$ and $\mathbf{t_{o.z}}$ - temperatures of the supplied air and air in the occupied zone,°C

$\mathbf{W_o}$ and $\mathbf{W_{o.z}}$ - humidity ratio, kg of water per kg of dry air in the supplied air and in the occupied zone air.

$\mathbf{C_o}$ and $\mathbf{C_{o.z}}$ - contaminant concentration in the supplied air and in the occupied zone air, kg/kg.

$\mathbf{C_p}$ - specific heat of air, J/kg.K .

The values of $\mathbf{K_w}$, $\mathbf{K_{vap}}$ and $\mathbf{K_g}$ depend on the method of air distribution, characteristics of the ventilated space, and activities within the space.

Mixing type air distribution systems create comparatively even distribution of temperature and gas concentration throughout the ventilating space. For the mixing type systems $\mathbf{K_w}$ and $\mathbf{K_g}$ have a value close to 1. This corresponds with the theoretical value in the Guide (Brouns and Waters, 1991) and the data in Table 9.1. In the system with a horizontal "piston flow" (also known as a "plug flow") the theoretical value of $\mathbf{K_g}$ parameter is 2 (Brouns and Waters, 1991). In the spaces with strong heat sources there is a tendency to the temperature stratification along the height. Displacement ventilation systems designed for stabilizing this stratification can be characterized by higher value of $\mathbf{K_w}$ coefficient, which can be as high as 2.5 and even higher (Shilkrot and Zhivov, 1992).

In the case when there is a shortcircuiting and the supply air is intercepted by the exhaust system without reaching the occupied locations $\mathbf{K_w}$, $\mathbf{K_{vap}}$ and $\mathbf{K_g}$ coefficient values are be below 1.

Analysis provided by Rymkevich (1990) showed that uneven temperature distribution along the height of ventilated space influence ventilation efficiency as well as heat and cold consumption. When $\mathbf{K_w} > 1$, the consumption of heat increases in the heating period, and the consumption of cold increases in the cooling period. When $\mathbf{K_w} < 1$, the consumption of heat decreases in the heating period and consumption of cold increases in the cooling period. Rymkevich has also showed the influence of unevenness of temperatures in the occupied zone on the energy consumption of HVAC systems and their elements.

Heat, moisture vapor and contaminant removal efficiency coefficients $\mathbf{K_w}$, $\mathbf{K_{vap}}$ and $\mathbf{K_g}$ values can be determined by laboratory and field tests. In some cases, $\mathbf{K_w}$ and $\mathbf{K_g}$ values can be obtained analytically or using Computational Fluid Dynamic (CFD) codes.

Measurements on scale models may provide one specific values, but this approach is restricted to specialist laboratories and would not provide the designer with a readily accessible tool (Liddament 1993).

Ventilation effectiveness field measurement techniques are too complex for routine use in buildings under normal conditions of operation. Also, results are building specific and are difficult to apply to naturally ventilated and leaky structures. The fundamental draw back of field measurements is that they can only be made in existing structures. Very little information can be used to provide guidance for other situations or alternative ventilation strategies.

These shortcomings can be overcomed by using numerical calculation models. In theory, CFD methods have the potential to replicate measurement methods and therefore ventilation efficiency may be evaluated as a part of design process. However, as it is stated in (Liddament 1993) numerical calculation models can be seen as a long term goal, since much more development is needed before they can be regarded as sufficient robust and accurate to simulate the complexities of fully furnished, occupied spaces. Review of CFD methods is presented by Liddament (1991).

Analytical models require the knowledge of airflow pattern in rooms applies the information resulted from air jet theory. These models are based on the system of zone-by-zone heat and mass balances, as well as the turbulent exchange between the zones. The above mentioned approaches to evaluate ventilation effectiveness and some results of their application are discussed in the following Sections.

9.2. EXPERIMENTAL STUDIES ON TEMPERATURE AND CONTAMINANT STRATIFICATION IN ROOMS

Originally, studies on temperature stratification were conducted for spaces with natural ventilation. Reitschil (1930) and Franchuk (1949) studied temperature gradients along the height of the space experimentally. Butakov (1962), Maksimov and Der'ugin (1972) found the experimental method of the temperature stratification evaluation to be not very reliable, due to influences from specific experimental conditions. Later Skåret (1986) shared similar experiences from his practice. He found that the temperature stratification observed in tests conducted in models is greater than in the field, which he explains by the larger air exchange between the upper and lower zones, due to more disturbances in real facilities. Cold convective downdrafts along walls, and air disturbances from the activity and process equipment in real plants are not accounted for in physical simulation and could be among the other reasons.

Seliverstov (1932) described heat flows in naturally ventilated spaces and stated that convective flows created by heat sources do not influence the temperature of the occupied zone directly. The air of the occupied zone is heated due to the secondary heat sources influenced by the radiant heat from the primary heat sources. Seliverstov (1934) provided detailed qualitative analyses of the convective flows in the hot spaces with natural ventilation and found, that when the convective flows created by heat sources are not drawn completely out of the ventilated space, a "hot sack" under the ceiling is created.

Kudryavtsev (1948, 1950) studied convective flows in confined spaces. He found that a heat source placed in the gas or liquid media creates temperature stratification with a dividing line located at the height of the low level of this source. The temperature of liquid or gas below this level is constant and it is not involved in the convective flow. Above this level there is a temperature gradient along the space height resulting in the "heat cushion". A temperature difference of 5-10°C between the upper and lower fluid layers is enough to make this stratification stable. It remains stable even when the lower zone is ventilated.

There were fewer studies done on temperature stratification in mechanically ventilated spaces. Baturin (1972) experimentally studied temperature stratification in glass manufacturing shops with mechanical heating and ventilating systems. Experimental work on the turbulent heat and mass exchange among the upper and lower zones in computer rooms with conditioned air supply through a false floor was done by Melik -Arakelyan (1978).

Experimental studies of air supply with inclined jets in a physical model room (Zhivov and Kelina 1989) have shown that the value of K_w, in the case of air supply in the rooms with insignificant heat gains (< 23 W/m^3), are approximately 1.0 and 1.1 for warm air supply and chilled air supply respectively (Table 9.2). This corresponds with the results of some other laboratory and field studies presented in Table 9.3. Approximate values of heat and vapor removal efficiency coefficients $\mathbf{K_w}$ and $\mathbf{K_{vap}}$ for other typical methods of air supply in industrial spaces are listed in Table 9.1, reproduced from (ASHRAE 1995).

Table 9.1. Coefficients of heat ($\mathbf{K_w}$) and vapor ($\mathbf{K_{vap}}$) removal efficiency for mechanically ventilated spaces with insignificant (less than 23 W/m^2) heat load.

Air Supply Method	K_w/K_{vap}			
	Air change rates, ach			
	3	5	10	>15
Concentrated air jets	0.95/1.1	1.0/1.05	1.0/1.0	1.0/1.0
Concentrated air jets and vertical and/or horizontal directing jets	1.0/1.0	1.0/1.0	1.0/1.0	1.0/1.0
Inclined air jets from a height				
greater than 4m	1.0/1.2	1.0/1.1	1.0/1.05	1.0/1.0
less than 4m	1.15/1.4	1.1/1.2	1.0/1.1	1.0/1.0
Through ceiling mounted air diffusers with				
radial attached jets	0.95/1.1	1.0/1.05	1.0/1.0	1.0/1.0
conical (compact) jets	1.05/1.1	1.0/1.05	1.0/1.0	1.0/1.0
linear (attached) jets	1.1/1.2	1.05/1.1	1.0/1.05	1.0/1.0

Table 9.2. Laboratory tests data on ventilation efficiency for air supply with inclined jets through grills into the space 36 m x 16.4 m x 16.4 m (Zhivov et al. 1989)

Grill size, $\sqrt{A_o}$, m	K_1	Air supply		Jet intercept with occupied zone, X_o/L	Airflow rate, m^3/s	Δt_o, °C	Heat load, W/m^2			Air change rate, 1/h	K_w
		height, h_o/h_r	direction, α_o, deg				occupied zone	upper zone	total		
1.2	3.1	0.67	39	0.3	3.80	3.6	6.9	13.8	20.7	1.4	1.06
1.2	3.1	0.67	39	0.3	5.36	3.9	10.5	21.1	31.6	2.0	1.07
1.2	3.1	0.67	39	0.3	5.51	3.9	10.8	21.7	32.5	2.0	1.08
1.2	3.1	0.33	18	0.3	5.38	3.7	10.0	20.1	30.1	2.0	1.18
1.2	3.1	0.33	18	0.3	5.37	3.6	9.7	19.5	29.2	2.0	1.17
1.2	3.1	0.33	23	0.3	6.34	3.6	11.5	23.0	34.5	2.3	1.15
0.6	3.1	0.67	25	0.6	14.66	4.8	35.4	70.9	106.3	5.4	1.05
1.2	3.1	0.67	22	0.6	10.80	4.1	22.3	44.6	66.9	4.0	1.08
1.2	3.1	0.67	22	0.6	10.81	4.2	22.9	45.7	68.6	4.0	1.09
1.2	3.1	0.33	9	0.6	10.85	4.3	23.5	47.0	70.5	4.0	1.11
0.84	3.1	0.33	12	0.6	14.93	4.6	45.8	91.6	137.4	5.5	1.01
0.6	4.2	0.67	43	0.3	3.12	3.7	5.8	11.6	17.4	1.2	1.09
1.2	4.2	0.67	39	0.3	5.37	4.1	11.1	22.2	33.3	2.0	1.03
1.2	4.2	0.33	18	0.3	3.81	3.6	6.9	13.8	20.7	1.4	1.13
0.84	4.2	0.33	23	0.3	3.37	3.7	6.3	12.5	18.8	1.2	1.09

Table 9.3. Experimental data on ventilation efficiency for air supply with inclined jets based on field and laboratory studies (Reproduced from Zhivov et al. 1989)

Air diffuser	Jet type	Year period	Ventilated space characteristics		Room size, m^2	Air supply		Ventilation efficiency coefficient, K_w	Reference
			Heating/ cooling load, W/m^2	Airflow rate, m^3/m^2h		direction α_o, deg	height, m		
VSP-4	compact	heating	130.8	52.3	48 x 17	25~30	15	0.96	GPI PPV, 1983
VSP-4	compact	heating	130.8	52.3	48 x 17	25~30	15	0.94	GPI PPV, 1983
VSP-4	compact	heating	139	58	30 x 60	10~15	13.6	0.94	GPI PPV, 1983
VSP-4	compact	heating	139	58	30 x 60	10~15	13.6	0.97	GPI PPV, 1983
VPES-10	swirl	heating	23.2	15.5	24 x 60	20	8	0.92	GPI PPV, 1983
VPES-14	swirl	heating	69.3	23.1	26 x 18	30	10	0.92	GPI PPV, 1983
VPES-14	swirl	heating	69.3	23.1	26 x 18	30	10	0.94	TsNIIPZ, 1985
NRV	incomplete fan-type	cooling	92.8	45	12 x 12	0	4	1.15	TsNIIPZ, 1985
NRV	incomplete fan-type	cooling	139.2	60	12 x 12	0	4	1.15	TsNIIPZ, 1985
VES	swirl	cooling	69.6	70	12 x 12	33	12	1.04	TsNIIPZ, 1985
VGK	compact	heating	89.8	27.6	24 x 12	20	4	1.04	TsNIIPZ, 1985
VGK	compact	heating	2.6	8.4	24 x 12	20	4	1.04	TsNIIPZ, 1985

9.3. ZONAL MODELS USING COMPUTATIONAL FLUID DYNAMICS CODES

Zonal models are used to calculate temperature and/or contaminant concentration in different zones of the building or a separate room. In a multizonal model approach one makes an idealization in the sense that a room or a building is subdivided into a number of zones (Etheridge and Sanberg, 1996). In each zone temperature and contaminant concentration is assumed to be uniform. Equations that constitute the multizonal model express the conservation of energy and mass (both air and contaminant). Subdivision of room into a number of zones depends on several factors and requires considerable experience (Etheridge and Sanberg 1996; Shilkrot 1993). The accuracy of prediction increase with a number of zones. When number of zones is greater than three one normally has to rely on numerical methods. Ventilation effectiveness indices can be derived using one and two dimensional analysis (Brouns and Waters 1991). However, a more detailed analysis of the room air flow can be obtained by solving relevant fluid flow equations in three dimensions. Commercially available software for such computation include programs "PHOENIX" (UK), "FLUENT" (UK) and "FLOWVISION" (Russia). Boundary descriptions complexities and computational requirements limit general acceptance in the design community, although speciality designs have benefitted from CFD simulations. Commercial and residential buildings constitute the majority of multizonal model applications (Lebrun 1978, Laret 1980, Howarth 1980, Inard 1988, 1996; Chen 1990, Togari et. al. 1993, Takemasa et al. 1996). Different multizonal models in application to large spaces (e.g., atriums) are discussed in the final report of the Annex 26 (Heselberg et al. 1996).

9.4. ZONAL MODELS BASED ON ANALYTICAL APPROACH

Temperature and contaminant stratification along room height also can be computed using the analytical method of zone-by-zone balances. This method is used in designing radiant/convective heating (Lebrun 1978, Bogoslovsky 1982), natural ventilation (Shepelev, 1962; Shilkrot, 1976), mechanical ventilation with different air supply schemes (Pozin 1983, Shilkrot 1993, Shilkrot et al. 1992, 1996a, 1996b, Bach et al. 1992, Dittes 1994). With this method the room is divided into separate zones, and within each of this zones the temperature and contaminant concentration values are assumed to be equal and can be calculated from a joint solution of mass and thermal balance equations for each zone.

A review of simplified one, two and four zonal models for idealized isothermal situations including cases of "complete mixing" "short-circuiting flow" and a "piston flow" and different locations of contaminant injection is presented in "A Guide to Contaminant Removal Effectiveness" (Brouns and Waters). Calculation of contaminant removal efficiency requires the knowledge of supplied and exhausted airflow rates for each zone, amounts of contaminants released into each zone and the, so-called, "**recirculation factor, β** ", which is the measure of the degree of internal mixing between each pair of zones. Graphs presented in the "Guide" resulted from calculations based on these simplified zonal models show, that the contaminant removal efficiency is significantly influenced by the recirculation factor value. However, contaminant removal efficiency value for all the presented cases is within 10% of that for the complete mixing case ($\mathbf{K_g} = 1$) when $\beta > 5$.

The process of dividing a room into zones and determining heat and contaminant exchange coefficients between them cannot be formalized. To divide a room into zones, one should define a physical pattern of air and heat flow propagation, using common considerations from aerodynamic and thermal physics laws and experimental data. Conductive, radiant and convective heat exchanges between zones are accounted for. In a general case the number of zones (and the numbers of heat balance equations) can be rather great.

Convective heat exchange can take place both on the air-to-surface boundary, and on the air-to-air boundary. Air-to-air heat exchange can be related to the mass transfer (air jets, thermal plumes or cold convective downdrafts crossing the boundary, air entrainment to the jet, etc.) or without it (turbulent heat exchange, when the resulting mass flow is equal to zero). The reliability of calculations using this method depends significantly on the proper choice of zones and exchange coefficients.

In mechanically ventilated rooms, one can consider three classes of zones: (1) surfaces of the building envelope and heat sources; (2) passive volumes: air in occupied zone and upper zone; and (3) active volumes; i.e. air jets and convective plumes.

Analytical method developed by Pozin (1975) of $\mathbf{K_w}$ determination is discussed in (Pozin 1980, 1983, 1984, 1993). The system of equations considers all heat flows within the room as well as in and out of the room, and air flows within the room. The model also includes equations based on diffuser jet theory (Grimitlyn 1993).

The model considers such parameters as indoor thermal conditions, total heat load, $\mathbf{W}$, heat loads from the internal sources located in the occupied zone, $\mathbf{W_{o.z}}$, heat losses, heat, $\mathbf{W_c}$, and mass, $\mathbf{Q_c}$, transported by the convective flows, air flow rate through the cross-section of the diffuser jet, $\mathbf{Q_j}$, and jets cross-sectional area at the point of its entering the occupied zone, $\mathbf{A_j}$, air flow rate at the critical cross-section of the confined jet (end of the Zone 2), $\mathbf{Q_{cr.j}}$,share of the inclined jet entering the occupied zone, $\boldsymbol{\theta}$, air flow rate evacuated by local exhausts, $\mathbf{Q_{l.exh}}$, heat gains due to hot water heating systems, infiltration and exfiltration and air flows from adjacent rooms.

This method determines the temperature of the evacuated air and air in the occupied zone, and, thus, the value of the $\mathbf{K_w}$ coefficient. In general, determination of $\mathbf{K_w}$ requires the use of computer (Pozin 1984). For the practical purposes the method can result in the following simplified equations for different methods of air supply:

- For air supply directly to the occupied zone (Figure 9.1a):

$$K_w = \frac{\alpha\left[\frac{Q_j}{Q_o}+\frac{Q_{o.z}}{Q_o}\left(1-\frac{W_{o.z}}{W}\right)\right]+\frac{Q_{o.z}}{Q_o}\left(\frac{Q_j}{Q_o}+\frac{Q_{o.z}}{Q_o}-1\right)\frac{W_c}{W}}{\alpha\left[\frac{Q_j}{Q_o}-\left(1-\frac{Q_{o.z}}{Q_o}\right)\right]-\left(1-\frac{Q_{o.z}}{Q_o}\right)\left(\frac{Q_j}{Q_o}+\frac{Q_{o.z}}{Q_o}-1\right)\frac{W_c}{W}} \tag{9.6}$$

where: $\boldsymbol{\alpha} = \max\{\mathbf{Q_c/Q_o},\ 1 - \mathbf{Q_{o.z}/Q_o}\}$; $\mathbf{Q_o}$ = supply air flow rate; $\mathbf{Q_{o.z}}$ = air flow rate into the occupied zone from adjacent rooms.

- For air supply by vertical downward projected jets (Figure 9.1b):

$$K_w = \frac{\alpha\left[\frac{Q_j}{Q_o}-\frac{W_{o.z}}{W_o}\left(1-\frac{A_j}{A_{o.z}}\right)\frac{Q_{o.z}}{Q_o}\right]+\frac{W_c}{W}\,\frac{Q_{o.z}}{Q_o}\left(\frac{Q_j}{Q_o}-1\right)}{\alpha\left[\frac{Q_j}{Q_o}+\frac{W_{o.z}}{W}\left(1-\frac{A_j}{A_{o.z}}\right)\left(1-\frac{Q_{o.z}}{Q_o}\right)-1\right]-\frac{W_c}{W}\left(\frac{Q_j}{Q_o}-1\right)\left(1-\frac{Q_{o.z}}{Q_o}\right)} \tag{9.7}$$

- For air supply through the ceiling mounted air diffusers by attached jets (Figure 9.1c):

$$K_w = \frac{\dfrac{Q_j}{Q_o} - \dfrac{W_{o.z}}{W}\dfrac{Q_{o.z}}{Q_o}\left(1-\dfrac{A_j}{A_{o.z}}\right)}{\dfrac{Q_l}{Q_l-Q_o}\left(\dfrac{Q_j-Q_{o.z}}{Q_o}\right)-\left(1-\dfrac{Q_{o.z}}{Q_o}\right)\left[1-\dfrac{W_{o.z}}{W}\left(1-\dfrac{A_j}{A_{o.z}}\right)\right]} \tag{9.8}$$

- For air supply by inclined jets (Figure 9.1d):

$$K_w = \frac{\theta\dfrac{Q_j}{Q_o} - \dfrac{W_{o.z}}{W}\dfrac{Q_{o.z}}{Q_o}}{\theta\left(\dfrac{Q_j}{Q_o}-1\right) + \dfrac{W_{o.z}}{W}\left(1-\dfrac{Q_{o.z}}{Q_o}\right)} \tag{9.9}$$

- Concentrated air supply by non attached jets - occupied zone is ventilated by the reverse flow (Figure 9.1e):

$$K_w = \frac{0.5\dfrac{Q_{cr.j}}{Q_o} + \dfrac{Q_{o.z}}{Q_o}\left(1 - 2\dfrac{W_{o.z}}{W}\right)}{2\dfrac{W_{o.z}}{W}\left(1-\dfrac{Q_{o.z}}{Q_o}\right) + 0.5\dfrac{Q_{cr.j}}{Q_o} + \dfrac{Q_{o.z}}{Q_o} - 1} \tag{9.10}$$

- Concentrated air supply by attached air jets (Figure 9.1f):

$$K_w = 1 - \frac{Q_o}{Q_l} \tag{9.11}$$

In Equation (9.11) $\mathbf{Q_l = Q_{cr.j}}$, if the jet length exceeds the distance from to the second critical cross-section ($\mathbf{l > X_{cr.j}}$).

From Equation (9.3), the greater is $\mathbf{K_w}$, less air flow $\mathbf{Q_o}$ is needed to be supplied into the ventilated space and more efficiently it is used.

Analysis of equations from (9.6) through (9.11) shows that the most effective method of air supply is directly into the occupied zone ($\mathbf{K_w}$ is from 1 to 3). When air is supplied by vertical jets, the value of $\mathbf{K_w}$ is also greater than 1 though less then when air is supplied into the occupied zone. The value of $\mathbf{K_w}$ is affected mostly by such factors as a diffuser jet ejection capability ($\mathbf{Q_j/Q_o}$) and the characteristics of the convective flows ($\mathbf{Q_c}$, $\mathbf{W_c}$).

When diffuser jet is attached to the ceiling or the wall, the air exhausts located in the jet region evacuate part of the fresh air supplied by the jet, which results in the lower value of $\mathbf{K_w}$ ($\mathbf{K_w}$ <

1). If air exhaust still has to be along the jet trajectory, Equation (9.8) can help to specify the distance to the location of the air exhaust, so that the $\mathbf{K_w}$ value would not be less than designed. In most of cases, $\mathbf{K_w}$ depends on the relative air flow through the jet cross-section ($\mathbf{Q_j/Q_o}$) in place of the exhaust location.

For air supply with inclined jets the $\mathbf{K_w}$ value will change in a wide range depending on the air flow pattern used, which reduces limitations on HVAC system control. The parameters which influence $\mathbf{K_w}$ value at air supply by inclined jets are θ and $\mathbf{Q_j/Q_o}$.

When air is supplied with concentrated air jets, the $\mathbf{K_w}$ value (Equations 9.10 and 9.11) in most cases is less than 1 and depends, mostly, on the relative air flow rate $\mathbf{Q_{cr.j}/Q_o}$ at the critical cross-section.

Similar model to calculate heat and contaminant loads on HVAC system was developed at the Stuttgart University (Bach et al. 1992, Dittes 1994). This model accounts for degree of heat/contaminant capture by local exhausts and a load factor that significantly depends upon the method of air supply. Researchers applied their model to air supply by inclined and vertical jets, displacement air supply, and air supply through openings in the floor. They emphasize an importance of considering impact of both convective and radiant components of heat produced by sources on the system load.

The model developed by Shilkrot (1993) considers not only conductive, radiant and convective heat exchanges between zones but turbulent exchange as well. The turbulent air exchange between the upper and occupied zones depends upon the energy introduced to the space by supply air jets, convective flows, human activity and process equipment operation (in industrial applications). 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 additional heating of the upper zone by 1-3°C as compared to the occupied zone air temperature (Shilkrot, 1993).

Shilkrot (1993) examined the influence of the turbulent exchange in the room on the temperature separation stability and stated that the heat flux density due to turbulent exchange, can be determined by the formula:

$$q_{turb} = A_{turb} C_p \, g \, \rho \frac{\delta t}{\delta z} \tag{9.12}$$

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

To calculate the turbulent exchange coefficient, A_{turb}, for the separation zone he used the V.H.Munk and E.R.Anderson relationship (Merrit et al., 1973), which is in a good agreement with empirical data:

$$A_{turb} = A_{ex} \, (1 + 3.3 \, Ri)^{-3/2} \tag{9.13}$$

The exchange coefficient, A_{ex}, has been evaluated experimentally (as a turbulent exchange coefficient in a convective flow above the heat release source at the separation level) and following the method proposed by Elterman (1980). According to Elterman, the turbulent exchange coefficient is proportional to the total energy of incoming (E_{jet}) and convective (E_{conv}) jets attenuating in the room and the energy introduced by moving objects ($E_{m.o.}$):

$$E_{room} = \Sigma E_{jet} + \Sigma E_{conv} + \Sigma E_{m.o.} \tag{9.14}$$

where:

$$E_{jet} \cong ACH * \frac{V_o^2}{2} \tag{9.15}$$

$$E_{conv} = \cong \frac{W_{conv}}{\rho * L * B * H} Z \tag{9.16}$$

$$E_{m.o.} \cong V_{m.o.}^3 * A_{m.o.} * \rho * \frac{\tau}{3600} \tag{9.17}$$

L, B and **H** = room dimensions; $\mathbf{W_{conv}}$ = convective component of the heat produced by the source; $\mathbf{V_{m.o.}}$ = velocity of the moving object; $\mathbf{A_{m.o.}}$= cross-sectional area of the moving object; τ = portion of a time per hour (sec) when the object is moving with a speed of $\mathbf{V_{m.o.}}$:

The energy supplied and/or produced in the space can be evaluated using criteria **K** introduced by Elterman (1980):

$$K = \frac{V^3 * C_p * \rho * L * B * H}{\beta * g * W_{conv} * I} \tag{9.18}$$

This criteria characterize the airflow in the stratified space and is a ratio of kinetic energy dissipating in the ventilated space to the energy used to suppress the buoyancy forces. The criteria **K** is similar to Archimedes number introduced in 1930 by Baturin and Shepelev (Elterman, 1980) to characterize air jets influenced by buoyancy or to the Richardson criteria used in meteorology to characterize the ratio of the turbulence suppression by the buoyancy forces over the turbulence generation by the Reynolds tension (Munk, 1948; Merrit et al., 1973). In the case of displacement ventilation, the Richardson criteria can be defined (Shilkrot et al., 1992) by the relationship :

$$Ri = \frac{g}{T}\frac{\delta t/\delta z}{(\delta V/\delta z)^2} \approx \frac{g}{T_{o.z}}\frac{\Delta t/\Delta z}{(\Delta V/\Delta z)^2} \tag{9.19}$$

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

The following section is based on the studies conducted jointly by Shilkrot (1996) and at the University of Illinois (Zhivov, 1997) and illustrates application of the multizonal model to heat and contaminant removal efficiency evaluation for mixing type air distribution with air supply through ceiling mounted air diffusers.

9.5. APPLICATION OF MULTIZONAL MODELS TO PREDICT HEAT AND CONTAMINANT REMOVAL EFFICIENCY WITH AIR SUPPLY THROUGH CEILING MOUNTED AIR DIFFUSERS

9.5.1. Heat removal efficiency

The analysis of air circulation patterns in rooms with a mixing type air distribution allows to consider five typical zones: the air of the upper zone and the occupied zones of the room (under and above the upper level, $h_{o.z.} = 1.8$ m, of the occupied zone, respectively, surfaces of the envelope (walls and ceiling) in those zones, and the air jet region, and to write heat balance equations for each of them. The total heat load introduced by each source:

$$W_o = W_{conv} + W_{rad} = \psi\, W_o + (1 - \psi) W_o \qquad (9.20)$$

The total radian component of the heat load introduced by each source:

$$W_{rad} = W_{rad\ low} + W_{rad\ up} = \varphi(1 - \psi) W_o + (1 - \varphi)(1 - \psi) W_o \qquad (9.21)$$

The total convective component of the heat load introduced by each source:

$$W_{conv} = W_{conv\ low} + W_{conv\ up} = \beta\psi W_o = (1 - \beta)\psi W_o \qquad (9.22)$$

where: ψ, φ, and β are dimensionless coefficients reflecting for each heat source the portion of the convective component of the total heat load released into the space, the portion of the radiant component of the total radiant heat load in the low zone, and the portion of the convective component of the total convective heat load in the low zone, respectively.

Heat balance for the upper zone surfaces:

$$W_{rad\ up} + W_{rad\ low\text{-}up} = \alpha_{up}\, A_{up}\, (\tau_{up} - t_{up}) \qquad (9.23)$$

Heat balance for the low zone surfaces:

$$W_{rad\ low} + W_{rad\ up\text{-}low} = \alpha_{up}\, A_{low}\, (\tau_{low} - t_{low}) \qquad (9.24)$$

Heat balance for the upper zone air:

$$\alpha_{up}\, A_{up}\, (\tau_{up} - t_{up}) + W_{conv\ up} + C_p\, G_o\, t_o - C_p\, G_{jet}\, \overline{t_{jet}} + C_p\, G_{jet}\, t_{low} - C_p G_o\, t_{up} = 0 \tag{9.25}$$

Heat balance for the low zone air:

$$\alpha_{low}\, A_{low}\, (\tau_{low} - t_{low}) + W_{conv\ low} + C_p\, G_{jet}\, \overline{t_{jet}} - C_p\, G_{jet}\, t_{low} = 0 \tag{9.26}$$

Heat balance for the air jet upper zone region:

$$C_p\, G_o\, t_o + C_p\, (G_{jet} - G_o)\, t_{up} = C_p\, G_{jet}\, \overline{t_{jet}} \tag{9.27}$$

Assuming,

$$\mu = \frac{G_{jet} - G_o}{G_o}, \quad k = \frac{\tau_{up} - \tau_{low}}{t_{up} - t_{low}} \tag{9.28}$$

one can derive the following equation for K_w coefficient from the system of equations (9.21) through (9.28):

$$K_w = \frac{1 + \dfrac{k\,(\alpha A)_{rad\ low}}{C_p\, G_o} + \mu}{[\varphi(1 - \psi) + \beta\psi] + \dfrac{k\,(\alpha A)_{rad\ low}}{C_p\, G_o} + \mu} \tag{9.29}$$

The relative airflow rate in jets G_{jet}/G_o can be calculated from the following equations:

for compact vertically projected jet:

$$\frac{G_{jet}}{G_o} = \frac{2}{K_1}\, \frac{H_r - h_{o.z.}}{\sqrt{A_o}} \tag{9.30}$$

for radial jet:

$$\frac{G_{jet}}{G_o} = \frac{\sqrt{2}}{K_1} \frac{\dfrac{\sqrt{L_r \times B_r}}{2} + H_r - h_{o.z.}}{\sqrt{A_o}} \tag{9.31}$$

The numerator of Equation (9.29) consists of two components reflecting a heat flux from the upper zone to the lower zone due to the radiant heat exchange and a mass transfer with a jet. The denominator of Equation (9.29) consists of three components: (1) reflecting the share of the total heat load from the source effecting the low zone air temperature directly; (2) and (3) same as in the numerator.

In rooms with a mixing type air distribution, the air flow pattern is influenced primarily by air jets. While in displacement ventilation systems, buoyancy plumes have the major effect on air circulation. E.g., when air is supplied through the ceiling mounted air diffuser, the parameter μ, reflecting heat flux from the upper zone to the low zone due to the mass transfer can be evaluated using Equations (9.29), (9.30) and (9.31):

for compact vertically projected jet:

$$\mu = \frac{G_{jet} - G_o}{G_o} = \frac{2}{K_1} \frac{H_r - h_{o.z.}}{\sqrt{A_o}} - 1 \tag{9.32}$$

for radial jet:

$$\mu = \frac{G_{jet} - G_o}{G_o} = \frac{\sqrt{2}}{K_1} \frac{\dfrac{\sqrt{L_r \times B_r}}{2} + H_r - h_{o.z.}}{\sqrt{A_o}} - 1 \tag{9.33}$$

The analysis of the Equation (9.29) shows, that for the mixing type air distribution systems $\mathbf{K_w}$ value is close to 1.

In the extreme case with the room where all heat sources are located only in the upper zone (e.g., light bulbs), and assume no radiant heat exchange between the upper and the lower zones, $\varphi = 0$ and $\alpha_{rad} = 0$. In the room with a height, $\mathbf{H_r} = 4$ m, air is supplied vertically through the air diffuser with $\mathbf{K_1} = 3$ and $\mathbf{D_o} = 6"$ (0.16 m). The parameter $\boldsymbol{\mu}$ value calculated using Equation (9.32) is:

$$\mu = \frac{2}{3}\,\frac{4 - 1.8}{\sqrt{\frac{0.16^2\,\pi}{4}}} - 1 = 9.3$$

From the Equation (9.29) the $\mathbf{K_w}$ coefficient value is equal to 1.1. The calculations for the same case with a radial jet air supply (room size 6m x 6m, $\mathbf{K_1}$ = 1.2 and $\mathbf{D_o}$ = 0.16 m) results in $\boldsymbol{\mu}$ = 16.3 and a $\mathbf{K_w}$ coefficient value is equal to 1.06.

9.5.2. Contaminant removal efficiency

Contaminant removal efficiency evaluation is similar to evaluation procedure of heat removal efficiency. It also requires the analysis of air circulation patterns in rooms and dividing the room space into typical zones. In case with air supply through ceiling mounted air diffusers, three distinctive zones can be selected: the occupied zone air, the upper zone air and the air jet.

The following information is required to evaluate contaminant removal efficiency:

Contaminant concentration in the supply air, $\mathbf{C_o}$;
Contaminant release into the occupied zone, $\mathbf{G_{o.z.}}$;
Contaminant release into the upper zone, $\mathbf{G_{u.z.}}$;
Room height, H_r, and floor area per one air diffuser, $\mathbf{A_r}$;
Type of air diffuser to be used and its $\mathbf{K_1}$ characteristic.

Contaminant removal efficiency coefficient value derivation is based on the following three equations.

Mass balance equation for the occupied zone:

$$C_{jet}\,Q_{jet} + G_{o.z.} - C_{o.z.}Q_{jet} = 0 \tag{9.34}$$

Mass balance equation for the upper zone:

$$C_{o.z.}Q_{jet} - C_{u.z.}Q_o + G_{up} - C_{u.z.}(Q_{jet} - Q_o) = 0 \tag{9.35}$$

Jet region:

$$C_oG_o + C_{u.z.}(G_{jet} - G_o) = C_{jet}G_{jet} \tag{9.36}$$

In equations (9.34) through (9.36) $\mathbf{C_{jet}}$ and $\mathbf{Q_{jet}}$ are average contaminant concentration and airflow rate in the jet at the point of its entering the occupied zone. The following equations allow for evaluation of the contaminant concentration in the occupied zone, $\mathbf{C_{o.z.}}$, the upper zone, $\mathbf{C}_{u.z.}$, and the contaminant removal efficiency, $\mathbf{K}_g$, as defined by Equation (9.2).

$$C_{exh} = C_o + \frac{G_{up} + G_{o.z.}}{Q_o} \tag{9.37}$$

$$C_{o.z.} = \frac{G_{up} + G_{o.z.}}{G_o}\left(1 - \frac{G_{up}}{G_{up} + G_{o.z.}} \times \frac{Q_o}{Q_{jet}}\right) \tag{9.38}$$

$$K_g = \frac{1}{1 - \frac{G_{up}}{G_{o.z.} + G_{up}} \times \frac{Q_o}{Q_{jet}}} \tag{9.39}$$

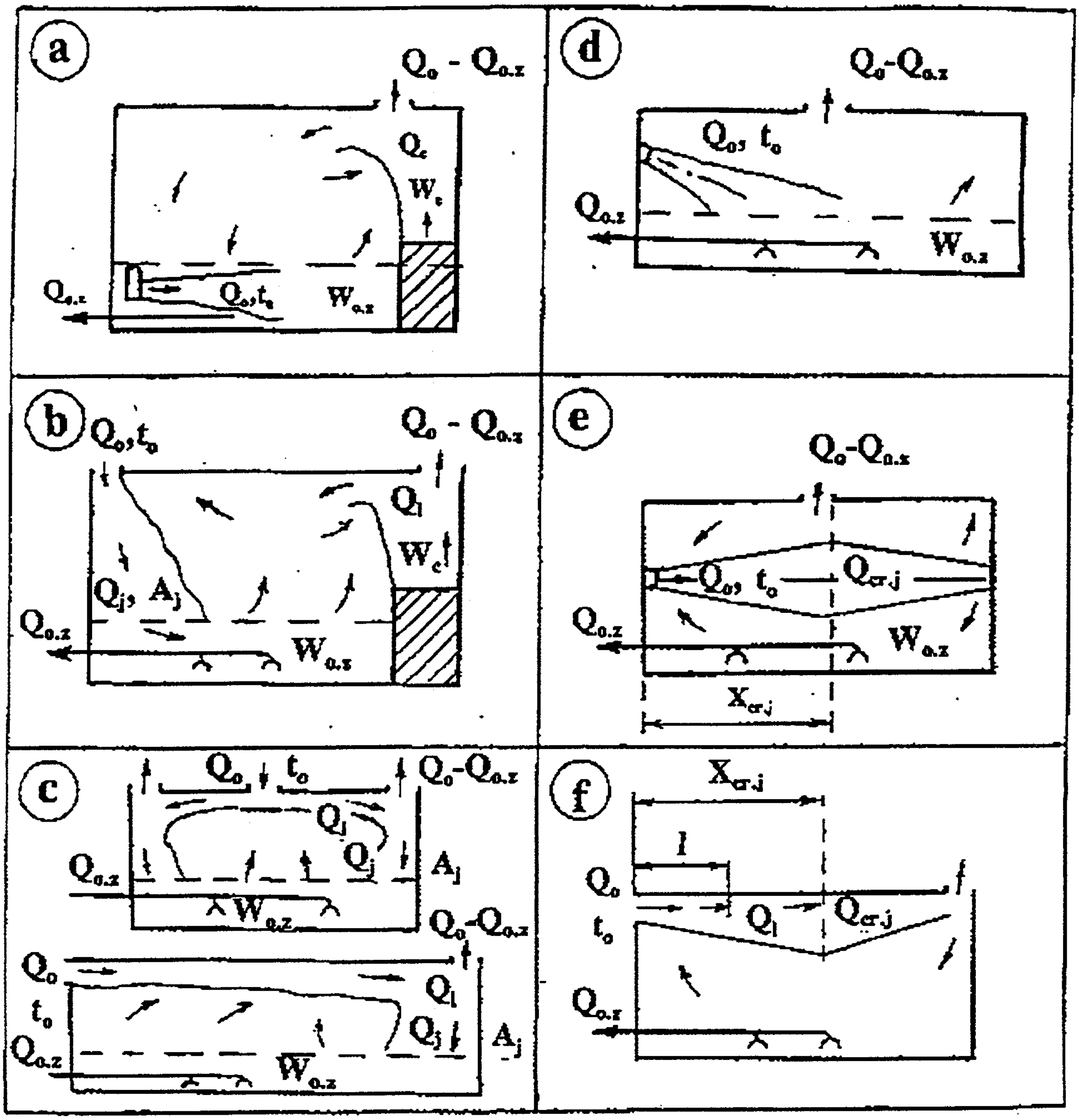


Figure 9.1. Schematics of the different methods of air supply:
a - directly into the occupied zone;
b - with vertically projected jets;
c - through ceiling mounted air diffusers with air jets attached to the ceiling;
d - by inclined air jets;
e - air supply through grilles and nozzles by the not attached jets and ventilating of the occupied zone by the reverse flow;
f - air supply through the grilles and nozzles by the attached to ceiling jets and ventilating of the occupied zone by the reverse flow.

10. REFERENCES

ABB. 1990. Industrial Ventilation. The right air to the right place in the right way. A presentation of the technical possibilities of today. E 2432B.

Abramovich, G.N. 1984. Theory of Turbulent Jets. Moscow: FIZMATGIZ.

Abramovich, G.N. 1960. Theory of Turbulent Jets. Moscow: FIZMATGIZ.

Abramovich, G.N. 1948. Turbulent Free Jets of Liquids and Gases. Moscow: GOSENERGOIZDAT.

Abramovich, G.N. 1940. Turbulent Free Jets of Liquids and Gases. Proceedings of TsAGI. No. 512.

ACGIH. 1996. Threshold Limit Values and Biological Exposure Indices. ACGIH.

AICVF. 1991. Aeraulique. Principles de l'aeraulique appliaues au genie climatique. Collection des guides de l'AICVF.

AIR-IX. 1987. Teollisuusilmanvaihdon suunnittelu. Kauppa-ja teollissusministerio. Helsinki.

AFS. 1990. Hygieniska gränsvärden. AFS 1990:13. Publikationsservice. Solna, Sweden.

Albertson M.L., Dai Y.B., Jensen R.A. Rouse H. 1948. Diffusion of submerged jets. Proceedings of the ASCE. V. 74. p. 1751.

Andersen, K.T. 1996. Beregning af Luftstråler og returstrømme i rum (Calculation of air jets and reverse flows in rooms). SBI-Report 248. Statens Byggeforskningsinstitut. 78pp.

Andersen, K.T. 1995. Vurdering af luftstrømningsmodeller for glasbygninger (Evaluation of air flow models in glass buildings). SBI-Report 247. Statens Byggeforskningsinstitut. 78pp.

Anderson, R., V. Hassani, A. Kirkpatrick, K. Knappmiller, and D. Hittle. 1991. Experimental and computational visualization of cold air ceiling jets. ASHRAE Journal. V.33(5).

Angel L.K. and B.M. Rudman. 1974. Ventilation of Non-ferrous Metallurgy Plants. Moscow: Metallurgy. 201 pp.

Annex 26. 1996. Ventilation of Large Spaces in Buildings. Part 3. Analysis and Prediction Techniques. Edited by P. Heiselberg, S. Murakami and C.-A. Roulet. IEA.

Aro, T. K. Koivula. 1992. Learning from experience with industrial ventilation. AIR-IX Consulting Engineeres. Center for the Analysis and Dissemination of Demonstrated Energy Technologies.

ASHRAE. 1997. Handbook of Fundamentals. Chapter 31. Atlanta. GA.

ASHRAE Handbook. Applications. 1995. ASHRAE. Atlanta, GA.

ASHRAE Standard 55. 1992. "Thermal environmental conditions for human occupancy". Atlanta, GA.

Awbi, H.B. 1991. Ventilation of Buildings. London: Chapman and Hall, UK.

Azer. N.Z. 1984. Design of Spot Cooling Systems for Hot Industrial Environments. ASHRAE Transactions. Vol.90. P.1B. pp.460-474.

Azer. N.Z. 1982. Design Guidelines for Spot Cooling Systems: Part 1- Assessing the Acceptability of the Environment. ASHRAE Transactions. Vol.88. P.1. pp.81-96.

Bach, H. 1973. Ähnlichkeitskriterien bei Raumströmungen (Analogy Criteria with space flow). Ki. Vol. 9. No.6, pp.37-42.

Bach H., W. Dittes, M. Madjidi, K. Becher, R. Detzer et al. 1992. Zonal Ventilation of Work Areas in Production Halls to Reduce Contaminant Load. Research report. HK 216.

Baharev, V.A. and V.N. Troyanovsky. 1958. The Basis of Heating and Ventilating Systems with the Concentrated Air Supply Design and Calculation. Moscow: PROFIZDAT.

Bakke, P. 1957. An experimental investigation of a wall jet. Journal of fluid mechanics. V.2. pp.467-472.

Balandina, L.Ya. 1996. Distribution of Air in the Working Zones. Proceedings of the 7th International Conference on Indoor Air Quality and Climate. INDOOR AIR'96. Vol. 1. July 1996. Nagoya, Japan.

Bašus, V. and Kočova V. 1963. Vzajemne Pusobeni Volnich Proudu. Zdravonti Technika a Vzduchotechnika. No. 4, pp.150-168.

Baturin, V.V. 1972. Fundamentals of Industrial Ventilation, 3rd English Edition. Pergamon Press. New York.

Baturin, V.V. 1965. Fundamentals of Industrial Ventilation. VtsSPS: PROFIZDAT. (In Russian)

Baturin, V.V. and V.I. Hanzhonkov. 1939. Air Circulation in Rooms Depending on the Location of Supply and Exhaust Openings. Heating and Ventilating. No. 4-5.

Baturin, V.V. and I.A. Shepelev. 1935. Heating and ventilation. No.5.

Becher, P. 1966. Lüftverteilung in gelüfteten Räumen. Heizung, Lüftung-Haustechnik. V. 17, No. 10. pp. 379-384.

Becher, P. 1950. Luftstrahlen aus Ventilationsöffnungen. Gesundheits-Inginieur. 71. P.139.

Becher, P. 1949. Calculation of jets and inlets in ventilation. Ph.D. Thesis. Technical University of Denmark. Copenhagen. 200pp.

Beck,E. 1992. Luftführung in Industrieräumen. TGA-Magazin. Vol.6. pp.36-42.

Beltaos, S. 1976. Oblique Impingement of Circular Turbulent Jets". Journal of Hydraulic Research. 1976. V. 14 (1). p 17-36.

Beyersdorfer, S. 1986. Proportionalität der Geschwindigkeitsfelder in Strömungssystemen mit Unterschiedlichen Werten der Ähnlichkeitskennzahlen. Luft- und Kältetechnik. No.2. pp.94-95.

Bogoslovsky V.N. 1982. Thermal Physics of Buildings. Moscow: Vis'shaya shkola. (in Russian)

Bourque, C., B.G. Newman. 1960. Reattachment of a Two-Dimensional Incompressible Jet to an Adjacent Flat Plate. The Aeromechanical Quarterly. Vol. XI. August. pp.201-232.

Bromley, M.F. 1946. Experimental studies of flows distribution. D.Sc. Theses. MISI. 152p.

Brouns, C. and Waters, B. 1991. A Guide to Contaminant Removal Effectiveness. Technical Note AIVC 28.2. December 1991.

Brunig, M. 1984. Bei Prallstromung senkrecht gegen eine Wand auftretende Umfeldströmung. HLH 35. Nr. 10. October. pp. 503-504.

Butakov S.E. 1962. Principles of Hot Shops Ventilation. Sverdlovsk: Metallurgizdat. (in Russian)

Carslaw and Jaeger. 1947. Conduction of Heat in Solids. Oxford University Press.

Chen, C.J., W.Rodi. Vertical Turbulent Buoyant Jets. A Review of Experimental Data. Pergamon Press. 79pp.

Chen Q., A. Moser and A. Huber. 1990. Prediction of buoyant, turbulent flow by a low Reynolds-number k-ϵ model. ASHRAE Transactions. Vol. 96 (1). p.15.

Chow, W.K. and W.Y. Fung. 1997. Experimental Studies on the Airflow Characteristics of Spaces with Mechanical Ventilation. ASHRAE Transactions. P.1.

Christianson, L.L. 1989. Building Systems: Room Air and Air Contaminant Distribution". ASHRAE, Atlanta.

CIBSE. 1988. CIBSE Guide. Installation and equipment data. Volume B. Chartered Institution of Building Services Engineers. London, UK.

Cole J.P. 1995. Ventilation Systems to Accommodate the Industrial Process. Heating, Piping, Air Conditioning - The magazine of mechanical systems engineering. May.

Coli, G. 1974. Distribuzione dell'aria con diffusori a lancio orizzontale. Condizionamento dell'Aria. No.7.

Corrsin, S. 1943. Investigation of flow in an axially symmetric heated jet of air. NACA wartime report. W-94.

Corrsin, S. and M.S. Uberoi. 1950. Further experiments on the flow and heat transfer in a heated turbulent jet. NACA Report No. 998.

Croom, D.J. and B.M. Roberts. 1981. Air Conditioning and Ventilation of Buildings. 2nd Edition. V. 1. Pergamon Press.

Curilev, E.S. and M.Z. Pechatnikov. 1966. Laws of Jets Distribution in Heavy Obstructed Spaces. Proceedings of the conference: "Heat Supply and Ventilation". Kiev: Budivelnik, pp. 137-141.

Davis, J.A. 1977. The unidirectional flow ventilation systems. Heating, piping/air conditioning. March. pp.63-67.

Delicieeux, P., H. Bouia and D. Blay. 1992. Simplified Modelling of Air Movements in a Room and its First Validation with Experiments. ROOMVENT'92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 1. Aalborg. pp.383-397.

Designer's Guide. 1992. Ventilation and Air-Conditioning. Moscow: STROIZDAT.

Detzer, R. 1993. Belüftung von Industriehallen. (Ventilation of industrial halls). Ki Klima-Kälte-Heizung. Vol. 3. pp.98-101.

Detzer, R. 1973. Beitrag über das Verhalten runder Luftfreistrahlen (Contribution to the behavior of round air jets). KI. Vol. 4/73. No. 6, pp. 9-15.

Didenko, V.G. 1972a. Studies of the process equipment location density on the ventilation effectiveness at different methods of air supply. Transactions on sanitary technique. Volgograd: Nizhne-Vozhskoye publishing house. Issue IV, p 115-117.

Didenko, V.G. 1972b. Studies of the process equipment sizes on the ventilation effectiveness at different methods of air supply. Transactions on sanitary technique. Volgograd: Nizhne-Vozhskoye publishing house. Issue IV, p. 124-126.

Dittes W. 1994. New concepts for air flow patterns in industrial halls - Calculation of the ventilation efficiency. Ventilation'94. Proceedings of the 4th International Symposium on Ventilation for Contaminant Control. V. 1. Stockholm: Sweden. p.85-91.

Drkal,F. 1990. Belüftung durch stabilisierte Lutströmung, kombiniert mit Strahlungsbänderheizung. HR. Vol.9.

Ebrahimi, I. 1968. Turbulentz in Isothemen Freistrahlen. Forsch.Ing.-Wes. Vol. 34. No. 6. pp.177-182.

Elbanna, H., S. Gahin and M.I.I. Rashed. 1983. Investigation of Two Plane Parallel Jets. AIAA Journal. Vol. 21. No.7. pp.986-991.

Elleson, J.S. and A.T. Kirkpatrick. 1997. Overview of the ASHRAE Cold Air Distribution System Design Guide. ASHRAE Transactions. Vol.107. P.1.

Elrod, H.G. 1954. Computation Charts and Theory for Rectangular and Circular

Jets. 1954. ASHVE Journal Section: Heating, Piping and Air Conditioning. March. pp.149-155.

Elterman, V.M. 1980. Ventilation of Chemical Plants. Moscow: KHIMIA.

Etheridge, D., M. Sandberg. Building Ventilation. Theory and measurements. Wiley and sons. 1996. 724pp.

Fanger, P.O., A.K. Melikov, H. Hanzawa and J.Ring. 1989. Turbulence and Draft. ASHRAE Journal. April. pp.18-23.

Fanger, P.O., A.K. Melikov, H. Hanzawa and J.Ring. 1988. Air Turbulence and Sensation of Draft. Energy and Building. Elsevier Sequoia. No. 12. pp.21-39.

Fanger P.O., Ipsen B.M., Langkilde G., Olesen B.W., Christensen N.K. and Tanabe S. 1985. Comfort limits for asymmetric thermal radiation. Energy Build. 8, 225-226.

Fanger, P.O. 1982. Thermal comfort. Malabar, FL: Robert E. Krieger, Publishing Company.

Farquharson, I.M.C. 1952. The ventilating air jet. JIHVE. V.19. pp.449-69.

Filney M. and A. Nosovitsky. 1967. Air Supply by Free Compact Jets. Proceedings of High School. Construction and Architecture. No. 2.

Finkelstein, W., K. Fitzner, Moog, W. 1973. Messungen von Raumluftgeschwindigkeiten in der Klimatechnik. Heizung-Luftung-Haustechnik 24. Nr. 2. Februar.

Fissore, A.A. and G.A. Liebeck. 1991. A simple empirical model for predicting velocity distributions and comfort in a large slot ventilated space. ASHRAE Transactions. V. 97(2). ASHRAE, Atlanta, GA.

Fleischbacker, G., W. Schneider. 1975. Experimentelle und Theoretische Untersuchungen über den Einfluß der Schwerkraft auf anisoterme, turbulente freistrahlen. Abhandl. Aus dem Aerodynamischen Institute. H. 22.

Etheridge, D., M. Sandberg. Building Ventilation. Theory and measurements. Wiley and sons. 1996. 724pp.

FlowVision. Aksenov, A.A. and Gudzovskii A.V. 1993. The software FlowVision for study of air flows, heat and mass transfer using numerical simulation. Proceedings of the 3rd Forum of the Association of Heating, Ventilation, Air-Conditioning , Heat Supply and Building Thermal Physics Engineers (AVOK), Moscow.

FLUENT. Flow Simulation Limited. Thermofluids Engineering Consultants. Sheffield. UK.

Fläkt. 1977. Svenska Fläktfabriken AB. 1977. Dirivent - Air treatment system for large premises. E1472A.

Fobelets,A.P.R. and Gagge, A.P. (1988). "Rationalization of the ET* as a measure of the enthalpy of the human environment." ASHRAE Transactions 94(1). ASHRAE, Atlanta.

Forstall, W. and A.H. Shapiro. 1950. Momentum and mass transfer in coaxial gas jets. Journal of Applied Mech. V.17, No. 4.

Franchuk, A.U. 1949. Method of Temperature Calculation at Different Room Height - Studies on Building Physics. Moscow - Leningrad: GOSSTROIZDAT. (in Russian)

Frean, D.H. and N.S. Billington. 1955. The Ventilating Air Jets. Journal of the Institution of Heating and ventilating Engineers. December.

Frings, P., J. Pfeifer. 1981. Einfluß der Raumbegrenzungsflächen auf die Geschwindigkeitsbnahme im Luftstrahl. HLH. Vol. 32. No. 2. pp. 49-61.

Fry, D. J., E.E. Adams. 1983. Confined Radial Buoyant Jet. Journal of Hydraulic Engineering. V. 109, No. 9, September.

Förthmann, E. 1934. Über turbulente Strahleasbreitung. Ing. Archiv. Vol. V. No.1. pp.42-54.

Gagge,A.P. Stolwijk,J.A.J and Nishi,Y. (1971) " An effective temperature scale based on a simple model of human physiological regulatory response." ASHRAE Transactions 77(1), ASHRAE, Atlanta.

Gartenmann, F.C. 1959. Versuche uber die Strahlausbreitung an Lüftungsdecken. Aerodynamnische Versuchsanstalt, Göttingen. Bericht: 59/A/12.

Gendrikson, V.A. and Ivanov Yu.V. 1973. Some regularities of axisymmetric jets supplied at an angle to the cross-draft. Proceedings of the Tashkent Polytechnical Institute. V. 101. pp. 184-198.

Gendrikson V.A. 1968. Mixing processes in the axisymmetric jet supplied in cross draft. Proceedings of the Acad.Sci. Estonian SSR., Tallinn

Ginevskii A.S. 1969. Theory of turbulent jets and wakes. "Mashinostroyeniye". Moscow.

Gobza, R.N. 1965. The Results of Heating Systems with Concentrated Air Supply Field Studies. Transactions of the scientific session of the institute. June 21-26, 1954. Issue 4. Concentrated Air Supply into Rooms. Leningrad: LIOT, pp. 83-106.

Gobza, R.N. 1947. Warm Air Heating with Concentrated Air Jet Supply. Moscow: STROIZDAT. Ser. 436. KTIS.

GOST. 1976. Occupied zone air. General sanitary-technical requirements. GOST 12.1.005-76. Moscow: GOSSTANDARD.

GPI PPV. 1983. State Institute for Industrial Ventilation Design (GPI Proektpromventil'yatsiya). Private communications.

Greenlaw, A.L, T.S. Hart. 1938. Air Paths from Grilles. Heating, Piping and Air Conditioning. July. pp. 452-455.

Grimitlyn, M.I. 1994. Air Distribution in Rooms. St. Petersburg.

Grimitlyn, M.I. and G.M. Pozin. 1993. Fundamentals of Optimizing Air Distribution in Ventilated Spaces. ASHRAE Transactions. V. 99(1).

Grimitlyn, M.I., A.M. Zhivov and S.Yu. Kondrashov. 1986. Specific air distribution in the premises with increased requirements for the uniformity of temperature field in the occupied zone. In: Intensification of Production and Increased Application of and Artificial Cold. Technological Institute of Refrigeration. St. Petersburg, Russia.

Grimitlyn M.I., G.A. Smirnova, V.I. Filatov, E.M. Elterman, L.E. Elterman and L.M. Brailovsky. 1983. Ventilating and Heating of Plastic Processing Shops. Leningrad: Chemia. Russia. 133 pp.

Grimitlyn, M.I. and A.M. Zhivov. 1983. Improvement of Air-conditioning Systems Effectiveness in the Operation Halls of the Automatic Telephone Exchange Stations. Proceedings of the conference: Improvements in the ventilation and air-conditioning systems effectiveness. Leningrad: LDNTP.

Grimitlyn, M.I., A.M. Zhivov, S.Yu. Kondrashov. 1981. Studies of room air change methods in typical modules of modern industrial buildings and recommendations for their selection. Studies of air distribution and air change in air conditioned industrial spaces with a special requirements to thermal parameters. Research Report T-5-81, Part II. All-Union Research Institute for Labor Protection in Leningrad.

Grimitlyn, M.I. and G.M. Pozin. 1975. Determination of Parameters for the Jets Developing in a Confined Space and Following a Blocked- or Through- Motion Pattern. Nauchniye raboti institutov ohranu truda VTsSPS. V.93.

Grimitlyn, M.I. 1973. Principles of air distribution in ventilated spaces. D.Sc. Thesis. Leningrad. VVISKU.

Grimitlyn, M. 1970. Zurluftverteilung in Raumen. Luft- und Kaltetechnik. Nr. 5.

Grimitlyn, M.I. 1969. Modeling and Designing of Air Distribution Devices. Filtration of Industrial Exhausts and Problems of Air Distribution. Institut ohranu truda VtsSPS. Leningrad.

Grimitlyn, M.I. 1965. Theoretical Fundamentals of Development and Ventilating Jets Design. Theory and ventilating jets design. Leningrad.

Grimitlyn, M.I. 1960. Air Distribution through perforated ducts. Leningrad.

Gunes, I.L. and I.L. Leshinskaya. 1977. Evaluation of the required air change rate with air supply through ceiling mounted air diffusers. The Issues of Sanitary Technique Systems Design and Installation. Leningrad: VNIIGS.

Gunes, I.L., I.L.Leshinskaya. 1974. Selection and Design of Air Diffusers for Combined Heating and Ventilating Systems. Transactions of VNIIGS: "Design and Installation of Sanitary-Technique Systems". V. 38. pp. 13-31. Leningrad: VNIIGS.

Gunes, I.L. 1970. Concentrated air supply in Industrial Spaces. Transactions of VNIIGS: "Design and Installation of Sanitary-Technique Systems". V. 28. pp. 3-15. Leningrad: VNIIGS.

Gunes, I.L. 1948. Ventilation of Rooms with Surplus Heat and Concentrated Air Supply. Ph.D. thesis. Leningrad: LISI.

Gutmark, E. and I. Wignanski. 1976. the planar turbulent jet. Journal of Fluid Mech. V. 73. p.3.

Görtler, H. 1942. Berechnung von Aufgaben der Freien Turbulentz auf Grund eines neunen Näherungsansatzes. Z. angewandte Math.Mech. Vol. 22. No.3. pp.245-254.

Hanel, B. 1994. Raumluftströmung. C.F. Müller Verlag. 178pp.

Hanel, B., B. Weidemann. 1991. Beitrag zur Berechnung nichtisothermer Freistrahlen.

Hanel, B. and E. Rechter. 1979. Das Verhalten von Freistrahlen in verschiedenen Reynolds-Zahlbereichen. Luft-und Käaltetechnik. Bd. 15. Nr. 1. pp.12-14.

Harris, P.R. 1967. The densimetric flows caused by the discharge of heated two-dimensional jets beneath a free surface. Ph.D. Thesis. University of Bristol. Department of Civil Eng.

Hassani, V., T.-G. Malmstrom, A.T. Kirkpatrick. 1993. Indoor Thermal Environment of Cold Air Distribution Systems. ASHRAE Transactions. V. 99 (1). pp. 1359-1365.

Hauesmann, K. 1966. Eigenschaften turbulenter Strahlenbündel. Chemie-Ing.-Techn. Vol. 38. No. 3. pp.293-297.

Hegge Zijnen, B.G. van der. 1958a. Measurement of the velocity distribution in a plane turbulent jet of air. Applied Sci. Res. A7. pp. 256-276.

Hegge Zijnen, B.G. van der. 1958b. Measurement of the distribution of heat and matter in a plane turbulent jet of air. Applied Sci. Res. A7. pp. 277-292.

Heins, T. 1996. Brandsimulation mit Mehrraum-Zonenmodellen. TAB. Vol. 94. No.4. pp.75-81.

Heiselberg P., S. Murakami and C.-A. Roulet. 1996. Annex 26. Air Flow Patterns in Large Enclosures. Ventilation of Large Spaces in Buildings. Part 3. Analysis and Prediction Techniques. IEA.

Helander, L., S.M. Yen, and W. Tripp. 1957. ASHRAE Research Report No. 1601. Outlet characteristics that affect the down throw of heated air jets. ASHVE

Transactions. V. 63:255.

Helander, L., S.M. Yen, and L.B. Knee. 1954. ASHVE Research Report No. 1511. Characteristics of downward jets of heated air from a vertical delivery discharge unit heater. ASHVE Transactions. V. 60:359.

Helander, L., S.M. Yen, and R.E. Crank. 1953. ASHVE Research Report No. 1475. Maximum downward travel of heated jets from standard long radius ASME nozzles. ASHVE Transactions. V. 59. pp. 241-259.

Helander, L. C.V. Jakowatz. 1948. Downward Projection of Heated Air. ASHVE Research Report No. 1327. ASHVE Transactions 59:71.

Heskestad, G. 1966. Hot-wire measurements in a radial turbulent jet. Transactions ASME. Journal of Applied Mechanics. No. 33. pp. 417-424

Heskestad, G. 1965. Hot-wire measurements in a plane turbulent jet. Transactions ASME. Journal of Applied Mechanics.

Holmes, M.J. and C. Caygill. 1973a. Air Movement in Rooms With Low Supply Air Flow Rates. The Heating and Ventilating Research Association. Laboratory Report No. 83. Bracknell. UK.

Holmes, M.J. and E. Sachariewicz 1973b. The Effect of Ceiling Beams and Light Fittings on Ventilating Jets. The Heating & Ventilating Research Association. Laboratory Report No. 79. The Heating and Ventilating Research Association. Bracknell. UK.

Hosni, M.H.; A-El Hassan, B. Mohamed, P.L. Miller. Airflow Characteristics of Jet Expansion for Non-isothermal Flow Conditions . ASHRAE Transactions. 1996. V. 102 (2).

Houghten, F.C., C.Gilbert and E.Witkowski. 1938. Draft temperatures and velocities in relation to skin temperatures and feelings of warmth. ASHVE Transactions. 44.

Howarth A.T. 1980. Temperature distribution and air movement in rooms heated with a convective heat source. Ph.D. Thesis. University of Manchester. 226 pp.

Hudenko, B.G. 1966. About the calculation of turbulent jet mixing based on the principle of superimposing. In: "Studies of Heat- a Massexchange in Production Processes and Equipment" Minsk: "Science and Technology".

Hwang, C.L.; Tillman, F.A.; Lin, M.J. " Optimal Design of an Air Jet for Spot Cooling", ASHRAE Transactions. 1984. V. 90 (1B). p 476-498.

Inard C., H. Bouia and P. Dalicieux. 1996. Prediction of air temperature distribution in buildings using a zonal model. Energy and Buildings.

Inard C. 1988. Contribution á l'étude du couplage thermique entre une source de chaleur et un local. Ph.D. Thesis. INSA de Lyon, France, 449 pp.

ISO 7933 (1989): "Hot environments - Analytical determination and interpretation of thermal stress using calculation of required sweat rate." International Organization for Standardization, Geneva, Switzerland.

ISO 8996 (1990). " Ergonomics - Determination of metabolic heat production." International Organization for Standardization, Geneva, Switzerland.

ISO TR 11079 (1993) "Evaluation of cold environments - Determination of required clothing insulation IREQ". International Organization for Standardization, Geneva, Switzerland.

ISO/DIS 9920 (1992). "Ergonomics - Estimation of the thermal insulation and evaporative resistance of a clothing ensemble" International Organization for Standardization, Geneva, Switzerland.

ISO 7730. 1984 (Revised version 1993). "Moderate thermal environments - Determination of the PMV and PPD indices and specification of the conditions for thermal comfort". International Standards Organization. Geneva.

Jackman, P.J. 1973. Air Movement in rooms with sill-mounted diffusers (including supplement A and B). BSRIA Laboratory Report No. 81. Bracknell, UK.

Jackman, P.J. 1971. Air Movement in rooms with sill-mounted grilles - A Design procedure. BSRIA Laboratory Report No. 71. Bracknell, UK.

Jackman, P.J. 1970. Air Movement in rooms with side-wall mounted grilles. BSRIA Laboratory Report No. 81. Bracknell, UK.

Jendroßek, J.U. 1992. Kanalloses Luftführungssystem. TAB. Vol.690. No.1. pp.45-49.

Karimipanah, T. 1996. Turbulent jets in confined spaces. Ph.D. thesis. Royal Institute of Technology. Center for Built Environment. Gävle. Sweden.

Karimipanah, T. 1996. Behavior of Jets in Ventilated Enclosures. Proceedings of the 5th International Conference on Air Distribution in Rooms. ROOMVENT'96. Vol. 1. July 1996. Yokohama, Japan.

Katz, P. 1973. Der Coanda-Effect. Gesudheits-Ingenieur. No. 6. pp.169-192.

Keagy, W.R. and A.E. Weller. 1949. Proceedings of the Heat Transfer and Fluid Mechanics Institute. University of California. ASME. pp.89-98.

Kirkpatrick, A., T. Malmstrom, K. Knappmiller, D. Hittle, P. Miller, V. Hassani and R. Anderson. 1991. Use of Low Temperature Air for Cooling of Buildings. Proceedings of the 1991 Building Simulation Conference.

Knaak, R. 1957. Velocities and Temperatures on Axis of Downward Heated Jet From 4-Inch Long-Radius ASME Nozzle. ASHVE Research Report No. 1619. ASHVE Transactions. V. 63. pp. 527-538.

Koestel, A. 1957. Jet Velocities from Radial Flow Outlets. ASHVE Transactions. V. 63. p 505-526

Koestel, A. Austin, J.B. Jr. 1956. Air Velocities in Two Parallel Ventilating Jets. ASHVE Transactions. V. 62. pp. 425-436.

Koestel, A. 1955. Paths of Horizontally Projected Heated and Chilled Air Jets", ASHVE Transactions. V. 61. p 213-232.

Koestel, A., Tuve, G.L. 1955. Performance and Evaluation of Room Air Distribution Systems. ASHVE Transactions. V. 61. pp. 533-550.

Koestel, A. 1955. Paths of Horizontally Projected Heated and Chilled Air Jets. Heating, Piping and Air Conditioning. July. pp. 221-226.

Koestel, A. 1954. Computing Temperatures and Velocities in Vertical Jets of Hot or Cold Air. ASHVE Transactions. V. 60. pp. 385-410.

Koestel, A, and C.-Y. Young. 1951. The Control of Air Streams from a Long Slot. Heating, Piping and Air Conditioning. July. pp. 111-115.

Koestel, A., P. Hermann and G.L. Tuve. 1950. Comparative study of ventilating jets from various types of outlets. ASHVE Research Report No. 1404. ASHVE Transactions 56: 459.

Koestel, A., P. Hermann and G.L. Tuve. 1949. Air Streams from Perforated Panels. ASHVE Journal Section: Heating, Piping and Air Conditioning. July. pp.107-113.

Kotsovinos, N.E. and E.L. List. 1977. Plane turbulent buoyant jets. Part 1. Integral properties. Journal of Fluid Mechanics. 81. Pt. 1.

Kotsovinos, N.E. 1977. A study of the entrainment and turbulence in a plane buoyant jet. Journal of Fluid Mechanics. 81. Pt. 1. pp.25-62.

Kraemer, K. 1971. Die Potentialströmung in der Umgebung von Freistrahlen.

Zeitschrift für Flugwissenschaften. Vol. 19. No. 3. pp.93-104.

Kristensson, J.A., O.A. Lindqvist. 1993. Displacement ventilation systems in industrial buildings. ASHRAE Transactions. V. 99(1). ASHRAE. Atlanta.

Kudryavtsev, E.V. 1948. Ventilation of Large Industrial and Commercial Buildings. Proceedings of the Acad. Sci. USSR, OTN, #3. (in Russian)

Kudryavtsev, E.V. 1950. Simulation of Ventilation Systems. Moscow-Leningrad: STROIIZDAT. (in Russian)

Kuzmina, L.V., A.M. Kruglikova, A.S. Gus'kov. 1986. Contaminant distribution in industrial hall with spiral vortex ventilation. Occupational Safety in Industry. Transactions of the All-Union Research Institutes for Labor Protection. Moscow: PROFIZDAT.

Kuzmina, L.V. 1968. On the Interaction of Parallel Supply Jets. Collected Papers of the Occupational Safety Institutes under the VTsPS. Moscow: PROFIZDAT.

Landau L.D. and Lifshits E.M. 1953. Mechanics of continuous medium. Gosidat.

Landis F. and Shapiro A.H. 1951. The turbulent mixing of coaxial gas jets. Heat Transfer and Fluid Mechanics Institute. Preprints and Papers. California. Stanford University Press.

Langkilde, G., L. Gunnarsen and N. Mortensen. 1985. "Comfort limits during infrared radiant heating of industrial spaces" CLIMA 2000, Copenhagen.

Laret L. 1980. Contribution au développement de modéles mathématiques du comportement thermique transitoire de structures d'habitation. Ph.D. Thesis. University of Liège. 113pp.

Laurikainen, J. 1995. Displacement Ventilation System Design Method. Seminar presentations. Part 2. INVENT Report 46. Helsinki: FIMET.

Lebrun, J. Etudes expérimentales des régimes transitoires en chambres climatiques: ajustment des méthodes de calcul. Journées Bilan et Perspectives en Génie. INSA de Lyon. 18pp.

Li, Z.H. 1994. Fundamental Studies on Ventilation for improving Thermal Comfort and IAQ. Ph.D. Thesis. University of Illinois. Urbana, IL.

Li Z., L.L.Christianson, J.S. Zhang, A. Zhivov. 1994. Cold Air Jets Systems Effects on the Occupied Regions. Proceedings of the Fourth International Conference on Air Distribution in Rooms "ROOMVENT'94". Vol.2. Krakow, Poland.

Li, Z.H., J.S. Zhang, A.M. Zhivov and L.L.Christianson. 1993. Characteristics of Diffuser Air Jets and Airflow in the Occupied Region of Mechanically Ventilated Rooms - A literature review. Transactions of ASHRAE. V.99(1).

Liddament, M.W. 1993. A review of ventilation efficiency. Technical Note AIVC 39. Air Infiltration and Ventilation Center. Warwick, UK. December 1991.

Liddament, M.W. 1991. A review of building air flow simulation. Technical Note AIVC 33. Air Infiltration and Ventilation Center. Warwick, UK. March 1991.

Liddament, M.W. 1987. A review and bibliography of ventilation effectiveness - definitions, measurements, design and calculation. Technical Note AIVC 21. Air Infiltration and Ventilation Center. Warwick, UK. July 1987.

Lin, Y.F. and M.J. Sheu. 1991. Interaction of Parallel Turbulent Plane Jets. AIAA Journal. Vol. 29. No. 9. pp.1372-1373.

Linke, W. 1957. Strömungsvorgänge in zwangsbelüfteten Räumen. VDI-Berichte. Bd. 21. pp.29-39.

Linke, W. 1966. Eigenschaften der Strahlüftung (Aspects of jet ventilation). Kältetechnik Klimatechnik. Vol. 18. No. 3. BSRAE Translation 103.

Livchak, A. J. Laurikainen, J. Pekkinen. 1993. A computer model for air distribution with non-isothermal axial jets. Proceedings of International Conference "INDOOR AIR' 93". Vol. 5.

Loitzanskiy, L.G. 1973. Mechanics of Fluid and Gas. Moscow: Nauka.

LVIS. 1996. Ilmastointi. LVIS -2000. Painopaikka: Kausalan Kirjapaino Oy.

Lyakhovsky, D. and S. Syrkin. 1939. Aerodynamics of the Torch Flowing out into the Medium of Another Density. Journal of Technical Physics. V.9 (9).

Madison, R. Elliot, W.R. 1946. Throw of Air from Slots and Jets. Heating, Piping and Air Conditioning. November. pp. 108-109.

Maksimov, G.A. and V.V. Der'ugin. 1972. Air Movement in Heated and Ventilated Spaces. Leningrad.

Malmstrom, T.-G. 1996. Archimedes Number and Jet Similarity. Proceedings of the 5th International Conference on Air Distribution in Rooms. ROOMVENT'96. Vol. 1. July 1996. Yokohama, Japan.

Malmström, T.-G. and V.Hassani. 1992. Use of Constant Momentum for Supply of Cold Air. Proceedings of the 3rd International Conference on Air Distribution in Rooms. ROOMVENT'92. Vol.3. Aalborg, Denmark.

Masuch, J. 1992. The influence of Air Outlet Directions on the Stability of Air Flow Patterns in Large Halls. Proceedings of the 3rd International Conference on Air Distribution in Rooms. ROOMVENT'92. Vol.1. Aalborg, Denmark. pp.539-552.

McRee, D.I and H.L. Moses. 1967. The Effect of Aspect Ratio and Offset on Nozzle Flow and Jet Reattachment. Advances in Fluidics. The 1967 Fluidics Symposium, ASME.

Melik-Arakelyan, A.T. 1978. Studies on Air-Change in Air-Conditioned Spaces. Computer Rooms in Computation Centers. Ph.D. theses. Moscow: MISI. (in Russian)

Melikov, A. and G. Zhou. 1996. Air Movement at the Neck of the Human Body. Proceedings of the 7th International Conference on Indoor Air Quality and Climate. INDOOR AIR'96. Vol. 1. July 1996. Nagoya, Japan.

Merrit, G. and Redinger, G. 1973. Measurements of Heat and Impulse Transfer Coefficient in Turbulent Stratified Flow. Journal of rocket technology and astronautics. No. 1 1.

Meshalin, V.S. 1974. About Turbulent Stress in Impinging Jets. In: "Mechanics of Gases and Fluids". Moscow.

Miller, P.L. Jr. 1991. Diffuser Selection for Cold Air Distribution. ASHRAE Journal, September, 1991. V. 33. pp. 32-36.

Miller, P.L. 1969. Room Air Distribution with Air Distributing Ceiling - Part 1. ASHRAE Research Report RP-55 No. 2100. ASHRAE Transactions. V.75. pp.118-131.

Miller, P.L. 1971. Room Air Distribution Performance of Four Selected Outlets", ASHRAE Research Report No. 2210 RP-88 Washington D.C. August.

Miller, D. R. and E.W. Comings. 1960. Force Momentum Fields in a Dual-Jet Flow. Journal of Fluid Mechanics. V. 7(2). p. 237.

Mitkalinniy, V. 1961. Gas jets flow in ovens. Moscow: METALLURGISDAT.

Mizuchina, T. 1982. An experimental study of vertical turbulent jet with negative buoyancy. Wärme- u. Stoffübertragung. 16. No. 1.

Moog, W. 1978. Ähnlichkeitstheoretische Überlegungen bei Raumstromüngen. (Theoretical considerations of similarity for airflow within the open space). KI 11/1978. Teil 4.3. pp. 267-270.

Munk, W.H. and E.R. Anderson. 1948. Notes on the Theory of the Termocline.

Journal of Marine Research. Vol.1.

Murakami, S. 1992. Proceedings of the International Symposium on Room Air Convection and Ventilation Effectiveness. Tokyo, Japan.

Murakami, S;, Kato, S.; Nakagawa, H. Numerical Prediction of Horizontal Non-isothermal 3-D Jet in Room Based on the κ-ϵ Model. ASHRAE Transactions. 1991. V. 97. p 38-48.

Müllejans, H. 1966. Über die Ähnlichkeit der nicht-isothermen Strömung und den Wandurbergang in Räumen mit Strahlüftung. Forschungsber des Landes NRW. No. 1656. Westdeutscher Verlag. Köln. The similarity between non-isothermal flow and heat transfer in mechanically ventilated rooms. BSRIA Translation 202, Bracknell, UK.

Müller , H.J. 1977. Einfluß der Geometrischen Verhältnisse auf die Raumunströmung bei der Strahllüftung. Luft-und Kältetechnik. No. 6.

Müller , H.J. 1975. Beitrag zum Thema der Luftführung in Tierproduktions-anlagen. Luft-und Kältetechnik. No. 2.

Nagasawa, Y., M. Nitadori, S Matsui. 1990. Characteristics of spiral vortex flow and its application to control indoor air quality. Roomvent' 90. Proceedings of the International Conference on Air Distribution in Ventilated Rooms. Oslo, Norway.

Nelson, D.W. and D.J. Stewart. 1938. Air Distribution from Side Wall Outlets. ASHVE Research Report No. 1076. ASHVE Transactions 44: 77.

Nielsen, P.V. 1995. Lecture Notes on Mixing Ventilation. University of Aalborg. Department of Building Technology. Denmark. 70pp.

Nielsen, P.V. 1994. Air Distribution in Rooms. - Research and Design Methods. Proceedings of the Forth International Conference on Air Distribution in Rooms. ROOMVENT'94. Krakow, Poland. V. 1.

Nielsen, P.V. 1992. Velocity distribution in the flow from a wall-mounted diffuser in rooms with displacement ventilation. ROOMVENT'92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 3. Aalborg.

Nielsen, P.V. and Åke T.A. Moller. 1988. Measurements on Buoyant Jet Flows from a Ceiling-mounted Slot Diffuser. The 3rd Seminar on Applications of Fluid Mechanics in Environmental Protection - 88. Selesian Technical University, Gliwice, Poland.

Nielsen, P.V. and Åke T.A. Moller. 1987. Measurement on Buoyant Wall Jet Flows in Air Conditioned Rooms. RoomVent 87, Stockholm.

Nielsen, P.V. 1981. Luftstrømning i Ventilerede Arbejdslokaler (ventilation of working areas). SBI-Rapport 128. Statens Byggefoskningsinstitut.

Nielsen, P.V. 1980. the influence of Ceiling-Mounted Obstacles on the Airflow Pattern in Air-Conditioned Rooms at Different Heat Loads. Building Service Engineering. research and Technology. Vol.1 No.4.

Nosovitsky, A. 1968. Single jet and system of multiple jets development between the walls. Proceedings of High School (Isvestia VUZOV). Construction and Architecture. No. 5.

Nosovitsky, A. and Posokhin V. 1966. Inclined Fountains of Heated and Chilled Air, Created by Nonisothermal Jets Supply. Heat-, Gas Supply and Ventilation. Kiev: Budivelnik.

Nottage, H.B. , J.G. Slaby, and W.P. Gojza. 1952. Exploration of a chilled jet. ASHVE Research Report No. 1459. ASHVE Transactions 58:357.

Nottage, H.B. 1951. Ventilation Jets in Room Air Distribution. Ph. D. Thesis. Case Institute of Technology.

Ogilvie, J.R. and E.M. Barber. 1989. Jet Momentum Number: An Index of Air Velocity at Floor Level. Building Systems: Room Air and Air Contaminant Distribution.

ASHRAE.

Olesen, B.W., A.M. Zhivov. 1994. Evaluation of thermal environment in industrial work spaces. ASHRAE Transactions. V. 100(2). ASHRAE. Atlanta.

Olesen, B.W., M.Scholer and P.O.Fanger. 1979. Discomfort caused by vertical air temperature differences. INDOOR CLIMATE. Danish Research Institute. Copenhagen.

Olesen B.W. 1977. Thermal comfort requirements for floors. International Institute of Refrigeration, Conf. Proc. Commissions B1, B2, E1, Belgrade. 4, 307-313.

Omelchuk V. 1966. Laws of Nonisothermal Jet Development, Banded by Buoyancy Forces. Water Supply and Sanitary Technique. No. 2.

Oosthuizen, P.H. 1983. An Experimental Study of Low Reynolds Number Circular Jet Flow. ASME Preprint 83-FE-36.

OSHA. 1989. Air contaminants - Permissible Exposure Limits. (Title 29 Code of Federal Regulations. Part 1910.1000). U.S. Department of Labor. OSHA 3112.

Padmanabham, G. and B.H.L. Gowda. 1991. Mean and Turbulence Characteristics of a Class of Three-Dimensional Wall Jets - Part 2: Turbulence Characteristics. Journal of Fluid Engineering. December. Vol.113. pp.629-634.

Pai, S.I. 1949. Two Dimensional Jet Mixing of a Compressible Fluid. Journal of Aeronautical Sciences. August. pp. 463-469.

Pechatnikov, M.Z. 1967. Principles of jet flows in cold rooms. Ph.D. Thesis. Leningrad: LTIHP. Russia.

PHOENIX. Cham Ltd. London, UK.

Posokhin, V. 1966. Interaction of Supply Air Jets. Water Supply and Sanitary Technique. No. 7. Moscow.

Poz, M.Y. 1994. Experimental Investigation of Planar Turbulent Jet in Enclosures. ASHRAE Transactions. V. 100 (1). pp. 1182-1194.

Poz, M.Y. 1993. Theoretical Investigation and Practical Applications of Non-Isothermic Jets for the Purpose of Ventilating Rooms. ASHRAE Transactions. V. 99 (1). pp. 950-959.

Pozin, G.M. 1993. Determination of ventilating effectiveness in mechanically ventilated spaces. Proceedings of the 6th International Conference on Indoor Air Quality (IAQ'93). Helsinki, Finland.

Pozin G.M. 1992. Mathematical Modelling of Heat and Air Processes in mechanically Ventilated Spaces. ROOMVENT'92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 1. Aalborg.

Pozin G.M. 1975. Principles of analytical evaluation of the air exchange efficiency coefficient. In: Research on Different Methods of the Air Exchange in Premises of Industrial Buildings. Moscow: Profizdat.

Prandl, L. 1942. Bemerkungen zur Theorie der freien Turbulenz. Z. für Angewandte Math., Mech. Vol. 22. No. 5. pp.241-243.

Priest, J.B. 1996. Ph.D. Thesis. University of Illinois. Urbana.

Rajartnam, N. 1976. Turbulent Jets. Elsvier, Amsterdam.

Rakoczy, T. 1977. Dimensionierung von Düsen als Luftdurchlässe für Raumlufttechnische Anlagen", HLH 28. May. pp. 173-175.

Regenscheit, B. 1981. Isotherme Luftstrahlen. Klima+Kälteingenieur. Verlag C.F.Müller. Karlsruhe. 205 pp.

Regenscheit, B. 1976. Einfluß der Reynoldszahl auf die Geschwindigkeitsabnahme turbulenter Freistrahlen. HLH. Vol.27., No.4. pp.122-126.

Regenscheit, B. 1975. Strahlgesetze und Raumströmung. Klima-Kälte-technik. Vol.6. pp.147-150.

Regenscheit, B. 1974. Der Einfluß von Raumwänden auf die Strahlgesetze der Lufttechnik The influence of the room enclosing walls on the laws of jets in ventilation technology). Ki. No.1. pp.9-16.

Regenscheit, B. 1971. Die Berechnung von radial strömenden Frei- und Wandstrahlen, sowie von Rechteckstrahlen. H. Krantz-Luttechnik. Aahen-Richterich. Gesundheits-Ingenieur. No. 7. pp.193-201.

Regenscheit, B. 1970. Die Archmedes-Zhal-Kenzah zur beurteilung von Raumstromungen. Gesund. Ing. Nr. 91(6).

Regenscheit, B. 1964. Modellversuche zur Erforschung der Raumströmung in belüfteten Räumen. Staub. Januar. No. 1. pp. 14-20.

Regenscheit, B. 1962. Das Wierderlengen eines ebenen Strahles an eine Wand. Hausbericht H. Krantz Lufttechnik. E. Bericht 2376.

Regenscheit, B. 1959. Die Luftbewegung in Klimatisierten Räumen (Movement of air in air-conditioned rooms). Kältetechnik. Heft 1. pp.3-11.

Regenscheit, B. Strahlen in Begrenzten Räumen. 27pp. From the B. Regenscheit archive at Essen University. Courtesy of Prof. Fritz Steimle.

Reichardt, H. 1944. Impuls und Warmeaustausch in freien Turbulenz. ZAMM. Bd.24, No.5,6.

Reichardt, H. 1942. Gesetzmäßigkeiten der freien Turbulentz. VDI - Forschungsheft 414.

REPUS. Technical Information and planing guide.

Rietschil, H. 1930. Leitfaden der Hier - und Luftungstechnik. Neunte Auflage.

Rodahl, E. 1977. The Point of Separation for Cold Jets Flowing along the Ceiling. CLIMA 2000. Belgrade, Yugoslavia.

Rodi, W. Turbulent Buoyant Jets and Plumes. 1982. Pergamon Press.

Rouse, H. C.S. Yih and H.W. Humphreys. 1952. Gravitational convection from a boundary source. Tellus 4. pp. 201-210.

Rozenberg, V.N. 1949. In: Lyakhovsky, D.N. 1949. Aerodynamics of the Jet and Torch Processes. TsKTI. Book 12. Heat transfer and aerodynamics. MASHGIZ.

Rubel, A. 1980. Computations of Jet Impingement on a Flat Surface. AIAA Journal. V. 18. No. 2. pp. 168-175.

Ruden, P. 1933. Turbulente Ausbreitungsvorgänge im Freistrahl. Naturwissenschaften. Bd. 21. No. 21-23.

Rydberg, J. and P. Norback. 1949. Air Distribution and Draft. ASHVE Research Report No. 1362. ASHVE Transactions, V. 55.

Rydberg, J. 1962. Luftinblåsning genom perforerade tak. VVS nr 6. 1962.

Rydberg, J. 1962. Maximala ventilationsluftmängder för olika insblåsninganodningar. VVS nr 12.

Rymkevich A.A. 1990. Optimization of General Ventilation and Air-Conditioning Systems Operation. Moscow: Stroizdat. (in Russian)

Sadovskaya, N.N. 1954. Air Circulation in Spaces with Concentrated Air Jet Supply. Proceedings of the conference. Concentrated air supply. Issue 4. Leningrad: LIOT.

Sadovskaya, N.N. 1950. Studies of Air Circulation in the Spaces with Heating and Ventilating by Concentrated Air Jet Supply. Ph.D. thesis. Leningrad VNIIOT VtsSPS.

Sandberg, M.B. Wirén and L. Claesson. 1992. Attachment of a cold plane jet to the ceiling - length of recirculation region and separation distance. Proceedings ROOMVENT'92. Aalborg, Denmark. V.1.

Sandberg, M. and Skåret, E. 1985. Air change and ventilation efficiency - new aids for designers. Swedish Institute for Building Research.

Sandberg, M. 1981. What is ventilation efficiency? Building Environment. Vol. 16. No. 2.

Sato, K., H. Osuka, J. Inone. 1981. Maximum penetration distance of a vertical buoyant jet. Int. Chem.Engin. V. 21, No. 3. pp. 435-442.

Sato, H. and F. Sakao. 1964. An Experimental Investigation of the Instability of Two-Dimensional Jet at Low Reynolds number. J.Fluid Mech. Vol. 20. P.2. pp.337-352.

Sawyer, R.A. 1963. Two-dimensional reattaching jet flows including the effects of curvature on entrainment. Journal of Fluid Mechanics. No. 17. pp.481-498.

Sawyer, R.A. 1960. The Flow due to a Two-Dimensional Jet Issuing Parallel to a Flat Plate. Journal of Fluid Mechanics. No. 9. Part 4. pp.543-561.

Schlanzke, G. 1975. Die Turbulente Vermischung in Freistrahlen unter Berücksichtigung von Dichteunterschieden. Luft-und Kaltetechnik. No. 2 pp. 71-75.

Schlichting, H. 1969. Boundary Layer Theory. Handbook of Fluid Dynamics. N.Y.

Schlünder, U. 1971. Über die Ausbreitung turbulenter Freistrahlen. Z. Flugwiss. Vol. 19. No. 3. pp. 108-113.

Schneider, W. 1975. Uber den Einfluß der Schwerkraft auf Anisotherme Turbulente Freistrahlen. - Abhandl. aus dem Aerodynamischen Institute der Rhein-Westf. Techn. Hochschule Aahen. H.22.

Schobesberger, R. 1985. Gezielte Zuluftverteilung in Industriehallen mit dem Hoval-Dralluftverteiler (Directed supply air distribution in large manufacturing shops with the HOVAL-Screw-Type air disffusers). Klima - Kälte - Heizung. 1/1985. Teil 6. 1408.

Schramek, E.R. Tachenbuch für Heizung und Klimatechnik. Recnagel - Sprenger - Hönmann. R. Olenbourg Verlag München Wien.

Schultz-Hausmann, F.K. 1985. Wechselwirkung ebener Freistrahlen mit der Umgebung. VDI-Verlag GmbH. Düsseldorf. 135 pp.

Schwarz, W.H. and W.P. Cosart. 1960. The two-dimensional turbulent wall jet. Journal of Fluid Mechanics. V.10. pp. 481-495.

Schwarz, W. 1995. Auslegung von Düsenstrahlbündeln. Ki Luft- und Kältetechnick. No.11. pp.518-521.

Schwenke, H. 1976. Zur Luftstromung in Raumen mit Wurfluftung. Luft und Kältetechnik. No. 1, pp. 11-14.

Schwenke, H. 1975. Uber das Verhalten ebener horizontaler Zuluftstrahlen im begrenzten Raum. Luft- und Kalteetechnik. No. 4.

Schädlich, S. 1993. der Einfluss Verschiedener Luftdurchlassgeometrien auf das Freistrahlverhalten. D.-Ing. thesis. Deutscher Kälte- und Klimatechnischer Vereins. No. 43.

Seban, R.A., M.M. Behnia, K.E. Abrea. 1978. Temperatures in a heated air jets discharged downward. International Journal of Heat and Mass Transfer. V. 21. pp. 1453-1458.

Sefker, T. 1989. Verallgemeinerte Darstellung des Verhaltens Isothermer Freistrahlen. D.-Ing. thesis. Deutscher Kälte- und Klimatechnischer Vereins. No. 27.

Seliverstov, A.N. 1934. Ventilation of Chemical Industry Plants. ONTI. Moscow: GOSSTRIIZDAT. (in Russian)

Seliverstov, A.N. 1932. Room Ventilation in Plants and Factories. ONTI. Moscow:

GOSSTRIIZDAT. (in Russian)

Sforza, P.M. and G. Herbst. 1970. A study of three dimensional incompressible turbulent wall jets. AIAA Journal. V. 8. pp. 276-293.

Shakerin, S, and P.L. Miller. 1996. Experimental Study of Vortex Diffusers. ASHRAE Transactions. V. 102 (2). ASHRAE, Atlanta, GA.

Shepelev I. Aerodynamics of Air Flows in Rooms. M., Stroyizdat. 1978. 145 pp.

Shepelev, I.A. and N.A. Gelman. 1966. Universal equations for velocity and temperature computation along the ventilation jets, supplied from rectangular outlets. Water- Supply and Sanitary Technique. No.7. (in Russian).

Shepelev, I.A. and Tarnopolsky. 1965. turbulent air jet spreading in the confined space. Heat-, Gas Supply and Ventilation. Proceedings of the conference. Kiev: Budivelnik.

Shepelev, I.A. 1962. New method for aeration calculation. Water Supply and Sanitary Technique Journal, No. L .

Shepelev I.A. 1961. Supply Ventilation Jets and Air Fountains. Proceedings of the Academy of Construction and Architecture of the USSR. No. 4.

Shilkrot, Eu. O., A.A. Naumov, A.B. Skachkov and A.M. Zhivov. 1996a. Universal equation for ventilation effectiveness evaluation in rooms with excessive heat gains. Proceedings of the 5th ABOK Congress. October 1996b. Moscow, Russia.

Shilkrot, Eu. O., A.M. Zhivov. 1996b. Zonal Model for Displacement Ventilation Design. 5th International Conference on Air Distribution in Rooms. ROOMVENT'96. Vol. 2. July 1996. Yokohama, Japan.

Shilkrot, Eu.O. 1993. Determination of design loads on room heating and ventilation systems using the method of zone-by-zone balances. ASHRAE Transactions. 99(1). ASHRAE. Atlanta.

Shilkrot, Eu.O. and A.M. Zhivov. 1992. Room ventilation with designed temperature stratification. ROOMVENT'92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 1. Aalborg.

Shilkrot, Eu.O. 1977. Natural ventilation of single-storey industrial buildings with a significant heat loads. Ph.D. Thesis. Moscow: MISI.

Shilkrot, Eu.O. 1976. Calculation of Required aeration air exchange rate by the zone-by-zone heat balances method. Proceedings of the Conference. Novelties in industrial ventilation systems design and operation. Leningrad: LDNTP. (in Russian)

Sichov, A.T. and G.Ya. Volov. 1981. Method of Confined Jets Calculation. Proceedings of Acad. Sci. BSSR. Ser. Phys.-Energ. Sci. No.1.

Sigalla, A. 1958. Experimental Data on Turbulent Wall Jets: A Correlation of Existing Data. Aircraft Engineering. May. pp. 131-134.

Skistad, H. I995. Industriventilasjon. Innblasning og avsug. Hefte 1. Scarland press AS.

Skistad, H. I994. Displacement Ventilation. Research Studies Press, John Willey & Sons, Ltd., West Sussex, UK.

Skåret, E. 1993. Advanced design of ventilation systems. Ventilation models. Lecture Series. Von Karman Institute for Fluid Dynamics.

Skåret E. 1986. Industrial Ventilation - Model Tests and General Development in Norway and Scandinavia. Proceedings of "Ventilation'85" Conference. Elsevier Science Publishers BV. Amsterdam.

SNiP 2.04.05-91. 1994. Building Norms and Regulations. Heating, Ventilation and Air Conditioning. Moscow: MINSTROY. (In Russian).

Sodec, F. 1986. Air Distribution Systems. Report No. 3554E. Krantz GmbH, Germany.

Sorokin, N.S. 1965. Ventilating, Heating and Air-Conditioning of Textile Factories. Moscow: Light Industry Publishing House.

Stein M. 1953. Studies on Air Jets. Journal of the Institute of Heating and Ventilating Engineers. V.21.

Stark S.V. 1950. Submerged turbulent flows mixing. Thesis (Ph.D.) Moscow Institute of Steel and Alloys (MISiS).

Stoecker, W.F. 1968. Principles for Air Conditioning Practice. Industrial Press Inc.

Ströder, R. 1991. RLT-Analagen für Fertigunshallen. TAB. Vol.2. pp.125-130.

Söllner, G., K. Klinkenberg. 1972. Leuchen als Störkörper im Luftstrom. HLH. Vol. 23. No.4.

Takemasa, Y., S.Togari and Y. Arai. 1996. Application of an unsteady-state Model for predicting vertical temperature distribution to an existing atrium. ASHRAE Transactions. 102(1). ASHRAE. Atlanta.

Taliev V.N. 1979. Aerodynamics of Ventilation. Moscow: STROYIZDAT. 295 pp.

Taliev, V. 1969. Laws of Free Non-isothermal Axially Symmetric Jet Development. Water Supply and Sanitary Technique. ("Vodosnabgeniye i Sanitarnaya Tekhnika"). No. 1.

Tarnopolsky, M. D. 1994. Approved Calculation of Air Distribution in an Auditorium. ASHRAE Transactions. V. 100 (1). pp. 1195-1209.

Tarnopolskiy, M.D. 1966. Concentrated Air Supply Design in Ventilated Rooms. Water Supply and Sanitary Technique. ("Vodosnabgeniye I Sanitarnaya Tekhnika"). No. 6.

Tavakkol, S., M.H. Hosni, P.L. Miller, and Straub, H.E. 1994. A Study of Isothermal Throw of Air Jets With Various Room Sizes and Outlet Configurations. ASHRAE Transactions. V. 100 (1). pp. 1679-1686.

Taylor, J.F., H.L. Grimmett and E.W. , E.W. Commings. 1951. Chemical Engineering Progress. No. 4. pp.175-180.

Teterevnikov, V.N., T.V.Kuksinskaya and Pavlukhin L.V. 1974. Industrial microclimate and air conditioning. Leningrad. (In Russian).

Timofeeva, O.N., Veksler G.S. 1972. Experimental Results of Concentrated Air Supply in Large Rooms. Transactions of the Institutes for Labor Protection of the VTsSPS. Moscow: PROFIZDAT. V. 76. pp. 18-23.

Togari, S., Y.Arai and K.Miura. 1993. A simplified model for predicting vertical temperature distribution in a large space. ASHRAE Transactions. V.99(1). ASHRAE. Atlanta.

Tollmien, W. 1926. Sonderabdruck aus Zeitschrift für angewandte Mathematik und Mechanik. Vol. 6. pp. 468-478.

Trapani, R.D. 1967. An Experimental Study of Bounded and Confined Turbulent Jets. Advances in Fluidics. The 1967 Fluidics Symposium. ASME. New York. pp. 1 - 13.

Trogisch, A. 1990. Zur Gestaltung und Dimensionierung von Lüftungstechnischen Systemen in Industriehallen (Designing ventilation systems for industrial halls). Ki Klima-Kälte -Heizung. No.2. pp.73-76.

Troyanovsky, V.N. 1969. Studies of the allowable initial air temperature difference of the ambient air and air supplied by the ventilation system with concentrated jets. Research report T-5-69. KazIOT. Kazan.

Trüpel, T. 1915. Uber die Einwirkung eines Luftstrahles auf die umgebende Luft. Z. für das gesammte Turrbinenwesen. Nr. 5-6.

TsNIIPZ. 1985. Develop recommendations on design of VAV heating and ventilation systems with inclined jet air supply. Final report # 02880089962. Central Research Institute for Industrial Buildings (TsNIIPZ). Moscow.

TsNIIPromzdanii. 1984. Recommendations on Heating and Ventilating Systems Design with Directing Nozzles", Moscow: Central Research Institute for Industrial Buildings.

TsNIIPZ. 1978. Air Exchange in Industrial Halls with Ventilation Systems Combined with Warm Air Heating. Research Project Report # 7608206. Moscow: Central Research Institute for Industrial Buildings.

Turner, J.S. 1973. Buoyancy effects in fluids. Cambridge University Press. Cambridge.

Tuve, G.L. 1953. Air Velocities in Ventilating Jets. Heating, Piping and Air Conditioning. January. pp. 181-191.

Tuve, G.L. and G.B. Priester. 1944. Control of Air Streams in Large Spaces. ASHVE Research report No. 1248. ASHVE Transactions. pp. 153-172.

Uspenskaya, L.B., L.S. Klyatchko and Z.I. Rashkovsky. 1975. Ventilation of textile shops of the synthetic fiber production plants. The Issues of Sanitary Technique Systems Design and Installation. Leningrad: VNIIGS.

Uspenskaya, L.B. and S.M. Slavina. 1970. Experimental studies of temperature distribution in modular industrial spaces with a non-uniform heat sources. The Issues of Sanitary Technique Systems Design and Installation. Leningrad: VNIIGS.

Vajda, J. 1989. Matematische abbildung des strömungsbildes der einblaswkonstruktionen. Wentylacja I klimatizacja. Proceedings of the VIII Conference on ventilation and air conditioning. Gdansk. June.

Vasilyeva, L.V. 1969. Studies on Parallel Air Streams Interaction. Ph.D. thesis. Moscow. NIIST.

VDI. 1994. Workplace Air. Reduction of Exposure to Air Pollutants. Ventilation Technical Measures. VDI 2262. Bt.3/Pt.3. VDI Handbuch Betriebstechnik, Teil 4. Düsseldorf.

Vialle, P., D. Blay and B. Lionnet. 1996. Decay Laws in the Case of 3D Vertical Free Jets with Positive Buoyancy. Proceedings of the 5th International Conference on Air Distribution in Rooms. ROOMVENT'96. Vol. 1. July 1996. Yokohama, Japan.

VNIIOT. 1982. Studies of room air change methods in typical modules of modern industrial buildings and recommendations for their selection. Studies of air distribution and air change in air conditioned industrial spaces with a special requirements to thermal parameters. Research Report T-5-81. Part II. All-Union Research Institute for Labor Protection in Leningrad.

VNIIOT. 1981. Ventilating and Heating of Shipbuilding Halls. General Requirements 103.040. VNIIOT-GSPI. Leningrad, Russia.

VNIIOT. 1970. In Grimitlyn M.I. 1973. Principles of air distribution in ventilated rooms. D.Sc. Thesis. Leningrad. VVISKU.

Vulis, L.A., V.G. Zhivov and L.P. Yarin. 1969. Transition Zone in a Free Jet. Inz.-Phyz. Zhurnal. XVII (2).

Vulis, L.A. 1960. Regarding free turbulent jets computation applying the heat conductivity method. Proceedings of Acad.Sci. Kaz.SSR. Ser."Energy". V. 2(18).

Waschke, G. 1974. Über die Lüftung mittels isothermer turbulente radialer Deckenstrahlen. Dissertation. RWTH. Aachen. Germany.

Wasikuk V., Jaskolski M., Olezsko J. 1980. Model testing a large-plane exhaust of inflowing air as a possible solution for ductless ventilation system "air piston". Clima 2000. 7th International Congress of Heating and Air Conditioning. Budapest.

Weidemann, B., B. Hanel. 1988. Untersuchengen zum verhalten des vertical von oben nach unten ausströmenden runden warmen Freistrahls. Luft- und Kältetechnik. No. 3. pp.119-124.

Weidemann, B., B. Hanel, G. Hopper. 1985. Experimentelle Unter-suchungen des Temperaturfeldes Nichtisotherme, Horizontal Ausstromender, Runder Fleistrahlen. Luft- und Kältetechnik. No. 4.

Weidemann, B., Gy. Makara, A. Trogisch. 1983. Zur Problematik der Lüftung von Industriehallen. Luft-und Kaltetechnik. 1983/3. pp. 123-128.

Weinhold, K., R. Dannecker, U. Schwiegk. 1969. Über Auslegungsverfahren von Lüftungsdecken. Luft-und Kältetechnik. 1969/2. pp. 78-84.

Wille. R. 1963. Beiträge zur Phänomenologie der Freistrahlen. Z.Flugwiss. Vol, 11. No.6. pp.222-233.

Wygnanski, I. 1964. The Flow Induced by Two-Dimensional and Axisymmetric Turbulent Jets Issuing Normally from an Infinite Plane Surface. The Aeronautical Quarterly. November. pp. 373-380.

Yakovlevskii O.V. and Sekundov A.N. 1964. Fluid Flows Generated by Turbulent Jets. Proceedings of the Academy of Science of the USSR. Mechanics and Machine-Building. "Nauka".

Zhang, G., J.S. Strom. and S. Morsing. 1996. Jet Drop Models for Control of Cold Air Trajectories in Ventilated Buildings. Proceedings of the 5th International Conference on Air Distribution in Rooms. ROOMVENT'96. Vol. 1. July 1996. Yokohama, Japan.

Zhivov, A.M., Eu. O. Shilkrot, P.V. Nielsen and G.L. Riskowski. 1997a. Displacement Ventilation Design. Proceedings of the 5th International Symposium on Ventilation for Contaminant Control "Ventilation '97". Vol. 1.Ottawa, Canada.

Zhivov A.M. 1997b. Development and evaluation of a practical model for predicting room and air contaminant distribution. Report for the Institute for research in Construction (National Research Council, Canada). University of Illinois. Urbana.

Zhivov, A., J.B. Priest and L.L. Christianson. 1996. Air Distribution Design for Realistic Rooms. Proceedings of the 5th International Conference on Air Distribution in Rooms. ROOMVENT'96. Vol. 1. July 1996. Yokohama, Japan.

Zhivov A.M., J.K. McLaughlin, J.B.Priest, L.L.Christianson, R.N.Kulp, and L.E.Sparks. 1994. International Differences in Indoor Environmental Quality Standards and Guidelines. Proceedings of the IAQ'94 Conference "Engineering Indoor Environments". ASHRAE, EPA. St.Louis.

Zhivov A.M. 1994. Air Supply with Directing Jets. Proceedings of the Fourth International Symposium on Ventilation for Contaminant Control "Ventilation'94". Stockholm, Sweden.

Zhivov A.M. 1993a. Theory and Practice of Air Distribution with Inclined Jets. Transactions of ASHRAE. V.99(1).

Zhivov A.M. 1993b. Principles of Source Capturing and General Ventilation Design for Welding Premises. Transaction of ASHRAE. V.99(1).

Zhivov A.M. and B.W. Olesen. 1993. Extending Existing Thermal Comfort Standards to Work Spaces. Proceedings of the 6th International Conference on Indoor Air Quality. IAQ'93. Helsinki, Finland.

Zhivov, A.M. 1992. Selection of general ventilation method for industrial spaces. Presented at 1992 annual ASHRAE meeting. ("Supply air systems for industrial facilities", Seminar).

Zhivov, A.M. 1990. Variable Air Volume Ventilation Systems for Industrial Buildings. Transactions of ASHRAE. V.96 (2).

Zhivov, A.M., Pozin, G.M. 1989a. Improvement of the Design for Air Distribution by Inclined Jets. Theses of Reports at the Eighth Conference on Ventilation and Air Conditioning. Gdansk, Poland. pp.182-191.

Zhivov, A.M., Kelina, E.L. 1989b. Evaluation of Air Exchange Rate with Air Supply with Inclined Jets. Occupational Safety Technique and Industrial Sanitary. Collected papers of the Occupational Safety Institutes under AUCCTU. Moscow: Profizdat. pp.33-38.

Zhivov, A.M. 1989. Variable Air Volume Ventilation Systems for Industrial Buildings. Theses of reports at the second world Congress on Heating, Ventilation, Refrigeration, and Air Conditioning. CLIMA-2000. Sarajevo. Yugoslavia. 1989.

Zhivov, A.M. 1985. The Interaction of Axially Symmetric Coaxial Jets in Free and Confined Conditions. Turbulent Jet Flows. Theses of Reports of Fifth All Union Scientific Conference on Theoretical and Applied Aspects of Turbulent Flows. Part I. Tallinn: Estonian Academy of Sciences. pp. 27-32.

Zhivov, A.M. 1983. Concentrated Air Distribution with Directing Jets. Ph.D. Thesis. All-Union Research Institute for Labor Protection. St. Petersburg, Russia.

Zhivov, A.M. 1982. Investigation of the Interaction of Axially Symmetric Turbulent Jets Supplied under Straight Angles to Each Other, Turbulent Jet Flows. Theses of Reports of the Fourth All Union Scientific Conference on Theoretical and Applied Aspects of Turbulent Flows. Part II. Tallinn: Estonian Academy of Sciences.