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PRINCIPLES OF LOCAL EXHAUST DESIGN

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INTRODUCTION

Local exhaust ventilation systems are normally the most cost efficient method for controlling air pollutants and excessive heat. For many manual operations, capturing pollutants at or near their source is the only way to insure compliance with threshold limit values in the workers breathing zone. Local exhaust ventilation optimize ventilation airflow thus optimizing system costs especially where recirculation is not used.

In some industrial ventilation designs, the main emphasis has been made on filtration of air captured by local exhausts prior to evacuating it outside the building or returning back to the production space (Chambers, 1993). As a result, these systems are evaluated by the efficiency of their filters. However, if only a small percentage of the emission is captured, the degree of separation efficiency becomes almost irrelevant.

The pollutant capturing efficiency of local ventilation systems depends on hood design, its positioning near the source of the contamination, and the exhaust air flow. The appropriate selection and layout of hoods has a significant influence on initial and operating costs of both local and general ventilation systems. This paper suggests a classification of typically used local exhaust systems, and discusses the principles of their design and application.

LOCAL EXHAUST SYSTEMS CLASSIFICATION

Knowledge of the process or operation is essential before a hood can be designed. The type and the size of the hood depends upon the type and geometry of the pollution source. Pollution sources can be classified as: (1) a **buoyant (heat) source**, (2) a **non-buoyant (diffusion) source**, and (3) a **dynamic source** (3). Contaminant movement created by these sources is different. The first type of source is characterized by contaminants movement in the space primarily due to the heat energy, as a buoyant plumes over the heated surfaces. The second type of source is characterized by contaminants diffusion in the room in all directions due to the concentration gradient (e.g., in the case of emission from a painted surface). The emission rate in this case is significantly affected by the intensity of the ambient air turbulence. The third type of source is characterized by contaminant movement in the space with an air jet (e.g., linear jet over the tank with a push-pull ventilation, or particle flow from the grinding wheel). In some cases, the above factors influencing contaminant distribution in the room are combined.

The geometry of the contaminant source can be **compact or linear**. Source geometry affects the hood geometry: round/rectangular or slot. Hoods are either **enclosing or nonenclosing**. Enclosing hoods provide better and more economical contaminant control because their exhaust rates and the effects of room air currents are minimal compared to those for nonenclosing hoods.

. nonenclosing hood can be used if access requirements make it necessary to leave all or part of the process pen. Nonenclosing hoods can be classified according to their location relative to the contaminant source s either **updraft coaxial**, **sidedraft (lateral)**, or **downdraft**.

.ocal exhaust systems with nonenclosing hoods can be **stationary** (i.e., having a fixed hood position), **noveable, portable**, or **built into the process equipment**. Moveable (turnable) hoods are used when process equipment must be accessed for repairing, loading, and unloading, (e.g., in electric ovens for melting steel). Hoods attached to flexible fume extraction arms are used for small, medium-sized, and large workplaces when the source of contamination is not fixed, as in arc welding (Zhivov 1993). Flexible fume extraction arms usually have a hood connected to a duct 140 to 160 mm in diameter and have higher efficiencies for lower air volumes compared to stationary hoods. When the source of contamination is confined to a small, poorly ventilated space, such as a tank, an additional flexible hose extension with a hood and a magnetic foot can be attached to the fume extraction arm instead of the standard hood.

The portable fume extractor with a built-in fan and filter and a linear or round nozzle attached to a flexible hose about 45 mm in diameter is commonly used for the temporary extraction of fumes and solvents in confined spaces or during maintenance. Built-in local exhausts, such as gun-mounted exhaust hoods and fume extractors built into stationary or turnover welding tables, are commonly used to evacuate welding fumes. Lateral exhaust hoods, which exhaust air through slots on the periphery of open vessels such as those used for galvanizing metals, are another example of built-in local exhausts.

Nonenclosing hoods should be located so that the contaminant is drawn away from the operator's breathing zone. Canopy hoods should not be used where the operator must bend over a tank or process (ACGIH 1995).

EFFECTIVENESS OF LOCAL EXHAUST

The most effective hood uses the minimum exhaust volumetric flow rate to provide maximum contaminant control. The capturing effectiveness should be high, but it would be difficult and costly to develop a hood that is 100% efficient. Makeup air supplied by general ventilation to replace exhausted volumes can dilute contaminants that are not captured by the hood (Posokhin 1984).

Capture Velocity.

Capture velocity is the air velocity at the point of contaminant generation due to the hood performance that forces contaminants to move along with the air into the hood. Capture velocity depends upon the hood volumetric flow rate and design and size of the hood. The ASHRAE Handbook (1995) lists ranges of capture velocities for several industrial operations. These capture velocities are based on successful experiences under ideal conditions.

Target Airflow Rate.

For nonenclosing hoods, the airflow rate that allows the contaminant capture is called a target airflow. The target airflow rate Q* is proportional to some characteristic flow rate $\mathbf{Q_o}$ that depends upon the type of contaminant source:

$$Q^* = K\,Q_o \tag{1}$$

where: K - is a dimensionless coefficient depending upon the hood design; $\mathbf{Q_o}$ = characteristic airflow rate depending upon the contaminant source (e.g., for buoyant source $\mathbf{Q_o}$ can be equal to airflow in the convective plume, for dynamic source $\mathbf{Q_o}$ can be equal to airflow rate in the jet). In the case of a primarily non-buoyant contaminant source, characteristic airflow rate $\mathbf{Q_o}$ is calculated as follows:

$$Q_o = V_o \times A_o \tag{2}$$

where: $\mathbf{V_o}$ = average hood face velocity based on a desired capture velocity at the particular location in front of the hood; $\mathbf{A_o}$ = hood face area.

Exhausted airflow rate lower than $\mathbf{Q^*}$ results in reduced contaminant capturing effectiveness and exhausted airflow rate greater than $\mathbf{Q^*}$ results in excessive capturing effectiveness.

Air flow near the hood can be influenced by drafts created directly by the supply air jets (spot cooling jets) or by turbulence of the ambient air caused by the jets, upward/ downward convective flows, moving people, and drafts from doors and windows. Process equipment may be another source of air movement in the room. These factors can significantly reduce the capturing efficiency of local exhausts and should be accounted for by the correction coefficient on room air movement $K_r > 1$ in Equation (1), which will then become replaced with the following:

$$Q^* = K_r \, K \, Q_o \tag{3}$$

The exhausted air may contain combustible pollutant-air mixtures. In this case, the exhausted air flow rate should be increased to dilute the combustible mixture to less than 25% of the lower explosive limit of the pollutant C_{exp}^{min} (NFPA 1991). Thus the exhausted airflow rate $\mathbf{Q}$ should exceed the following:

$$Q = \frac{G}{0.25X \, C_{exp}^{min}} \tag{4}$$

For enclosing hoods, the exhaust airflow rate $\mathbf{Q}$ is the product of the hood face area $\mathbf{A_o}$ and the velocity $\mathbf{V_o}$ required to prevent outflow of the contaminant and can be calculated using Equation (2). The inflow velocity $\mathbf{V_o}$ for enclosing hoods is typically 0.5 m/s.

AIR MOVEMENT IN VICINITY OF LOCAL EXHAUST

Airflow near the hood can be described using the incompressible, irrotational flow (i.e. potential flow) model. The total pressure P_{tot} in the area upstream of the hood remains constant and can be described with the following equation:

$$P_{tot} = P_{st} + P_d = Constant \tag{5}$$

where: P_{st} and $P_d = \rho V^2/2$ are static and dynamic pressure respectively at any point of the flow, ρ = air density; V = air velocity.

At some distance from the hood, the total pressure in the air flow P_{tot} is equal to the ambient air pressure, e.g., $P_{tot} = 0$. Thus,

$$P_d = \frac{\rho V^2}{2} = - P_{st} \tag{6}$$

The above discussion does not apply to the wakes with a vortex air movement.

Air velocities in front of the hood suction opening depend upon the exhausted airflow rate, geometry of the hood, and the surfaces comprising the suction zone. Velocity contours (expressed as a percentage of the hood face velocity) were found to be similar for the hoods with similar geometry (Dallavalle 1952).

Air and contaminant distribution with non-buoyant sources.
Theories of hood performance with non-buoyant pollution sources are based on the equation of the turbulent diffusion and the following equation for contaminant concentration decay in the flow created by the hood

$$C_x = C_o \, e^{-\frac{V}{D}X} \tag{7}$$

where: C_o = contaminant concentration at the source; C_x = contaminant concentration at the distance X from the source; V = velocity in the flow created by the hood near the source; D = the coefficient of the turbulent diffusion, e = 2.71 - base for a natural logarithm.

The value of the coefficient of turbulent diffusion, D, depends upon the air change rate in the ventilated space and the method of air supply. Studies by Posokhin (1984) show that the approximate D values for locations zones outside supply air jets is equal to 0.025 /s. Air disturbance caused by the operator or the robot results in an increase of the D value at least two times. Studies by Zhivov et al. (1997) showed that D value is affected by cross draft velocity and direction against the hood face, and the presence of an operator: e.g., at a cross draft directed along the hood face with a velocity of V = 0.5m/s D = 0.15m²/s (with the presence of an operator); cross draft velocity increase to V = 1.0 m/s results in D = 0.3m²/s.

Numerical simulation of hood performance is complex and results are dependent upon hood design, flow restrictions by surrounding surfaces, source strength and other boundary conditions. Thus, most of the currently used methods of hood design are based on analytical models and experimental studies.

Airflow near a round or square/rectangular shape hood can be approximated by the point sink and the slot hood by the linear sink with a vanishingly small dimensions. The point source will draw air equally from all directions. The velocity $\mathbf{V_x}$ at any distance **X** can be calculated by the following equation:

$$V_x = \frac{Q}{4 \pi X^2} \tag{8}$$

Linear sources will create a two-dimensional flow with the velocity $\mathbf{V_x}$ calculated as follows:

$$V_x = \frac{Q}{2\,\pi\,X} \qquad (9)$$

where: Q = is exhaust airflow rate per unit of slot length (m or ft).

Centerline velocities for different realistic hoods are presented in Table 1. The comparison of relative velocity changes for realistic hoods and a point source made by Posokhin (1984) show, that at a distance greater than X/R = 1, velocities induced by realistic hoods and a point source are practically equal. It means that in some cases airflow in front of realistic hoods can be described using the simplified point source model.

Typically, velocity distribution in the hood face area is not uniform. Wakes formed close to the hood sides, reduce "effective suction area" or vena contracta of the hood. The size of these wakes and velocity uniformity level depend upon hood design. Vanes, baffles, perforation and other inserts can be used to control the size of vena contracta and velocity uniformity at the hood face area.

Air and contaminant distribution with buoyant sources.

Hood size and exhaust airflow rate for a buoyant contaminant source requires the knowledge of thermal plumes and physical size of the process equipment. Convectional heat and pollutants from the hot process are presumed to be contained in the thermal plume above the source., so the capture of the air transported with this plume will ensure the efficient capture of the contaminant (Burgess et al., 1996).

The analytical equations to calculate velocities, temperatures, air flow rates and other parameters in thermal plumes over spot and linear heat sources with given heat loads were derived by Zeldovich 1937, Schmidt 1941, Morton et al. 1956, and Shepelev 1961. These equations are based on the momentum and energy conservation equations and assuming Gaussian velocity and excessive temperature distribution in thermal plume cross-sections. These equations correspond with those received experimentally by other researchers (Popiolec, 1981; Tapola et al., 1987) e.g., the equation for the air flow rate in the thermal plume is as follows:

$$Q = C\,W_{conv}^{1/3}\,Z^{5/3} \qquad (10)$$

where: Q = air flow rate; W_{conv} = convective component of the heat source; Z = height above the source level; C = dimensionless coefficient.

Equation (10) was derived with the assumption that the heat source size was very small and did not account for the actual source dimensions.

The adjustment of the point source model to the realistic sources using the virtual source method gives a reasonable estimate of the air flow rate in thermal plumes (Ivanitskaya et al. 1974, Elterman 1980, Holman

1989, Mundt 1992). The weak part of this method according to Skistad (1994) is how to estimate the location of the virtual point. The method of a "maximum case" and a "minimum case" (Skistad 1994) provides a tool for such estimation. According to the "maximum case", the real source is replaced by the point source such that the border of the plume above the point source passes through the top edge of the real source (e.g., cylinder). The minimum case is when the diameter of the vena contracta of the plume is about 80% of the upper surface diameter and is located approximately 1/3 of the diameter above the source. For low temperature sources, Skistad (1994) recommends the "maximum case", whereas the "minimum case" best fits the measurements for larger, high temperature sources.

Interaction of the thermal plume with a wall and with another plume was studied by Kofoed (1991a, 1991b). In the case of the wall plume, the air flow rate should be decreased by a factor of 0.63; and increased by a factor of 1.26 for interaction with another equal plume.

Another approach to evaluate the thermal plume parameters is based on computational fluid dynamics (Nielsen, 1993b; Shaelin et al., 1992; Davidson, 1989; Aksenov et al., 1994). According to this approach, the air flow in the thermal plume is described by the system of Navier-Stokes equations and the equation for energy.

The hood's height above the buoyant source should be kept to a minimum to reduce the total exhaust air rate. A low canopy hood positioned within 1 m of a process (within the thermal plume transition zone) requires the least exhaust airflow rate. According to Posokhin (1984), the canopy hood is effective when:

$$\frac{V_r\,(Z + Z_o)}{V_z\, b} \le 0.35$$

where: V_r = room air velocity; Z_o = distance from the virtual source to the upper source level; V_z = air velocity on the thermal plume axis at the hood face level, b - source width.

The angle of the hood cone should be smaller than 90 deg. Hoods with a greater cone angle have an increased vortex zones near the edges which reduces the "effective suction area". Velocity distribution in the hood face should be nonuniform following the profile of the incoming flow. This can be achieved by incorporating vanes in the hood

Sidedraft hoods.

Sidedraft hoods are typically used when the contaminant is drawn away from the operator's breathing zone. In the case of a buoyant sources, sidedraft hood requires a higher exhaust volumetric flow rate than a low canopy hood. When a low canopy hood restricts the operation, the application of sidedraft hoods may be more cost effective than of the high canopy hood. Multislot "pickling hood" near the weld bench, flanged hood, and slot hood on tanks are some examples of sidedraft hoods. Sidedraft hoods should be installed with low edge of the suction area at the top level of the heat source. The distance b from the hood to the source may vary from 0 to B from the source. Based on studies by Kuz'mina (1959), Designer's guide (1992) recommends the following airflow rate through the sidedraft hood:

$$Q_o = c\, W_{conv}^{1/3}\, (H + B)^{5/3} \tag{11}$$

where: c = non-dimensional coefficient depending upon hood design and its location relative to contaminant source; W_{conv} = convective component of the heat source; H = vertical distance from source top surface to the hood center; B = source width.

For the hood without screen:

$$c = 280 \left(\frac{l}{B + H} \right)^{2/3} \tag{12}$$

For the hood with a side screen

$$c = 280 \left(\frac{l}{B + H} \right)^{1/2} m \tag{13}$$

where, m = 1, when b/B = 0; m = 1.5, when b/B = 0.3; m = 1.8 when b/B = 1, and m = 2 when b/B > 1.

For open vessels, contaminants can be controlled by a lateral exhaust hood, which exhausts air through slots on the periphery of the vessel. The hood capturing effectiveness depends upon the exhausted airflow rate and hood design, and is not influenced by the air velocity through the slot. There are hood designs with air exhaust from one side of the vessel and from two sides. Air exhaust from two sides requires a smaller exhausted airflow rate. In most typical designs, the hood is positioned with a vertical face where the distance between the vessel edge and the liquid level, h_l, is smaller than 100mm (Designers guide, 1992). When h_l > 100mm, hoods with a slot-tipped over to the liquid surface are more effective. A more cost effective alternative to a one- or two-sided lateral hood is a push-pull hood.

Downdraft hoods.

Downdraft hoods should be considered only when overhead or sidedraft hoods are impractical. Air can be exhausted through the slotted baffle (e.g., downdraft cutting table) or through a circular slot with a round source or two linear slots along the long sides of the rectangular source. To achieve higher capturing effectiveness, the exhaust should be located as close to the source as possible. Capturing effectiveness drops with a source height increase, and increases when the source top is located below the hood face surface. For a buoyant source air velocity induced by exhaust should be equal or greater than the air velocity in the plume above the source (Posokhin, 1984). The target airflow rate for a circular downdraft hood is

$$Q_o = 0.0314\,(W_{conv}\, d^5)^{1/3}\,\left(1 - 0.06\frac{W_{conv}^{vertical}}{W_{conv}^{horizontal}}\right) K_1\, K_v \tag{14}$$

for a double linear slot downdraft hood

$$Q_o = 0.05\ W_{conv}^{1/3}\ l\ b\ K_1\ K_v \tag{15}$$

where l = source length; d = source diameter; $W_{conv}^{vertical}$ = convective heat component from the source vertical surfaces; $W_{conv}^{horizontal}$ = convective heat component from the source horizontal surface; K = coefficient accounting for hood geometry; K_v = coefficient accounting for room air movement V_r:

for circular downdraft hood

$$K_v = 1 + 44.7 \left(V_r^3 \frac{d}{W_{conv}} \right)^{1/2} \tag{16}$$

for a double slot downdraft hood

$$K_v = 1 + 44.7 \left(V_r^3 \frac{bd}{W_{conv}} \right)^{1/2} \tag{17}$$

Air movement created by dynamic sources.

Air jets utilized in push-pull systems are supplied in the contaminated zone. They inject contaminated air and direct it towards the hood. Air jets in push-pull systems can be compact or linear. To reduce the effect of room air movement on the hoods performance, the push air jet centerline velocity at the critical "cross-section" , where the push air jet becomes weak, and the influence of the hood is not strong enough, should be from one to 2 m/s (Designer's guide, 1992). In the case of push-pull hood over the tank, the supply air velocity should not exceed 10 m/s to avoid waves on the liquid surface.

Influence of air movement on local exhaust performance.

Air movement caused directly by supply air jets and turbulence of the ambient air resulted from general ventilation system operation, convective plumes, moving people and process equipment is at least as important as hood face velocity in controlling spillage of contaminant. Caplan and Knutson (1978) recommend that air movement caused by the above factors should be less than 1/2 to 2/3 the hood face velocity.

Studies of the hood with a vertical face area by Zhivov et al. (1997), showed that when the direction of the cross draft is known, the preferable of orientation of the hood relative to the most likely direction of the cross-draft is 135°. This achieves both the lowest contaminant concentration in the operator's breathing zone and the highest capturing effectiveness. A moderate draft from behind the operator significantly increases the contaminant concentration in the operator's breathing zone. A cross draft from the side has minimal effect on the operator exposure, but the contaminant removal by the hood is low.

To reduce the influence of cross-drafts greater that 0.4 m/s on the performance of the canopy hood above the buoyant source, it was recommended (Designer's guide, 1992) to have a supply hood with one-, two- or three-sided removable shields attached to the hood dropping to a height of 0.8 of the equivalent diameter of the source size in the plan. The hood function can be improved with the help of air jets creating air curtains in front of the hood.

Jet-Assisted Hoods.

Jet-assisted hoods are nonenclosing hoods combined with compact, linear or radial air jets. They are used to

- Increase capturing effectiveness of hoods by transporting contaminants towards their face with supply jets (ACGIH 1995, Heinsohn 1991, Elinskii 1989, Designers Guide 1992, Sciola 1993);
- Separate contaminated zones from relatively clean zones in working spaces (Boshnyakov 1975, Romeyko et al. 1976, Stoler and Savelyev 1977, Posokhin and Broida 1980, Anichkhin et al. 1984, Cesta 1988, and Zhivov 1993 (e.g., jet- assisted hoods can be used over welding robots);
- Prevent contaminated air from moving into clean zones by creating positive static pressure (Strongin and Marder 1988), for example, in drying chambers and cooling tunnels of casting conveyors;
- Protect hoods from room air movement (e.g., laboratory hoods and industrial ovens, hoods can be used in combination with air curtains).

The jets cause swirling airflows near the exhaust hood, increasing its capturing efficiency (Ljungqist and Waering 1898). The capturing efficiency of jet-assisted hoods is 15 to 20% higher than that of conventional hoods for the same operating costs (Strongin and Marder 1988). For the same capturing efficiency, they are 25 to 30% more cost-effective due to lower airflow. When exhausted air must be cleaned, reduced cleaning costs make jet-assisted hoods 100 to 150% more cost-effective. If compensating-type jet-assisted hoods are used, operating costs for heating and cooling in general ventilation systems are also reduced.

SUMMARY

Numerous studies of local exhausts and common practices (Posokhin 1984, ASHRAE 1995) resulted in the following hood design principles:

- The hood should be located as close as possible to the source of contamination;
- The hood opening should be positioned so that it causes the contaminant to deviate the least from its natural path;
- The hood should be located so that the contaminant is drawn away from the operator's breathing zone;
- The hood must be the same size as or larger than the flow entering the hood. If the hood is smaller than the flow, a higher volumetric flow rate will be required;
- The velocity distribution in the hood opening cross section should be nonuniform, following the velocity profile of the incoming flow. This can be achieved by incorporating vanes in the hood opening. In the case of a stationary hood and a contaminant source that is not fixed (e.g., welding or soldering), the air velocity along the hood must be uniform; this can be achieved by using vanes or perforations.

REFERENCES

ACGIH. 1995. Industrial ventilation. A manual of recommended practice, 22st ed. Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists.

ASHRAE. 1995. ASHRAE Handbook. HVAC Applications. Atlanta, GA..

Boshnyakov, E.N. 1975. Local exhaust with air curtains. Water supply and sanitary techniques, #3. Moscow (in Russian).

Braconnier, R. 1988. Bibliographic review of velocity field in the vicinity of local exhaust hood openings. American Industrial Hygienist Association Journal 49(4):185-98.

Caplan, K.J. and G.W. Knutson. 1978. Laboratory Fume Hoods: Influence of Room Air Supply. ASHRAE Research Project RP-70. ASHRAE Transactions 82(1).

Caplan, K.J. and G.W. Knutson. 1977. The effect of room air challenge on the efficiency of laboratory fume hoods. ASHRAE Transactions 83(1): 141.

Chambers, D.T. 1993. Local Exhaust Ventilation. A Philosophical Review of the Current State-of-the-Art with Particular Emphasis on Improved Worker Protection. Leicester. UK.

Davidson, L. 1989b. Numerical Simulation of Turbulent Flow in Ventilated Rooms. Ph.D.-Thesis, Chalmers University of Technology, Sweden.

DallaValle, J.M. 1952. Exhaust hoods, p.22. Industrial Press, New York.

Designer's guide. 1992. Ventilation and air conditioning. 4th edition. Part 3(1). Moscow: Stroiizdat (In Russian).

Elinskii, I.I. 1989. Ventilation and heating of galvanic shops of machine-building plants. Mashinostroyeniye, Moscow (in Russian).

Elterman, V.M. 1980. Ventilation of Chemical Plants. Moscow: KHIMIA.(in Russian)

Flynn, M.R. and M.J. Ellenbecker. 1985. The potential flow solution for air flow into a flanged circular hood. American Industrial Hygiene Journal 46(6):318-22.

Heinsohn, R.J. 1991. Industrial Ventilation: Engineering Principles. John Wiley & Sons, New York.

Heinsohn, R.J., K.C. Hsieh, and C.L. Merkle. 1985. Lateral ventilation systems for open vessels. ASHRAE Transactions 91(1B):361-82.

Hemeon, W.C.L. 1963. Plant and process ventilation, p. 77. Industrial Press, New York.

Holman, J.P. 1989. Heat Transfer. McGraw Hill Book Company. Singapore.

Huebener, D.J. and R.T. Hughes. 1985. Development of push-pull ventilation. American Industrial Hygiene Association Journal 46(5):262-67.

Ivanitskaya, M. Yu. and V.I. Kunitsa. 1974. Experimental Studies of Thermal Plumes above the Round Heat source. Proceedings of TsNIIPromzdanii. V.37. Moscow: TsNIIPromzdanii. (in Russian)

Kofoed, P. 1991a. Thermal Plumes in Ventilated Rooms. Ph.D. thesis, Aalborg University, Aalborg.

Kofoed, P. and P.V. Nielsen. 1991b. Thermal Plumes in Ventilated Rooms - Vertical Volume Flux Influenced by Enclosing walls. 12th AIVC Conference, Ottawa.

Ljungqist, B. and C. Waering. 1988. Some observations on "modern" design of fume cupboards. Proceedings of the 2nd International Symposium on Ventilation for Contaminant Control, Ventilation'88. Pergamon Press, United Kingdom.

Morton, B.R. 1959. Forced Plumes. J.Fluid Mechanics. Vol. 5. pp.151-163.

Mundt, E. 1992. Convection Flows in Rooms with Temperature Gradients - Theory and Measurements. ROOMVENT'92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 3. Aalborg.

NFPA. 1991. Standard for ovens and furnaces. Design, location and equipment. National Fire Protection Association, Standard 86A, Item 4-2.1.

Paoiolec, Z. 1981. Problems of testing and mathematical modeling of plumes above human body and other extensive heat sources. A4-seria. No. 54. KTH, Stockholm.

Posokhin, V.N. 1984. Design of local ventilation systems for the process equipment with heat and gas release. Mashinostroyeniye, Moscow (in Russian).

Posokhin, V.N. and V.A. Broida. 1980. Local exhausts incorporated with air curtains. Hydromechanics and heat transfer in sanitary technique equipment. KHTI, Kazan (in Russian).

Romeyko, N.F., N.E. Siromyatnikova, and E.V. Schebraev. 1976. Design of air curtains near an oven opening supplied with a hood. Heating and Ventilation. Proceedings of the A.I. Mikoyan Institute of Civil Engineers (in Russian).

Schaelin, A. and P. Kofoed. 1992. Numerical Simulation of Thermal Plumes in Rooms. ROOMVENT'92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 1. Aalborg.

Sciola, V. 1993. The practical application of reduced flow push-pull plating tank exhaust systems. 3rd International Symposium on Ventilation for Contaminant Control, Ventilation '91 (Cincinnati, Ohio).

Sepsy, C.F. and D.B. Pies. 1973. An experimental study of the pressure losses in converging flow fittings used in exhaust systems. Document PB 221 130. Prepared by Ohio State University for National Institute for Occupational Health.

Schmidt, W. 1941. Turbulente Ausbreitung eines Stromes erhitzer Luft ZAMM. Bd. 21 # 5. (in German)

Shepelev, I.A. 1961. Turbulent Convective Stream above the Heat Source. Proceedings of Acad.Sci. USSR. Mechanics and Machinery Construction. #4. (in Russian)

Shibata, M., R.H. Howell, and T. Hayashi. 1982. Characteristics and design method for push-pull hoods: Part 1. Cooperation theory of air flow; Part 2. Streamline analysis of push-pull flow. ASHRAE Transactions 88.

Silverman, L. 1942. Velocity characteristics of narrow exhaust slots. Journal of Industrial Hygiene and Toxicology 24 (November):267.

Skistad, H. 194. Displacement Ventilation. Research Studies Press. John Willey &Sons, Ltd. West Sussex. UK.

Stoler, V.D. and Yu. L. Savelyev. 1977. Push-pull systems design for etching tanks. Heating, Ventilation, Water Supply, and Sewage Systems Design 8(124). TsINIS, Moscow (in Russian).

Strongin, A.S. and M.L. Marder. 1988. Complex solution of painting shops ventilation. Proceedings of the conference "Utilization of Natural Resources and New Ventilation and Dust Transportation Systems Design". Penza (in Russian).

Sutton, O.G. 1950. The dispersion of hot gases in the atmosphere. Journal of Meteorology 7(5):307.

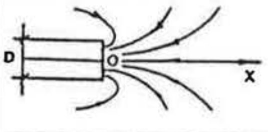
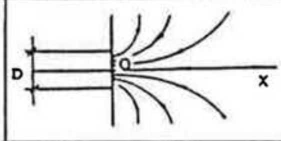
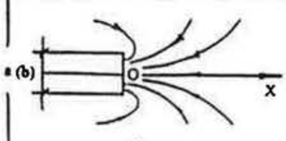
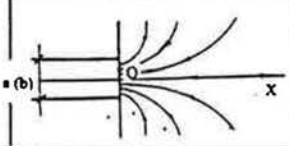
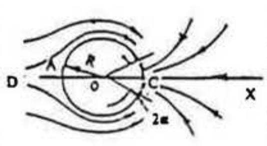
Zarouri, M.D., R.J. Heinsohn, and C.L. Merkle. 1983. Numerical computation of trajectories and concentrations of particles in a grinding booth. ASHRAE Transactions 89 (2A):119-35.

Zeldovitch, Y.B. 1937. Fundamental Principles for Free Convective Plumes. Journal of the Experimental and Technical Physics. Vol.7(12). Moscow. (in Russian)

Zhivov, A.M. 1993. Principles of source capturing and general ventilation design for welding premises. ASHRAE Transactions 99(1):979-86.

Zhivov, A.M., L.L.Christianson and G.L. Riskowski. 1997. Influence of Space Air Movement on Hood Performance. ASHRAE Research Project RP-744. ASHRAE. Atlanta, GA.

Table 1. Centerline air velocities induced by non-enclosing hoods performance

Hood type	Schematic	Equation	Applicable range	Refere
Round free-standing hood, unflanged		$V_x/V_o = (1 + 10X^2/A)^{-1}$	$X \le 1.7\sqrt{A}$, $\delta \le 30°$	Dallavall 1952
Round free-standing hood, flanged		$V_x/V_o = 1.1(0.07)^{-X/D}$ $V_x/V_o = 0.1(X/D)^{-1.6}$	$0 \le X/D \le 0.5$, $C \ge D$ $0.5 \le X/D \le 1.5$, $C \ge D$	Garrison, 1977
Rectangular free-standing hood, unflanged		$V_x/V_o = (0.93 + 8.58\alpha_F^2)^{-1}$ $\alpha_F = (X/\sqrt{A})(a/b)^{\beta F}$ $\beta_F = 0.2(X/\sqrt{A})^{-1.3}$	$1 \le a/b \le 16$, $0.05 \le X/\sqrt{A} \le 3$, $\delta \le 30°$	Fletcher, 1977
Rectangular free-standing hood, flanged		$1 - 2/\pi\ \mathrm{Arctg}\ [2X(X^2 + a^2 + b^2)^{1/2}/ab]$	$1 \le a/b \le 16$, $0 \le X/\sqrt{A} \le 1.6$, $C/\sqrt{A} \ge 1$	Tyaglo an Shepelev, 1970
Slot in the pipe wall		$\frac{V_{CX}}{V_o} = \frac{V_{AD}}{V_o} = \frac{2\ R}{\pi\ X}\ Arctg\left(\frac{X + R}{X - R}\ tg\frac{\alpha}{2}\right)$	$\|X\| \ge R$	Posokhin, 1984