5605

Calculation Method for Airflow Rate in Displacement Ventilation Systems

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ABSTRACT

The operation of displacement ventilation is based on the natural upward movement of air as it warms up. Different kinds of equipment warm the room air and cause an upward thermal current. The same heat sources might also produce impurities and contaminants, which will rise with the convective flow. Temperature and concentration gradients are naturally formed in displacement ventilation. Contaminants and excess heat are exhausted at a high level. The aim is to keep the occupied zone as clean and comfortable as possible. At a certain level, the amount of convective flow is equal to the fresh airflow introduced to the space at floor level. This is the so-called shift zone. The main goal is to keep this shift zone above the occupied zone and achieve a clean and comfortable occupied zone.

The design method for displacement ventilation is based on calculation of the total convective flow generated by the heat sources. Different kinds of heat loads cause different amounts of convection. This problem has been taken into account by selecting the most important heat-source parameters affecting the convective flow. These parameters were found in 140 tests conducted under both laboratory and field conditions. Correlations for the airflow rate calculations for displacement systems were developed by using these measurement results.

INTRODUCTION

Displacement ventilation has become a commonly used ventilation method in Scandinavia during the past five years. It is a very efficient system for certain applications and gives many benefits, especially in high spaces. The operation of the system differs from mixing ventilation, and this must be taken into account in the airflow rate calculation method. The method of airflow rate calculation for displacement ventilation systems is based on research done from 1986 to 1988. The equations used for computations are based on both the utilization of theoretical models and measured data. A displacement ventilation system was simulated in a laboratory in several cases. Many different cases were included in the testing program to find out the relationships between the most effective characteristics. Primary interest was given to heat loads. Measurements were made at different heights, surface temperatures, and locations in the room. The height and airflow rate were also changed.

LABORATORY MEASUREMENTS

Measurements were carried out by a computerized system. Temperature measurements were made at 40 points located at 10 different heights in the space, in the supply and extract air, and at the surfaces of the heat loads and walls. Surface temperatures of the walls were measured to analyze the energy balance.

Ventilation effectiveness and stratification of concentration caused by convective flow were determined using a tracer gas technique. The tracer gas was dosed to the surfaces of the heat loads. Concentrations of the tracer gas transported by the convective flow were measured at 10 different heights in the space. Extract air and supply air concentrations were also measured. Dosage of the tracer gas was kept constant until steady-state conditions were achieved, and measurement continued at least 30 minutes. Gradient curves were drawn by using average values obtained during this 30-minute period. The decay of the concentration was also measured. Gradient curves were drawn by using measured tracer gas concentrations at different heights. Analyzing these curves, the shift zone position could be defined. The shift zone position values were fitted to the theoretical model.

Several measurements were made in real applications during the research. These case-study measurement results and the results measured in the laboratory were compared and found to be similar. The field measurements have shown that the operation of a displacement ventilation system can be calculated and designed by using developed equations. Some additional data and precise knowledge about the heat loads in the space are required.

GROUNDS FOR CALCULATION

In thermal displacement ventilation, the main intent is to send excess heat and contaminants toward the ceiling so that the occupied zone is as clean as possible. Temperature at floor level will be lower than at the higher level because of the natural air movement caused by convective flow together with low mixing-air distribution directly to the occupied zone.

The calculation method for removal of impurities is based on control of the layered height, and it can be used with thermal displacement when contaminants are warm. Theoretically, the shift zone will be at a height where the volume of the convective flow is equal to the supply airflow rate. In this model, the shift zone is defined as the height where the concentration of tracer gas is one-third of the concentration in the extract air. It means that the concentration in the occupied zone is equal to or less than one-third of the concentration in the extract air.

The definitions are relatively close. In theory, the extract air concentration is equal to the concentration in a mixing system with the same airflow rate. Practically, this means that air quality is about three times better than in mixing ventilation with the

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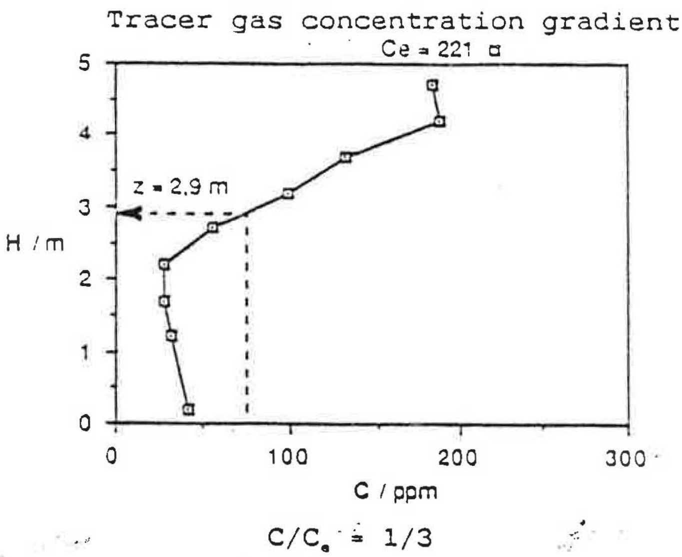

Figure 1 *Determination of the shift zone position from measured tracer gas concentrations at different heights*

same airflow rate. The requirement to keep the shift zone above the occupied zone can be a basis for calculating good air quality.

SHIFT ZONE POSITION

The layering phenomenon is based on flows of natural convection caused by heat sources. The correlation for convective flow caused by different kinds of heat load has been introduced in the literature. The volume flow is a function of the main flow direction and the convective heat power of the source. This function can be described as follows (Tapola and Koivula 1989):

$$Q \approx P_c^{1/3} \cdot X^{5/3} \tag{1}$$

or

$$X \approx Q^{3/5} / P_c^{1/5} \tag{2}$$

where

Q = volume flow of convection (L/s)
P_c = convective heat power of the heat load (W)
X = distance from the heat load (m).

The characteristics of the heat source affect the spreading of impurities into the space. The interdependence may now be expressed as follows:

$$X = k \cdot Q^{3/5} / P_c^{1/5} \tag{3}$$

where

k = coefficient of correlation made to contain the effect of the heat load characteristics.

CHARACTERISTICS OF THE HEAT LOADS

The characteristics of the heat sources affect the amount of convective flow. A low surface temperature will cause a greater volume flow than a source having a higher surface temperature if the convective power of the heat sources is equal. This is due to the larger exothermal area resulting in a convective flow. Distance from the heat source affects the amount of convective flow as well.

In Equation 3, X is a distance from the heat source. Floor level is a natural zero point for the height of the heat and the height of the occupied zone. In this model, the distance from the heat source (X) was replaced with a term Z, which is the height of the shift zone and includes the height of the heat load and distance from the heat load to the height of Z.

Coefficient k is obtained from measurement so that the calculated airflow rate will transport and keep the convective flow caused by heat load effect P at the height of Z above floor level. The equation for coefficient k is assumed to be a function of the temperature difference between the surface temperature of the heat load and room temperature and the height of the heat source. The equation can be expressed as follows:

$$k = C_1 \cdot (t_f - t_r)^{C2} + C_3 \cdot h. \tag{4}$$

The whole adapted equation is

$$Z = (C_1 \cdot (t_f - t_r)^{C2} + C_3 \cdot h) \cdot Q^{3/5} / P_c^{1/5} \tag{5}$$

where

Z = height of the shift zone above floor level (m)
C_1, C_2, C_3 = parameters adapted from measurement
t_f = surface temperature of the heat load (°C, average surface temperature value for all heat loads)
t_r = room temperature in the occupied zone (°C)
h = height of the heat load (m, average height of upper surface of all heat loads)
Q_v = supply air volume flow rate per floor area (L/s, m^2)
P_c = convective power of all the heat loads calculated as an average value per floor area (W/m^2).

The following values were found in the laboratory measurements for the adapting parameters:

C_1 = 0.0075
C_2 = 1.02
C_3 = 0.54.

This formula can also be shown as a diagram (Appendix A).

REMOVAL OF EXCESS HEAT

The calculation method for removal of excess heat is based on total heat balance and takes into account the thermal balance caused by the heat sources. The convective flow does not necessarily contain any hazardous impurities, only excess heat. In these cases, the removal of this excess heat is of primary interest. The thermal balance in the displacement system is the same as in a mixing system. The main difference is the determination of extract air temperature. The cooling load of the ventilation is a function of the airflow rate and temperature difference between the extract air and the supply air.

$$P_{cool} = Q_v \cdot \rho \cdot C_p \cdot (t_e \cdot t_s) \tag{6}$$

where

P_{cool} = cooling load of ventilation (W)
Q_v = air volume flow rate (L/s)
ρ = density of air, 1.20 kg/m^3
C_p = specific heat of air, 1.006 kJ/(kg, K)
t_e = extract air temperature (°C)
t_s = supply air temperature.

In the displacement system, the extract air temperature cannot be defined without taking into account the effect of the heat load characteristics. The room temperature and the extract air temperatures are not equal because of the layering effect. Taking into account the effect of the surface temperature and the height, Equation 4 can be used as a basis. Requirements for the temperature gradient are not greater than 3°C/m with the limitations for the supply air temperature and the acceptable temperature difference between the extract and supply air temperatures. The diagram in Appendix B takes these comfort criteria into account and gives an estimation of the temperature difference in the different kinds of spaces at different heights.

DISCUSSION

This method for displacement ventilation design and calculation requires some extra information to be obtained before the air volume flow calculation. These values can be estimated despite the mathematical method to determinate needed data, such as average surface temperature and height. All heat loads are handled together, which offers the possibility of making a rough calculation by estimating the average values.

This paper includes two design methods to determine the total airflow rate in the displacement system. These two methods have similarities, and both were developed by using the same measurement results. These two methods give two answers for airflow rate. The final answer depends on the requirements and the type of space and quality of impurities. In the spaces that do not include any source for impurities, the design can be temperature-based. When the requirement is to keep a low concentration of contaminants in the occupied zone, the design should be based on the shift zone.

Measurements were made mainly by using one or two horizontally located points because the differences were noticed to be very small. There will be much higher concentrations inside the convective flow above the heat load.

CONCLUSIONS

Determination of airflow rates in displacement ventilation systems differs from the mixing system. This paper has described a method to calculate airflow rate so that advantages of the system can be realized. This design method offers two bases to determine the airflow rate in a displacement ventilation system, one for removal of impurities and the other for removal of excess heat. The selected airflow rate depends on the requirements. If the removal of impurities is more important, the shift-zone-based design should be selected.

The calculation method is based on a large series of laboratory measurements and several measurements in real applications. Collected information for the measurements has been adapted to the theory. Results from the developed calculation model have been compared to the case studies (Appendix C). This method has been used for airflow rate computation in Scandinavia for the last two years.

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APPENDIX A

DIMENSIONING DIAGRAM FOR DISPLACEMENT VENTILATION

Determination of airflow rate to transport contaminants at height Z.

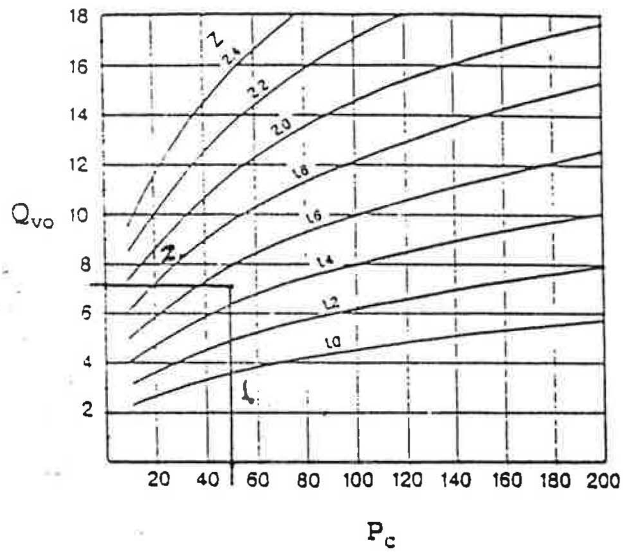


Diagram A.1

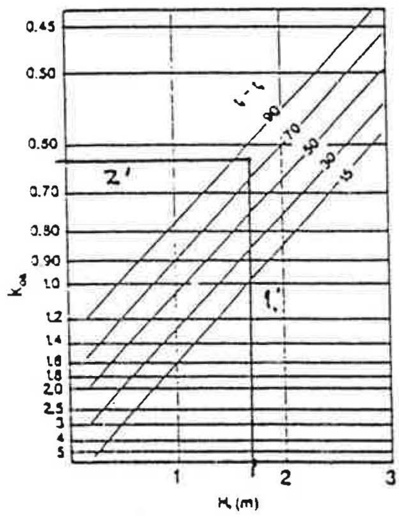


Diagram A.2

where

Q_{vo} = air volume flow rate before correction (L/s)

P_v = convective power of the heat load per floor area (W/m^{-2})

$t_f - t_r$ = temperature difference between the room temperature and average surface temperature of all the heat loads (°C)

$H_1(m)$ = average height of the heat load's warm upper surface (m)

Z = height of the shift zone (m)

K_{OK} = correction coefficient. Takes into account the effect of the surface temperature and height of the heat load.

APPENDIX B

DETERMINATION OF THE TEMPERATURE DIFFERENCE BETWEEN THE EXTRACT AIR AND SUPPLY AIR IN A DISPLACEMENT SYSTEM

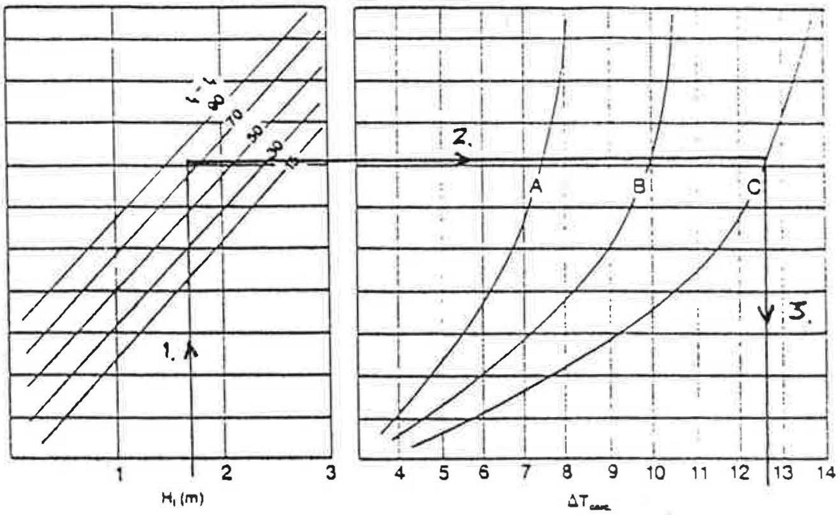


Diagram B.1

where

$t_f - t_r$ = temperature difference between the room temperature and average surface temperature of all the heat loads (°C)

H_1(m) = averaged height of the heat load's warm upper surface (m)

ΔT_{calc} = determined temperature difference (°C).

TABLE B.1

CURVE	TYPE or HEIGHT OF SPACE	RECOMMENDED SUPPLY AIR TEMPERATURE °C	DEGREE OF COOLING, SUPPLY AIR TO ROOM TEMPERATURE °C	MAXIMUM TEMPERATURE GRADIENT °C/m
A	- OFFICE - H_r < 3 m	> 18	0.5-3	2.0
B	- FOYER - AUDITORIUM - H_r = 3-6 m	> 16	1 - 3	3.0
C	- INDUSTRIAL SPACE - H_r > 6 m	> 15	3 - 6	3.0

APPENDIX C

CASE STUDY: LAUNDRY

The main object was to measure the operation of an existing displacement ventilation system. The measured space was a laundry with a room area of 450 m^2 and a height of 5 m.

The measurements were made by using the tracer gas technique. Measurement points were at seven different heights. Supply and extract air concentrations were also measured. Temperatures were measured in the same points.

The heights and surface temperature of the heat loads were measured before the tracer gas measurements. Measured temperature and tracer gas concentrations were as follows:

TABLE C.1 Measured Heat Loads

heat source	number*effect	conv. effect	t_f(°C)	H_1(m)
W mach PK1	4*1.5=6.0	0.5*6.0=3.0	90	1.3
W mach PK2	2*2.0=4.0	0.5*4.0=2.0	40	2.1
pressing	1*1.0=1.0	0.5*1.0=0.5	90	1.3
hot water	14	14	120	1.5
people	5*0.1=0.5	0.5	35	1.2
lighting	3.6	0.5*3.6=1.8	70	4.0
total	29.1 kW	21.8 kW	102 *	1.7 *

*weighted average figures (weighing factor convection effect)

Comparison between the Calculated and Measured Data

Temperature-based design: Diagram B.1

$H_1 = 1.72$ m, $t_f - t_r = 81$°C with curve C gives $\Delta T_{calc} = 12.5$°C, airflow from Equation 6 gives $Q_v = 1{,}935$ L/s $= 4.3$ L/s.

Shift-zone-based design: Diagrams A.1 and A.2

Convective effect based on previous table is 48.4 W/m^2. Based on measured results, one-third of the extract air concentration was met at a height of 1.5 m. Based on the design method, the required airflow rate to lift the shift zone at height 1.5 m is as follows:

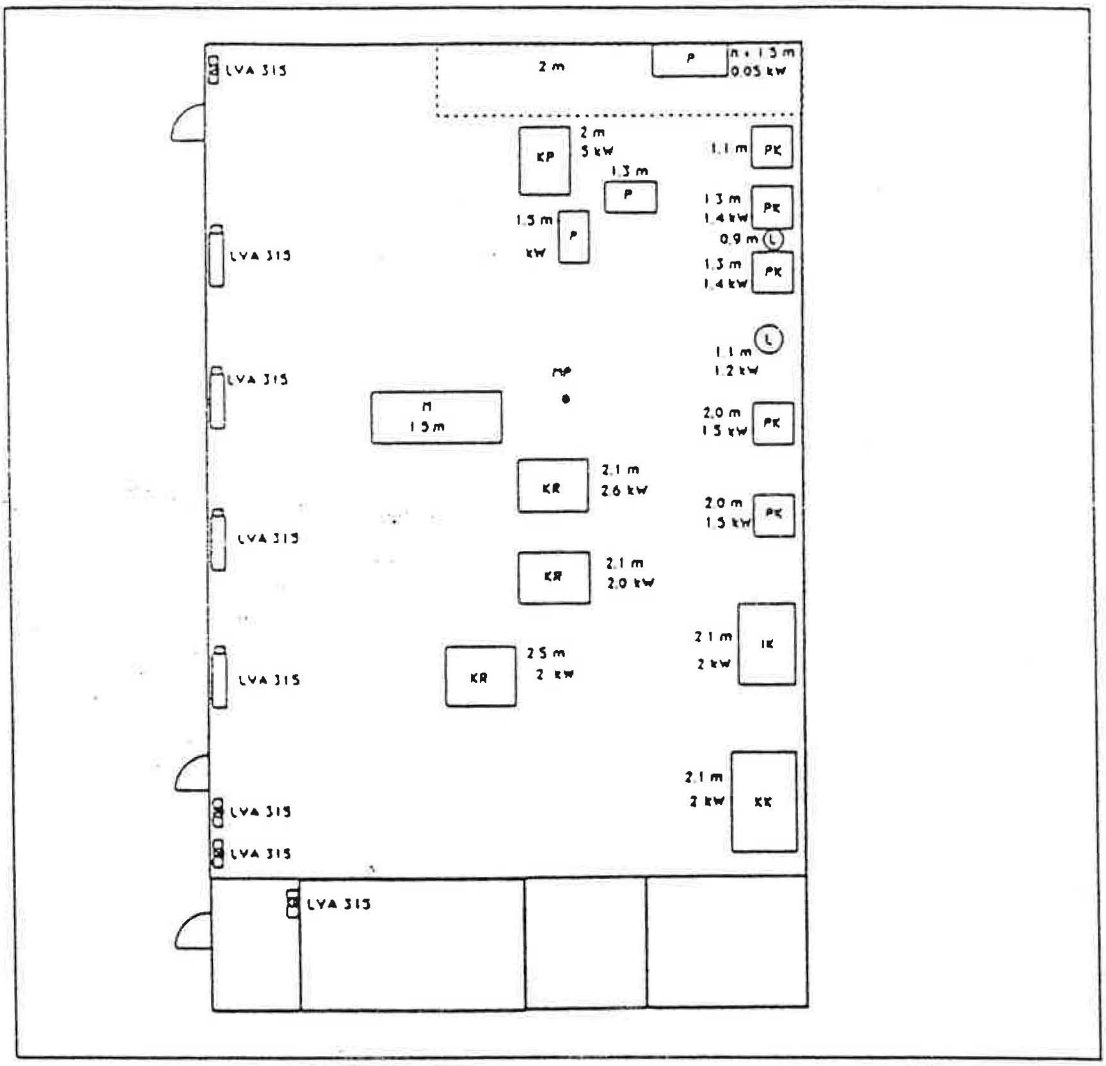
Diagram A.1: $P_c = 48.4$ W/m^2 and $z = 1.5$ gives $Q_{VO} = 7.2$

Diagram A.2: $H_1 = 1.7$ m and $t_f - t_r = 102 - 21$°C gives $K_{OK} = 0.62$

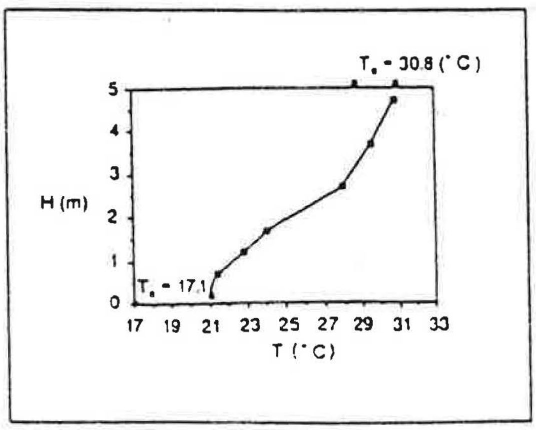
$Q_V = 7.2 \cdot 0.62 = 4.46$ L/s, m^2.

In the measured situation, it was 4 L/s, m^2.

Lay out:



TEMPERATURE GRADIENT



IMPURITY CONCENTRATION