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EXPERIMENTAL MEASUREMENT OF INDUSTRIAL VENTILATION EFFECTIVENESS AND THERMAL COMFORT

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ABSTRACT

The evaluation of the ventilation effectiveness and thermal comfort for various industrial ventilation schemes has been carried out by 1:4 scale model experimentation. Measurements of air speed, temperature, and contaminant concentration allowed the contaminant removal and thermal comfort to be quantified using ventilation effectiveness and thermal comfort indices, respectively. Archimedes number scaling was used to convert the small scale measurements to full scale conditions. The ventilation efficiency generally increased when the heat load was increased and/or the flow rate decreased. Increasing the number of diffusers in the occupied zone increased the ventilation effectiveness. The thermal comfort results depended on the diffuser configuration and the activity level of the worker. Most of the configurations produced acceptable thermal comfort results for a seated worker condition, and unacceptable conditions at an increased activity and clothing level.

KEYWORDS

Industrial ventilation, ventilation effectiveness, thermal comfort, effect of supply and return location

INTRODUCTION

The subject of this paper is the experimental determination of the ventilation and thermal comfort performance of a variety of configurations of industrial ventilation systems. The ventilation of industrial facilities is important for two main reasons. The ventilation of industrial spaces is important to ensure a supply of fresh air, and also to maintain thermal comfort. The introduction of fresh air by a ventilation system reduces the level of contaminants produced by machinery and occupants. The type of ventilation process examined in this paper is mixing ventilation, in which the fresh and cool ventilation air will mix with the existing room air, as opposed to displacement ventilation, in which the ventilation system is designed to displace the room air with fresh air.

The impact of the process equipment on the ventilation performance and thermal comfort of a given HVAC system is not well understood. The objective of this research is to experimentally determine the ventilation and thermal comfort characteristics of a scale model industrial space containing process equipment. Specific goals are to compare the performance of supply and return diffusers located at different heights for a range of flow rates and heat loads.

REPESENTATIVE PREVIOUS WORK

Fissore and Liebecq (1991) used a 1:3 scale model to predict velocity distribution and thermal comfort in a slot ventilated space. Chan et al. (1993) used measurements made on a 1:5 scale model to predict HVAC system performance in a full scale prototype. Zhang et al. (1993) found that a scale model predicted the mean velocity and temperature of a full scale enclosure to within 11% and 15% respectively. Irwin and Besant (1994) investigated contaminant removal by two ventilation systems in a 1:2.5 scale model. Heiselberg (1996) reported measurements of local ventilation effectiveness with a high supply and high exhaust. Chen et al. (1992) numerically studied the performance of a different ventilation orientations.

VENTILATION AND THERMAL COMFORT INDICES

The performance of the ventilation systems is assessed with two indices: the ventilation effectiveness and the thermal comfort. The ventilation or contaminant removal effectiveness at a point p in a mechanically ventilated space is $VE_p = (C_e - C_s)/(C_p - C_s)$ where C is the contaminant concentration, s is supply, e is exhaust, (Skaret and Mathisen, 1983). The thermal comfort model used in this work is the ISO comfort standard 7730 (ISO 1995). This model includes the occupied zone air temperature, air speed, mean radiant temperature, humidity, and the occupant clothing, metabolic rate, and external work. The turbulence intensity can also be included in the model. The output of the model is the Predicted Mean Vote (PMV), and the associated Percent Persons Dissatisfied (PPD)

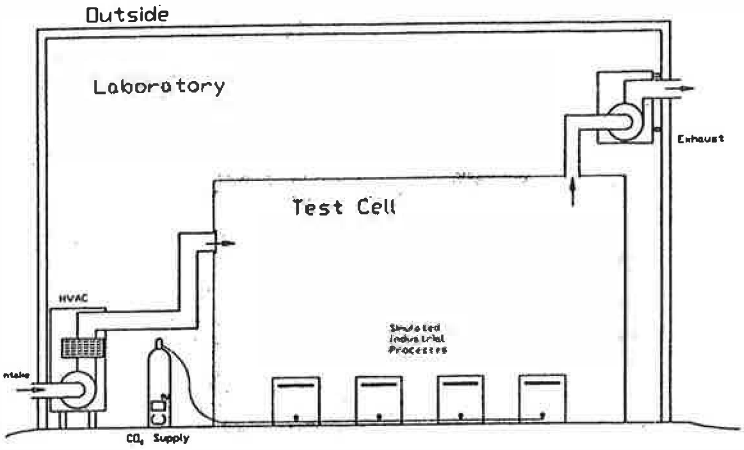
In the research, two different levels of clothing and metabolic activity conditions were chosen and termed **ppd-1** and **ppd-2**. The ppd-1 condition is a typical condition for light industrial work, such as seated assembly or computer work (1.2 met) in which the person is sitting, and wearing a shirt and pants (0.8 clo) . The second condition, ppd-2, is also a common industrial condition. It corresponds to heavier industrial work requiring standing and a higher degree of physical activity (2.0 met), and working in conditions that require an additional gown worn over the shirt and pants (1.3 clo).

DESCRIPTION OF EXPERIMENT

The experiments were performed in an instrumented test room, shown in Figure 1. A one quarter scale experiment was used in this research. The dimensions of the test room are 4.6m by 7.6m with a ceiling height of 2.7m. The test cell corresponds to a full size industrial environment with floor dimensions of 18.2m x 30.4m and a ceiling height 11m. The scale test room's occupied zone, i.e. that space occupied by workers, is the space 76cm above the floor and 15cm from each wall. The HVAC system consisted of an air handler unit, two heaters, and a system of ducts. The system was designed to supply tempered air from the outdoors and exhaust to the outdoors, with no internal recirculation, and to keep the static pressure in the test cell close to ambient pressure. The outside air inlet and exhaust were located on opposite sides of the laboratory building to minimize contamination of the intake flow by the exhaust stream.

The geometries were chosen to provide a simple benchmark comparison of widely used configurations. There were four main ventilation configurations: Single Wall (SW): two wall grilles located on a single wall, Opposite Wall (OW): four wall grilles on opposing walls, Column Drop (CD): two columns in the space, each with two diffusers, and Round (R): two circular diffusers located in the ceiling. The diffuser height arrangements were: High supply/High exhaust (HH), Low supply/High exhaust (LH), and Low supply/Low exhaust (LL). A opposite wall grille, low supply, and high exhaust is shown in Figure 2.

Figure 1: Schematic of test cell, HVAC system, and loads



The industrial processes were modeled with a series of 60 cm sheet metal cubes with interior heaters and a CO_2 source. Carbon dioxide was used as a surrogate contaminant. The cube size corresponded to 2.4 x 2.4 x 2.4 m full-scale processes. Each cube had an open slot on either side to serve as heat and contamination outlets. Measurements of air speed and temperature in the test cell were made in two planes: the diffuser plane and a center plane perpendicular to the diffuser plane. The instrumentation on a movable test stand included four constant temperature omni-directional anemometers, nine thermocouples, and one CO_2 sampling tap. The anemometers measured the mean air speed at small-scale heights of 15, 36, 56, 76 cm. Also at each of these four heights a type-T thermocouple measured the air temperature. The CO_2 concentration sample tap was positioned at a height of 0.38m, equivalent to the 1.5 m height of the full scale breathing zone.

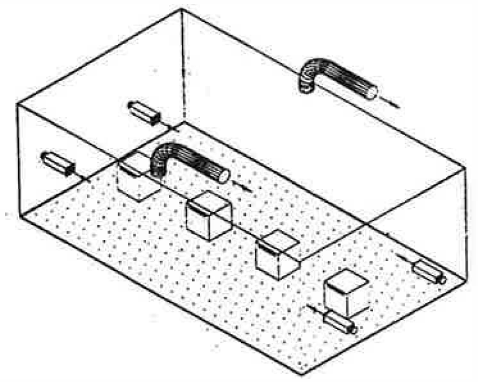


Figure 2: Opposite wall grilles in low supply- high return arrangement

EXPERIMENTAL RESULTS

Forty experiments involving nine to ten ventilation arrangements each with three different supply air flow rates (5, 10, and 15 L/s·m^2) and two different process heat loads (65 and 130 W/m^2) were performed. The nominal supply outlet temperature was 60 F (15.6 C). The temperature distribution for the opposite wall low high case is shown in Figure 3, and for the opposite wall high-high case is shown in Figure 4. In both Figures, the full scale flow rate is 10 L/s·m^2 (2 cfm/ft^2) and the heat load is 130 W/m^2 (40 Btu/hr ft^2), twice the small scale values. For the high-high case the supply jets are fully mixed with the enclosure air above the occupied zone, and for the low-high case the supply jets directly ventilate the occupied zone.

The ventilation and thermal comfort indices: ppd-1, ppd-2, VE are tabulated in Tables 1 and 2 for the ten diffuser configurations for two flow rate and heat load combinations. The ventilation effectiveness shown in Table 1 and 2 is the average of the twenty four ventilation effectiveness measurements taken at the breathing height in the diffuser plane. Also tabulated are the diffuser plane normalized average temperature and average air speed for the occupied zone.

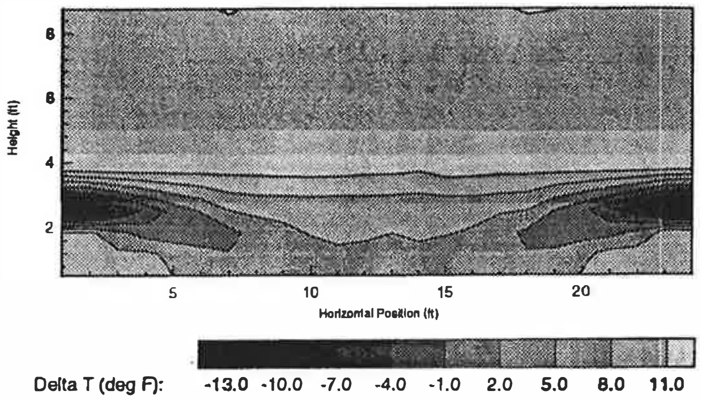


Figure 3: Temperature distribution for opposite wall low-high configuration

The largest ventilation effectiveness occurred with the low-high configuration, with values above 1.6, followed by the high-high configuration with values in the range from 1.0 to 1.2. The low-low configuration had values of about 1.0. The higher VE values for the low-high configuration are due to two factors. The low supply delivers fresh air directly to the occupied zone. The high return causes the ventilation flow to be in the same direction as the natural convection flow from the cubes, so that the mixing of the contaminant with the room air in the occupied zone is reduced.

As the number of supply diffusers increases, the ventilation effectiveness increases. This result is reasonable, since with an increased number of diffusers, fresh air is delivered to more locations in the occupied zone. In general, for a given configuration, the ventilation efficiency increased when the heat load was increased and/or the flow rate decreased. This occurred because the increased heat load created a more intense convection plume above each industrial process. These intensified plumes were able to transport contaminant upward and to the exhaust with less mixing. With increased supply flow rate, there was increased mixing of the supply flow with the process flow resulting in higher contaminant concentrations in the occupied zone.

The thermal comfort results are dependent on the flow rate-load combination. The results are complementary, in the sense that a configuration acceptable for a ppd-1 condition is not acceptable for a ppd-2 condition, and vice versa. Most of the configurations produced acceptable results for the ppd1 condition, and unacceptable conditions at the ppd-2 condition. For example, in the 1-20 tests, all of the diffuser configurations are acceptable for a ppd-1 condition with the exception of the column drop low-low, column drop low-high, and the single wall low-low. However, at the ppd-2 condition, only the column drop low-low and column drop low-high configurations are acceptable. With increased activity and clothing levels, the only diffuser configurations that performed acceptably were those that delivered supply air directly to the occupied zone.

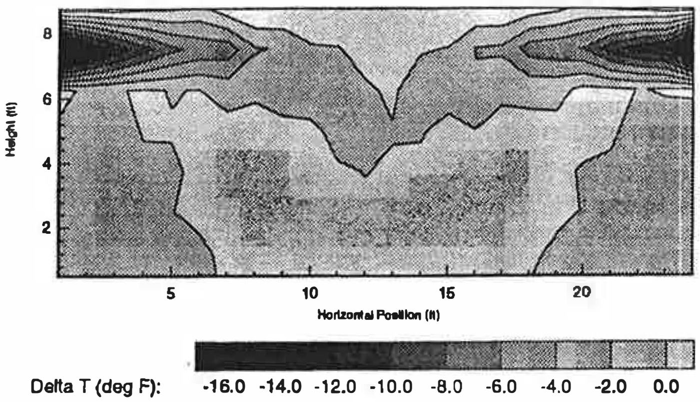


Figure 4: temperature distribution for opposite wall high-high configuration

These results are reasonable, since the ISO thermal comfort model predicts that to maintain thermal comfort as the clothing and activity level is increased, an increase in the airspeed and or a decrease in the occupied zone temperature is required. If the occupied zone temperature is too low and or the air speed too high, the ppd-1 conditions are not acceptable. Likewise, if the occupied zone temperature is too high and or the air speed too low, the ppd-1 conditions are not acceptable. Further information about the experimental results is given in Kirkpatrick and Strobel (1999).

SUMMARY AND CONCLUSIONS

The evaluation of the ventilation effectiveness and thermal comfort for various industrial ventilation schemes has been carried out by scale model experimentation. The largest ventilation effectiveness occurred with the low-high configuration, followed by the high-high configuration. Increasing the number of diffusers in the occupied zone increased the ventilation effectiveness. For a given configuration, the ventilation efficiency generally increased when the heat load was increased and/or the flowrate decreased.

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TABLE 1
RESULTS FOR AIR FLOW RATE =10 L/S M^2 AND LOAD = 65 W/M^2

Diffuser Type	Inlet	Exhaust	T_{oc} °F (°C)	V_{oc} fpm (m/s)	PPD -1	PPD -2	TE	VE
Round	High	High	65 (18)	38 (0.19)	32	17	1.28	1.19
Column Drop	High	High	71 (22)	43 (0.22)	7	35	1.09	1.18
Opposite Walls	High	High	76 (24)	46 (0.23)	9	54	0.99	1.02
Single Wall	High	High	73 (23)	78 (0.39)	6	38	0.95	0.96
Column Drop	Low	High	73 (23)	120 (0.61)	9	36	1.49	1.91
Opposite Walls	Low	High	72 (22)	140 (0.71)	11	34	1.64	1.65
Single Wall	Low	High	68 (20)	170 (0.86)	41	19	0.98	1.01
Column Drop	Low	Low	66 (19)	110 (0.56)	60	15	1.02	1.05
Opposite Walls	Low	Low	68 (20)	140 (0.71)	42	18	1.06	0.99
Single Wall	Low	Low	76 (24)	180 (0.91)	5	38	0.93	0.99

TABLE 2
RESULTS FOR AIR FLOW RATE =10 L/S M^2 AND LOAD =130 W/M^2

Diffuser Type	Inlet	Exhaust	T_{oc} °F (°C)	V_{oc} fpm (m/s)	PPD -1	PPD -2	TE	VE
Round	High	High	76 (24)	50 (0.25)	8	53	0.84	1.20
Column Drop	High	High	76 (25)	50 (0.25)	7	52	1.15	1.15
Opposite Walls	High	High	78 (26)	56 (0.28)	16	63	1.09	1.10
Single Wall	High	High	82 (28)	96 (0.49)	35	76	1.00	1.00
Column Drop	Low	High	69 (21)	120 (0.36)	30	24	1.72	1.84
Opposite Walls	Low	High	73 (23)	150 (0.76)	9	37	1.69	1.84
Single Wall	Low	High	77 (25)	160 (0.81)	6	52	1.25	1.33
Column Drop	Low	Low	80 (27)	120 (0.61)	18	65	1.08	1.13
Opposite Walls	Low	Low	71 (22)	140 (0.71)	15	31	1.05	1.02
Single Wall	Low	Low	78 (26)	170 (0.86)	8	57	0.95	0.97