A newly designed supply diffuser for industrial premises

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Abstract

The results of the investigation revealed the air-flow distribution from the new supply diffuser under non-isothermal conditions. To illustrate the indoor climate parameters in the occupied zone for both heating and cooling seasons, the experimental investigation has been carried out in the industrial premises. Indoor climate was explored at ankle, waist and neck levels for standing person at different positions, which present the variation of the thermal comfort indexes and draught rating (DR) due to changes of measuring point in the facility. The observed PPD and DR values indicate acceptable levels of thermal comfort in the facility for both summer and winter cases. The conclusion can be drawn that well-distributed air-flow save energy by removing the need for an additional heating and cooling system during the cold and hot season.

Key words: Industrial ventilation, Experimental investigation, Indoor climate, Supply diffuser

1. Introduction

The ventilation system is an important process in industrial premises as it is related to energy efficiency usage while maintaining a healthy indoor climate with a good supply of fresh air.

This paper explores indoor air quality strategies and control of flow in facilities at target levels, to provide an environment that is acceptable and does not impair the health and performance of the occupants. The main target levels inside industrial premises are the variations of the air velocity, air-flow rate, temperature, humidity and contaminant concentration, ref. [1].

The objective of this study focuses on evaluation of the realistic case for predictions of the basic features of the indoor climate with the newly designed supply diffuser in industrial premises. The supply diffuser under consideration with appellation Confluent Jets can be described as a number of free circular jets issuing from diffuser apertures at the inlet of the supply device that are fitted on the body of the cylinder. The diffuser is positioned vertically and the jets are directed against a target wall, see Figure 1, 3a. The HVAC system used constant air volume flow (CAV) with total ventilation air flow rate 3.6 m^3/s, keeping a constant inlet temperature during the heating season, while for the cold season the trend of inlet air temperature is mostly based on the outdoor temperature, allowing fast response to changes. It should be noted that no external cooling and heating system has been installed for the facility.

2. Experimental procedure

2.1 Physical model

The facility is located in Tallinn, Estonia with 4 890 m^2 floor area, 81.5 m long and 60 m wide, the highest height is about 7.4 m. The newly designed supply diffuser has been installed in the facility. Figure 2 shows the position of the measuring points for the indoor climate parameters in the occupied zone for both heating and cooling seasons.

Air is supplied through 14 supply devices. Each device includes two supply diffusers. The jets issue from the 9 parallel columns of 58 nozzles of each supply diffuser. The space between nozzles are $s = 3d$, where d is the diameter of the round nozzles. The supply devices are located against the target wall with the side dimension, 0.56 × 0.56 m, the first device is installed at 0.29 m over the floor. The exhaust air devices are placed on the two parallel long ducts along the facility at two different

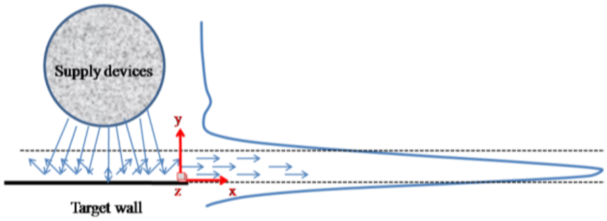

Figure 1. Top view, flow pattern, base on the result of velocity measurement, winter case.

positions at the same level, see Figure 2. The facility has external walls with average *U* - value of 0.3 and 0.25 W/m^{2}K below and above the window respectively, and the interior walls, which connect the offices and the other side of the facility, are insulated with an average *U* - value of 0.6 W/m^{2}K. Fifty-seven windows with *U* - value 1.4 W/m^{2}K, are located on the external walls: the frame are not included in the *U* - value. Three doors 4 m in length and 4.8 in height with the 0.8 *U* - value W/m^{2}K are located in the external walls which are used to export and import the product. Two different types of lamps with 42.8 kW power were illuminated in the facility and 14.6 kW power from different machines for cutting and banding together. The occupant load was from 7:30 a.m. to 5 p.m.

2.2 Field measurement

Thermography is a technique for visualization of the relationship between air temperature distribution and air-flow patterns, which can record an image of radiation emitted from the objects, see ref. [2]. Thermal image was used to register the images of the general flow and temperature field close to the supply air diffuser. Agema S60 (FLIR system) which is sensitive to infrared radiation in range 7.5-13 µm was used with the accuracy of screen surface temperature ±2 °C and with a resolution of 320×240 number of pixels. The IR-camera was calibrated with the manufacture before use. The IR camera was also used to determine the surface temperature of all machines, the storage and the building envelope. Detailed air velocity measurements were logged close to supply devices with the hot wire, the probe was connected to the multi-measuring instrument. Measurement was done at different distances and levels from the floor to capture the velocity profile in two directions of the flow, and for each point the velocity was measured during 3 minutes with a sampling rate 1 Hz with accuracy ±(0.03 m/s + 5% avg mv). The gradient temperature and humidity were measured continuously at two positions in the center of the facility and measurement domain to monitor any increase or decrease during the measurement. Portable data loggers recorded the average value of

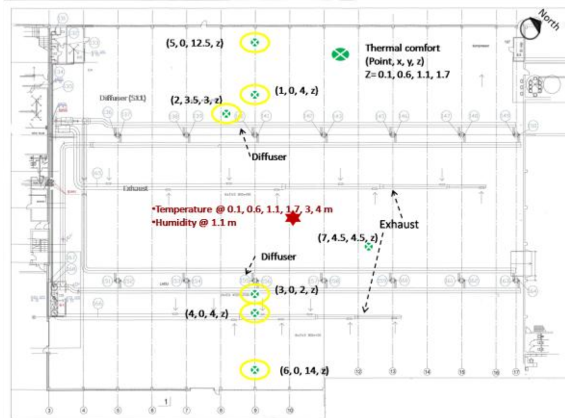

Figure 2. Geometry and position of the measuring points.

temperature and humidity every 15 minutes during the measuring date, which was calibrated at the manufacturer.

The Innova transducers have been used to investigate air velocity, air temperature, humidity and the thermal comfort as well as DR values according to ISO 7730, see ref. [3,4]. The temperature transducers register the results of air temperature and operative temperature with different accuracy ±0.2 °C and ±0.3 °C for range 5-40 °C, respectively. Air temperature was registered with minimal thermal radiation interface from the surroundings. PMV values were calculated directly based on humidity, air velocity, activity level and clothing. Air velocity transducer with accuracy ±(0.05Va+0.05) m/s logged the air velocity based on the constant temperature difference anemometer principle. Humidity was measured for the absolute humidity of air. Average measuring values were recorded every 3 minutes, and to reach the steady state condition, 20 minutes delay was applied between each measurement position. These measurements were made at ankle, waist and neck level for a standing person. The locations of the measuring points were chosen to figure out the thermal comfort in the occupied zone, see Figure 2.

Air velocity and air-flow rate were measured inside the duct for each supply device before starting the measurement that was located in the measuring zone.

3. Case study

Two different cases have been studied in this paper in order to present indoor climate parameters. The measurement results were used to quantify level of thermal comfort and ventilation strategy for cooling and heating seasons. The evaluation of these levels depends on the parameters of the measurements such as air velocity, temperature, and thermal comfort in the occupied zone and near zone of the supply device.

- Summer case, measurement in July with total air-flow rate 3.6 m^3/s from supply devices, and ventilation working hours 02:00-06:00 / 07:30-18:00, see Figure 7a.
- Winter case, measurement in January with total air-flow rate 3.6 m^3/s from supply devices, and decreasing ventilation working hours 07:00 – 16:00, see Figure 7b.

4. Result

4.1 IR image

The supply temperature has been changed during registration of the image of the flow pattern from supply diffuser with infrared camera, which was the easiest way to capture flow from the well-distributed supply devices, A screen with emissivity 0.91 was stretched at the edge of the wall, see Figure 3a, that did not disturb the air-flow pattern and temperature distribution from supply devices, Figure 3 c and d.

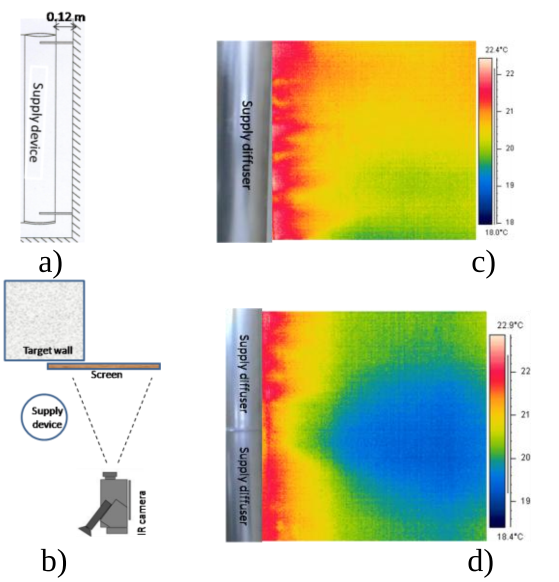

Figure 3. IR image visualization - supply air temperature about 22 °C , air-flow rate 0.25 m^3/s, winter case a) side view b) top view set up c) one supply device d) two supply devices.

4.2 Velocity profiles and characteristic of the jets

Experimental result was carried out with hot wire to measure one dimension velocity field in enclosure; the highest average mean velocity was recorded from the inlet device 8.2 and 8.8 m/s for the summer and winter cases, respectively. Figure 4 shows the, U_x, velocity profile in the y direction. The velocity of interest was registered at 0.8 m height and 0.2 m distance from the edge of the target wall, where the jets strike the target wall.

Figure 5 shows the profile of the local velocity, U_x, and the air temperature, T, as a function distance from the floor in the outward direction of the flow at the middle distance between the supply device and the target wall, i.e. distance in y direction and x direction is 0.06 m and 0.2 m, respectively from the edge of the target wall. The results were registered at different vertical heights from the first nozzle at the top of the lower diffuser to the last nozzle near the floor. Velocity profile at a certain distance from the supply devices explores the behavior of the jet as the confluents jets phenomena, see ref. [5]. Vertical air temperature gradient almost seems constant due to dominating region by issuing jet.

Two different plots depict the empirical expression of the decay measured velocity for the summer and winter cases. For comparison the two cases were carried out at the same level from the floor. The same behavior of the plane wall jet was explored by the measured velocity profile, see Figure 6. As shown in the Figure 6 the highest velocity has grown at one-half meter from diffuser center with a slight and normal decrease of jet average velocity thereafter.

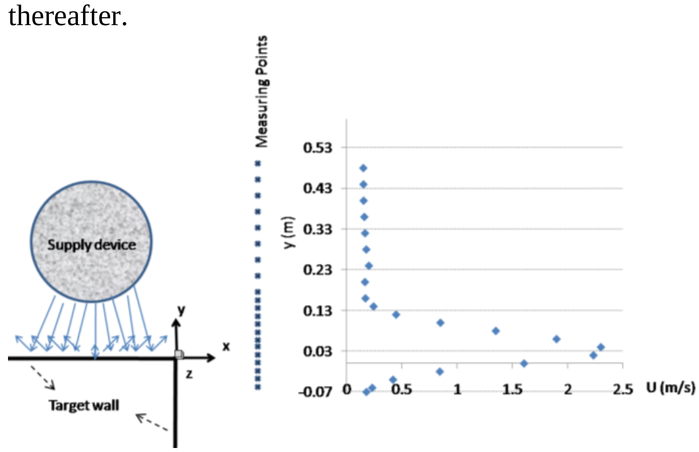

Figure 4. Profile of x velocity component in y direction, winter case.

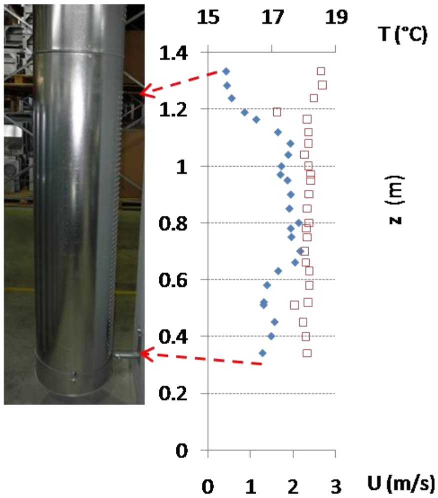

Figure 5. Profile of x velocity component and air temperature in z direction, winter case.

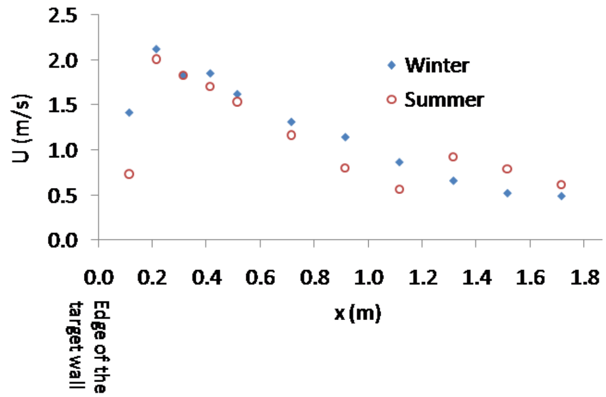

Figure 6. Decay of x velocity component in x direction at height 0.8 m from the floor.

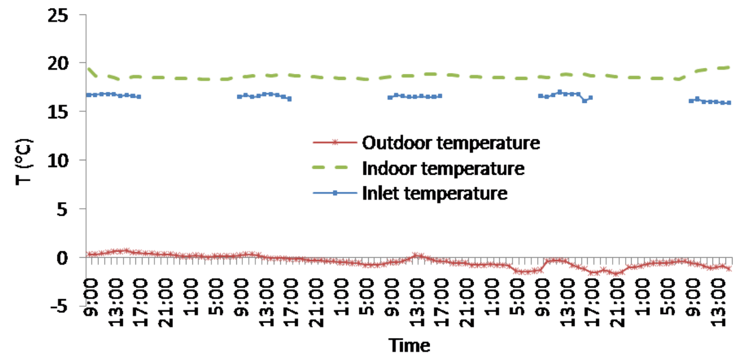

Figure 7b. Measured outdoor, indoor and inlet temperature, winter case.

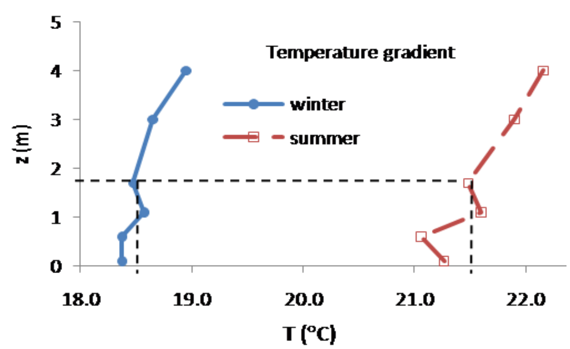

Figure 8. Temperature gradient in the center of the facility.

4.3. Indoor climate analysis

This study includes two climates extremes, winter and summer cases; Figure 7a and b. shows the indoor temperature profiles during 5 days measurement in the facility. In the summer case, in order to provide comfortable indoor environment before workers come to the facility, the supply inlet temperature was adjusted by using outdoor air from 02:00 a.m. to 06:00 a.m., see Figure 7a. This control strategy both save energy and provides proper indoor environment. During the measurement, the mean temperature was measured continuously at heights 0.1, 0.6, 1.1, 1.7, 3 and 4 m from the floor. Average temperature gradient and relative humidity observed 0.8, 1.1 °C and 29, 55 % for the winter and summer cases, respectively, during working hours for measurement days in the center of the facility, with the relative humidity sensor placed at height 1.1 m from the floor see Fig. 8.

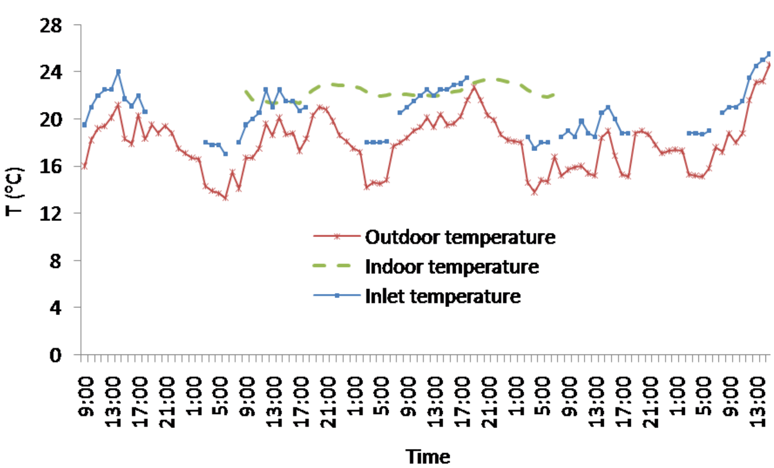

Figure 7a. Measured outdoor, indoor and inlet temperature, summer case.

Figure 9. The measurement configuration, point 3.

Table1. Set up for supply devices.

	Summer case		Winter case	
Measuring point	1,2,5	3,4,6	1,2,5	3,4,6
Q* (m^3/s)	0.283	0.250	0.311	0.270
U* (m/s)	3.64	3.21	4.0	3.5
T* (°C)	22.1	21.9	17.4	17.2

* Q, U, T are air-flow rate, velocity and inlet temperature inside the supply devices

Table 2. Average indoor climate parameters for different heights for all measurement points (1, 2, 3, 4, 5 and 7) .

	Summer case			Winter case		
Measuring point	*Ankle*	*Waist*	*Neck*	*Ankle*	*Waist*	*Neck*
u_l (m/s)	0.28	0.25	0.32	0.30	0.20	0.14
T_l (°C)	21.9	21.7	21.7	18.3	18.6	18.7
DR (%)	41	30	34	48	31	20
PMV (-)	0.8	0.8	0.8	0.2	0.3	0.3
PPD (%)	20	18	19	7	7	8

Ankle (0.1 m), waist (1.1 m) and neck (1.7 m) from the floor

Table 3. Average indoor climate parameter for each measurement point for all heights (ankle, waist, neck), summer and winter cases.

	Summer case						Winter case					
Measuring point	1	2	3	4	5	6	1	2	3	4	5	6
u_l (m/s)	0.21	0.14	0.75	0.27	0.12	0.10	0.18	0.05	0.66	0.27	0.03	0.12
T_l (°C)	21.4	21.4	21.9	21.8	21.9	23.2	18.5	18.6	18.3	18.6	18.3	18.18
DR (%)	27	13	94	49	11	7	32	5	98	55	0	17
PMV (-)	0.8	0.8	0.6	0.7	0.9	1.1	0.2	0.4	0	0.2	0.3	0.3
PPD (%)	17	20	13	17	24	30	6.8	7.8	6	6.7	7.1	6

Ankle (0.1 m), waist (1.1 m) and neck (1.7 m) from the floor

In order to estimate thermal comfort of the personnel, the measurements were carried out for working hours in the facility. Two levels of clothing insulation were setting to a distinct winter and summer 0.9 and 0.8 clo, respectively. It should be note that medium activity for the standing person was, 2 met for both cases. The measured boundary condition could be seen in Table 1.

The comparison of average thermal climate value for different points at different levels, shown in Table 2, remains in an acceptable region for the waist and neck of the standing person for the winter case and with small difference for the summer case. An explanation for higher average DR value at ankle position is due to air in the cold air-flow stream. The measuring points have been chosen as shown in Figure2.

Table 3 illustrates variation of average thermal climate at different height ankle, waist and neck level (see ref. [3]) for a standing person at different positions in the facility.

In Table 2, 3 and Equation (1), u_l and T_l are the local mean air velocity and local air temperature, respectively. The DR has been chosen to represent the percentage of people predicted to be dissatisfied in the occupied zone

$$DR = (34 - T_l)(\overline{u_l} - 0.05)^{0.62}(0.37\overline{u_l}Tu + 3.14), \quad (1)$$

Where Tu is the turbulence intensity (%)

The PMV values have been explored by measuring temperature, air velocity and relative humidity that give rise to PPD index (predict percentage of dissatisfied) as the following equation:

$$PPD = 100 - 95 \cdot \exp - \left(0.03353\, PMV^4 + 0.2179\, PMV^2\right) \quad (2)$$

Measurement around two different supply devices shows that at a farther distance from the diffuser (comparison between points 3, 4 and 6), the PMV and PPD values increase for the summer case, the supply air cannot reach far from the diffuser, but the results are acceptable for the winter case. It is obvious that the mean velocity and draught reduce by increasing the distance from the supplied devices, see Table 3. Max values of mean velocity and draught occurred at 2 m from supply device. The results represent a temperature rise of about 1 °C for the summer case and slight rise for the winter case by comparing at increasing distances. By looking at winter case it seems there is not too much variation of PPD and PMV between different points of measurement.

5. Conclusion

The new proposed air supply device provides good indoor thermal comfort conditions in the occupied zone by creating a well-distributed air-flow. The supply devices also saves energy by removing the need for any additional heating and cooling systems during the cold and hot season. As a consequence of the thermal comfort values observed in the facility at different positions for the winter and summer, the comfort indexes (PMV, PPD) remain in an acceptable range according to ISO 7730, with the results of the thermal environment becoming more comfortable for the winter. The evaluation of the DR was also good according to ISO 7730 far from the diffuser or not in the direction of the flow.

Average temperature difference between ankle and neck level for the standing person is about 0.2 and 0.4 for the winter and summer case, respectively.

For both summer and winter cases, the result of the velocity profile issuing from the non-isothermal confluent jets has explored the same behavior of the plane wall jet for the zone of special interest, which is most typical for a working place.

6. Acknowledgment

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