Impact of heat load distribution and strength on airflow pattern in rooms with exposed chilled beams

Risto Kosonen[1], Arsen Melikov[2], Boryana Yordanova[2] and Lyuben Bozkhov[2]

[1]Halton Oy
[2]International Center for Indoor Environment and Energy, Department of mechanical Engineering, Technical University of Denmark

Corresponding email: risto.kosonen@halton.com

SUMMARY

The impact of the heat load strength and distribution on the performance of lengthwise installed exposed chilled beam was studied in a test room. The cooling capacity varied between 56 W/m^2 - 111 W/m^2, which are typical cooling loads in office buildings. In addition, two experiments at 84 W/m^2 were performed with all of the heat sources close to the window side of the room and with the heat sources split on both sides of the room. The results from the physical measurements and the performed smoke visualization showed that the flow from the chilled beams was not disturbed with the heat load of 56 W/m^2. However higher loads depicted significant impact of the convective flow from the heat sources on the distribution of the air in the room. The maximum velocity increased from 0.20 m/s to 0.26 m/s when the cooling capacity increased from 56 W/m^2 to 111 W/m^2.

INTRODUCTION

The local air velocity, temperatures of the room air and air jets, and fluctuations in air velocity are the key factors that determine occupants' thermal comfort in general and in particular the draught discomfort. In this respect the design of ventilation systems focuses on evaluation of the maximum velocity in the occupied zone which can be obtained under different sollutions. The aim is to achieve velocity lower than the maximum velocity recommended in the present standards and guidelines (ISO 7730 1994 [1], ASHRAE 55 2004 [2], EN CEN 1752 1998 [3])

The effect of the convection flows on the flow range in the room is usually ignored and the assessment focuses only on the flow supplied to the room from the air terminal device. However, the convection flows caused by heat sources may significantly affect the supplied flow as well as the airflow distribution in the whole room.

This article addresses the effect of thermal loads on the operation of an exposed chilled beam system. The impact of the heat load strength and distribution on the performance of an exposed chilled beam was studied based on measurements of the room air distribution.

METHODS

Experimental design

Two sets of experiments were designed in order to study: 1) the impact of the heat load strength on the air distribution in the room and 2) the impact of heat load distribution on the airflow characteristics in the occupied zone.

Experimental facilities

The experiment was conducted at Halton Research Centre, Finland. A test room (L *x* W *x* H = 6.5 m *x* 2.75 m *x* 3m (H)) was used to simulate a real office. Two active chilled beams, type CBQ-A-2500/2200 for exposed installation were installed in series as shown in Figure 1. The chilled beams were installed at the height of 2.5 m. The inlet water temperature was 14 ºC. The mass flow rate of the water for one beam was in range of 0.03 to 0.1 kg/s. The primary airflow rate was 2.0 l/s per m^2 and supply air temperature was 18^oC.

Real computers (123 W per computer) were used to simulate the heat load from office equipment. A window with size of 1.3 m x 1.9 m (H) was simulated by heated/cooled panels. In most of the experiments the heat load from the window was 350 W (surface temperature of 41 ^oC) except one experiment when the window was cooled to produce -144 W effect (surface temperature 20 ^oC). Heat gain from solar radiation was also simulated by heating 5.5 m^2 area of the floor close to the window by electric panels (300 W). Lightning was integrated into the chilled beams and gave heat gains of 140 W. The heat generated by occupants was simulated with dummies (120 W each).

The number of the used dummies was changed during the experiments when the impact of the heat load was studied. The experimental conditions for during these experiments are listed in Table 1 and Table 2. During these experiments, the cooling capacity was varied between 56 W/m^2 - 111 W/m^2 by balancing the number of dummies and the heat gain of the simulated solar load (Table 1.). The experiments with the impact of heat load distribution were performed only at one heat load level in the room, 84 W/m^2 as specified in Tables 1 and 2.

Table 1. Distribution of heat loads in the test room

Heat loads W/m²	Heat gain from windows, W	Heat gain from PCs, W	Heat gain from dummies, W	Heat gains from floor panels, W	Heat gain from lightning, W
56W/m²	-144	3x123	3x120	300	140
84W/m²	350	3x123	3x120	300	140
98W/m²	350	3x123	5x120	300	140
111W/m²	350	3x123	7x120	300	140

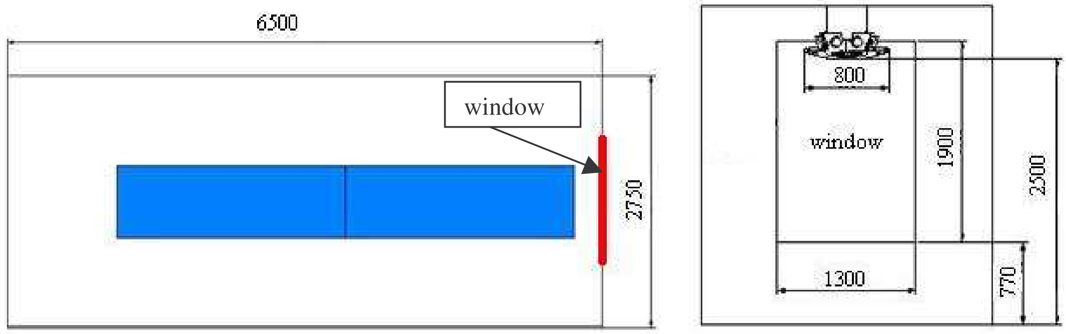

Figure 1. Test arrangement used to study the effect of convection flow on an exposed beam.

Table 2. Experimental conditions in the test room

Heat loads W/m²	Airflow rate, l/s per m^2	Temperature in the room, ºC	Surface temperature of window, ºC
56W/m²	2	23.5	20
84W/m²	2	24.1	41
98W/m²	2	24.3	41
111W/m²	2	24.9	41

Mean air velocity, air temperature and turbulence intensity were measured at 15 locations within the occupied zone at the heights of 0.1m, 0.2m, 0.5m, 1.1m, 1.5m and 1.8m above the floor, e.g. at 90 measurement points (Fig. 2). Smoke visualization of the airflow pattern was performed during the experiments.

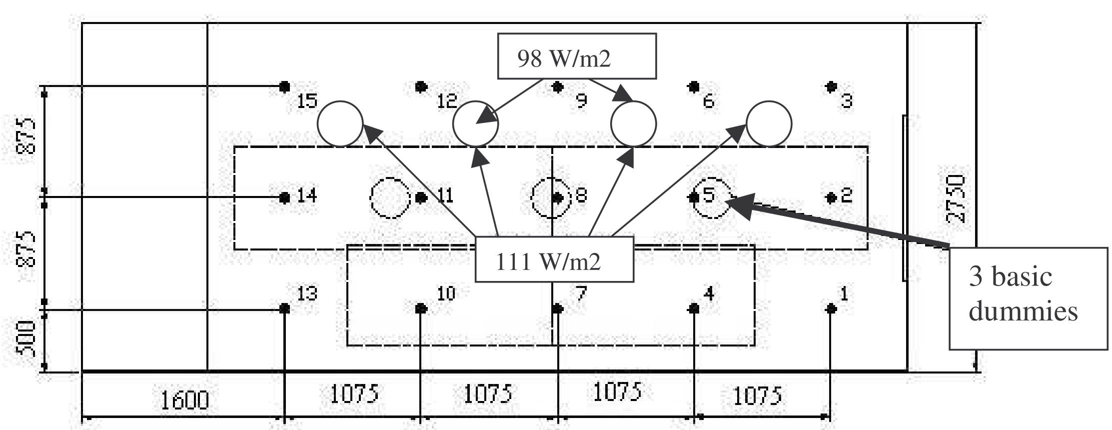

Figure 2. Measurement grid and the location of dummies.

Air flow velocities were measured with 8–channel low velocity anemometer HT-400 with omnidirectional velocity sensor, type sensor HT 412. The used 6 omni-directional velocity probes have spherical velocity sensor with a diameter of 2 mm, ensuring a fast response. The temperature sensor is shielded against radiation. Instantaneous values of velocity and temperature are measured simultaneously. Measurements of velocity and temperatures were time averages over 180 s. The characteristics of the instrument complied with the requirements in the standards (ASHRAE 113 2005 [4],ISO 1993 [5], ISO 1998 [6]). The room air temperature, inlet water temperature, supply and exhaust air temperatures were measured with temperature sensors of type PT100 class A. The water flow rate was measured with Krohne Electromagnetic Flowmeter IFC010 with accuracy less than ± 1% of the readings. The airflow rate was measured with differential pressure transmitter Furness Controls FCO33 with accuracy less than ± 0.5% of the readings.

RESULTS

Impact of the heat load strength

The impact of the heat load strength was studied by changing the number of the used dummies. The position of the dummies was shown in Fig. 2. The results from the physical measurements and the performed smoke visualization shown in Figure 3 reveal that the flow from the chilled beams was not disturbed significantly by the convection flows in the experiment with the lowest heat load (56 W/m²). For the experiments with heat loads 84, 98 and 111 W/m² a significant impact of the convective flow from the heat sources on the distribution of the air supplied to the room was identified. The higher heat load from the simulated window at the right side of the test room created strong convective flow (Fig 3).

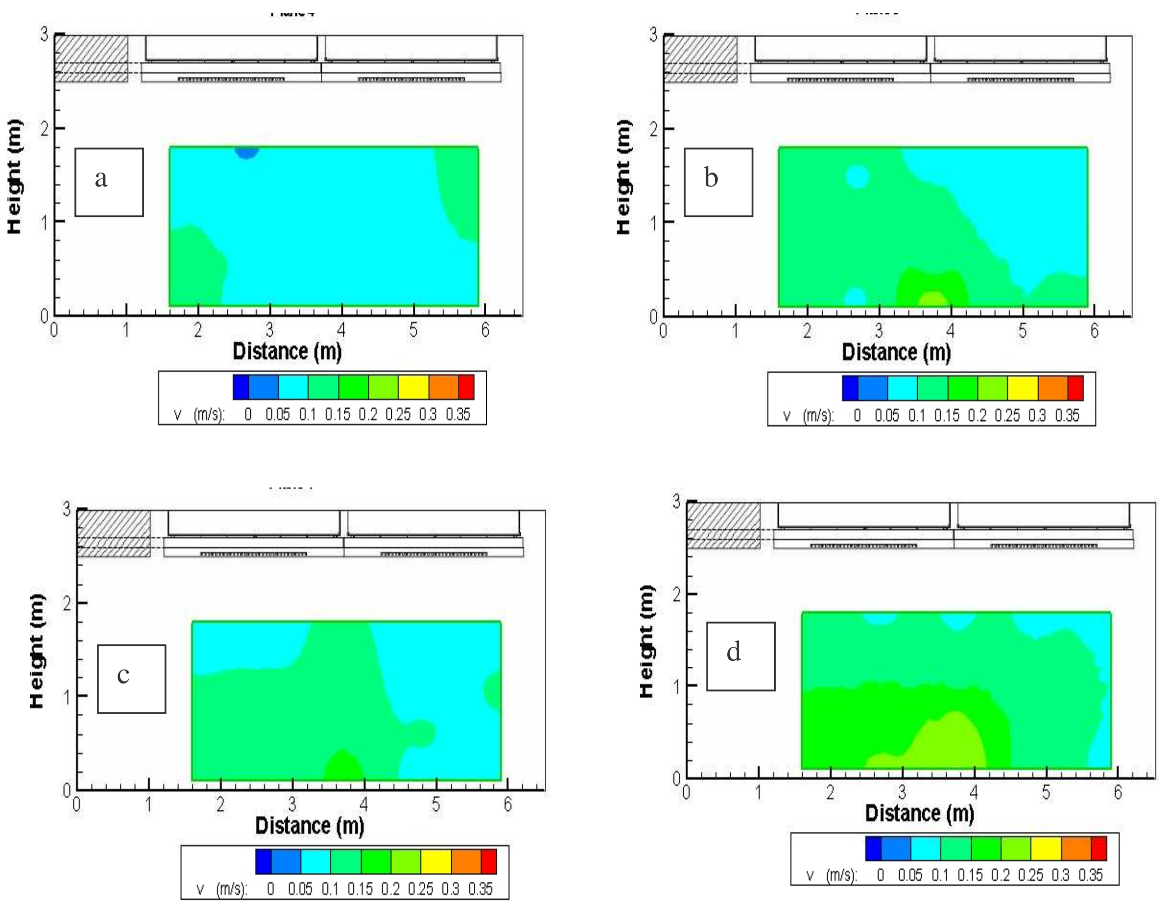

Figure 3. Velocity profile in a vertical plane across the middle of the test room: a) heat load of 56W/m², b) heat load of 84W/m², c) heat load of 98W/m², d) heat load of 111W/m²

In Table 3, the results for the average, maximum and minimum values of the temperature, velocity and draught rating in the occupied zone are presented. The values are obtained from all of the measurement location over the occupied zone.

Table 3. Minimum, maximum and average temperatures, velocities and DR-index .

Heat loads (W/m^2)	v_{max} (m/s)	v_{min} (m/s)	v_{avg} (m/s)	T_{max} (ºC)	T_{min} (ºC)	T_{avg} (ºC)	DR_{max} (%)	DR_{min} (%)	DR_{avg} (%)
56 W/m^2	0.20	0.04	0.096	24.0	22.9	23.5	15	2.5	7.7
84 W/m^2	0.24	0.05	0.098	24.5	22.9	23.7	18	2.3	7.8
98 W/m^2	0.22	0.04	0.098	24.9	22.8	24.0	18	0	7.8
111W/m^2	0.26	0.05	0.11	25.0	23.1	24.1	19	0	9.1

The results in the table show that there was significant increase of the maximum velocities in the occupied zone of the room when the heat load increased. Similar result was observed for the maximal temperatures. However, there was no or very small increase in the maximum and average draught rating. In all cases studied, the air velocities were not too high (maximum 0.26 m/s). There was only couple of location with the highest cooling capacity (111 W/m^2) where the air velocity was higher than the value of 0.25 m/s recommended in the standards for summer conditions. In practice it should be possible to adjust the air flow rate supplied from chilled beam, which will decrease the velocity in the occupied zone and thus will reduce the risk of a draught.

Impact of heat load distribution

The effect of the heat load distribution on the airflow field in the occupied zone was studied at 84 W/m^2 and outdoor air flow rate of 2 l/s per m^2 (see Table 1 and Table 2). Three different thermal load distributions were applied as shown in Figure 4: 1) normal office room with thermal load on the window and side walls,;2) 50% on the window wall and 50% on the corridor wall; and 3) all load on the window wall.

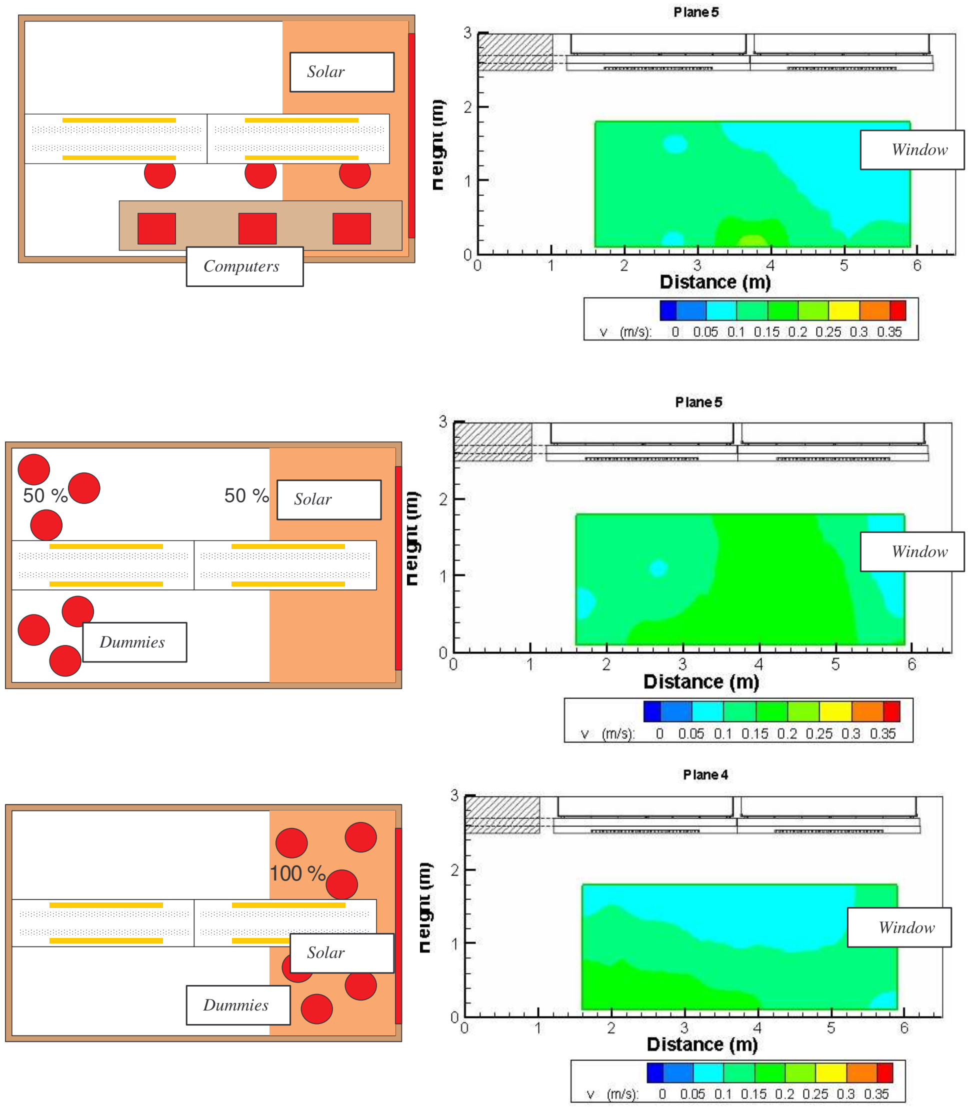

Figure 4. Velocity profiles with different heat load distribution (84 W/m^2).

In all cases studied, the air velocities were relatively low (less than 0.24 m/s). The highest air velocities occurred in cases where the load had been altered from that of a normal office. By locating the heat sources close to the window wall, the maximum velocity was found in the proximity of the corridor wall, while, when the heat sources were evenly distributed between the window and corridor walls, the maximum speed occurred in the middle of the room. The performed smoke visualization further depicted the measured results. Figure 5 presents visualization in the case when the heat sources are located close to the window wall.

Figure 5. Smoke visualization when all heat sources are located closet to the window wall.

Overall in the measured cases, the location of the thermal load did not significantly affect the maximum velocity. However, the location of the maximum velocity is strongly depending on how the loads are located in the room space.

DISCUSSION AND CONCLUSIONS

The local air velocity, temperatures of the room air and air jets, and fluctuations in air velocity are the key factors that determine the risk of a draught. Indoor climate standrads recommend maximum allowable velocity in the occupied zone needed for occupants' comfort. Therefore in designing an indoor environment, it is important to evaluate different solutions in regard to the maximum velocity in the occupied zone of rooms. This is particularly important for office buildings and on other business premises with relatively high requirements for cooling capacity.

It is still not acknowledged during the design of indoor environment that convection flows caused by heat sources may significantly affect the flow distribution in rooms. Most often the focus is only on the flow generated by the air supply terminal devices. However as the present results show the point of occurrence of the maximum velocity in the occupied zone depends on the heat sourse streangth and its distribution in the room, as well as on the flow pattern of terminal device. Thus, the air flow interaction in rooms ventilated with chilled beams is of great importance for occupants' comfort.

The results from the physical measurements and the performed smoke visualization showed that the flow from the chilled beams was not disturbed with the heat load of 56 W/m^2. However higher loads depicted significant impact of the convective flow from the heat sources on the distribution of the air in the room. The maximum velocity increased from 0.20 m/s to 0.26 m/s when the cooling capacity increased from 56 W/m^2 to 111 W/m^2. With the conducted installation, the room air velocity was within the acceptable level (below 0.24 m/s) when the cooling capacity was below 100 W/m^2. Higher cooling capacity increases the draught risk and local velocities in the occupied zone. Experience shows that at high cooling load it may possible to adjust the air flow rate supplied from chilled beam, which will diminish the maximum and average velocity in the occupied zone and thus will reduce the risk of a draught.

Simple analytic modelling of the jets and thermal loads that affect the air flows of a room space has limitations due to the complex interaction of the flows in rooms. Therefore, full-scale tests and CFD predictions are recommended. The design tools currently available enable studying the flow ranges of room devices but without taking account of the effect of the convection flows from thermal loads. It should be noted that there are no standardised methods for presenting air velocities, and each supplier has its own way to present velocity field in the occupied zone. This makes it difficult to compare the performance of different air supplied devices and installations.

ACKNOWLEDGEMENT

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REFERENCES

1. ISO 7730, Moderate thermal environments- Determination of the PMV and PPD indices and specification of the conditions for thermal comfort, 1994
2. ASHRAE Standard 55-2004: Thermal Environmental Conditions for Human Occupancy, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.
3. CEN CR 1752-1998 – "Design for Buildings – Design Criteria for the Indoor Environment", European Committee for Standardization, December 1998.
4. ANSI/ASHRAE Standard 113-2005: Method of testing for room air diffusion. American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA.
Vol.25, No.2, pp. 127-132.
5. ISO Guide to the Expression of Uncertainty in Measurement, First Ed. International Organization for Standardization, Geneva, Switzerland, 1993, ISBN 92-67-10188-9.
6. ISO/DIS/7726 International Standard: Ergonomics of the thermal environment - Instruments for measuring physical quantities. International Organization for Standardization, Geneva, Switzerland, 1998.