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Evaluation of thermal comfort criteria of an active chilled beam system in tropical climate: a comparative study

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ABSTRACT

Air-conditioning systems are employed to provide thermal comfort inside buildings. Therefore, several air-conditioning and mechanical ventilation (ACMV) systems have been tested and applied in different climates. Chilled beam systems are a new addition to it and have been in the application in Europe and North America since last two decades. In the present study, comparative performance of active chilled beam (ACB) system with conventional fan coil unit (FCU) system has been appraised in regards to thermal comfort for the tropical region of Singapore. Experiments were carried out in a simulated office building with approximately 80% glazed area under real environmental conditions. Room air temperature, air velocity, mean radiant temperature (MRT) and relative humidity which are important thermal comfort parameters were measured in perimeter zone inside the cells and assessed. It was perceived that the ACB system was providing acceptable thermal comfort as per the standard of SS 554. General thermal comfort has been assessed based on an index of PMV-PPD model and graphical method. With regard to the local thermal discomfort: indices of air draught, vertical air temperature difference (VATD) and radiant temperature asymmetry have also been evaluated. The entrance of solar radiation through window façade may have a significant impact on the asymmetry of the radiant field and can enhance thermal discomfort of the occupants. Therefore, in the study radiant temperature asymmetry has been calculated extensively. The results show that ACB system yields satisfactory thermal environment as per ISO 7730 standard and sometimes performs even better than conventional FCU system.

Keywords: Active chilled beam, conventional fan coil unit, PMV-PPD model, Radiant asymmetry, Thermal comfort, Vertical air temperature difference

1. Introduction

Air-conditioning and mechanical ventilation (ACMV) systems uphold a comfortable and healthy environment inside buildings. A recent addition to the several ACMV systems available in the market, chilled beam technology is gaining popularity due to its positive outcomes concerning energy conservation, thermal comfort and reduced cost. Since the last two decades, chilled beams are widely used in Europe and North America [1-3].

An Active Chilled Beam (ACB) receives pre-treated (primary) air through the air handling unit (AHU) and chilled water (CHW) flow into the cooling coil within the terminal unit mounted on the ceiling. The schematic of an ACB terminal unit is represented in Fig. 1 to understand its working principle. The primary air is injected through a series of nozzles within the beam to condition the room air. Because of low local static pressure region created near the nozzle outlet, the room air rises upwards. The secondary (induced) air takes away the sensible load of the conditioned zone while passing over the cooling coils. It then mixes with the primary air in the mixing chamber, and this mixed air then discharges back through the slots to the zone. The ACB terminal units mounted on the ceiling maintain a sufficient discharge velocity to ensure a well-mixed room air distribution. The primary air is responsible for the removal of the latent heat load.

In literature, the authors have validated the performance of the ACB system [4-5]. Filipsson et al. [6] developed a numerical model of the ACB system to compute its cooling capacities. The model distinctly includes the effect of buoyant forces acting on the cooling coil in the beam. The model was calibrated with measured data and showed good agreement in a broad range of working conditions.

In terms of energy-savings, ACB systems can achieve savings of up to 20-30% related to traditional air conditioning systems, depending upon different locations [7-9]. Betz et al. [10] reported the issues emerging while modeling the ACB system using various simulation tools. It was asserted that factors like induction ratio, manifold chilled water loops, humidity regulation and ACMV system configurations should be taken into consideration while doing simulations to attain accurate results. It was suggested by Stein and Taylor [11] that the ACB system designed for low supply air discharge rate along with intermediate CHW temperature, results in improvement in its energy performance compared to variable airflow with reheat systems. Authors also argued that the ACB system performs well under high sensible load.

Kosonen and Tan [12] conducted an experimental study to explore the usefulness of the ACB system in hot and humid climate. It was shown that condensation in the beam system could be prevented when infiltration is low, and supply air discharge is ample to remove humidity caused by occupants. Loudermilk and Alexander [13] suggested that chilled beams can be better alternatives compared to air-water distribution systems when it comes to humidity controls. It was also concluded that humidity and condensation risk could be overcome using chilled beams in a humid climate.

Invariable temperature, ample rainfall and high humidity all over the year are the main characteristics of tropical climate [14]. In Singapore, which is located near the equator, the diurnal limit of temperature lies between 25°C to 33°C. The diurnal range of relative humidity (RH) range is around 60% in the middle of the afternoon and could be as high as 90% in the early morning. The typical dew point temperature is near 24°C, and the absolute humidity is 19.5 g/kg in the daytime and 18.5 g/kg in nights all over the year [12, 15]. Cooling and dehumidification is a challenge in the tropical climate. For this reason, it is essential to maintain the RH together with room temperature.

In the view of thermal comfort, it has been reported that similar or better thermal environment can be produced by the ACB system, related to the other conventional air-conditioning systems [7]. Authors have assessed general thermal comfort in perspective of comfort indicator like predicted mean vote (PMV) and; it has been found that it can assure satisfactory thermal environment. However, in addition to general thermal comfort, it is also essential to assess local thermal discomfort, in terms of air draught, vertical air temperature difference (VATD), etc. [16-17].

Zboril et al. [18] conducted an experimental study in a test-bed facility consisting of multiple chilled beams under induced airflow rates and varied heat loads and investigated the draught rate and VATD. The result indicated that the draught rate was below 20% and VATD was below 3°C for all test conditions, which conforms to the thermal comfort criteria. Koskela et al. [19, 20] carried out an experimental study with test-bed facility exposed in a real environment and multiple numbers of ACBs. In the study, tests were carried out with varied cooling loads replicating summer, spring/autumn and winter conditions, and for which airflow patterns and mean airspeeds were investigated. The results showed that the internal cooling loads could impact the airflow patterns and draught possibilities.

In order to investigate the consistency of the thermal environment produced by the ACB system, a test-bed facility was developed by Rhee et al. [21]. A comparative study was made between traditional air-conditioning systems and the ACB system. The performance indices of air diffusion parameter index (ADPI), air velocity, and VATD were adopted in order to study the consistency in thermal environment of the ACB system. The results indicated that uniform thermal environment could be produced by the ACB system with low air discharge rate through AHU compared to traditional systems.

The present study has been carried out in the tropical region of Singapore. The objective of this study is to evaluate thermal comfort characteristics of the ACB system and compare it with a conventional FCU system. Experiments have been conducted using a test-bed facility, representative of an office building with about 80% glazed area, exposed to real outdoor conditions. In order to assess general thermal comfort, indices of predicted mean vote – people percentage dissatisfied (PMV-PPD) and graphical method have been used. The indices of air draught, VATD along with radiant temperature asymmetry have been adopted to evaluate local thermal comfort. The impact of solar radiation on radiant temperature asymmetry has also been assessed extensively to study their influences on indoor thermal comfort.

2. Experimental Set-up

In order to assess the thermal comfort criteria of ACB system compared with the conventional FCU system, each of these systems was installed in two cells of the test-bed of BCA SkyLab, Singapore. Conventional FCU and ACB system were employed in the reference and test cell, respectively. The schematic of the test-bed facility is presented in Fig. 2. Both cells are a replica of each other, and their dimensions are 8.41 m (L) $\times$ 5.54 m (W) $\times$ 3.47 m (H) up to the ceiling. The orientation of the cells can be changed as the test-bed facility is constructed over a rotating platform.

The supply and return air schematic of the conventional FCU and ACB systems are depicted in Figs. 3 and 4, respectively. The conventional FCU system employed an AC-motor FCU to provide air-conditioning in the reference cell. The CHW supply temperature was constant at 7.9°C, but the CHW flow rate was varied as per the cooling demand. The modulation of CHW flow rate was controlled by a thermostat (Flaktwoods STRA 17, Accuracy $\pm 0.05^\circ\text{C}$ at 15 – 30°C) installed in the cell. The conditioned air was supply through four 4-way spread type ceiling mounted air diffusers. Room air was returned through six ceiling mounted air return

grills. The locations of the supply air diffusers and return air grills are shown in Fig. 5 (a). The ACB system employed a DC-motor FCU to provide air-conditioning in the test cell. The conditioned air was supplied through six chilled beams mounted on the ceiling. Temperature of the supply air was controlled by modulating the CHW flow rate through the FCU. The ACB system was divided into three zones, each consisting of two ACB terminals as shown in Fig. 5 (b), which enabled zonal control of room air temperature. The air temperature in each zone was controlled by modulating the CHW flow rate through the two ACB terminals of the zone. The modulation of ACB CHW flow rate was controlled by a thermostat (Flaktwoods STRA 14, Accuracy $\pm 0.05^{\circ}\text{C}$ at $15 - 30^{\circ}\text{C}$) installed in each zone. The inlet CHW temperature for the ACB was controlled at 17°C (higher than the room dew point temperature) to avoid condensation. The CHW for ACB terminals was supplied through a secondary loop which was separated from the primary loop that supplied to the FCU by a heat exchanger (Alfalaval model no.: M3-FG). Design conditions of both ACMV systems have been summarized in Table 1.

The plug loads (due to computers, other equipment), occupancy load in both cells were balanced. These loads are summarised in Table 2. For human occupancy load, four thermal dummies and two humidifying devices were installed in each cell (as presented in Fig. 5 (b)) to represent the sensible and latent loads of four people. The lighting load may vary depending on the arrangement in different Tests, as summarised in Table 2. In tests that involve auto dimming lighting, the lighting power level was controlled by a Digital Addressable Lighting Interface (DALI) [22] system.

For the evaluation of thermal comfort criteria of conventional FCU and ACB systems, a set of experiments were conducted as summarized in Table 3. Tests 1 and 2 were conducted to compare the FCU and ACB system without influences from window blinds (all retracted) and variable lighting (fixed at 100% power). For Tests 3 and 4, more parameters were taken into account. In the test cell, the Venetian window blinds were lowered and the angles of the motorised slats were controlled automatically by an in-house developed control algorithm with the main purposes of keeping glare to below 0.35 in Daylight Glare Probability (DGP) index (imperceptible range) and to maximise the daylight penetration into the room. The building management system (BMS) monitored the real-time sky conditions based on a roof-top illuminance sensor (Reinhart, light intensity sensor) and façade orientation inputs as it executed the algorithm. West and North window façade orientations were covered in the experiments to

study the impact of orientations. Each experiment was conducted from 9:00 am to 6:00 pm for three days.

For thermal comfort measurement, air temperature sensors (Precon, ST-S3EW-XPA) were mounted at different height levels (as per ISO 7726 [23]) of a stratification tree. At the 1.1-m height level of the stratification tree, an air velocity anemometer (Swema, Swema 03), a globe temperature sensor (Kimo, TM110) and an RH sensor (Vaisala, HWM90) were installed (Fig. 6). The stratification tree was installed at the seating position of one of the two workstations that are closer to the window (Fig. 5). The measurement position represents the occupant sitting in this workstation, which is more susceptible to the thermal stress due to heat gain through the window compared to other workstations that are further away from the window. The global and diffuse solar radiation at the rooftop (on the horizontal plane) were also measured during the experiments using a pyranometer (AT Delta-T Devices, SPN1). All sensors were connected to an on-site data acquisition system, recording readings at one minute interval. The technical specifications of the sensors are listed in Table 4.

3. Evaluation of thermal comfort criteria

Thermal comfort can be impacted by a number of parameters including clothing level (I_{cl}), metabolic rate (M), room air temperature (T_a), air velocity (V_a), RH and mean radiant temperature (MRT) [17]. Damiani et al. [24], suggested that the clothing level of 0.57 clo is representative of offices in Singapore. The metabolic rate of 1.1 met was taken, for an occupant sitting and typing as described by ASHRAE handbook [25]. MRT was calculated based on the relation to the room air temperature, air velocity and globe temperature [25]. The general and local thermal comfort can be classified into three categories (A, B, and C) as per ISO 7730 [26].

3.1 General thermal comfort

To assess the general thermal comfort criteria meant for both the ACMV systems, two methods were adopted in the study. One is the PMV-PPD model [26] and the other is graphical method [17]. PMV articulates the state of cold and warm discomfort for the whole body using a thermal sensation scale ranging from -3 (cold state) to +3.0 (hot state) with '0' representing the neutral state. In the graphical method, assessment is made by plotting the dataset graphically on a psychometric chart.

3.2 Local thermal discomfort

A person feeling thermally neutral according to the general comfort criteria may still be subjected to undesirable localised over-cooling or heating of a certain portion of the body, which is considered as local thermal discomfort. Factors of draught, VATD and radiant temperature asymmetry are adopted in the study to evaluate the possibility of local discomfort. Draught rate (DR) and people dissatisfied due to it can be calculated from mean air velocity, room air temperature and turbulence intensity [27]. Room air temperature stratification was monitored by measuring air temperatures at the elevation of 1.1 m and 0.1 m from the finished floor, representing the neck and ankle level, respectively, of the seated occupant. The difference in air temperatures at these two height levels was the VATD [26]. Thermal discomfort due to the asymmetry of a radiant field because of a cold or warm ceiling and a cold or warm wall was evaluated by radiant temperature asymmetry [26]. To estimate plane radiant temperature, the measurements of outdoor global and diffuse solar radiation on the horizontal surface were measured on the rooftop by a pyranometer (AT Delta-T Devices, SPN1). The transmitted radiation through the glazed area was also measured after blinds using a pyranometer (Kimo, CR110-PN) facing parallel to the window façade as shown in Fig. 5. The case of warm wall is considered since the glazed surface has a temperature of more than 23°C [28] and consequently, the percentage of people dissatisfied (PD) due to the warm wall in relation to radiant temperature asymmetry could be estimated.

4. Results and discussion

4.1 General thermal comfort criteria

4.1.1 PMV-PPD model

The mean room air temperature, air velocity, relative humidity and MRT for each of the four tests are computed and summarized in Table 5. Measured room air temperatures are summarized in Fig. 7. It can be observed that for the FCU system, the range of air temperature is 22–26 °C, meaning that the air temperature in the FCU system fell outside the comfort range of 23–25°C recommended by [31] sometimes. The percentage of time that fell outside the comfort range was 28.1%, 21.3%, 24.2% and 18.5% for the respective Tests 1, 2, 3 and 4. For the ACB system, air temperature ranged between 23–25.5 °C. The percentage of time that the air

temperature went outside of the comfort range is 8.8%, 6.1%, 5.7% and 3.0% for the respective Tests 1, 2, 3 and 4, much lesser than that found in the FCU system. The more-precise zonal control of the room air temperature in the ACB system could contribute to the better control of room air temperature in the comfortable range than the FCU system.

Statistical distributions of measured air velocities are summarized in Fig. 8. Air velocity did not exceed the recommended comfort maximum of 0.3 m/s [31] in all Tests. ACB system led to slightly lower air velocity as compared to the FCU system. In the FCU system, third quartiles are 0.21, 0.21, 0.23 and 0.23 m/s for the respective Tests 1, 2, 3 and 4. While for the ACB system, third quartiles are 0.16, 0.15, 0.15 and 0.16 m/s for the respective Tests 1, 2, 3 and 4. The lower air velocities measured in the room served by the ACB system can be primarily attributed to the lower air discharge by the ACB system compared to higher discharge from the FCU.

Statistical distributions of measured RH are summarized in Fig. 9. For the FCU system, RH ranges between 55 – 75%. Third quartiles are 70.9%, 70.0%, 68.8% and 67.4% for the respective Tests 1, 2, 3 and 4. While, for the ACB system, RH ranges between 55 – 68%. Third quartiles are 64.2%, 63.7%, 64.0% and 61.6% for the respective Tests 1, 2, 3 and 4. It suggests that the ACB system maintained the RH within the comfort range of less than 65% [31] for the majority of the time. However, for the FCU system, the RH went outside the comfort range more often. Since the ACB system receives primary air with an actively controlled supply air temperature (designed to handle the latent load) it is more capable to accurately control the RH to a comfortable range. The conventional FCU's supply air RH varies passively based on the room air temperature.

MRT of both the systems is summarized in Figs. 10 (a) and (b). It can be observed that for conventional FCU system, third quartiles are 25.9, 25.8, 26.0 and 26.0°C for the respective Tests 1, 2, 3 and 4. The range of minimum and maximum MRT is 23 – 27.5 °C. While, for the ACB system, third quartiles are 25.3, 25.7, 25.9 and 26.2°C for the respective Tests 1, 2, 3 and 4. The range of minimum and maximum MRT is 22.5 – 27.0°C. Thus, it can be observed that the range of MRT of the ACB system is slightly lower as compared to the conventional FCU system due to the lower air velocity recorded at the occupant station.

PMV and PPD can predict the thermal comfort environment generated by both the ACMV systems. For conventional FCU system, PMV values are falling between -0.7 to 0.7 for

most of the time. Fig. 11(a) shows the frequency of PMV below -0.7 is 6.2%, 4.4% and 8.6% for the respective conducted Tests 1, 2 and 3. This may cause a cold environment for occupants inside the cell. For the ACB system, most of the points are falling between -0.5 to 0.5, barring a few as shown in Fig. 11(b). Therefore, it can be inferred from the results that the ACB system is providing better thermal comfort in contrast to the conventional FCU system. The category of the thermal environment can be categorised based on the average PPD for different test conditions as shown in Table 6. It can be perceived that both the conventional FCU and ACB systems are performing well and satisfying the criteria of thermal comfort as per ISO 7730 and ASHRAE 55. Moreover, ACB system is performing better as compared to conventional FCU system, resulting in category A of ISO 7730 as compared to conventional FCU system resulting in category B of ISO 7730 for the conducted tests

4.1.2 Graphical method

For assessment in the study, comfort zone has been proposed considering clothing level of 0.57 clo as well as air velocity less than 0.2 m/s. RH has been fixed at 65% as mentioned in the standard of SS 554 [31]. As per the study, low humidity environment (dew point less than 2 °C) may also cause discomfort especially to eyes [32]. Therefore included in the proposed comfort zone.

ASHRAE psychometric chart tool was used to plot the comfort zone for both the two systems. The plot for Test 1 is depicted in Figs. 12 and 13, which illustrate that the data points are falling either inside the zone or nearby the proposed comfort zone for the ACB system. Only 12% of the data points were found outside the comfort zone. In case of the conventional FCU system, 53% of data points are falling outside the proposed comfort zone. This can be attributed to the relatively high RH recorded in the room served by the conventional FCU as discussed in section 4.1 leading to discomfort to the occupants. A similar trend was observed for other tests as well.

A comparison of results obtained from the PMV-PPD model and Graphical method is conducted. Based on the results of the PMV-PPD analysis, for conventional FCU system, only 5% of the data points were falling outside the comfortable range, and for ACB system, no data point was outside the comfortable range. However, according to the results of the Graphical method, it can be observed that 53% data points are falling outside the proposed comfort zone for the conventional FCU system and 12% of data points are outside the proposed comfort zone

for the ACB system. A possible reason for this deviation in the results is the adaptability of the PMV model with different parameters. In PMV-PPD model, only personal parameters like clothing insulation and metabolic rate are fixed whereas the remaining environmental parameters are varied. However, in the graphical method, the personal parameters and an environment parameter (i.e., air velocity) are fixed. The variation in air velocity could have led to the overestimation of occupant discomfort by the graphical method. Therefore, it can be said that PMV-PPD model performs better and more adaptable compared to the graphical method.

4.2 Local thermal comfort criteria

4.2.1 Air draught

It can be perceived that for the case of conventional FCU system, PD due to draught is less than 20% for the conducted tests (Table 7), which results in the category B of ISO 7730. For the case of the ACB system, the PD due to draught is less than 10% for the conducted tests (Table 7), which results in category A of ISO 7730.

It can be observed that average air velocity for conventional FCU system is higher compared to ACB system which is 0.19 and 0.14 m/s for Test 1 (Table 5). Sometimes, air velocity goes up to 0.27 m/s for conventional FCU system that may cause a slight draught, although it is lower than acceptable air velocity of 0.3 m/s as per SS 554. For the ACB system, the highest air velocity measured for ACB is 0.19 m/s, which is quite within the comfortable range as per SS 554. The average PD caused due to air draught by conventional FCU and the ACB system is 13.1% and 8.3% respectively. It means more people are feeling comfortable in environment produced by the ACB system as compared to conventional FCU system.

4.2.2 Vertical air temperature difference, VATD

Figs. 14(a) and (b) show the vertical air temperatures profile at different levels for conventional FCU and ACB systems respectively for the conducted tests. In Fig. 14(a), it can be observed that for conventional FCU system, room air temperature increases from ankle level to neck level and then decreases afterward. Considering temperature deviation within the occupied zone (up to a height of 2 m), the difference between the highest and lowest vertical temperatures are 0.8, 0.8, 0.8 and 0.7°C for Tests 1, 2, 3 and 4 respectively. For the ACB system (Fig. 14(b)), no significant deviation of vertical temperature was noticed. Within occupied zone (up to a

height of 2 m), the difference between highest and lowest vertical temperatures are 0.2, 0.2, 0.4 and 0.3°C for Tests 1, 2, 3 and 4 respectively. Therefore, it can be observed that the mixing of room air was adequate in the vertical direction for both ACMV systems. Although, ACB system performs even better compared to conventional FCU system.

The range of VATD and the PD caused due to temperature stratification is determined by the temperature difference between neck and ankle level and shown in Figs. 15 and 16 for FCU and ACB systems respectively. It can be observed that the temperature gradient in the vertical direction for both ACMV systems is found within 2°C for conducted tests and the corresponding PD is found within 1%, resulting in Category A of ISO 7730. The ACB system, despite its lower supply airflow rate compared to the FCU system, does not lead to stratified air in the vertical direction and no thermal discomfort. The results also indicate that the induced air of the ACB leads to proper mixing of air in the vertical direction.

4.2.3 Radiant temperature asymmetry, ΔT_{pr}

Calculation of plane radiant temperature is based on the internal surface temperature of the building envelope. In the conducted experiments, the internal surface temperatures of the walls, ceiling and floor were measured. It was also assumed that the occupant is seated at the same location of the measuring point depicted in Fig. 17, also representing the isothermal planes of window façade w.r.t. the occupant. However, assessment of radiant temperature asymmetry w.r.t. window façade wall has only be made because it is the only source of inflow of solar radiation. Radiant temperature asymmetry has been evaluated for each of the test mentioned in Table 3, considering two conditions: one with internal surface temperature and other includes solar radiation.

For Singapore, it was reported that there is a transient pattern in solar insolation due to fast-moving clouds [33]. It is also evident from the radiation plot for Tests 1, 2, 3 and 4 depicted in Figs. 18 (a), (b), (c) and (d) respectively. Fig. 18 (a) illustrates that for the north orientation, the vertical façade does not receive direct radiation during the entire day but for the west orientation, the vertical façade receives direct radiation in the afternoon time (Figs. 18 (b), (c) and (d)).

In reference cell, for Tests 1 and 2, blinds were retracted. The blinds were closed after 2 PM and were closed for the whole day for Tests 3 and 4 respectively. In the test cell, blinds were retracted in Tests 1 and 2 similar to the reference cell. However, blinds were in auto mode for

Tests 3 and 4. For closed and auto blind case, the room air temperature of the perimeter area is considered as blinds temperature for calculation.

In Table 8 and 9, radiant temperature asymmetry is provided for both rooms installed with conventional FCU and ACB systems respectively. It can be observed that for both rooms, radiant asymmetry for the Tests 1 and 2 without considering solar radiation, is more than 23°C resulting in category B as per ISO 7730. On the other hand, it is less than 23°C for Tests 3 and 4, which results in category A as per ISO 7730.

For conventional FCU cases, radiant asymmetry when solar radiation is taken into consideration, is higher for all tests, apart from Test 4. Accordingly, the percentage of people dissatisfied with it also higher. This is due to Test 4 being conducted with blinds closed throughout the day and less solar radiation being transmitted inside the cell. For the ACB cases also, the radiant asymmetry when solar radiation is taken into consideration and the percentage of people dissatisfied increases for all the tests. This increase is not significant for Tests 3 and 4 since the blinds were being operated in auto mode, leading to less transmitted solar radiation.

In order to see the impact of solar radiation on radiant asymmetry, mean hourly variation has been plotted w.r.t. time for each of the tests. It can be perceived that its value is increased significantly when solar radiation has been taken into consideration. For reference cell, it has been depicted in Figs. 19 (a), (b), (c) and (d).

It is observed that when the window façade was north orientated, there is a maximum increase of 3°C during afternoon time when solar radiation is taken into consideration (Fig. 19a). Reason for this increase is diffuse radiation, as no direct radiation is received at this orientation (Fig. 18a). When the window façade was west orientated, the maximum increase in 8°C can be seen during the afternoon time (Fig. 19b). Reason for this increase both direct and diffuse radiation is received at this orientation (Fig. 18b). Both orientations received similar amount of diffuse radiation during this period, but the quantum of direct radiation was significantly higher for West. It can be inferred that direct radiation has more impact on the asymmetry of the radiant field. For Test 3 (Fig. 19(c)), the window façade was west oriented and blinds were closed after 2 PM. It can be seen that there is a maximum increase of about 3°C in radiant asymmetry, which is mainly caused by the diffuse radiation. There is a decrease in radiant asymmetry can be observed after 2 PM, the reason is shielding of solar radiation from blinds. In Test 4 (Fig. 19(d)), the window façade was west oriented and the blinds were closed during the whole day. The

maximum increase in radiant asymmetry never goes beyond 1°C, because blinds shield the direct solar radiation on the seated occupant.

For test cell, a variation of radiant asymmetry has been depicted in Figs. 20 (a), (b), (c) and (d). For Tests 1 and 2, the test conditions were similar to the reference cell. It can be perceived that there is a maximum increase of 3°C during afternoon time when the window façade was north orientated (Fig. 20(a)). Reason for this increase is an inflow of diffuse radiation, as no direct radiation is falling at on the vertical surface. For Test 2, the increase pattern is similar to the reference cell and the maximum increase in radiant asymmetry is of 8°C (shown in Fig. 20(b)). In Tests 3 and 4, the window façade was west orientated and the blinds were in auto mode. The maximum increase of 2°C in radiant asymmetry can be seen during the afternoon time (Figs. 20 (c), (d)). Reason for this increase is transmitted diffuse radiation.

The results indicated that the incident solar radiation would increase the asymmetry of radiant field remarkably in the occupant zone closest to the window, causing more occupants to be dissatisfied. The impact of direct radiation causing this discomfort will be more compared to diffuse radiation. It has also been seen that by intercepting solar radiation through the window using blinds, we can reduce the influence of radiant asymmetry on the occupants.

5. Conclusions

In the study, the assessment of conventional FCU and ACB systems with regards to thermal comfort criteria has been carried out. Thermal comfort parameters of room air temperature, air velocity, RH and MRT produced by both systems has been assessed. It was found that the parameters are within the acceptable limit as specified by SS 554. Within the acceptable framework, ACB system performs better compared to conventional FCU system.

General thermal comfort has been investigated using PMV-PPD model and graphical method. It was perceived that for ACB system, mean PPD was less than 6% for conducted tests, resulting in Category A of ISO 7730. For conventional FCU system, mean PPD was less than 10% for conducted tests, resulting in Category B of ISO 7730. As per ASHRAE psychometric chart, the data points falling well within or closer to proposed comfort zone for ACB system compared to the conventional FCU system. It was also reported that PMV-PPD model performs better compared to the graphical method.

The factors of draught, VATD and radiant temperature asymmetry were adopted to investigate the local thermal comfort. Due to low airflow rate, the ACB system produced less air

draught compared to the conventional FCU system. In terms of VATD, the temperature gradient in the vertical direction for both ACMV systems was within 2°C for conducted tests. The corresponding PD was also not more than 1%, resulting in Category A of ISO 7730. It means the mixing of conditioned air was adequate in the vertical direction for both the systems.

Solar radiation can be the primary reason for thermal discomfort in highly glazed office buildings. Therefore, the influence of solar radiation on radiant temperature asymmetry was also investigated. It was established that direct radiation has more impact on radiant asymmetry compared to diffuse radiation. It was found that radiant asymmetry was more than 23°C for both the systems, resulting in the category B of ISO 7730. This increases while incorporating the impact of solar radiation. However, it does not increase by more than 2 °C, when blinds were employed (closed/ auto mode). Therefore, it was recommended that blinds can be used to curtail the impact of solar radiation on radiant asymmetry.

It can be concluded that both conventional FCU and ACB system have accomplished the required comfort criteria as per ISO 7730. ACB system can produce satisfactory thermal environment inside any building, with low air discharge compared to traditional air conditioning (here, it is Conventional FCU) system and sometimes provide even better thermal environment.

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Table 1

	Conventional FCU system	ACB system
Room set-point	Room air temperature: 24°C, Relative humidity: 65%	Room air temperature: 24°C, Relative humidity: 65%
Chilled water temperature	Supply= 7.9°C Return= 19.4°C	Supply= 7.9°C, Return= 19.4°C (for FCU) Supply= 17°C Return= 19.5°C (for ACB terminal)
Supply air temperature	Variable (14.9 to 26.7°C)	16°C
Supply air flowrate	1942 m ³ /hr	670 m ³ /hr (111 m ³ /hr per chilled beam)
Fresh air flowrate	120 m ³ /hr	120 m ³ /hr
Induction ratio	-	3.04

Table 2

Parameter	Wattage	Heat generated (W)
Equipment (Sensible)	16 W/m ²	720
People (Sensible)	75 W / thermal dummies	300
People (Latent)	110 W / humidifying device	220
Lighting (Sensible)	7.8 W/m ² (For T-5)	340
	6.1 W/m ² (For LED)	270

Table 3

	Reference Cell (Conventional FCU system)	Test Cell (ACB system)
Test 1	Window façade orientation= North Lighting = 100% T5 florescent lamps Blinds = Retracted Internal load = 1.58 kW	Window façade orientation = North Lighting = 100% T5 florescent lamps Blinds = Retracted Internal load = 1.58 kW
Test 2	Window façade orientation = West Lighting = 100% T5 florescent lamps Blinds = Retracted Internal load = 1.58 kW	Window façade orientation = West Lighting= 100% T5 florescent lamps Blinds= Retracted Internal load= 1.58 kW
Combination with other parameters		
Test 3	Window façade orientation = West Lighting = 100% T5 florescent lamps Blinds = Retracted + closed after 2 PM Internal load= 1.58 kW	Window façade orientation = West Lighting = LED + Auto dimming Blinds = Auto Internal load = 1.51 kW
Test 4	Window façade orientation = West Lighting = 100% T5 florescent lamps Blinds = Closed Internal load = 1.58 kW	Window façade orientation = West Lighting = LED + Auto dimming Blinds = Auto Internal load = 1.51 kW

Table 4

Parameter	Brand and Model	Range	Accuracy	Output Resolution
Air velocity	Swema, Swema 03	0.05 to 3 m/s at 15-30 °C	± 0.03 m/s at 0.05 to 1m/s	0.001 m/s
Room air temperature (stratification)	Precon, ST-S3EW-XPA	-40 °C to 105°C	$\pm 0.2^\circ\text{C}$	0.1°C
Globe temperature	Kimo, TM110	0°C to 60°C	$\pm 0.71^\circ\text{C}$	0.1°C
Relative humidity	Vaisala, HWM90	0-100% RH	$\pm 1.7\%$ RH for 0 to 90% RH	0.1% RH
Transmitted solar radiation	Kimo, CR110-PN	0 to >1500 W.m-2	$\pm 5\%$	1 W.m ⁻²
Global and diffuse radiation	AT Delta-T Devices, SPN1	0 to >2000 W-m-2	$\pm 5\%$	0.6 W-m ⁻²
Outdoor illuminance	Reinhardt, light intensity sensor	0 to 150,000 Lux	$\pm 6\%$	1 Lux

Table 5

	Conventional FCU system				ACB system			
	Room air temp., °C	Air velocity, m/s	Relative humidity, %	MRT, °C	Room air temp., °C	Air velocity, m/s	Relative humidity, %	MRT, °C
Test 1	23.9	0.18	68.0	25.4	24.5	0.15	62.5	25.0
Test 2	23.8	0.19	67.8	25.4	24.4	0.14	62.5	25.4
Test 3	23.8	0.20	65.5	25.6	23.9	0.14	62.6	25.6
Test 4	24.3	0.20	65.3	25.8	24.2	0.15	60.7	26.0

Table 6

	Conventional FCU system		ACB system	
	PPD (%)	Category	PPD (%)	Category
Test 1	8.5	B	5.5	A
Test 2	8.2	B	5.3	A
Test 3	8.6	B	5.9	A
Test 4	6.6	B	5.2	A

Table 7

	Conventional FCU system		ACB system	
	PD (%)	Category	PD (%)	Category
Test 1	12.7	B	8.6	A
Test 2	12.5	B	7.5	A
Test 3	13.8	B	8.3	A
Test 4	13.4	B	8.8	A

Table 8

	Conventional FCU system					
	Radiant temperature asymmetry, ΔT_{pr}					
	Without radiation, °C	PD (%)	Category	With radiation, °C	PD (%)	Category
Test 1	23.4	4.1	B	24.6	4.5	B
Test 2	23.2	4.0	B	25.1	4.7	B
Test 3	22.8	3.8	A	24.1	4.3	B
Test 4	22.0	3.6	A	22.2	3.6	A

Table 9

	ACB system					
	Radiant temperature asymmetry, ΔT_{pr}					
	Without radiation, °C	PD (%)	Category	With radiation, °C	PD (%)	Category
Test 1	23.3	4.0	B	24.5	4.5	B
Test 2	23.5	4.1	A	25.2	4.8	B
Test 3	22.4	3.7	A	22.9	3.9	A
Test 4	22.2	3.6	A	22.7	3.8	A

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Figure 1

Figure 2

Figure 3

Figure 4

(a)

(b)

Figure 5

Figure 6

Figure 7

Figure 8

Figure 9

Figure 10

(a)

(b)

Figure 11

Figure 12

Figure 13

(a)

(b)

Figure 14

(a)

(b)

Figure 15

(a)

(b)

Figure 16

Figure 17

Figure 18

Figure 19

Figure 20