Impact of thermally activated furniture system on the occupant thermal comfort – A study using thermoregulation model and computational fluid dynamics.

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Abstract

Air-conditioned buildings are conventionally designed and operated to maintain homogeneous thermal conditions. Maintaining the occupied and unoccupied zones at the same thermal conditions leads to higher energy consumption. More importantly, homogeneous thermal conditions do not address the need for individual thermal comfort preferences. A personal comfort system (PCS) allows the occupants to create desired localized thermal conditions around workstations in the office environment. Using computational fluid dynamic (CFD) and multi-node thermoregulation models, this paper evaluates the feasibility of a PCS with a radiantly cooled partition panel system to achieve thermal comfort. All input parameters for the model were derived from real-life measurements, including thermal characteristics of the room, work desk with radiantly cooled partition, and HVAC systems. A combination of scSTREAM™ and scTETRA™ was used to model the room and the human body (Cradle MSC Software, 2017). The simulation model had a mannequin in a seated position having summer clothing values and office activity metabolic rate. Combinations of three ambient room air temperatures and five panel surface temperatures were investigated to estimate the impact of radiant panels on overall thermal comfort and various body parts of the mannequin. The body parts like the thighs, chest, back, and pelvis showed a low thermal variation in the range of 0.9-1.2°C. The parts such as the head, neck, shoulders, arms, and legs showed a thermal variation in the range of 1.6-2.7°C, while the body parts farthest from the warm torso - the feet, experienced the highest variation in the range of 4.4-4.5°C. It was observed that the back side of the body was distinctly warmer than the front side of the body throughout the studied cases due to the action of front-placed radiant panels. It also indicates that at a given room air temperature with an increase in the difference between surface and air temperature, from 0°C to 8°C, all the body parts experience a reduction in the body part surface temperature.

Introduction:

It is an established fact that human activities have resulted in global warming of approximately 1.0°C above pre-industrial levels (Masson-Delmotte et. al, n.d.). In the absence of mitigation activities, this number is likely to reach 1.5°C between 2030 and 2052. Lowering the energy

intensity to reduce CO₂ emissions is one of the effective mitigation measures. 2.5% of the global final energy use is towards space cooling in buildings (IEA, 2018).

In many parts of the world, real estate developers do not have the opportunity to construct offices keeping the end user in context. In many countries, energy code enforcement is also in a very nascent stage, in such contexts, the users do not have the choice of adopting personal comfort system based approach to achieve thermal comfort without changing the building envelope or changing the HVAC system of the leased space. Space cooling system based on radiative heat transfer, known as the radiant space conditioning system, is an efficient method to reject energy from space for cooling (Feustel and Stetiu, 1995; Stetiu, 1999; Imanari et. al, 1999; Sastry and Rumsey, 2014). The technology based on radiant space heating or cooling has been in existence since the 1930s, but recent advancements in installation engineering, availability of materials, and betterment in sensor-control technologies have renewed the interest in this domain.

Workstation-incorporated PCS focus on environmental conditioning of the occupied indoor area with personal comfort and control systems while allowing the unoccupied area to not meet thermal comfort criteria, using separate or dedicated HVAC systems. The overall goal of such systems is to provide indoor environmental thermal comfort for occupied areas and optimize energy consumption. This leads to environmental parameters such as dry-bulb temperature, relative humidity, air velocity to be maintained in the occupied area more suitably than in the unoccupied zones. Open-plan office layouts with a homogeneous thermal environment leave many occupants dissatisfied (Honnekeri et. al, 2014; Brager and de Dear, 1998; Manu et. al, 2014). PCS provide the opportunity to the individual user to customise their immediate thermal environment based on their thermal preference. The thermal environments of the unoccupied zones may deviate from thermal comfort conditions, hence it may lead to significant energy savings (Arens et. al, 1998; Zhang, 2011). Since PCS allow the occupants to have more control over their immediate environment, it is likely to lead to a higher workspace satisfaction too. In cooling dominated climates, raising the cooling set point results in a considerable amount of energy savings for space cooling. An elevated cooling set

point in air-conditioned spaces can be augmented with the PCS capable of providing appropriate cooling closer to the occupied area. Multiple researchers have studied the feasibility of PCS and their associated energy benefits (Schiavon and Melikov, 2008; Watanabe et. al, 2009; Yang et. al, 2009; Yang et. al, 2010). Many of these studies are based on the convective heat transfer method. Use of localized air movement with the help of a personal fan, ceiling fan, or floor embedded air diffusers was studied by researchers with specific conclusions about the preferred air velocities (Zhang et. al, 2011; Amai et. al, 2007; Han et. al, 2007; Melikov et. al, 2002; Pasut et. al, 2014; Tsuzuki et. al, 1999). Most of the studies offer an insight into the ability of convection-based systems to provide a thermally comfortable local environment to the occupants. In regard to using radiation-based PCS to provide thermal comfort, some researchers have studied the use of radiant heating panels (Bolashikov et. al, 2013), while some have studied the impact of radiant cooling panel (Khan et. al, 2015; Khare et. al, 2015; Sharma et. al, 2015). Over a period of about 20 years of research, various categories of PCS have been developed and studied. Some of them are aimed at improving or innovating the system hardware, providing personal comfort, while others focused on control and operation of available hardware systems to achieve personalised comfort. Based on the purpose of the system, PCS are given names such as personalized air conditioning system (PAC), individually controlled system (ICS), task conditioning (TC), personal environment module (PEM), localised thermal distribution (LTD), localised ventilation (LV), task ventilation (TV), etc.

Studies undertaken to evaluate PCS may be categorized into the following groups based on the methods adopted for study:

- (1) Studies conducted in a thermal comfort chamber in the presence of human subjects.
- (2) Studies conducted in a thermal comfort chamber using thermal mannequin that represents the thermo-regulatory process of the human body.
- (3) Studies based on virtual thermal mannequin models in conjunction with dynamic thermal modelling and computational fluid dynamic (CFD) models.

A large number of studies relied on the method which involved a thermal comfort chamber and human subjects. Based on the literature, studies paying attention to air flow interactions, thermal comfort, and perceived air quality (PAQ) are more in number. The literature also helps understand the relation between thermal comfort and work productivity. However, the absence of literature investigating the impact of personalized radiant cooling on thermal comfort and energy consumption provides an opportunity to study it. More so, the opportunity also lies in studying the subject using thermoregulation models and CFD, which is a less explored area. Personalised ventilation has been studied using numerical thermal mannequin and CFD before (Gao, Niu, & Zhang, 2017). The studies talk about the usefulness of thermo-regulation models coupled with CFD in the study of personalised

comfort systems. Another study focused on evaluating the performance of ceiling-mounted personalised ventilation with desk and chair fans using CFD (Habchi, Chakroun, Alotaibi, Ghali, & Ghaddar, 2016). This study coupled CFD with a bio-heat model to predict body part sensation and over all body comfort sensations.

The study presented in this paper is the second part of a larger, two-part study. The larger study involves thermal chamber-based experiments using thermal mannequin and simulations using a CFD tool with a thermoregulation model. This paper is an attempt to summarise the study to understand and quantify the impact of personalised radiant cooling on human thermal comfort in terms of Predicted Mean Vote (PMV), the change in average body skin temperature, the impact on individual body parts, and the variation in total sensible heat loss at different operational conditions.

Methodology:

This study was conducted to understand the change in PMV, body skin temperatures and body heat flux of various body parts using the combination of room air temperature and surface temperature of the radiantly cooled furniture. To correctly capture the behaviour of the human body and its parts in the presence of various radiant and air temperatures, it was essential to use CFD coupled with human body thermoregulation models. Since the larger study used a thermal comfort chamber (TCC) with a thermal mannequin and CFD coupled with thermoregulation model, it was necessary to select a numerical tool which could provide the opportunity of validating the model using physical experiments. Based on the literature review and the physical thermal mannequin used in the laboratory experiments, it was found that the body-surface characteristics such as the surface area to volume ratio of the physical and virtual mannequins were similar to the scTETRA™ model (Kobayashi and Tanabe, 2013). This was one of the key decisions behind the selection of scSTREAM™ and scTETRA™ models. scTETRA™ is a general-purpose, unstructured mesh tool capable of solving complex geometries such as the human body. The inbuilt JOS-2 model (Kobayashi and Tanabe, 2013) was capable of simulating the human body surface temperature and heat flux based on the heat balance equations for divided body segments. It uses skin temperature and thermal resistance of the clothing to generate body-surface boundary conditions of temperature to be simulated in the CFD environment. It is also capable of accounting for the air temperature and partial pressure of water vapour at body skin as one of the boundary conditions. For the ease of understanding, the research method adopted for the part of the study involving CFD and thermoregulation model has been described in the sections on (a) Modelling approach and boundary conditions (b) Model calibration (c) Grid independence test.

Modelling approach and boundary conditions:

Figure 1 shows the thermal comfort chamber (TCC) setup, with the dimensions of 5.5×4.4×2.7m. This included the HVAC system and the radiantly cooled

Thermally Activated Furniture (TAF). The TAF consisted of a horizontal desk (1200×600mm) and a vertical partition divided into two equal parts of 1200×600mm each, as shown in Figure 2. As mentioned earlier, we used the same geometry and boundary conditions for the simulations as used in the real laboratory experiments. It resembled the layout of an open plan space office – a layout predominantly used in the service industry. Design of the TAF was based on contemporary industry practice. The TAF design represents a typical office desk design found in a majority of offices having open plan office layout. The 1200×600mm desktop serves as a working platform often used to place desktop or laptop computer, accessories and notepads. A 1200mm high partition works as a visual barrier for the seated human being, while providing an opportunity to increase the collaboration amongst building occupants. These partitions were placed in the front of a seated mannequin as radiant cooling panels. At the time of the laboratory experiment, the panel temperature was regulated by passing cooled water through the embedded coils. For the simulation, we considered the entire panel surface to be at a constant temperature for each case. The walls, ceiling and floor were considered as adiabatic. They were assigned the conditions of ‘stationary wall shear stress’ and ‘adiabatic heat transfer’. The air exchange in the chamber was maintained through four supply and two return air diffusers with areas modelled after the real-life equivalents.

Figure 1: Layout of the Modelled Thermal Comfort Chamber.

Figure 2: Thermally Activated Furniture and the mannequin.

The TAF was placed in the centre of the chamber with a thermal mannequin seated on it. The modelled thermal mannequin was specified a metabolic rate of 1.1 MET (~65W/m²), a clothing insulation value of 0.612 clo, and a body fat percentage of 14.4% (Schellen et. al, 2013).

The mannequin was divided into 17 body parts (Kobayashi and Tanabe, 2013) as listed in the discussion section of this paper. It had front and side view factors of 0.224 and 0.180 respectively. To account for the ambient heat gains, the lighting and computer loads were taken as 38.12W and 44.5W respectively.

The simulation was run for 15 cases, which included three room air temperature settings and five panel surface temperatures at each room air temperature. Figure 3 summarises the 15 simulation cases. The temperature difference between the room air and the panel surface (ΔT) was restricted to 8°C as per ASHRAE guidelines (ASHRAE, 2009).

Figure 3. Simulation cases.

The mannequin surface was specified with a tetrahedral mesh, while the rest of the domain area was specified hexahedral mesh, thereby cumulatively identifying as a hybrid meshing scheme (Gao and Niu, 2005; Yang et. al, 2017). The mesh for the mannequin surface was given the size of 0.6 mm. In order to capture the values around complex geometries of head, hands, and feet, they were assigned 10 prism layers of size 0.6 mm and a growth rate of 1.1. The remaining zones were given a mesh size of 92 mm after a grid independence test, as described in the later section. Simple geometries like the ceiling, floor, and walls were specified assigned 3 prism layers of size 0.1 mm and a growth rate of 1.1.

In order to capture the low air velocity and turbulence around the mannequin, low Reynolds number turbulence model was used (Gao and Niu, 2005; Yang et. al, 2017; Pan and Xia, 2014; Kurabuchi et. al, n.d.; Oh and Kato, 2016; Omori et. al, n.d.; Kajiya et al., 2011). Second order Monotonic Upwind Scheme for Conversion Laws (MUSCL) was used to calculate the convective terms. The convergence criteria for the model was specified as 1e-10.

Given that the point of focus of this study was to account for radiant heat transfer, the floor and ceiling were considered as adiabatic. To account for the radiation, emissivity (ϵ) of each of the indoor surfaces was measured as per (ASTM C1371-15, n.d.). The composite aluminium coated wall panels ($\epsilon=0.85$), carpeted floor ($\epsilon=0.81$), gypsum false ceiling ($\epsilon=0.92$), laminated table top ($\epsilon=0.89$), expanded carbon radiant panels with felt-surface finish ($\epsilon=0.92$), and windows ($\epsilon=0.84$) were the measured radiative surfaces. In order to reduce the mesh count, the walls were modelled as heat conduction panels, with the initial surface temperature as 30°C and RH as 50% for all cases.

Before determining the air flow, the authors measured air flow at the diffuser level in TCC using Flow Hood (SKU8380 by TSI). Based on these measurements, 141 CFM of air was modelled to be supplied through each of four air inlets parallel to the ceiling surface, while 282 CFM of air was modelled to exit through the two return air diffusers.

Model calibration:

To verify the accuracy of the modelled airflow, real-life measurements were taken at the supply and return diffuser of the TCC. This section presents the measurement procedure and compares the measured and modelled airflow.

The airflow angle and air velocity of the supply and return diffusers were measured using TSI manufactured VelociCalc multifunction meter 9565-P, and air velocity probe-964. The range, accuracy, and resolution of these instruments was 0-50 m/s, ± 0.015 m/s, and 0.01m/s respectively. The measurements were carried out on one of the four air inlet diffusers, and one of the two air outlet diffusers.

Figure 4 shows the twelve measurement locations around square diffuser in the four directions and the position of the velocity probe at 0° (along the ceiling), 15°, 30° and 60°. It was assumed that discharge of air from the diffuser was parallel to the ceiling. This assumption was based on the design of diffuser.

Figure 4. Measurement points for the diffuser.

A simulation was run to validate the air discharge velocities through the diffuser. The simulated air velocity was matched with the average of the experimentally measured air velocities. As shown in Table 1, the simulated values matched well with the average experimental values, which confirmed the flow direction of the air flow through the diffuser to be at 0° (along the ceiling).

Table 1. Comparison of measured and simulated velocities.

Angle	Measured Velocity (m/s)		Simulated Velocity (m/s)
	Diffuser 1	Diffuser 2	
0°	2	2	1.9
10°	1.2	1.5	1.1
15°	0.6	0.7	0.5
30°	0.2	0.2	0.2
60°	0.2	0.2	0.1

Grid independence test:

To minimize the impact of grid size on the solution, the grid independence test was carried out. The thermal mannequin was the key subject of interest in this study, therefore the average skin temperature of the mannequin was selected as a parameter to subject to the grid independence test. Five cases of grids sizes were examined before selecting one for further analysis. The first test was conducted with a mesh size of 0.15 m generating 1.1 million elements in the model. The mesh size was reduced to 0.112 m (2.0 million elements), 0.092 m (3.2 million elements), 0.078 m (4.8 million elements) and 0.0625 m (7.3 million elements). As shown in Figure 5, the skin temperature showed significant variation when the number of elements changed from 1.1 million to 2.0 million. However, it remained unchanged when the elements were increased beyond 3.2 million. CFD simulations involve heavy computation, therefore the optimization of computational time without compromising on results was crucial. It was observed that the convergence of cases with 3.2, 4.8, and 7.3 million elements took 48, 72, and 98 hours respectively. Given these inferences, the model with 3.2 million elements (mesh size 0.092 m) was deemed suitable for further simulations and is shown in Figure 6.

Figure 5: Grid Independence test, skin temperature at different mesh counts.

Figure 6: Cross section of TCC showing CFD mesh.

Predicted Mean Vote Calculations:

This paper presents Predicted Mean Vote Calculations (PMV) of the entire body as well as includes the observations pertaining to the skin temperature and heat flux of the individual body parts. scTETRA™ integrates PMV calculation in accordance with ISO 7730. Following parameters are considered (i) metabolic rate (W/m^2) (ii) effective mechanical power (W/m^2) (iii) clothing insulation (m^2K/W) (iv) clothing surface area factor (v) air temperature ($^{\circ}C$) (vi) mean radiant temperature ($^{\circ}C$) (vii) relative air velocity (m/s) (viii) water vapour partial

pressure (Pa) (ix) convective heat transfer coefficient ($\text{W}/\text{m}^2\text{K}$) (x) clothing surface temperature ($^{\circ}\text{C}$)

Results and Observations:

The mean skin temperature of all the body parts was found to be the lowest at $T_{\text{room}}=26^{\circ}\text{C}$, when the surface temperature of the radiant panels was maintained at 8°C lower from the room ambient air temperature ($\Delta 8^{\circ}\text{C}$). Whereas, the highest mean skin temperature was observed at $T_{\text{room}}=30^{\circ}\text{C}$, when the radiant panels were not operated at all ($\Delta 0^{\circ}\text{C}$). The difference in T_{room} and T_{RP} (indicated by Δ) had the most visible impact on the skin temperatures of the body parts at $T_{\text{room}}=26^{\circ}\text{C}$. The highest thermal fluctuation due to the variation of Δ was observed in the feet – the T_{skin} changed by -1.9°C as the Δ increased from 0°C to 8°C . The back, pelvis, and thighs were the least affected body parts with a T_{skin} change of -0.2°C corresponding to an increase in Δ from 0°C to 8°C . In comparison, at $T_{\text{room}}=30^{\circ}\text{C}$, as the Δ increased from 0°C to 8°C , the T_{skin} of the feet, back, pelvis, and thighs changed by -0.5°C , $+0.1^{\circ}\text{C}$, $+0.1^{\circ}\text{C}$, and -0.2°C respectively.

The variation of skin temperature across the 15 studied cases for all the body parts is shown in Figure 7. The body parts like the thighs, chest, back, and pelvis showed a low thermal variation in the range of $0.9\text{--}1.2^{\circ}\text{C}$. The parts such as the head, neck, shoulders, arms, and legs showed a thermal variation in the range of $1.6\text{--}2.7^{\circ}\text{C}$, while the body parts farthest from the warm torso – the feet, experienced the highest variation in the range of $4.4\text{--}4.5^{\circ}\text{C}$. In order to understand the reasoning behind this variation, one must understand the contributing factors to local thermal fluctuations in the body.

The skin temperature and its fluctuation across various body parts are dependent on the heat exchanged by the body part with the ambience. With the body's internal temperature being regulated in the range of $36.1\text{--}37.2^{\circ}\text{C}$, the individual body parts act as mechanical fins with their surface temperatures majorly dependent on the area of contact with the cooling media. In this case, the extent of heat lost through each body part depended on the thermal mass of the body part and skin surface area. The thighs, chest, back, and pelvis have a high fat content in comparison to the other body parts, which adds on to the insulation and restricts drastic thermal fluctuations. They also have a low 'skin surface area: thermal mass' ratio in comparison to other body parts, indicating a smaller interface for heat transfer for a higher content of heat. The head, neck, shoulders, arms, and legs, with a relatively lower fat content and higher 'skin surface area: thermal mass' ratio, therefore indicated a higher thermal fluctuation. The feet, being farthest away from the warm body parts and having the least fat content with the highest 'skin surface area: thermal mass' ratio, showed the most drastic thermal fluctuations.

Understanding the variation of T_{skin} of each body part at the three-room temperatures offers a deeper insight into the results. At $T_{\text{room}}=26^{\circ}\text{C}$, the left and right feet were found to be the coolest body parts with an average skin temperature for the five Δ variations ($T_{\Delta,\text{avg}}$) as 31.7°C and

31.6°C respectively. At $T_{\text{room}}=28^{\circ}\text{C}$ and 30°C , the head was found to be the coolest body part with a $T_{\Delta,\text{avg}}=32.8^{\circ}\text{C}$ and 33.6°C respectively. At $T_{\text{room}}=26^{\circ}\text{C}$, the right thigh was found to have the highest skin temperature with a $T_{\Delta,\text{avg}}=35.0^{\circ}\text{C}$, the left thigh was found to be cooler with $T_{\Delta,\text{avg}}=34.7^{\circ}\text{C}$. At $T_{\text{room}}=28^{\circ}\text{C}$, the chest, pelvis, and the right thigh were found to be the warmest body parts with $T_{\Delta,\text{avg}}=35.6^{\circ}\text{C}$, 35.5°C , and 35.4°C respectively. At $T_{\text{room}}=30^{\circ}\text{C}$, the left and right thighs ($T_{\Delta,\text{avg}}=35.6^{\circ}\text{C}$ and 35.5°C respectively), along with the left and right hands ($T_{\Delta,\text{avg}}=35.5^{\circ}\text{C}$ each) were found to be the warmest body parts.

Figure 7: Skin surface temperature for the 17 body parts across all temperature cases.

Figure 8 shows the simulated skin surface temperature for all the 15 cases studied. The figure is representative of the fact that the back was distinctly warmer than the front portion of the body throughout the 15 cases due to the action of front-placed radiant panels. It also indicates that at a given T_{room} , with an increase in Δ from 0°C to 8°C , all the body parts experience a reduction in surface temperature.

Figure 8: Variation of skin surface temperature on the digital thermal mannequin for the 15 simulated cases.

Figure 9 helps understand the possible relation between the surface area and the skin temperature of an individual body part. It shows the difference in skin surface temperature of the 17 body parts of the digital mannequin

with the difference in the air temperature and the radiant panel surface temperature (Δ). The size of the circle indicates the skin surface area of the respective body part – the parts above the waist are indicated in red, while the ones below are indicated in green. It is evident that no significant trend is visible to establish a relation between the surface area of skin and its temperature.

Figure 9. Variation of skin surface temperature of the 17 body parts with the Δ (difference in the air temperature and the radiant panel surface temperature).

The overall average skin surface temperature varies by less than 0.2°C, while the feet temperature changed by over 1.1°C across the cases. The body parts close to the central region – chest, back, pelvis, exhibited a negligible change of 0.1°C.

The scope of heat exchange with the ambience (within the comfort limit) directly relates to the extent of perceived thermal comfort and is reflected through the variation in T_{skin} . In principle, the greater the temperature difference between the body and the ambience, the greater will be the heat loss, therefore it was expected that as the avg. skin temperature reduced and T_{room} remained constant, the sensible heat loss would have increased - Figure 10 confirms this and shows the inverse relation between the average skin temperature of the entire body ($T_{avg,skin}$) and the total sensible heat loss from the body ($H_{sensible}$) for the five Δ values at the three T_{room} conditions. The figure also shows the variation of thermal comfort through PMV (Predicted Mean Vote) for the respective simulation conditions.

Figure 10: Variation of average skin temperature, total sensible heat loss and PMV values for each simulated case.

As can be seen in the figure, at $T_{room}=26^\circ\text{C}$, 28°C , and 30°C , with no operation of the radiant panels ($\Delta=0^\circ\text{C}$), $T_{avg,skin}$ was maintained at 33.8°C , 34.8°C , and 35.0°C , with $H_{sensible}$ at 38.4 W/m^2 , 34.4 W/m^2 , and 24.7 W/m^2 , and PMV at 1.0, 1.5, and 2.2 respectively. At a fixed T_{room} , $H_{sensible}$ increased linearly with a linear reduction of $T_{avg,skin}$ with respect to an increase in Δ in steps of 2°C . At $T_{room}=26^\circ\text{C}$, 28°C , and 30°C , as the Δ increased from 0°C to 8°C along the X-axis, $T_{avg,skin}$ experienced a reduction of 0.5°C , 0.3°C , and 0.1°C , with an increase in $H_{sensible}$ by 2.8 W/m^2 , 2.4 W/m^2 , and 4.8 W/m^2 respectively. At the three respective T_{room} conditions, the PMV also indicated a linear decrease to the neutral (0) state by 0.3, 0.2 and 0.3 votes with an increase in Δ .

The results indicate that a 2°C increase in T_{room} translates to a non-linear increase in $T_{avg,skin}$ by 1°C (transition from $T_{room}=26^\circ\text{C}$ to 28°C) and 0.2°C (transition from $T_{room}=28^\circ\text{C}$ to 30°C). This non-linearity can be explained by the fact that the body in itself is a heat source with its core temperature moderated in the range of $36.1-37.2^\circ\text{C}$. The skin surface attains thermal equilibrium with this core temperature and the ambient temperature through complex thermoregulatory processes. The reduction of the range of variation of T_{skin} at elevated T_{room} levels is indicative of these thermoregulatory processes in action – these processes naturally restrict the skin temperature from going beyond the permissible limit, which, if absent, could lead to unregulated body temperature rise and pose severe health complications.

At fixed room temperature, as the radiant panel was cooled by up to 8°C , the T_{skin} was reduced as a direct response to the decrease in the mean radiant temperature. In order to maintain the thermal equilibrium, the heat content of the skin surface was dissipated in the ambience, thereby accounting for an increasing $H_{sensible}$ trend. With the reduction in T_{skin} , the PMV, originally indicating a sensation in the range of 'slightly warm' to 'warm' at $\Delta=0^\circ\text{C}$, eventually came closer to the 'neutral' state.

Conclusion:

This study demonstrates the application of the latest advancements in thermoregulation models and CFD to highlight the importance of the simulation technology - through which, we can derive affordable and effective results within a defined time-frame. It also establishes the necessity of experimental validation prior to simulations to provide appropriate physical context as a simulation input. Ultimately, it translates the 8°C reduction in the surface temperature of the radiant panels to a ~0.6 vote reduction in the PMV at each of the three room temperatures. The reduction in comfort was near-constant across the three cases due to the low front view factor (low skin surface area exposed directly to the radiant panels). This study demonstrates the ability of thermally activated furniture to provide comfort conditions while keeping the ambient room temperature higher than the thresholds.

Acknowledgment:

The project was performed within the framework of the International Energy Agency - Energy in Buildings and Communities Program (IEA-EBC) Annex69 "Strategy and Practice of Adaptive Thermal Comfort in Low Energy Buildings".

Abbreviations:

T_{room} = Room Ambient Air Temperature
 T_{RP} = Radiant Panel Surface Temperature
 Δ = (Room Ambient Air Temperature – Radiant Panel Surface Temperature)
 T_{skin} = Skin Surface Temperature
 $T_{\Delta, \text{avg}}$ = Average Skin Surface Temperature for a body part at a given Room Ambient Air Temperature for $\Delta=0, 2, 4, 6, \text{ \& } 8^{\circ}\text{C}$
 $T_{\text{avg, skin}}$ = Whole-body Weighted-average Skin Temperature
 PMV = Predicted Mean Vote

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