A review of human thermal comfort model in predicting human–environment interaction in non-uniform environmental conditions

Yat Huang Yau1,2,3 · Hui Sin Toh1,2 · Bee Teng Chew1,2,3 · Nik Nazri Nik Ghazali1

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Abstract
This paper presented a review of the literature on the human thermal comfort model, which can be employed to predict the response of a human towards the environmental surroundings. An important premise of this paper is that governments in tropical regions have taken proactive action in minimizing energy consumption by air-conditioning through elevated room temperature. However, would such an action worsen the quality of interior conditions, particularly the thermal comfort? To answer this question, developing a human thermal comfort model under stratum ventilation mode can become a reference model for air-conditioning system design in all tropical buildings and indirectly reduce the emission of carbon dioxide (CO2) from heating, ventilation, and air-conditioning (HVAC) system that caused a warmer environment. For this purpose, there are two critical processes to identify the role of human thermal comfort, namely human reaction towards the thermal ambient (thermoregulation) and the heat transfer and air movement that occur in the enclosed space due to natural and forced convection.

Keywords Computational thermal manikin · CFD · Heat transfer · Thermal comfort · Model coupling · HVAC

Abbreviations

| Abbreviation | Description |
|--------------|-------------|
| CFD          | Computational fluid dynamics |
| Cld          | Cold |
| CSP          | Computer simulated person |
| DV           | Displacement ventilation |
| Err          | Error |
| GCI          | Grid independence index |
| PMV          | Predicted mean vote |
| IAQ          | Indoor air quality |
| MV           | Mixed ventilation |
| SV           | Stratum ventilation |
| Wrm          | Warm |
| CFD          | Computational fluid dynamics |

Greek letters

| Symbol | Description |
|--------|-------------|
| $\rho$ | Air density (kg m$^{-3}$) |
| $u$    | Flow velocity |
| $u_t$  | Friction velocity |
| $v$    | Air kinematic viscosity (Pa s) |
| $\tau_w$ | Wall shear stress (Pa) |

Symbols

| Symbol | Description |
|--------|-------------|
| $E_{sw}$ | Evaporative heat transfer rate from skin, W m$^{-2}$ K$^{-1}$ |
| $h_c$  | Convective heat transfer coefficient, W m$^{-2}$ K$^{-1}$ |
| $h_r$  | Radiative heat transfer coefficient, W m$^{-2}$ K$^{-1}$ |
| $h_{fg}$ | Heat of vaporization of water, kJ kg$^{-1}$ |
| $k$    | Thermal conductivity, W m$^{-1}$ K$^{-1}$ |
| $T_a$  | Ambient air temperature, °C |
| $T_{sk}$ | Temperature of skin layer, °C |
| $T_w$  | Wall surface temperature, °C |
| $Q_c$  | Convective heat transfer rate, W m$^{-2}$ |
| $Q_r$  | Radiant heat transfer rate, W m$^{-2}$ |
| $Q_T$  | Total heat transfer rate of human body, W m$^{-2}$ |
| $y$    | First inflation layer thickness, mm |
| $y^+$  | Dimensionless wall thickness |

1 Department of Mechanical Engineering, University of Malaya, 50603 Kuala Lumpur, Malaysia
2 UM-Daikin Laboratory, Department of Mechanical Engineering, Faculty of Engineering, University of Malaya, 50603 Kuala Lumpur, Malaysia
3 Centre for Energy Sciences, University of Malaya, 50603 Kuala Lumpur, Malaysia
Introduction

People spend time in air-conditioning interior space is significant in hot and humid climates, where the heat and moisture released from the human body are the dominating factors influencing the thermal environment [1]. It is often common to find an indoor environment in buildings that provide comfortable space—where people live and work that rely on natural or mechanical ventilation. In the early days, air-conditioning started to control the air humidity in a manufacturing plant before spreading to buildings for cooling purposes [2]. According to ASHRAE 55-2004/ISO Standard 7730, thermal comfort can be expressed as “the condition of mind that expresses satisfaction with the thermal environment” [3]. The thermal acceptability helps to predict the subject’s acceptance level towards the ambient thermal environment. However, it could not identify the effectiveness of different personal comfort systems under various environmental conditions [4].

In recent years, there has been an increased interest in exploring methods for assessing human comfort inside the building to help with indoor design [5] due to overall population growth and urbanization [6]. The condition of the indoor space is strictly affected by the overall building energy performance, as stated in the 2002/91/EC European Directive [7]. In other words, there is an antagonistic relationship between the quality of the indoor environment (IEQ) and building energy consumption. Therefore, it is reasonable to expect if the thermal comfort is implemented well, the energy efficiency in a building would be improved as well.

For this purpose, it is preferable to predict and control the criteria of thermal comfort in the initial stage of building design and heating, ventilation, and air-conditioning (HVAC) system installation in complete compliance with international sustainability standards of the acceptable thermal environment. In addition, any changes in these measures cause unhealthy indoor air and thermal discomfort to the indoor occupants. The problems of indoor air quality (IAQ) are sometimes called sick building syndrome, where occupants complain about indoor space’s health and comfort issues. They might have asthma, respiratory illness, allergies, and increased symptoms of Sick Building Syndrome (SBS), namely fatigue, headache, dry eyes, irritated throats, etc. Thus, unhealthy indoor space may significantly impact workers’ task performance and productivity [7]. To deal with these IAQ problems, many building operators, managers, and designers have spent efforts on the aspect of the building’s design to avoid any potential IAQ problems. Several years back, these concerns were analysed to be addressed by using thermal comfort practices involving ASHRAE 55, EN 1521, and ISO 7730. Based on the adaptive and rational thermal comfort methods, these standards have decided the indoor design inputs for the operative temperature and comfort equations as presented in Table 1 [8].

Moreover, as the second largest economy in Southeast Asia, Malaysia has promised to reduce its greenhouse gas (GHG) release intensity of Gross Domestic Product (GDP) by 45% in Paris Agreement 21st Conference of the Parties (COP21) with a given implementation timeline from 2021 to 2030. The employment of sustainable energy-saving programs is an important national strategy to control the energy used while achieving sustained economic growth. About 2/3 of global carbon dioxide (CO₂) was emitted by the building sector, and carbon dioxide is a significant contributor to greenhouse gas (GHG) that causes global warming and is related to human activities, namely anthropogenic. The anthropogenic sources result from energy-related activities such as the combustion of fossil fuels, transportation, agriculture, waste management, and various industrial processes (U.S. EPA). So far, nearly 42.1% of the final energy consumption in Malaysia is consumed by the building sector, according to Energy Commission Malaysia. The cooling system in the building contributes significantly to carbon emission [9]. For example, the typical energy breakdown in Malaysia’s office building, where 50% of the energy is due to air-conditioning, small power equipment has an energy consumption of 25% and 25% for electrical lighting. Hence, the essential way to minimize carbon emission is to reduce the energy use for air-conditioning in buildings [10].

| Table 1 Thermal comfort standard practice [8] |
|-----------------------------------------------|
| **Standard** | **Year** | **Thermal comfort approach** | **Operative temperature summer** |
| ASHRAE 55 | 2004 | Rational: $0.5 < PMV < 0.5$, PPD < 10% | 24.5 °C–28.0 °C |
| ISO 7730 | 2005 | Rational: $0.5 < PMV < 0.5$, PPD < 10% | 23 °C–26 °C |
| EN-15521 | 2007 | Adaptive | $T_a = 0.3021RMT + 19.39; TMRT > 10$ |
| ASHRAE 55 | 2010 | Adaptive | $T_a = 22 : 88; TMRT \leq 10$ |

$$T_a = 0.317T_e + 17.8$$
In addition, governments in East Asia have recently proposed a rule to elevate the indoor room temperature in the summer for energy conservation. Table 2 shows the elevated room temperature in some of the countries. Yet would such implementation deteriorate the quality of the indoor environment, particularly thermal comfort? The answer is to develop a human model to assess the indoor thermal comfort acceptance level and apply this model to air-conditioned building design systems in tropical climate countries. The human thermal comfort model acts as the setpoint or dead band to decrease the building energy use, and this model could indirectly reduce the emission of carbon dioxide (CO₂) from the HVAC system that causes a warmer environment. For example, one set-point increment in indoor temperature saves up to 10% of building energy [9]. The human model focuses on the tropical country because individuals who live in regions of the world with hot and warm climates may respond differently to warm circumstances than those who live in areas with cold and cool climates [11]. Many field studies showed that the indoor comfort level could be identified using heat balance models like PMV. Still, this approach did not agree well with the comfort condition in the actual field, as the comfort temperature differed between groups. Hence, the calculated PMV should come with the presence of indoor air-conditioned and involve different areas of the world as there is an interaction between the human body and surrounding air movement [2].

This review provides an overview of hot thermal conditions in tropical countries and how their indoor air-conditioning demands differ from those in non-tropic countries. The literature is reviewed to examine the available information based on the CFD method that could be used to study the micro-environment close to the body in the indoor environment followed by a brief introduction of the non-uniform environments that occurred in the indoor space and the development of ventilation from the conventional air distribution to the advanced ventilation mode. This study highlights the heat transfer from the human body to the environment. Thus, the CFD setups of the micro-environment around the human body are then reviewed, including human body geometry, boundary condition, grid generation, turbulence model, and validation.

### Non-uniform environments

Thermal non-uniformities always occur inside buildings, inner vehicles, and in the outdoor environment. It is not enough to examine the non-uniform conditions that happen in real life by using a steady-state approach. Activities such as people moving around, changing their clothing and activity level, office space with large windows, and temperature gradient may create transient conditions. Ahmed [12] stated that non-uniformities in the air temperature, airflow, surface radiation, and surface conduction made a complex connection unexpectedly [13]. Before this, there were three types of conventional ventilation modes have been broadly adopted in an air-conditioned indoor room, namely mixing ventilation (MV), displacement ventilation (DV), and personalized (task) ventilation (PV).

Mixing ventilation (MV), a traditional air distribution that treats the room with uniform air distribution, is still widely used. This mixing mode supplies the air into the room via a high-velocity jet and entrains the air in the enclosure space. However, energy wastage was found when the air was distributed uniformly to conditioning the unoccupied upper zone of the room [14], and the air draught occurred in the occupied zone, which leads to a poor indoor environment for the occupants [15]. Hence, an alternate way to shift the uniform air distribution to the non-uniform air distribution is by adopting an advanced ventilation mode to save the energy used for conditioning the empty upper zone [16]. The annual energy consumption of MV was diminished to at least 44.37% and 25.61% by the SV and DV, respectively [14].

Displacement ventilation (DV) has recently been used as a method of ventilation in industrial and non-industrial rooms to give better indoor air quality (IAQ) and energy savings [15]. Unlike mixing ventilation, the buoyancy forces govern the airflow in displacement ventilation. Likewise, a building has high internal heat loads where cooling is needed. Upward displacement ventilation is mainly adopted, and the air inlet is situated close to the floor. In terms of coping with the high internal heat loads, mixing ventilation performed better than displacement ventilation because it can be used for heating and cooling. However, the effectiveness of displacement ventilation is greater than the mixing ventilation [17], and the breathing air quality in the occupied zone is performed better than in the mixing system [15]. Furthermore, Schellen et al. [17] discovered that displacement ventilation has a sizeable vertical temperature difference of 6 °C.

| Elevated indoor temperature | Authorities concerned                  | Temperature/°C |
|-----------------------------|----------------------------------------|----------------|
| Countries                  |                                        |                |
| Hong Kong                  | Hong Kong S.A.R Government             | 25.5           |
| China                      | Chinese State Council                  | 26             |
| Taiwan                     | Standard of Energy Management          | 27             |
| Korea                      | Ministry of Knowledge and Economy      | 26–28          |
| Japan                      | Ministry of the Environment (MoE)      | 28             |
| Singapore                  | Singapore Standard CP 13               | 22.5–25.5      |
and a low temperature of about 18 °C close to the floor area. The vertical temperature difference and draught may bring about local discomfort to the occupants [17]. Moreover, the DV system has its restriction for the ventilation operation as the air temperature supply must be at least 18 °C, and the airflow velocity must be less than 0.2 m s⁻¹ [18].

Personalized ventilation (PV) is another type of ventilation by adjusting the preferred personal environment in terms of airflow rate, direction, temperature distribution, and turbulence intensity, which is outlandish in a conventional air-conditioning system. This ventilation works with the fresh air supplied to the breathing zone, directly influencing the local air movement close to the mouth and nose [19].

In 2005, those mentioned above stated that the mixing and displacement ventilations have limitations. Hence, Lin et al. [20] have suggested a new non-uniform ventilation mode for small to medium-sized spaces to overwhelm these problems, known as stratum ventilation. Stratum ventilation (SV) supplies a layer of fresh air into the breathing zone. The air is provided to the domain through the supply inlet that is placed along the room's sidewall at the height just above the head of the occupants [20]. The speed of supply air from the sidewall is strong enough to deliver fresh air into the breathing zone without space mixing [21], and the ventilation efficiency would be maximized. Furthermore, strati-fied thermal comfort is a promising innovation that aims to address the trouble of today’s energy savings and deliver good quality air in the breathing zone [22]. It is essential to explain the local heat transfer in detail when predicting occupants' thermal sensation in a non-uniform environment [23]. Hence, concentrating on the non-uniform thermal conditions is essential because the space conditioning systems with less energy-intensive manage to condition the occupied space and not the whole room [24].

Predicted Mean Vote (PMV) Model

A great deal of research was gained on Fanger’s PMV model in applying it to the new building design or being an assessment for the existing building in an attempt to provide a comfortable environment. Fanger’s PMV thermal sensation predicts the model expression of thermal comfort conditions as a whole human body feels thermally neutral [25] with minimum sweat secretion [26]. The PMV-index is an objective approach depends on an investigation of the heat balance equation of the human body. It was integrated with four indoor environmental parameters, and the other two refer to the occupant parameters. This model was established from a climate chamber research without windows in a controlled condition room and laboratory in 1970, [24] where participants were subjected to different thermal environments to rate their thermal comfort conditions on a sensation scale with 7-point, as seen in Table 3.

However, asking participants to report their thermal sensation might have disadvantages as the participants interfere with others, their feelings or activities. Parsons [11] claimed that there was relatively little concerning why people varied responses. Little is thought about the behavioural responses (working practice, changing posture, taking shelter, putting on clothing), what makes up a thermal pleasure, what provides a new thermal condition or adds to a comfortable ambient, except those thermal conditions that influence these mental reactions. Physical test research presented indicated that they might be some significant discrepancies between the thermal states delivered by a real human and the predicted state of comfort provided by indices such as PMV, PPD, TSV and \( t_{eq} \) [27].

Moreover, this PMV model developed in steady-state experimental for uniform environments cannot predict the transient response [26, 28] since comfortable skin temperature and sweat rate is assumed when calculating the heat exchange between the human body and surrounding ambient. Each human body segment has different physiological and geometric characteristics when occupants are constantly exposed to non-uniform thermal environments. Thus, thermoregulation of the human body allows the detailed consideration of a non-uniform climate to identify environmental conditions such as vehicle cabins, air-conditioned spaces, and outdoor [29].

Current research status of thermal comfort

The thermal comfort of humans is important to research in the health and safety field. Research on the human thermal response was conducted through subjective tests, human thermoregulatory model, thermal manikin, and CFD simulation. The results obtained can be used to predict the human thermal responses, to assess the thermal comfort in different environmental conditions, and it can also be further evaluated where there is an interest in guiding the emergency rescue, test, and design of personal protective equipment, heat strain, so on and so forth. This study mainly focused on the indoor occupants in tropical countries where the ambient is high humid and hot throughout the year. The work is

| 7-point thermal sensation Scale [25] |
|-------------------------------------|
| +3 | Hot |
| +2 | Warm |
| +1 | Slightly warm |
| 0  | Neutral |
| −1 | Slightly cool |
| −2 | Cool |
| −3 | Cold |
first carried out with thermal physiological experiments in the artificial climate chamber to obtain human physiological parameters and analyse the influence of environmental temperature on these parameters. Next, the human thermoregulatory mathematical model is established and is coupled with the thermal manikin and CFD simulation. Both coupling models were used to obtain the human physiological parameters and predict thermal comfort levels in different ambient conditions. Afterward, the data results from both coupling systems are compared with the data collected from the human physiological experiment to verify the accuracy of the numerical model and coupling systems.

1. Subjective tests (human physiological experiment)

The human thermal physiology experiment is commonly used to study human thermal response. Multiple subjects are selected to conduct experiments in an artificial climate room with controllable indoor temperature and humidity or outdoors to measure human thermal physiological parameters, such as skin temperature, metabolic rate, sweat rate, etc. Human thermal physiology experiments can provide a large amount of critical primary data, which can be used for model verification, measurement of clothing thermal insulation and moisture insulation performance, statistical analysis to obtain human thermal physiological control parameters, etc.

i) Measure the thermal physiological parameters

The human body temperature, core temperature, sweat volume, metabolic heat production, heart rate, and other human thermal physiological parameters under different working conditions are measured through the control of human body–environmental parameters.

ii) Evaluate human thermal comfort

Subjects are selected to the climate chamber under different setups such as various environmental temperatures, humidity, wind speed, and human activity level, and questionnaire surveys are distributed to them to obtain a thermal comfort index and then analyse the heat exchange between the human body and the surrounding environment.

In summary, the existing human thermal physiological experiments mainly focus on the changes in human thermal physiological parameters and thermal comfort under normal temperature environments. However, the thermal physiological parameters and thermal comfort of the human body under normal temperature and high-temperature environments are very different. The thermal physiological experiments conducted under normal temperature environments cannot provide exact data support for analysing human thermal physiological indicators and model verification in humid and high-temperature tropical countries. In addition, most of the existing human thermal physiology experiments measure the temperature of subjects as a whole instead of local body parts. Local skin temperatures and perceptual such as face and neck cooling have improved thermal comfort [30]. Therefore, it is necessary to conduct thermal physiology experiments on the human body in a high-temperature environment, to measure the skin temperature of different body parts when subjects are not wearing protective clothing, and then analyse the influence of varying temperatures on human thermal response.

2. Human thermoregulatory model

Mathematical models have been widely used in the research of human thermal response. It can predict human body temperature, blood flow, and heat exchange between the human body and the environment through metabolic heat production, clothing parameters, environmental temperature, humidity, wind speed, etc. The model can be divided into a one-node model (taking the human body as a whole) according to the physiological structure of the human body, a two-node model (taking the human body as a whole, divided into core and skin layers from the inside to the outside), and a multi-node model (the human body is divided into multiple layers).

The most representative ones are the two-node model and Stolwijk’s 25-node model. Stolwijk’s 25-node model adopted the system engineering method to establish the human thermoregulatory model by dividing the human body into control (active) and controlled (passive) systems. The National Aeronautics and Space Administration (NASA) improved the Stolwijk model by dividing the model into six local parts. Each part was made up of 4 layers, where the central blood node connects the layers of other parts through blood circulation. The most significant contribution of Stolwijk’s research results is to realize the quantitative expression of the human body control system based on the negative feedback tuning theory so that the human sweating, tremor, blood vessel expansion, and contraction can be expressed as a functional relationship between the human body temperature and set point. The hypothalamus regulates thermal physiological responses through temperature differences in negative feedback regulation [31]. The Stolwijk model is the combination of theoretical modelling and engineering requirements. It greatly influences the modelling of human thermoregulatory and has become a milestone in the research of human thermoregulatory. However, the Stolwijk model also has many shortcomings: it does not consider the differences in thermal physiological responses between the human bodies; the blood circula-
Murakami et al. [32] conducted a numerical study. The two-node model treats the human body as a whole and is not divided into multiple parts according to the physiological structure. The model is simple to calculate and easy to understand and apply. Still, the model has a poor calculation accuracy and cannot be used to analyse the physiological parameters of each part of the human body. The limitations can be seen, especially in evaluating the local thermal comfort and human physiological parameters when wearing different clothing materials.

Murakami et al. [32] conducted a numerical study on the airflow around a standing manikin by coupling a two-node model and CFD to each body part. Although the original concept of the two-node model cannot be applied to analyse the local heat transfer, the results show the use of the two-node model locally on the body surface obtained stable heat transfer characteristics in the analysis [32]. Teixeira et al. [5] coupled CFD code with a multi-node thermoregulatory model to determine body skin temperature via two methods: constant temperature and different temperature in different local parts. However, the differences between simulations and experimental results were not analysed [5].

In summary, the established human thermoregulatory model of the human body ranges from steady state to transient state, from whole to parts, and from one-dimensional to three-dimensional. To make the application of models more targeted, some models are applied to specific groups of people, and other models are used for specific environmental conditions. Each model has advantages and disadvantages; the most representative ones are the two-node model, which lays the foundation for other models, but the effect of sweat dripping is not considered in the calculation of evaporative heat exchange, and the clothing model is not accurate enough, resulting in low accuracy of the model in high-temperature environments. The three-dimensional model can get more detailed information. Still, it takes a long time to solve the model, and many parameters need to be input, which limits the application and promotion of the model.

3. Thermal manikin

Non-perspiring thermal manikins are subjected to an artificial climate chamber with controllable temperature and humidity to simulate human body heat transfer and estimate the comfort of indoor thermal environments. People and thermal manikins can detect heat loss changes on local body parts, discrete measurements of relevant parameters around the thermal manikin, which are less expensive, repeatable, and commonly practised [33].

The sensible thermal manikin is divided into 16 body segments that are electrically heated and individually controlled, which can accurately measure the heating power of the manikin. The thermal manikin control system can be operated in three ways [34]. However, this study will use only two methods for the control system due to the limited resources. One is to keep the supplied electrical power constant (constant heat control method), and the other is to keep the surface temperature of the segment constant (constant temperature control method). These two control methods will be done in the experimental analysis to study the stability and uniformity of the manikin surface temperature.

i) Measure the exchange rate between the human body and the environment and the heat transfer coefficient

The human body heat transfer coefficient is a crucial input parameter to the human thermoregulatory model, which calculates the heat transfer rate between the human body and the surrounding environment. Oliveira et al. [35] investigated the convective heat transfer coefficient for the manikin in walking movements. A standing articulated thermal manikin with 16 body segments was placed in the wind tunnel, where experimental conditions varied from natural convection to forced convection. Different wind velocities were considered, and the mean convective heat transfer coefficient for the whole body was 5.74 W m$^{-2}$C$^{-1}$, 2.48 W m$^{-2}$C$^{-1}$ and 9.43 W m$^{-2}$C$^{-1}$, for 0.51, 2.48, 9.43 m s$^{-1}$ in free convection [35]. Li et al. [36] measured the skin temperature using wireless temperature sensors at 1 min sampling intervals and collected air speed data at 0.1 m, 0.6 m, 1.1 m, and 1.7 m heights with hot wire anemometers. Mao et al. [37] studied the airflow and moisture transportation inside an experimental bedroom equipped with a sleeping person under task/ambient air-conditioning (TAC). Heat dissipation from the sleeping body is mainly generated by the heating wire on the surface of the thermal manikin. The study shows that higher temperature region near the manikin at the TAC system compared to an ordinary full air-conditioning (FAC) system [37].
ii) Couple with the thermoregulatory model

Human body experiments and mathematical modelling can study the body’s thermal response, such as skin temperature, core temperature, heart rate, and heat exchange between the human body and the environment. Testing subjects would require time-consuming experiments, significant individual differences, and large sample sizes needed, and conducting human experiments in a high-temperature environment could endanger subjects' health.

In real time, the manikin can measure the heat exchange between the human body and the environment. Still, its disadvantage is that it has no thermoregulatory function and cannot realize a dynamic response to the environment. Still, the thermoregulatory numerical model can simulate the physiological regulation function of the human body. Therefore, the combination of the human thermoregulatory numerical model and the thermal manikin is not only simulating the physiological regulation function of the human body, but also will obtain a real-time heat exchange with the surrounding, thereby making up for the thermal manikin’s inability to realize the characteristics of physiological forms and enabling the manikin to “feel” the environment's heat and dynamically adjust its heat production.

In short, the existing research mainly uses a dummy for measuring thermal resistance and moisture resistance of clothing, and the measurement procedure has been standardized. It is only necessary to control the climate chamber’s temperature, humidity, and wind speed within the range specified by the standard and then measure according to the procedure. Yet, the dummy’s heat transfer and thermal response coefficient are mainly used for human thermal comfort and heat stress assessment, clothing protection performance testing, etc.; the coupling system between the dummy and the model has only been developed for more than ten years. Stability and accuracy are still in the exploratory stage. There is no recognized system that the academic community can generally accept. Each system has its scope of application, advantages, and disadvantages. In addition, the existing research and development coupling system is used in a thermally neutral environment. Under the conditions, the accuracy is high, and the accuracy for high-temperature environments is poor; the average skin temperature obtained by the single cylindrical dummy is higher than that obtained by the multiple dummies, but the local skin temperature cannot be obtained.

4. CFD simulation

Computational fluid dynamics (CFD) analysis is used to simulate the flow and heat conduction of fluid flowing. Compared with human and manikin experiments, the CFD simulation has the characteristics of fewer parameter restrictions and no interference in the flow field. It can visually display the calculation results and carefully analyse the flow field.

i) Simulate the heat exchange between the human body and the environment

The three-dimensional geometry of the manikin is established, and different working conditions in the CFD environment are set up to simulate and calculate the amount of heat exchange between the human body and the environment. On this basis, the human body heat transfer coefficient can be obtained according to the temperature difference between the human body and the environment. Sorensen et al. [38] performed three-dimensional laser scanning of the warm body dummy to obtain a numerical dummy. On this basis, the block division, the selection of solution models, and the setting of boundary conditions were performed to calculate the convection and radiation exchange between the human body and the environment and the exchange rate. The thermal coefficient, comparing the computed results and the experimental values, shows that the simulation results have higher accuracy. The human body's convective heat transfer coefficient obtained in this study is 3.13 W m\(^{-2}\)K\(^{-1}\), which is close to the dummy experimental measurement value of 3.30 W m\(^{-2}\)K\(^{-1}\); the calculated human body radiation heat transfer coefficient is 4.83 W m\(^{-2}\)K\(^{-1}\), which is relative to the dummy measurement value of 4.70 W m\(^{-2}\)K\(^{-1}\) [38].

ii) Simulate the flow field around the human body

The three-dimensional geometry of manikin in the indoor environment is applied to analyse the influence of environmental factors such as temperature and wind speed on the flow field around the human body. Mao et al. [37] studied the effect of indoor air flow and moisture transport on a sleeping manikin experimentally and numerically. The results show the averaged absolute temperature differences between the measured and the simulated ones were
0.01 °C in the unoccupied zone and 0.02°C in the occupied zone, respectively, while using the SST \( k - \varepsilon \) (SST) model. Thus, the maximum deviation of the relative humidity ratio is about 3%, which is acceptable for comparison in this study, and the CFD method was validated [37]. Yan et al. [39] studied the significant effect of virtual thermal manikin simplification on thermal buoyancy flow in indoor environments. It was found that the simplified and smoothed model was very close to the original 3D scanned model resulting in a better computational efficiency while achieving an acceptable predictive accuracy [39].

iii) Simulate human thermoregulatory

Through the secondary development function of the CFD platform, the mathematical model provides boundary conditions for the geometry manikin and calculates the thermal physiological parameters of the human body in different environmental conditions. Yang et al. [31] coupled the CFD code with a multi-node thermal model to predict the heat transfer and human physiological parameters in real time. However, the maximum discrepancy between the simulated skin temperature and test results was more than 1.0 °C at the arm [31]. Tanabe et al. [29] established a 65 multi-node thermoregulation model based on the Stolwijk model and coupled it with the CFD simulation of skin temperature and skin wetness for every local part of the computational human body [29]. There were a lot of past studies that used the multi-node model to obtain human physiological parameters. Kilic et al. [40] adopted Gagge’s two-node model in the numerical coupling analysis due to its simplicity and well-performance. The predicted results were in agreement with available experimental and theoretical data in the literature, where the radiative heat transfer coefficient for the whole body was 4.6 \( Wm^{-2}K^{-1} \), closely matching the generally accepted whole-body value of 4.7 \( Wm^{-2}K^{-1} \) [40].

iv) Evaluation of human thermal comfort

Sevilgen and Kilic [41] researched thermal comfort and moisture transport inside a room with two heated radiators using the computational fluid dynamics (CFD) technique. The results show that wall and window materials with high heat transfer coefficients presented high heat losses from the outer surfaces and a relatively higher room energy consumption. Hence, the insulated wall and surface generated predicted mean vote (PMV) were very close to the neutral condition (PMV ≈ 0) and achieved about 60% energy savings [41].

Summarizing the existing CFD simulation research, it is found that the focuses are mainly on the heat exchange between the human body and the environment, the flow field distribution around the human body, the prediction of human thermal physiological parameters, and the evaluation of human thermal comfort. However, most of these studies analyse the normal temperature environment or the human body as a whole and rarely examine the temperature environment in tropical countries and divide the human body into multiple parts according to the physiological structure. Therefore, the existing research is unsuitable for studying heat transfer and thermo-physiological responses of various human body parts in a high-temperature environment. It is necessary to divide the numerical human body into multiple pieces according to the physiological structure and use the human body thermal response model as the numerical human body control system to achieve a high temperature. Therefore, the overall environmental simulation of the human body and its local thermal physiological response is required to be investigated.

**Overview of the Past CFD Application in Indoor Built Environment**

In the past decades, the CFD simulation has become a powerful research method widely used in indoor air distribution to estimate parameters needed to compute room air distribution, thermal comfort level and indoor air quality (IAQ) indices [42]. In most indoor spaces, human bodies play a central role, and the thermal plume significantly affects the airflow pattern around human bodies [39]. The CFD is a simple yet effective alternative to studying heat transfer, a physiological response, and airflow surrounding the numerical human model surface and the convective human body plume induced by the skin surface [31, 43].

The aim to predict the flow phenomena around a person and personal exposure raises the demand for a proper computational model of a person [44]. Some detailed information that is very difficult to obtain from experiments between the thermal environments and the human body can be better known [15]. Thus, the thermal comfort level of the numerical model in indoor space is reviewed. For this purpose, fifty-five journal articles which focus on the CFD analyses of thermal comfort in the indoor built environment in terms of their types of HVAC systems, CFD codes, target parameters, and country zones as shown in Table 4 have been comprehensively reviewed.
| No. | Year | Author | Type of HVAC system | CFD code | Target parameters | Country (Zone) | Season | Tropical country |
|-----|------|--------|---------------------|----------|-------------------|----------------|--------|-----------------|
| 1   | 1996 | [44]   | MV NS               | AV       | Contaminant concentration (ppm), Personal exposure assessment | Denmark        | NS     | Non-tropical    |
| 2   | 1999 | [45]   | DV NS               | AT, AV   | CHTC              | Japan          | NS     | Non-tropical    |
| 3   | 1999 | [46]   | MV NS               | AT, AV, RH (comfort level) | Singapore | –      | Tropical        |
| 4   | 2000 | [32]   | NS NS               | AT, AV, R, $T_{d}$ | Singapore | –      | Tropical        |
| 5   | 2001 | [47]   | DV VORTEX           | AT, AV, ACH | UK          | NS     | Non-tropical    |
| 6   | 2002 | [1]    | MV NS               | AT, AV, RH, ET, WBGT | Singapore | –      | Tropical        |
| 7   | 2002 | [29]   | NS                  |            | Japan          | NS     | Non-tropical    |
| 8   | 2003 | [12]   | FLUENT              | AT, CHTC, EDT, EDT, PMV, PD | Sweden   | NS     | Non-tropical    |
| 9   | 2003 | [48]   | DV ANSYS CFX        | AV, AT   |                   | Japan          | NS     | Non-tropical    |
| 10  | 2003 | [38]   | NS STAR-CD          | AT, AV, CHF, CHTC, RHF, RHTC | Japan | NS     | Non-tropical    |
| 11  | 2004 | [13]   | PV NS               | AV (airflow rate) | Denmark | NS     | Non-tropical    |
| 12  | 2004 | [19]   | DV, PV FLUENT 6.1   | AF, CHTC, PV | Hong Kong | NS     | Non-tropical    |
| 13  | 2004 | [49]   | NS                  | AT, AV, $T_{d}$, CHTC, RHTC |            |        |                 |
| 14  | 2006 | [50]   | NS                  |            | Japan          | NS     | Non-tropical    |
| 15  | 2007 | [51]   | NV, DV ANSYS CFX    | AT, AV, CHTC, RHTC | UK        | NS     | Non-tropical    |
| 16  | 2007 | [23]   | DV NS               | AT, AV, CHF, RHF | Japan | NS     | Non-tropical    |
| 17  | 2008 | [40]   | FLUENT              | AT, RH, CHF, CHTC, RHF, RHTC | Malaysia | –       | Tropical        |
| 18  | 2009 | [52]   | MV ANSYS FLUENT     | AT, AV, PMV, PPD | Malaysia | –       | Tropical        |
| 19  | 2009 | [53]   | DV FLUENT           | AT, Concentration | China | NS     | Non-tropical    |
| 20  | 2009 | [54]   | NS                  |            | UK             | NS     | Non-tropical    |
| 21  | 2010 | [55]   | NV ANSYS CFX        | AT, AV, RH, |            |        |                 |
| 22  | 2010 | [56]   | Front-Flow/Red      |            |                |        |                 |
| 23  | 2010 | [57]   | MV FLUENT           | PMV,$T_{d}$ |              |        |                 |
| 24  | 2011 | [58]   | DV NS               | AT, AV, AF, $T_{d}$, CHF | Hong Kong | NS     | Non-tropical    |
| 25  | 2011 | [41]   | FLUENT              | AT, AV, RH |              |        |                 |
| 26  | 2012 | [59]   | NV, MV ANSYS CFX    | AT, AV, CHF, CHTC, RHF, RHTC | Portugal | NS     | Non-tropical    |
| 27  | 2012 | [60]   | DV FLUENT           | AT, AV, RH |              |        |                 |
| 28  | 2012 | [5]    | ANSYS FLUENT        | AT, AV, RH |              |        |                 |
| 29  | 2013 | [42]   | NV ANSYS CFX        | AV, AT   |              | Ireland | Spring | Non-tropical    |
| 30  | 2013 | [61]   | MV FLUENT 12.1      | PMV, PPD, EHT | Korea | NS     | Non-tropical    |
| 31  | 2014 | [62]   | NV FLUENT 12.1      | AV, CHF, RHF, CHTC | Japan | NS     | Non-tropical    |
| 32  | 2014 | [63]   | MV ANSYS CFX        | AT, AV, Exhaled | Malaysia | –       | Tropical        |
| 33  | 2014 | [64]   | NS                  |            |                |        |                 |
| 34  | 2015 | [65]   | DV, PV ANSYS        | AT, AV, Ar, PMV, PPD | India | NS     | Non-tropical    |
| 35  | 2015 | [66]   | ANSYS FLUENT        | AT, $T_{d}$ |              |        |                 |
| 36  | 2016 | [18]   | DV ANSYS FLUENT     | AV, AT, $CO_{2}$ | Lebanon | NS     | Non-tropical    |
| 37  | 2016 | [6]    | DV NS               | AT, AV, Pollution Concentration |         |        |                 |
| 38  | 2016 | [37]   | SV FLUENT           | AT, AV, RH |              |        |                 |
| 39  | 2016 | [67]   | NS                  |            |                |        |                 |
| 40  | 2016 | [68]   | SCRUY/Tetra         |            |                |        |                 |
| 41  | 2016 | [39]   | MV ANSYS CFX        | AV, AT   |              | Australia | NS     | Non-tropical    |
| 42  | 2017 | [27]   | MV NS               | AV, AT, RH, PMV, PPD | Romania | NS     | Non-tropical    |
| 43  | 2017 | [69]   | SV NS               | AT, AF, CHTC,$T_{d}$ | China | NS     | Non-tropical    |
| 44  | 2017 | [70]   | ANSYS FLUENT        | AT, AV, PMV, PPD | UK       | Winter | Non-tropical    |
| 45  | 2017 | [71]   | ANSYS FLUENT        | AT, AF, CHTC | Australia | NS     | Non-tropical    |
| 46  | 2017 | [31]   | ANSYS 14.5          |            | China      | Summer | Non-tropical    |
| 47  | 2018 | [72]   | SOLIDWORKS          | AV, AT, PMV, | Malaysia | –       | Tropical        |
The air flow estimation depends on the solution of the flow equations, namely the mass equation, three momentum equations, and the energy equation.

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These equations are all-time averaged. Most of these past studies have neglected the influence of heat and moisture produced by every part of the human body in an indoor space, where, for instance, a high number of students in an auditorium, a business occasion, or a social gathering is held. Under tropical weather conditions, the suitable parameters of temperature, moisture content airflow as well as the body and skin temperature for an acceptable thermal environment in the enclosure room must be known to achieve energy conservation. Ansys Fluent is currently the most widely used method in predicting indoor air parameter distributions [43] and is used for heat transfer, air flow field, and moisture transport analysis [40]. It is a commercially available CFD code as it utilizes a control volume strategy to change coupled nonlinear equations into algebraic equations, which can be solved numerically [12].

**Hot climate in tropical countries**

A hot environment triggers the body temperature rise and causes skin vasodilation, where sweat is emitted on the skin surface to enable heat loss by evaporation to maintain a stabilized core temperature. This thermoregulatory function was performed by eccrine glands, which can be found in the trunk, forehead, neck, back of forearm, thigh, hands, soles, and palms. In practice, the temperature of the skin is often recorded as the mean skin temperature \( T_{sk} \), commonly calculated from the weighted average of temperature taken from several body parts [11].

In the hot atmosphere, the concern is always given to how to lower interior temperature and offer a comfortable environment to the occupants by using air-conditioning ventilation or raising the air movement [11]. Occupant’s preferences for the hot season were low humidity and high air wind speed. These preferences were due to the high moisture evaporation rate over the skin surface and frequent convective heat exchange between the skin and surrounding air. Moreover, the air temperature significantly influences the preferences towards the surrounding [26]. However, the study by Xue et al. [46] claimed that increasing the ventilation velocity has a minor effect on helping the human body emit heat. Research studies have shown that the human average body temperature is decreased from 35.56 °C at 25.5 m³/h to 35.41 °C at 34.0 m³/h. This implies that the two ventilation rates recommended by ASHRAE applied in other areas of the world may not be suitable for tropical countries due to the high temperature and moisture levels throughout the year [46]. From here, the effect of heat and moisture generated by the human body cannot be neglected in crowded indoor spaces, and the indoor condition is affected by heat release from the body. The convective heat released from the body generates a microclimate, which consists of the boundary layer around the human body and an ascending heat plume above the head [64].

It can be seen that the thermostat set point for the air conditioner in tropical climate countries is usually tuned to the lowest temperature, which is about 16 °C. However, the desired low temperature was not in line with the recommendation stated in Malaysia Standard (MS) 1525 (2007). The national standard has suggested that the indoor temperature should fall in the scope of 23 °C to 26 °C and that every 1 °C decline in the indoor regulator will bring about an 8% increase in electricity cost [52]. For example, the air-conditioning system has high energy consumption under most circumstances and minimizing it would help reduce carbon emissions. The past studies focused on using a validated CFD simulation model and computational human manikin to study micro-environmental control systems and restore people’s thermal comfort, as shown in Table 4. However, only a few tropical countries focus on thermal comfort in indoor built environments, as shown in Fig. 2.

Throughout the review, only a few tropical countries focus on stratum ventilation (SV), as most still use mixing ventilation in the indoor space. The mixing ventilation is not encouraged due to the factor of hygiene. For example, the outbreak of Coronavirus is rapidly spreading throughout the world. So far, no cure has been found, and humans are encouraged to stay home or work away to avoid themselves from crowded places. From here, the high volume of occupancy in the building aims to protect from the outbreak of diseases in a tropical country and requires a better ventilation system design. Hence, understanding the movements of air particles surrounding the human body may lead to effective control measures and ensure the comfort level of occupants when they spend their days indoors for long hours.

**Modelling computational thermal manikin**

Before this, one of the first questions to arise is how to quantify thermal comfort. The human comfort model has emerged over the past 30 years as a promising and versatile method to help with the design and assessment of both indoor rooms and the open-air surrounding environment [77]. Occupants are one of the buildings’ major heat sources or flow obstacles [80]. Their role will increase in the future due to energy-savings efforts in buildings and the growth of low-power-consuming office equipment. Moreover, the thermal plume generated from the human body has proved dramatically influenced the room air movement, transport of pollutants, thermal comfort level, and inhaled air quality. Employing the real human body when studying the thermal plume over the body is problematic due to the unstable human behaviour and measurement duration of up to several weeks [34, 80]. Hence, using a human simulator with
a computational thermal manikin (CTM) instead eliminates some of the difficulties.

Heat transfer around the human body

When subjecting computational thermal manikin in stratum ventilation (SV) modelling, some critical features should be considered, such as the geometry of CTM, heat transfer model, boundary conditions, and thermoregulation system.

Human body geometry

There is no standard for designing computational thermal manikins (CTMs), and it is based on the real human body’s shape [43]. Gao and Niu [19] mentioned that a simple human body geometry requires fewer computational sources and a shorter time to run the modelling and grid generation. However, a definite geometry can provide a better result that is close to the real situation. Thus, it is necessary to understand how complicated is a shape for human body geometry required to yield the ideal CFD results. In general, three different postures are applied according to the situation: sleeping, standing, and seated. The degree of geometric complexities of human modelling relies upon the purpose of simulations involving them. A less detailed geometry is required when the numerical study emphasizes global airflow patterns in a ventilated indoor space. At the same time, an accurate geometry is necessary when the study focuses on the local flow pattern around every part of the human body [81].

In 1996, three different CFD models were proposed by Brohus and Nielsen [44] that made up of simple rectangular geometry. The study aimed to predict personal exposure to a contaminant source in a ventilated room. They claimed that it was essential to consider the local influence of the person itself in a ventilated room when carrying out the full-scale measurement, or else significant errors might occur in the personal exposure assessments [44]. Another study revealed that accurate and detailed human body geometry could be developed by employing a 3-D laser scanning method. A nude and seated female occupant is presented in CAD with a 4 mm small patch size, where it has a total number of 2000 patch sizes, and the surface area is 1.5696 m². Some patch sizes are refined to save computational resources. For example, the surface patches are coarsened for the back and abdomen [19].

Furthermore, in the research work of Nielsen [13], the difference between different shapes of computing simulated person (CSP) geometry was investigated numerically. The simulated human body geometries were made up of a simplified rectangular CSP1 (1.3 m height and area of 1.61 m²) and a detailed CSP2 (1.36 m height and area of 1.52 m²) concerning what horizontal field (0.05 m s⁻¹). The high vertical speed at the back of CSP1 was found to have a huge concentration near the face part, most probably because of the absence of two separate legs [13]. Therefore, it is important to work with a complex model. This statement agreed well with studies of Brohus and Nielsen [44], where the inclusion of “legs” and “head” in a manikin.

![Fig. 1 CFD code used in analyses of thermal comfort](image1)

![Fig. 2 Respective countries focused on previous studies](image2)
model has a significant influence on the flow fields close to a person. When a 1.7 m height of the “torso” model is exposed to a relatively high horizontal distance of air to the breathing zone, the entrainment occurs along the entire body. The model with “leg” and “head” only exerts the entrainment in the height of 0.8 m to 1.47 m, which is the upper part of the body [44]. But, they do not mention what complexity the computer simulated person (CSP) must be to examine local flow close to the person accurately. A complex CSPs need fine and unstructured grids to mesh the geometry model precisely (≈ about $10^5 - 10^6$ grids), and the calculation time for a single node personal computer can be up to several days [53].

Table 5 shows the human body geometries adopted by other researchers. In the review studies, modelling of computational thermal manikins plays the main role in identifying the distributions of air velocity and air temperature in the enclosure room of buildings, which inevitably affect indoor air quality, thermal comfort level, and energy quality used. The microenvironment around an individual is greatly affected by the convective heat transfer, radiant heat transfer, and buoyancy-driven plume surrounding the human body [43]. Using computational thermal manikin is essential as it is practically challenging to quantify the detailed skin temperature distribution. Those estimations cannot account for independent outcomes of radiation and convection [23]. Moreover, the manikin was used as a heat source and obstacle to observe the temperature and flow conditions [64]. In the simulation study of Yan et al. [82], three simplified computational thermal manikins (CTM) were rebuilt based on the 3D scanned manikin. They found out that the smoothed manikin yields an error of less than 5% compared to the baseline case, whereas the predictive errors were 14.1% and 18.1% concerning skeleton based and surface area based [39].

Thus, the upper body of the human model is considered the most vital section in evaluating thermal comfort and the airflow conditions around the body can be the inputs for a comfort model to assess the thermal sensation and thermal comfort of occupants in transitional spaces, where to determine how comfort can be attained via the air movement while changing the attire properties of the upper body part. The research has focused on an upper body manikin due to the upper part’s thermal state being the most influential area over the thermal sensation of the overall body and discomfort in hot and warm conditions, mainly the trunk, neck and forehead [7].

Thermal manikin with a rectangular shape was adopted in the SV system because the simple geometry is adequate and efficient in assessing the global airflow pattern [80]. Thus, this study focuses only on the upper body part. The head and torso were found to have high skin temperature distribution compared to others, in which it ranges from 27.8 °C and 35.9 °C. Large deviations were found in the legs [38], and the simulation method could not precisely estimate the skin temperature of the limbs, even though it was in a uniform room condition [23].

**Heat transfer between human body and microenvironment**

Recent CFD modelling has demonstrated heat transfer processes between the body and the environment instead of within the human body; for instance, the computational thermal manikin does not have the thermoregulation function to react with the stimulus from the surroundings as a real human body does [43]. The computational thermal manikin is mainly employed for the body heat dissipation in CFD, including heat loss of sensible and latent via convection and heat loss by radiation. Radiative heat transfer occurs inside the surface of the chamber, such as walls, seats and attire of the occupants, due to temperature differences [50]. The flow formation in internal space can arise from natural convection caused by the temperature difference, buoyancy forces, or forced convection generated by an air-conditioning system. Thus, the computational thermal manikin was used to evaluate the thermal comfort of occupants in an enclosed room. The CFD simulation approach can be adopted to calculate the radiation and convection of each body segment [61].

Concerning thermal comfort, heat transfer on the skin surface is a dominant factor in verifying the skin surface temperature [62]. In human physiology, it is crucial to identify the heat transfer distribution over the entire body in detail as the local thermal sensation of each body segment is directly heated or cooled [23]. Moreover, the heat transfer values obtained from the CFD simulation are then employed as inputs to the human thermoregulation model to predict the physiological reactions of the occupant. The radiative heat flux is determined by the CFD simulation method. The radiative heat flux was subtracted from the total sensible heat flux (measured in the physical field) to obtain convective heat flux [9, 31, 53, 62].

\begin{equation}
Q_c = Q_T - Q_r
\end{equation}

\begin{equation}
h_r = \frac{Q_r}{(T_{sk} - T_w)}
\end{equation}

\begin{equation}
h_c = \frac{Q_c}{(T_{sk} - T_a)}
\end{equation}

where $Q_r$ is the convective heat flux (W m$^{-2}$); $Q_T$ is the total sensible heat flux (W m$^{-2}$); $Q_c$ is the total radiative heat flux.
| Year | Author | CFD models | Model Posture | Height/m | Surface area/m² | Head load (W) | Convective heat flux/W m⁻² | Radiative heat flux/W m⁻² | BS (no.) | Physical Manikin | Summary |
|------|--------|------------|---------------|----------|-----------------|--------------|-----------------------------|-----------------------------|----------|----------------|---------|
| 1996 | [44]   | Rectangular geometry | Standing | 1.7 | 1.62 | 40.5 | 25 | | | Fibre-armed polyester shell Heat: Nickel wires | Inclusion of leg may be important |
| 1999 | [45]   | Standing | | 1.651 | 1.688 | 20 | NS | | | | |
| 1999 | [46]   | Rectangular | Sitting | 1.69 | NS | NS | NS | 25 nodes | NS | Applied Stolwijk thermoregulation model |
| 2001 | [47]   | Detailed | Standing/Sitting | | 1.60 | | | | | 1 mm Al sheet, heating elements |
| 2002 | [1]    | Simple body lump | Sitting | 1.69 | | | | 61 | | |
| 2002 | [29]   | Simplified | Standing | 1.7 | 1.53 | | | | | |
| 2003 | [48]   | Rectangular versus detailed model | Sitting | 1.6 | | | | 16 | | |
| 2003 | [38]   | Seated | | 1.594 | 89.67 | 4.83 W m⁻³ K | 16 | | |
| 2004 | [13]   | Simple rectangular VS detailed model | Sitting | 1.3–1.36 | 1.61–1.52 | NS | NS | 1 | NS | Necessary work with a detailed model |
| 2004 | [19]   | Detailed human model | Sitting | 1.65 | 1.5696 | | | 16 | | |
| 2004 | [49]   | 3 | | | | | | | | |
| 2006 | [50]   | Detailed | Seated | 1.87 | | | | 62 | | |
| 2007 | [51]   | Detailed | Sitting | | | | | 59 | | |
| 2007 | [23]   | Detailed | Seated | | | | | | | Surface covered: heating wire and temperature sensors |
| 2008 | [40]   | Gagge's two-node model | Standing | 1.70 | 1.81 | 33.5/14.9 | 14.9/21.2 | 17 | | |
| 2009 | [53]   | Detailed CSP, Simple CSP, Rectangular CSP, Improved CSP | Standing | 0.706–0.935 | 38 | 26.912–53.824 | 53.2 W | 4 | | |
| 2009 | [54]   | Details, IESD-Fiala model | Standing | | | | | | 65 | |
| 2010 | [55]   | Adopt IESD-Fiala model | Standing | 1.86 | 3.07 | 4.77 | 59 | | | |
| Year | Author | CFD models | Model Posture          | Height/m | Surface area/m² | Head load (W) | Convective heat flux/W m⁻² | Radiative heat flux/W m⁻² | BS (no.) | Physical Manikin | Summary                                                                 |
|------|--------|------------|------------------------|----------|-----------------|---------------|-----------------------------|---------------------------|----------|------------------|--------------------------------------------------------------------------|
| 2010 | [56]   |            | Standing               |          |                 | 1.532         |                             |                           |          |                  |                                                                          |
| 2010 | [5]    |            | Sleeping               |          |                 |               |                             |                           |          |                  |                                                                          |
| 2011 | [58]   | Detailed   | Sleeping               |          |                 | 1.7           | 1.19                        |                           |          |                  |                                                                          |
| 2011 | [41]   |            | Sitting                |          |                 | 1.68          | 1.6–1.63                   |                           |          |                  |                                                                          |
| 2012 | [59]   | Detailed   | Sitting                |          |                 | 1.68/1.2      | 1.6–1.63                   |                           |          |                  |                                                                          |
| 2012 | [34]   | Cylinder, Rectangular, dummy and TM | Sitting |       | 1.7           | 1.19          |                             |                           |          |                  | Cylinder and rectangular are not recommended, leg and thigh are important |
| 2013 | [42]   | Simple     | Sitting                |          |                 | 1.68          | 1.596–1.638                | 22.8                      | 53.2     |                  | Useful to use a simple model for thermal evaluation                     |
| 2013 | [61]   | Simple versus Detailed | Standing | 166.8–167 | 1.723–1.702 | 76            | 22.8                      | 53.2                      | 16       | NA               |                                                                          |
| 2014 | [62]   | Detailed   | Sitting                |          |                 | 1.350         |                             |                           |          |                  |                                                                          |
| 2014 | [63]   | Boxman, Detailed | Seated | 1.7           | 1.19          |                             |                           | 53.2     |                  |                                                                          |
| 2014 | [35]   | Detailed   | Standing               |          |                 | 1.7           | 1.64                       |                           |          |                  |                                                                          |
| 2015 | [65]   | Blocks (head, torso and two legs) | Sitting | 1.83         |               |                             |                           | 53.2     |                  |                                                                          |
| 2015 | [66]   |            | Sitting                |          |                 | 1.89          |                             |                           |          |                  |                                                                          |
| 2016 | [80]   | Rectangular | Sitting               |          |                 | 1.2           |                             |                           |          |                  |                                                                          |
| 2016 | [6]    | Detailed   | Sitting                |          |                 | 1.68          |                             |                           |          |                  |                                                                          |
| 2016 | [68]   | Detailed   | Standing               |          |                 | 1.71          | 1.64                       |                           |          |                  |                                                                          |
| 2017 | [37]   | Sleep      | Sleeping               |          |                 |               |                             |                           |          |                  |                                                                          |
| 2017 | [69]   | Cylinder, Details | Sitting | 1.96         |               |                             |                           |          |                  |                                                                          |
| 2017 | [71]   |            | Standing               |          |                 |               |                             |                           |          |                  |                                                                          |
| 2017 | [39]   | Detailed, smooth feature, skeleton and rectangular | Sitting | 1.23         | 1.596–1.638 | 35.6          | 22.31–21.73                |                           |          |                  | 3D-scanned CTM Manikin with surface feature smooth was recommended        |
| 2017 | [31]   | Detailed   | Sitting                |          |                 | 1.68          | 2.82 W m⁻² K              |                           |          |                  | 3D-laser scanning                                                        |
| 2018 | [73]   | Complex    | Sitting                |          |                 | 1.68          | NS                        | NS                        | NS       |                  | Heat: Nickel wires                                                        |
(W m\(^{-2}\)); \(h_r\) is the radiative heat transfer coefficient (W m\(^{-2}\) K); \(h_c\) is the convective heat transfer coefficient (W m\(^{-2}\) K). The total sensible heat load released from the manikin is equivalent to the heat loss from an average human body [6] or heat supplied to each body segment of the thermal manikin in steady-state conditions [62].

The sensible and evaporative heat loss between the human body and ambient conditions can be identified through the coupling of the field and thermoregulation model [46]. To couple, the representative points towards particular body segments of the thermoregulation model are picked for inputting the values of air velocity, moisture content and air temperature. It is noticed that the human body always acts as a heat and moisture source in an experimental chamber to predict the moisture transfer rate between the human skin surface and thermal condition. The mass fraction of the water vapour for skin surfaces is chosen as 10 g H\(_2\)O/kg air [41]. Moreover, it is assumed that the heat flux \(q\) and evaporative heat loss \(E_{rsw}\) are uniformly released from the skin surface to the surrounding. The thermal conductivity of the body segment was defined as a function of temperature using user-defined function (UDF). Heat dissipation from the skin surface of the body segment is input to the thermoregulation model [79].

**Temperature:** The heat flux \(q\) is released based on Fourier’s law:

\[
q = -k\frac{dT}{dx_i}
\]

(4)

Moisture content: The mass transfer by diffusion of the skin surface is caused by concentration gradient based on Fick’s law of diffusion:

\[
G = \frac{E_{rsw}}{h_{fg}} = \rho D \frac{\partial \omega}{\partial x_i}
\]

(5)

where enthalpy of vaporization \(h_{fg}=2430.5\) kJ kg\(^{-1}\); \(D\) is the mass diffusion coefficient.

In the numerical work, moisture generated from the human body was eliminated through diffusion into the uprisling air in the enclosed room. There is high absolute humidity in the unoccupied top areas of the room compared to the occupied floor areas. Thus, absolute humidity is decreased with raising airflow. The calculated mass fraction of H\(_2\)O varied from 9.6 to 9.8 g H\(_2\)O/kg air [41]. The sum of the enthalpies of dry air and the water vapour is the total enthalpy of the moist air as follows:

\[
H = m_a h_a + m_v h_v
\]

(6)

Enthalpy of air per unit mass of dry air: \(h_a = C_p T + \omega h_v\)

Humidity ratio \(\omega = m_v/m_a\)
At the same temperature: Enthalpy of water vapour, \( h_v \approx \)

\[ \text{Enthalpy of saturated vapour:} \quad h_v \approx 1.82(T - 273.15) + 2501.3 \text{kJkg}^{-1} \]

### Boundary condition

In both experimental and simulation work, the computational manikin is always used as a heat generator, and its thermal boundary condition can be demonstrated in three different methods, namely consistent skin surface temperature, constant skin surface heat flux, and total heat power production via the volume called a constant volumetric heat output for the body [43]. There are many ratios of convection to radiation (C:R); heat transfer was found in the literature review to obtain an accurate temperature distribution over the human body in a room chamber 30:70 for standing CSPs [53]; 30:70 for a standing simulated human model [61], or 30:70 for a standing person [51].

In the research of Hajdukiewicz et al. [42], the measured outdoor climate gave the boundary conditions to the simulated chamber. At the same time, a network of wireless sensors inside the room provided air temperature and air speeds for model calibration [42]. At the inlet boundary conditions, the values of the turbulent kinetic energy, \( k_0 \), and the dissipation of the turbulent kinetic energy, \( \epsilon_0 \), can be calculated from [46, 83].

\[ k_0 = 1.5(u_0)I_0 \quad (7) \]

\[ \epsilon_0 = C_\mu \frac{k_0^{1.5}}{l_0} \quad (8) \]

where \( u_0 \) is applied at \( x = 0 \), \( u_0 \) = Uniform Velocity (m s\(^{-1}\)), \( I_0 \) = Turbulence intensity (%), 40% base on [83]. \( l_0 \) = Turbulent length scale (m), 0.5 m base on [83].

### Turbulence model

A proper turbulence model must be selected in the CFD solver to predict the turbulent field accurately. The airflow distributions were determined by three-dimensional, steady-state Reynolds averaged Navier–Stokes (RANS) equations combined with the mass and energy equations [69]. Moreover, the SIMPLE algorithm employs a second-order upwind scheme for the convective terms. The coupled equation governs the energy interaction between the body and the surrounding for mass, momentum, and energy [12]. The controlled equations are as follows [8, 37, 69, 82, 84].

#### Conservation of mass

\[ \frac{\partial}{\partial x}(\rho U) + \frac{\partial}{\partial y}(\rho V) + \frac{\partial}{\partial z}(\rho W) = 0 \quad (9) \]

#### Conservation of momentum

For X-direction (U-momentum)

\[ \frac{\partial}{\partial x}(\rho U^2) + \frac{\partial}{\partial y}(\rho UV) + \frac{\partial}{\partial z}(\rho UW) = -\frac{\partial P}{\partial x} + \mu \frac{\partial^2 U}{\partial x^2} \quad (10) \]

- For Y-direction (V-momentum)

\[ \frac{\partial}{\partial x}(\rho UV) + \frac{\partial}{\partial y}(\rho V^2) + \frac{\partial}{\partial z}(\rho VW) = -\frac{\partial P}{\partial y} + \mu \frac{\partial^2 V}{\partial y^2} + \rho g \quad (11) \]

- For Z-direction (W-momentum)

\[ \frac{\partial}{\partial x}(\rho UW) + \frac{\partial}{\partial y}(\rho VW) + \frac{\partial}{\partial z}(\rho WW) = -\frac{\partial P}{\partial z} + \mu \frac{\partial^2 W}{\partial z^2} \quad (12) \]

where \( P \) is the static pressure and \( \mu \) is the fluid viscosity.

#### Conservation of thermal energy

\[ \frac{\partial}{\partial x}(\rho U\bar{T}) + \frac{\partial}{\partial y}(\rho V\bar{T}) + \frac{\partial}{\partial z}(\rho W\bar{T}) = \frac{\partial}{\partial x}\left( \mu \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y}\left( \frac{\mu}{\sigma} \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z}\left( \frac{\mu}{\sigma} \frac{\partial T}{\partial z} \right) \quad (13) \]

where diffusion coefficient \( r = \frac{\mu}{\sigma} \) and \( \sigma = \frac{C_k}{\mu} \) is the Prandtl number of fluids.

Table 6 lists the turbulence models used in past studies of the personal microenvironment. The behaviour of the turbulence flow in a non-uniform environment can be accurately predicted by two turbulence \( k \) – \( \epsilon \) models. These models are simple yet robust and have always been employed by past and current studies to simulate indoor airflow. Temperature distributions and air velocity characterize the heat transfer and airflow within the indoor space. They are identified by the solution of governed equations stated above until a converged solution is obtained [12]. In the airflow simulation near the human body, the most frequently utilized turbulence models are the three \( k \) – \( \epsilon \) models, namely the standard (SKE), realizability (RKE) and re-normalization group (RNG) models [73].

Most research studies have followed the references given in the CFD manuals for indoor CFD applications and selected a suitable turbulence model among a well-known one. For the standard \( k \) – \( \epsilon \) model (SKE) model, the equation of SKE was derived from the phenomenological assumptions and empiricism as per ANSYS 2015 to examine the non-uniform cases. However, this model has come with some limitations, such as the inaccurate outcome of the spreading rate of round planes.
and the expectation of $k$ can be unphysical close to stagnation positions [73]. In addition, the SKE model is an ideal option among the numerous turbulence models, and the lower Reynolds number is precise in predicting the heat generated from the human skin surface. Thus, SKE and RNG $k - \varepsilon$ model are adequate to predict air movement in indoor areas [81].

Among the RANS models, the RNS is broadly used because the RNG model covered the better dissipation rate equation for quickly strained flow, choices of the swirl-dominated flows, and different viscosity models [73]. The experiment was done by Yan et al. [82] applied the RNG model to forecast the indoor airflow surrounding the surface of the computational thermal manikin due to its well-documented performance [39]. In the studies of Lin et al. [21], they examined the performance of three parameters, namely the air temperature, air movement, and CO$_2$ concentration in a personal office room under the SV ventilation mode. The field data of the three parameters were collected to validate RNG turbulence model and found out the simulated outcomes from the model were agreed well with the field data [85]. In addition, the RNG can better simulate the indoor air than the SKE [12] because of its excellent simulation outcomes for the air velocity, air temperature, and contaminant distributions in the indoor space [53].

RKE is also popular because its modified transport equation for the term $\varepsilon$, which is expressed from the transport equation of the mean square vorticity fluctuation [73]. For the SST model, it is best to agree with the experiment compared to LRNKE and $v^2 - f$ model while studying the convective heat transfer coefficient of the body under high velocities in the case of front-way direction. However, it has performed poorly in the direction of the sideways, especially under the velocities higher than 6 m s$^{-1}$ [62]. A good agreement is found in the SST model than the SKE model when comparing its simulated data with the field data [59].

An investigation analysed the SKE and RNG to anticipate the airflow in a food-processing tidy room. It indicated that the re-normalization group (RNG) $k - \varepsilon$ model was more appropriate in clean room simulation when compared to the SKE due to its ability to present three-dimensional flow with anisotropy [84]. Another research assessed the performance of the SKE, RKE, and RNG by comparing the air velocity, air temperature and ventilation effectiveness. Every turbulence model is verified with two pressure interpolations, broadly adopted in the past studies, namely the second order and PRESTO schemes. They found out that the RKE model best agrees with the measured data of the air temperature and ventilation effectiveness. However, the air velocity agrees well with the RNG model [73].

In Fig. 3, the performance of various turbulence models in modelling indoor air flows has been continuously evaluated by many researchers. No particular turbulence model can ideally control all the flow components such as the jet flow, momentum-driven flow, stratified flow and buoyancy flow. Hence, it is usually necessary to adopt a more accurate model to predict the microenvironment that is close to the human body and is characterized by a complex flow [43]. Up to the present moment, the RNG $k - \varepsilon$ turbulence model is still the most commonly employed, which is 42% as seen in Fig. 3 in practical engineering applications. Encouraged by these results, the RNG $k - \varepsilon$ turbulence model is adopted in the current research to simulate the airflow field.

**Table 6 Turbulence model**

| No. | Year | Author | Turb. |
|-----|------|--------|------|
| 1   | 1996 | [44]   | SKE  |
| 2   | 1999 | [45]   | LRNKE|
| 3   | 1999 | [46]   | HRNKE|
| 4   | 2000 | [32]   | LRNKE|
| 5   | 2002 | [1]    | HRNKE|
| 6   | 2003 | [12]   | RNG  |
| 7   | 2003 | [48]   | RNG  |
| 8   | 2003 | [38]   | LRNKE|
| 9   | 2004 | [19]   | SKE, RNG|
| 10  | 2007 | [23]   | LRNKE|
| 11  | 2008 | [40]   | RNG  |
| 12  | 2009 | [53]   | RNG  |
| 13  | 2010 | [55]   | $k - \omega$ shear stress transport|
| 14  | 2011 | [58]   | LRNKE|
| 15  | 2011 | [41]   | RNG  |
| 16  | 2012 | [59]   | SKE, SST|
| 17  | 2012 | [60]   | SKE  |
| 18  | 2012 | [5]    | RNG  |
| 19  | 2013 | [42]   | SKE  |
| 20  | 2013 | [61]   | RNG  |
| 21  | 2014 | [62]   | LRNKE, SST, $v^2 - f$ model|
| 22  | 2014 | [63]   | RNG  |
| 23  | 2014 | [64]   | RNG  |
| 24  | 2015 | [65]   | SKE  |
| 25  | 2016 | [18]   | SKE  |
| 26  | 2016 | [68]   | LRNKE|
| 27  | 2017 | [27]   | RNG  |
| 28  | 2017 | [69]   | SST  |
| 29  | 2017 | [39]   | RNG  |
| 30  | 2017 | [31]   | LRNKE|
| 31  | 2018 | [73]   | SKE, RKE, RNG|
| 32  | 2019 | [77]   | SKE  |
Grid generation

Grid verification investigation occurred after the first CFD model was created after the field visits and measurements by checking the mesh quality [24]. The grid creation of the body has some particular demands on grid factors such as the grid size, grid topology, grid shape, consistency of the grids with the body geometry so on and so forth because the geometry of the human body is complex. The accuracy of a simulation outcome highly relies upon the quality of the grid and complex grid generation that require plenty of time and human labour in the entire process of CFD calculation [81]. Table 7 shows the grid distribution in the three-dimensional computational domain and geometry of thermal manikins carried out by previous research.

To ensure the simulation outcomes do not change significantly (or converge) in rising mesh cell numbers, numerous runs of the initial CFD model are conducted on various mesh sizes. The outcomes are compared to obtain the grid independence solution [42]. If a significant modification is made to the corresponding model, grid independence or a convergence generation should be repeated [24]. Thus, the grid convergence index (GCI) is adopted by Hajdukiewicz et al. [42] studies. They set the target to achieve a satisfactory convergence by using the criteria 0.01% of the root mean square residuals for continuity and momentum equations, while 1% for the energy preservation [42].

Moreover, grid independence is tested in the research of Yan et al. [39] to check the mesh quality. The three-dimensional computational domain is discretized by employing unstructured tetrahedron mesh, and the mesh’s quality is kept uniform for the three simplified CTM to avoid any possible errors. The entire grid numbers of three simplified cases are within the range of 1.9–2.4 million [39].

The ventilated room is usually divided into two sections (inner and outer):

i. The cuboid enclosing the human body is broken with an unstructured grid (Tetrahedron), and the total

\[
\text{Wrm}(i,j) = \text{Err}(i,j), \quad \text{Cld}(i,j) = 0
\]

\[
\text{Cld}(i,j) = -\text{Err}(i,j), \quad \text{Wrm}(i,j) = 0
\]

where \(i\) represents the segment number; \(j\) represents the layer number. All the balance equations are numerically solved by means of the Runge–Kutta–Merson approach in Fortran. This technique is selected to solve a different kind of differential equations due to its simplicity and robustness. The Fortran program helps to estimate the temperatures and the water vapour concentration as the function of time in various body segments [5].

As stated previously, the computational human manikin does not incorporate the thermoregulation models to determine the heat produced from the body by default and could not respond to the environment as humans do [31]. But the non-uniform environment and transient conditions are constantly encountered by occupants in the buildings and places, where tasks involving different temperature sources or surfaces and the occupants’ thermoregulation greatly rely on the local heat transfer characteristics [43]. Thus, the thermal model is developed to decide the thermal reactions of the human body with the inputs such as the human activity level, attire properties, and ambient conditions predicted from CFD. In addition, the human thermoregulatory model is used to simulate heat production [51].

To better understand the heat transfer and physiological responses in a hot environment, it is essential to couple the CFD with the thermoregulation function [31]. Undoubtedly, the coupled system is a valuable method to help researchers and consulting engineers evaluate the effect of design decisions on individual thermal comfort [55]. Coupling numerical models in simulation work are essential because simulation outcomes are presented for academic and real-life applications [50].
| Year | Author | CDD | Surface of manikin | New wall treatment (NWT) | Rad Iteration | Grid independence solution | TCN | Max $y^+$ | NBL (FLH, mm) |
|------|--------|-----|---------------------|-------------------------|--------------|---------------------------|-----|----------|---------------|
| 1996 | [44]   | 2.44×1.2×2.46 | Rectangular grid | SWF | NS | NS | NS |  | <5 |
| 1999 | [45]   | 2.6×2.2×2.7 | NS | NS | NS | 5000 |  |  |  |
| 1999 | [46]   | 19.6x8×2.8 | NS | NS | NS |  |  |  |  |
| 2000 | [32]   | 2.6×2.2×2.7 | NS | NS | NS |  |  |  |  |
| 2001 | [47]   | 2.78×2.78×2.3 | NS | NS | NS |  |  |  |  |
| 2002 | [1]    | 2.6×2×2.5 | NS | NS | NS |  |  |  |  |
| 2002 | [20]   | 6×2.5 | NS | NS | NS |  |  |  |  |
| 2003 | [12]   | 3.5×3×2.8 | NS | NS | NS |  |  |  |  |
| 2003 | [48]   | 4×2×2.7×3.4 | NS | NS | NS |  |  |  |  |
| 2004 | [38]   | 2.95×2.95×2.4 | Prism, Unstructured grid (TET), structure grid (HEX) | SWF | 20 (0.2) |  |  |  |  |
| 2006 | [5]    | 4×2×1.4×2.4 | Prism shape cell, TET | S2S |  |  |  |  |  |
| 2008 | [39]   | 2.2×2.6×2.7 | TET, Structured | S2S |  |  |  |  |  |
| 2009 | [49]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2010 | [50]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2011 | [51]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2012 | [52]   | 4×4×1×4 | Prism, TET, HEX | S2S |  |  |  |  |  |
| 2013 | [53]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2014 | [54]   | 4×4×1×4 | Prism, TET, HEX | S2S |  |  |  |  |  |
| 2015 | [55]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2016 | [56]   | 4×4×2.4 | Prism, TET, HEX | S2S |  |  |  |  |  |
| 2017 | [57]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2018 | [58]   | 4×4×2.4 | Prism, TET, HEX | S2S |  |  |  |  |  |
| 2019 | [59]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2020 | [60]   | 4×4×2.4 | Prism, TET, HEX | S2S |  |  |  |  |  |
| 2021 | [61]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2022 | [62]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2023 | [63]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2024 | [64]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2025 | [65]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2026 | [66]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2027 | [67]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2028 | [68]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2029 | [69]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2030 | [70]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2031 | [71]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2032 | [72]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2033 | [73]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2034 | [74]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2035 | [75]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2036 | [76]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2037 | [77]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2038 | [78]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
| 2039 | [79]   | 3×3×2.5 | TET, Structured | S2S |  |  |  |  |  |
ii. The remaining space is discretized with structured grids, and the total cell number outside the cuboid is 223,228 [19, 31, 53].

Connect the inward and external areas by the “arbitrary couples” [31] in which they assure a proper interpolation of the dependent variables across the interface [38].

Grid independence is verified by checking the mesh quality in ANSYS ICEM and GCI. Another study evaluated the surface of CSP using prism grids and tetrahedral grids and pointed out that prism grids are ineffectively for multiple linked domains. Prism grids are used only when the domain areas have viscous boundary layers. This could make the grid-generation process time-consuming and tedious compared to the tetrahedral grids [61]. Another study has done this in which the extruded triangular prisms were put over the boundary layer around the computational manikin [31]. During the coupled simulation convergence, the skin surface temperatures of the thermoregulation model and flow models have various values during the iterative plan, yet they quickly converge to one another [50].

The surface-to-surface radiation type model (S2S) is used to compute radiative heat loss modelled by the discrete beam approach [5, 31]. The net radiation methodology simplifies the S2S model, where the surface is divided into a limited number of big panels with uniform radiation flux, temperature, and emissivity [50]. Moreover, the energy exchange between the two surfaces relies on parts of various sizes, separation distance, and orientation [40]. In fact, the near-wall formulation decides the exactness of the wall shear stress and the heat transfer prediction, and it significantly affects the growth of boundary layers [59]. Inflation layers accurately estimate the flow performance at the boundary layer. In addition, non-dimensional wall distance in manikin surface $y^+$ [18, 43, 59] is defined as:

$$y^+ = \frac{u_T \times y}{v}$$

$$u_T = \sqrt{\frac{\tau_w}{\rho}}$$

where $y^+$: non-dimensional wall distance in manikin surface. $u_T$: friction velocity. $v$: kinematic viscosity. $\rho$: air density, kg m$^{-3}$. $\tau_w$: wall shear stress, Pa.

| Year  | Author       | CDD, computational domain dimension (L×W×H) | TCN, total cells number (MILLION) | LCN, local zone cell number around the human body | FLH, first layer height (mm) | NBL, number of inflated prismatic layers | $y^+$ | RadIteration | Grid independence solution | New wall treatment (NWT) |
|-------|--------------|---------------------------------------------|-----------------------------------|-----------------------------------------------|-----------------------------|----------------------------------------|------|--------------|--------------------------|-------------------------|
| 2017  | [31]         | 5×3×2.7                                     |                                   |                                               |                             |                                        |      |              |                          | S2S                     |
| 2018  | [33]         | 3×3×2.44                                    |                                   |                                               |                             |                                        |      |              |                          | Unstructured tetrahedral |
| 2018  | [54]         | 3×3×2.44                                    |                                   |                                               |                             |                                        |      |              |                          | Polyhedral cells         |
| 2018  | [34]         | 3.5×3×2.1                                   |                                   |                                               |                             |                                        |      |              |                          | Unstructured tetrahedral |
| 2018  | [57]         | 4.8×3×3.69×3.05                             |                                   |                                               |                             |                                        |      |              |                          | Unstructured tetrahedral |
| 2018  | [73]         | 3×3×2.44                                    |                                   |                                               |                             |                                        |      |              |                          | EWT S2S                 |
| 2018  | [75]         | 3×3×2.44                                    |                                   |                                               |                             |                                        |      |              |                          | Polyhedral cells         |
| 2018  | [74]         | 3.5×3×2.1                                   |                                   |                                               |                             |                                        |      |              |                          | Unstructured tetrahedral |
| 2018  | [77]         | 4.8×3×3.69×3.05                             |                                   |                                               |                             |                                        |      |              |                          | Unstructured tetrahedral |
| 2019  | [60]         | 3×3×2.44                                    |                                   |                                               |                             |                                        |      |              |                          | HEX                     |
| 2019  | [9]          | Unstructured tetrahedral                    |                                   |                                               |                             |                                        |      |              |                          | Prism, TET, HEX          |
| 2020  | [9]          | Unstructured tetrahedral                    |                                   |                                               |                             |                                        |      |              |                          | Prism, TET, HEX          |
Conclusions

This paper studied the research of CFD analysis techniques in indoor ambient environments. In the first place, the current study found that the critical increment in the quantity of MV, DV, PV, and SV system research appeared using the CFD program with robust ventilation system types related to various backgrounds of HVAC systems. For the human body, this study mainly concentrates on the CTM, employing different types of ventilation in the CFD to better know its effect on the local thermal state. Overall, the CFD past studies investigated are a valuable method for predicting occupants’ thermal comfort. Nevertheless, the current investigations do not inspect the stratum ventilation that would influence the overall or local thermal comfort of the individual occupant in the high-rise buildings in tropical regions.

Moreover, the proper CFD outcomes require deep consideration of the numerical structure. Knowing this is essential for enhancing the SV performance linked with other structured systems and advancing its use. It is precious to experience what environmental conditions create thermal comfort in the building design. If a simple model of humans can predict a level of dissatisfaction with the ambient condition, then this is a real bonus. Moreover, designing a thermally comfortable indoor space requires determining thermal comfort levels through modelling that designers can adopt to compare various design performances and enhance the perceived level of thermal comfort. Besides, policymakers can adopt the predictions made by the model to support technologies that produce satisfactory comfort levels while consuming less energy. In other words, if the thermal comfort is implemented well, the energy efficiency in a building could be improved as well. For this purpose, it is preferable to predict and control thermal comfort criteria in the initial stage of designing a building and its HVAC systems in full compliance with sustainability and international standard of the acceptable thermal environment.

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Authors’ contributions Y.H. Yau PhD (Mech) (Canterbury, NZ), PEPC (Malaysia), IntPE, FIE Aust CPEngNER (Australia), APEC Engr, FIEM, MASHRAE is a professor at the Department of Mechanical Engineering, University of Malaya, Kuala Lumpur, Malaysia. Professor Yau is the principal investigator (PI) of the current project. H.S. Toh MEng (UTM) is currently a PhD candidate at the Department of Mechanical Engineering, University of Malaya. B.T. Chew PhD (Mech)(UM) is a senior lecturer at the Department of Mechanical Engineering, University of Malaya. Dr. Chew is the research collaborator of the current project. N.N.N. Ghazali PhD (Mech)(UM) is an associate professor at the Department of Mechanical Engineering, University of Malaya. A/Prof. Nik Nazri Nik Ghazali is the Co-PI of the current project. All authors contributed equally in the preparation of this manuscript.

Declarations

Conflict of interest On behalf of all authors, the corresponding author states that there is no conflict of interest.

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