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The impacts of the thermal radiation field on thermal comfort, energy consumption and control—A critical overview

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ABSTRACT

Thermal comfort is determined by the combined effect of the six thermal comfort parameters: temperature, air moisture content, thermal radiation, air relative velocity, personal activity and clothing level as formulated by Fanger through his double heat balance equations. In conventional air conditioning systems, air temperature is the parameter that is normally controlled whilst others are assumed to have values within the specified ranges at the design stage. In Fanger's double heat balance equation, thermal radiation factor appears as the mean radiant temperature (MRT), however, its impact on thermal comfort is often ignored. This paper discusses the impacts of the thermal radiation field which takes the forms of mean radiant temperature and radiation asymmetry on thermal comfort, building energy consumption and air-conditioning control. Several conditions and applications in which the effects of mean radiant temperature and radiation asymmetry cannot be ignored are discussed. Several misinterpretations that arise from the formula relating mean radiant temperature and the operative temperature are highlighted, coupled with a discussion on the lack of reliable and affordable devices that measure this parameter. The usefulness of the concept of the operative temperature as a measure of combined effect of mean radiant and air temperatures on occupant's thermal comfort is critically questioned, especially in relation to the control strategy based on this derived parameter. Examples of systems which deliver comfort using thermal radiation are presented. Finally, the paper presents various options that need to be considered in the efforts to mitigate the impacts of the thermal radiant field on the occupants' thermal comfort and building energy consumption.

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1. Introduction

Thermal radiation effects of the surrounding surfaces in indoor settings have been known to affect thermal comfort of the occupants of buildings. In the heat balance model this factor is termed the mean radiant temperature (MRT). This is an environmental parameter defined as “the uniform surface temperature of an imaginary black enclosure with which an occupant would have the same radiant energy exchange as in the actual temperature space” ([1], p. 3). The presence of MRT can be illustrated in the following examples. When outside temperature in winter is far below the normally acceptable room temperature, the interior surfaces of the building wall of an occupied room can be significantly colder than the room air temperature resulting in occupants feeling colder. Solar radiation that penetrates through the transparent glass windows will help to compensate for the colder interior wall surface or even the colder air temperature. Conversely, during summer, solar radiation penetration into the room is usually to be avoided in order to prevent overheating. In conventional air-conditioning the effect of existence of one degree difference between MRT and air temperature amounts to approximately raising air temperature by one degree [2]. In other words, this one degree difference becomes a load to the air-conditioning system. A heating load increase of 18.7% was noticed when operative temperature replaced the air temperature as a controlled parameter [3]. Similarly, an increase between 14 and 29.5% of cooling load in the perimeter zone was estimated, depending on the type of glazing used which affected MRT [4]. In recent decades, new methods of room heating or air-conditioning have been introduced based on considerations that radiant heat or ‘coolth’ can provide thermal comfort. The radiant ceiling cooling is an example of such a method of radiant comfort delivery.

In recent literature reviews on thermal comfort research and practice [5–7], the thermal radiant field has received little attention, which to a large extent, was due to the limited number of developments in the domain. Therefore, this paper presents a critical overview of the literature that deals with thermal radiant field in the forms of the mean radiant temperature and radiant asymmetry.

The paper is structured in a way that it first deals with the thermal radiant field and its effect on thermal comfort, energy consumption, and the control of air-conditioning systems in a successive order. This is followed by an overview of the implications of the presence of the thermal radiant field in naturally ventilated buildings. Thereafter, the paper also discusses a number of misconceptions in the application of the equations related to this parameter in the literature. Finally, the paper discusses the potential and drawbacks of various strategies to minimise the impact of the thermal radiant field on thermal comfort and energy consumption.

2. The physics of the mean radiant temperature

The physical explanation of thermal radiation field can be found in many standard textbooks on heat transfer (for instance [8,9]); however, its relevance and impacts on human thermal comfort were explained in Refs. [10–13]. Earlier, Gagge [14] introduced the term “operative temperature”. One of the relevant terms encountered in the literature is irradiance. Irradiance is defined as the energy flux per unit area falling on a surface from all directions. It is a function of both direction and location. Energy radiated by an enclosure from all directions will reach the human body living in the enclosure. Likewise, a person standing outside of an enclosure is exposed to radiation coming from the surfaces of the enclosure and surrounding surfaces that “see” the person, including that coming from the sky in term of diffuse and direct radiation. In thermal comfort, the temperature resulting from these radiation exchanges on the human body is termed the mean radiant temperature (MRT). Prior to McIntyre [14], Fanger [15] had given ample treatment of the mean radiant temperature; however, clear explanation, quantification and measurement of asymmetric radiation can be regarded as McIntyre’s clear contribution in the field.

The MRT is the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual non-uniform enclosure [1]. For an enclosure, the calculation of the MRT is based on the radiant heat exchange between all radiating surfaces and the point of interest, in this case the human body. Thus [1]:

$$\overline{T}_r^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + \cdots + T_N^4 F_{p-N} \quad (1)$$

where T_r is the mean radiant temperature, K; T_N is the temperature of the surface N, K; F_{p-N} is the angle factor between a person and surface N.

In the case of small differences in the temperature of the surfaces of the enclosure, the following equation can be used, resulting in a slightly lower mean radiant temperature than calculated by Eq. (1).

$$t_r = t_1 F_{p-1} + t_2 F_{p-2} + \cdots + t_N F_{p-N} \quad (2)$$

where t is the temperature in °C.

In an enclosure, the angle factor between a person and a surface depends on the position and orientation of the person (Fig. 1) [16]. In a study by d’Ambrosio Alfano et al. [17] it was found that the measurement method based on the angle factors is reliable.

Since the MRT is a result of the radiation exchanges between the surfaces and the point of interest, i.e., the human body, its value depends on the complex interaction of these factors.

3. Radiation asymmetry

Due to the nature of non-uniform radiative environments surrounding an enclosure, an object of interest within an enclosure is often exposed to asymmetric field of radiation. A heating element standing in front of an occupant will make the occupant feel warmer from that direction than from other directions. Likewise, a person sitting closer to a cold wall in the midst of winter

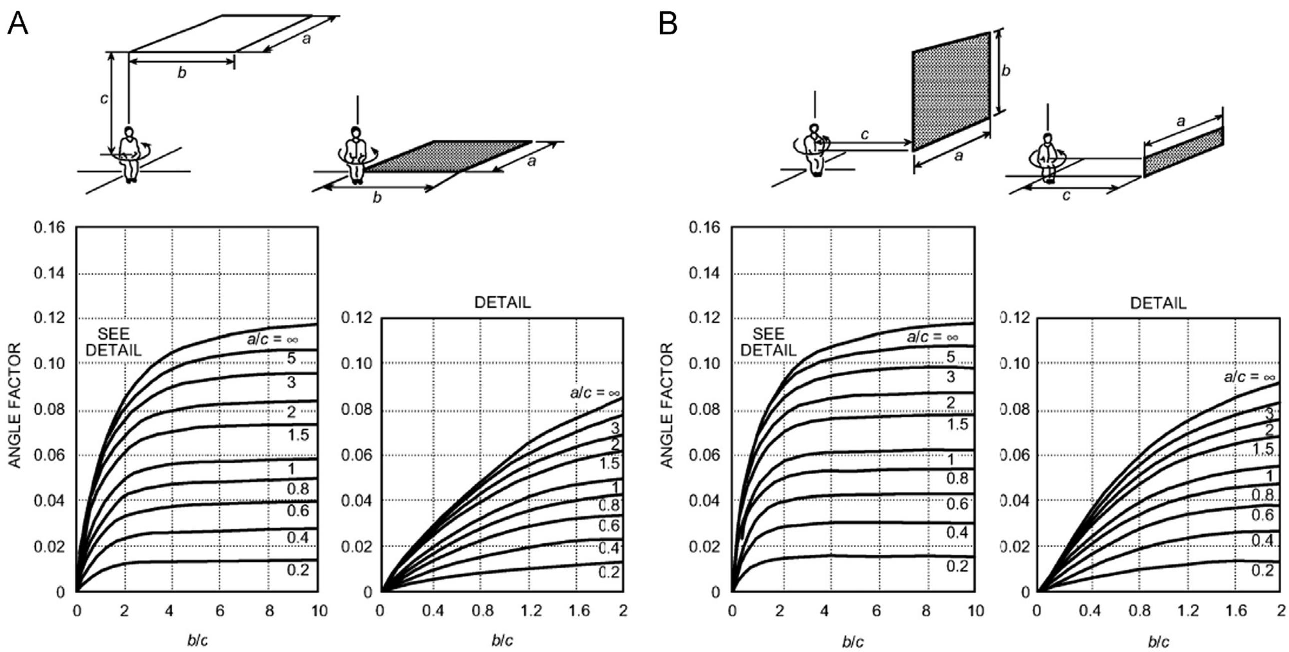

Fig. 1. Mean value of angle factor between a seated person and a horizontal or vertical rectangle when the person is rotated around a vertical axis [16]. (a) Horizontal rectangle (on ceiling or floor) and (b) Vertical rectangle (above or below center of person).

Table 1
Recommended levels of acceptability for 3 classes of environment [21] for local discomfort caused by radiant temperature asymmetry (Δt_{pr}) [20].

Category	PD[a] [%]	Δt_{pr} [K] warm ceiling	Δt_{pr} [K] cool ceiling	Δt_{pr} [K] cool wall	Δt_{pr} [K] warm wall
A	<5	<5	<14	<10	<23
B	<5	<5	<14	<10	<23
C	<10	<7	<18	<13	<35

[a] PD = percentage of dissatisfied.

will feel colder from the direction of the cold wall than from other directions.

When there is a difference between the plane radiant temperature of the opposite sides of a small plane element or of the environment on opposite sides of a person, this difference is termed radiant temperature asymmetry, Δt_{pr}, [1,18,19]. This parameter is especially important in comfort conditions. Asymmetric or non-uniform thermal radiation in a space may be caused by cold windows, uninsulated walls, cold products, cold or warm machinery, or improperly sized heating panels on the wall or ceiling [1].

Radiant temperature asymmetry is calculated from the plane radiant temperature (t_{pr}) [10]. The plane radiant temperature is the uniform temperature of an enclosure in which the incident radiant flux on one side of a small plane element is the same as that in the actual environment. The plane radiant temperature describes the thermal radiation in one direction, and its value thus depends on the orientation of that plane. In comparison, the mean radiant temperature describes the thermal radiation to the human body from all directions [1]. Because the radiant temperature asymmetry is defined with respect to a plane element, its value depends on the positions and the orientation of all surfaces in relation to the human body.

There are recommendations for the three classes or categories of indoor environmental quality set in ISO Standard 7730 [20] and ASHRAE Standard 55 [21] (Table 1). These are based primarily on studies reported by Fanger et al. [18,19] and they come with certain bandwidths.

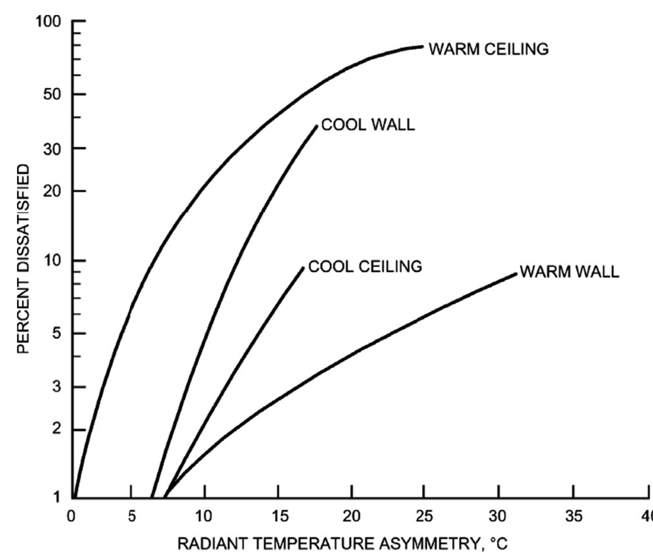

Fig. 2. Percentage of occupants expressing discomfort due to radiant temperature asymmetry [27].

Among the studies conducted on the influence of asymmetric thermal radiation are reported in Refs. [13,18,22–26]. In these studies all subjects were seated and they were always in a state of thermal neutrality and exposed only to the discomfort resulting from excessive asymmetry. The subjects gave their reactions on their comfort sensation, and a relationship between the radiant temperature asymmetry and the number of subjects feeling dissatisfied was established (Fig. 2).

People, both men and women, tend to be more sensitive to radiant asymmetry due to warm overhead surfaces. Warm vertical surfaces also affect people's comfort; however, their effect is much less than that of cool overhead and cool vertical surfaces. Table 1 and Fig. 2 show that the recommended radiant temperature asymmetry due to warm ceiling for 10% PD is not to exceed 7 K, much lower than the recommended value for radiant temperature asymmetry due to cool ceiling (18 K), cool wall (13 K) and warm wall (35 K). These data are particularly important when applying radiant panels to provide comfort in spaces with large cold surfaces or cold windows.

Despite the likely impacts of radiant asymmetry on human thermal comfort, the number of studies conducted on the fundamentals of radiant asymmetry and the perception of the mean radiant temperature is, however, surprisingly small. The thoroughness of the work, and hence its practical implications, should therefore be treated with some caution. For instance, McIntyre [24] presented work on overhead radiation, exposing a total of 148 participants to one of four levels of overhead radiation (0–5–9–14 K), up to a maximum ceiling temperature of 45 °C. Air and wall temperatures were held equal to each other, and reduced to compensate for the raised ceiling temperatures, so that perceived warmth was constant across the conditions. After 15 min exposure, the subjects rated the environment on seven scales. Participants were "ready to attribute discomfort to unusual aspects of the environment". A maximum asymmetry of a vector radiant temperature of 10 K was, therefore, suggested as a design criterion. In this study [24], this level did not actually increase discomfort, but was noticeable and in practice levels greater than this are likely to produce complaints.

Kähkönen and Ilmarinen [28] continuously measured important thermal comfort parameters in 13 shops and stores during working days in the winter and summer. Workers rated their subjective sensations. They concluded not only that the radiant temperature asymmetry is difficult to measure, but also that it seemed that in real situations the effect of radiant temperature asymmetry on comfort is either smaller than assumed, or that the comfort limits are too high. Moreover, they conclude that not enough research had been done when the limits were specified, and more scientific data were, therefore, needed. Finally, Kähkönen and Ilmarinen recommended that more attention should be paid to low air temperatures, great differences in vertical air temperatures, and cold airflow than to radiant temperature asymmetry or mean air velocities. In contrast, Hodder et al. [29] presented the work on climate chamber experiments on combined chilled ceiling/displacement ventilation environments. Vertical radiant temperature asymmetry was found to have an insignificant effect on the overall thermal comfort of the seated occupants for the typical range of ceiling temperatures that would be encountered in practice in such environments. It was concluded that existing guidance regarding toleration of radiant asymmetry is valid for thermal comfort design of chilled ceiling/displacement ventilation environments. Existing guidance by Fanger et al. [18] was considered to be valid, without modification, for thermal comfort design in such environments.

In relation to windows, a number of studies have been conducted. Lyons et al. [30] concluded, based on modeling, that long-wave exchange between the body and the window is the most significant except for the case where the body is in direct sun, in which case the impact of solar load can be more significant. For most residential-sized windows, draft effects exist but are typically small and generally, windows are not the primary element affecting the comfort of a building's occupants. However, when a window is very hot or cold, the occupant is very close to the window, or other factors resulting in thermal conditions near the edge of the comfort zone, the effect of the window can become quite significant.

Furthermore, it is believed that current methods may under predict discomfort caused by windows. Gan [31] analysed the effect of glazing on the mean radiant temperature and thermal comfort in rooms. It is shown that a tall and narrow window performs better than a square window in terms of indoor thermal comfort. In line with theory and practice, double-glazing is effective in reducing thermal discomfort due to radiant asymmetry. For a room with a cold or hot surface such as a large single-glazed window or radiant heating panel in winter, thermal discomfort may exist as a result of radiant asymmetry and localized cold or hot spots in terms of radiant temperature, which may not be evident with air temperature alone. In case of commercial buildings with curtain wall systems with large windows, Rowe [32] found that radiant asymmetry and associated diurnal temperature shifts had a significant impact on thermal comfort in proximity to the windows. A difference of one comfort vote interval could be attributed to a frontal radiant temperature difference of about 2.8 K whereas a 3.3 K difference brought about a one vote difference for persons side on to the radiant source/sink. Also a change in operative temperature in perimeter zones as the sun moved around the building had the effect of moving the comfort vote approximately one interval for every two degrees of shift over approximately three hours. This study [32] suggested that the thermal comfort standards understate the significance of asymmetric heat exchange with hot or cold vertical surfaces. Likewise, Underwood and Parsons [33] studied the thermal comfort of subjects seated "side on" to a vertical cold window, in an otherwise thermally neutral environment. The small study, comprising eight male participants who were exposed to a cold window with a surface temperature of 5 °C ± 1 K in a climate chamber, found a significant radiant asymmetry across participants' bodies. Predictions with three models (Predicted Mean Vote (PMV), Draft Rating, and Radiant Asymmetry) were then compared with the actual percentage of dissatisfied participants, and PMV was found to be the most accurate predictive model for these conditions.

Not all studies relate to the design of windows as a source of radiant asymmetries. The effects of radiant heating panels have also been studied and documented. For instance, Dudkiewicz and Jezowiecki [34] described the effects of radiant heaters near work stations. Chih-Chun and Shao-Yi [35] investigated the influence of horizontal radiant temperature asymmetry on the thermal sensation of subjects and the correlative variation of environmental parameters in a warm sitting area. The evaluation results show that the PMV index can be used to predict the overall thermal sensation of a group of sedentary subjects in a warm environment with horizontal radiant temperature asymmetry. The study found that subjects felt local discomfort on exposed parts of their body. Barna and Bánhidi [36] studied the combined effect of the radiant temperature asymmetry and warm floors in a climate chamber set-up. They found that the thermal sensation and skin temperatures of human subjects were affected more by the radiation from the vertical cold surface than by the warm floor.

What the studies summarised above show, is that sample sizes are small (few participants), the studies have often been conducted in climate chambers, and the results are not consistent. Some studies support existing models and standards, whereas others do not, and this leads to uncertainties or even conflicting outcomes. Some researchers have deployed methods such as thermal manikins [37] or (validated) simulations [38,39], instead of working with (a large set of) real participants.

It can be argued that practically in all situations the thermal radiant field is perceived by the building occupants in the combined form of radiant asymmetry and mean radiant temperature where the latter is the result of the former. Yet the only 'aspect' of thermal radiant field accounted for in the double heat balance equation is the MRT. This is not surprising since the term "radiant asymmetry" was introduced a decade later [19] than the formulation of double heat balance equation [15], although the research on radiant asymmetry had actually been performed earlier in 1950s by Chrenko [40] and later by McNall and Biddison [25] as mentioned earlier. It is, therefore, legitimate to question the various strategies that focus to mitigate the impacts of mean radiant temperature when in fact both are inseparable.[1]

4. The measurement or estimation of mean radiant temperature and radiant temperature asymmetry

The measurement or the estimation of the value of the mean radiant temperature is required for the design, assessment, and development of control strategy to address the effects of the thermal radiation field.

For measurement of the mean radiant temperature, ISO Standard 7726 [42] listed three methods, i.e., (1) through the measurement of globe temperature using black globe thermometer, (2) the use of two sphere radiometer, and (3) the use of constant air temperature sensor. The estimation methods listed in the Standard include: (1) view factor between the person and the surrounding surfaces—see Section 2, and (2) plane radiant temperature—see Section 3.

4.1. Globe thermometer

The use of a sphere to measure the temperature actually sensed by humans was first suggested by Vernon [43]. Vernon used some hollow globes of different sizes to sense the effect of radiation. The original globe thermometer was a thin-walled copper sphere of 15 cm diameter of which the outer and inner surfaces were painted matt black. The bulb of a mercury-in-glass thermometer was located at its centre. The globe thermometer accepts radiant heat from all directions. The equilibrium temperature at the centre of the globe is called the globe temperature. The time to reach the steady-state globe temperature following a change in conditions, which is about 20 to 25 mins places a severe restriction on the use of the globe thermometer for control applications.

Bedford and Warner [44] developed a formula relating absolute MRT, $\bar{t}_r$, absolute air temperature, t_a, air velocity, v_a, and globe temperature, t_g, as follows:

$$\bar{T}_r^4 = T_a^4 + C\sqrt{v_a}(t_g - t_a) \tag{3}$$

An adequate representation of Eq. (3) over the temperature range of interest is:

$$\bar{t}_r = t_g + k\sqrt{v_a}(t_g - t_a) \tag{4}$$

Eq. (3) is the form used in the ASHRAE Handbook of Fundamentals [1] while Eq. (4) is the form used in ASHRAE Standard 55-2010.

4.1.1. Globe temperature and operative temperature

Gagge et al. [12] indicated that the globe temperature is approximately equal to the operative temperature defined as "the temperature of a radiantly black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual non-uniform temperature environment" [45]. This term was introduced earlier [14] to account for the combined effect of mean radiant temperature, air temperature and air velocity.

Mathematically operative temperature can be defined as follows [45]:

$$t_o = \frac{h_c t_a + h_r \bar{t}_r}{h_c + h_r} \tag{5}$$

or

$$t_o = A t_a + (1 - A)\bar{t}_r \tag{6}$$

where

$$A = \frac{h_c}{h_c + h_r} \tag{7}$$

The similarity between globe temperature and operative temperature can be seen from the values of the weighting coefficient A given in [45], which are closely similar to those calculated from Eq. (4) after rearrangement in the form:

$$t_g = \frac{k\sqrt{v_a}}{\sqrt{v_a}} t_a + \frac{1}{1 + k\sqrt{v_a}} \bar{t}_r \tag{8}$$

which can be written as:

$$t_g = A^* t_a + (1 - A^*)\bar{t}_r \tag{9}$$

where

$$A^* = \frac{k\sqrt{v_a}}{1 + k\sqrt{v_a}} \tag{10}$$

Table 2 shows the values of the weighting coefficient A^* for various air velocities, v_a, based on Eq. (10) from [45]. Similar values of A given in ISO 7730 as quoted in [46] and also in [21] are also shown for comparison in Table 3.

In cases where accurate air velocity sensors suitable for practical application are not available, the value of A in Eq. (6) can be taken as 0.5 which gives the operative temperature for conditions normally encountered in practice [45]:

$$t_o = 0.5\ t_a + 0.5\ \bar{t}_r \tag{11}$$

where

$$t_o = t_g$$

Thus, the MRT is often estimated indirectly from its relation with air and globe temperatures. The MRT is also often assumed equal to the room air temperature and as a result, the calculated operative temperature (as shown in Eq. (11)) is assumed to equal the room air temperature.

While Vernon's globe thermometer, comprising a 15 cm diameter hollow black sphere with a thermometer at its centre, is recommended in HVAC (heating, ventilation and air-conditioning) applications for estimating mean radiant and operative temperatures [45], and hence has become the de facto standard used when commissioning or testing HVAC systems, it is impractical for use in the control of HVAC systems because of its obtrusive size and long-time constant.

Table 2
Values A^* (Eq. (9)) and A [45].

v_a [m s^{-1}]	0.05	0.15	0.2	0.3	0.4	0.5	0.6	0.7	0.8
A^* [–]	0.33	0.41	0.5	0.55	0.58	0.61	0.63	0.65	0.66
A [–]	0.42	0.42	0.46	0.52	0.55	0.58	0.61	0.63	0.65

Table 3
Values of weighting coefficient A given in ISO 7730 and [21]:

v_a [m s^{-1}]	< 0.2	0.2–0.6	0.6–1.0
A [–]	0.5	0.6	0.7

[1] In a controlled setting, these two aspects can be separated as demonstrated by Olesen et al. [41] who created a test chamber in which radiant asymmetry could be set as high 40 K while the MRT in the centre of the chamber remained equal to the air temperature.

The standard suggests the use of smaller spheres to minimise the drawbacks of Vernon's globe; however, there are no clear and sufficient guidelines given on the use of the non-standard spheres. In addition, Halawa [2] indicated that smaller spheres suffer from 'uncertainties' due to the effect of air velocity. He found there were significant differences among convection heat transfer coefficients obtained for parallel, cross and counter flow orientations in the 'high' velocity region. For the low velocity region this does not seem to be the case. Consequently, the provision of elevated air velocities in air-conditioned spaces will make smaller spheres unsuitable for this purpose.

The heat stress monitoring/metering devices available in the market use such a small sphere for measuring globe (operative) temperature. However, specification of such a device does not seem to be based on any existing standards, such as ISO Standard 7726 [42].

According to this standard [42], MRT measurement through the use of globe thermometer is only an "approximation due to the difference in shape between a person and a globe" ([42], p. 27). Due to this shape factor, impact of radiant field coming from ceiling or floor received by a standing or seated person will be overestimated. In order to minimise the impact of the shape factor, ISO Standard 7726 [42] recommends the use of an ellipsoid sensor.

4.1.2. Misconceptions and misinterpretations

It is worth noting that Bedford and Warner [44] developed Eq. (4) for air velocities above about 0.1 m s^{-1}. At lower air velocities, they observed that natural convection became important, causing non-linearity in the plot of $(h_c/\sqrt{\upsilon})$ as a function of $(t_g - t_a)$, from which plot the velocity correction coefficient k was determined. Thus the application of the formula near zero air velocity is unjustified. This fact needs to be stressed to avoid misinterpretation and possible recurrence of misleading statements such as the following which appears in an air conditioning text book ([47], p. 97):

> It (the globe thermometer) is affected by dry-bulb temperature, air velocity and mean radiant temperature but in still air reads the mean radiant temperature exactly.

In a much recent note a similar misinterpretation appears:

> MRT this is the solid-angle-weighted average temperature of surrounding surfaces. It cannot be measured directly, but it can be estimated from GT readings. In still air MRT=GT, but a correction for air movement of velocity (in m/s) is possible… ([48], p. 21).

At very low or zero air velocities, the heat transfer between the sphere and its surroundings occurs predominantly by both natural convection and conduction modes; and at a zero air velocity the sphere is immersed in otherwise stagnant fluid where a constant Nusselt number $Nu = 2$ is observed [8].

4.2. Two sphere radiometer

The working of this device is based on the different emissivities of the two spheres used (one black, the other polished). The two spheres exposed to the same radiant and convective environment will be subject to the same convective heat loss. The emissivity difference between the two results in the difference in heat supplied to each sphere. Using the radiometer, the MRT can be estimated as follows [42]:

$$\overline{T}_r^4 = T_s^4 + \frac{P_p - P_b}{\sigma(\varepsilon_b - \varepsilon_p)} \tag{12}$$

where T_r is the mean radiant temperature, K; T_s is the sensor temperature, K; P_p is the heat supplied to the polished sensor, W m^{-2}; P_b is the heat supplied to the black sensor, W m^{-2}; ε_p is the emissivity of polished sensor; ε_b is the emissivity of the black sensor; σ is the Stefan–Boltzmann constant $= 5.67 \times 10^{-8}$ W m^{-2} K^{-4}.

As in the case of globe thermometer, the ISO Standard [42] recommends the use of ellipsoid shaped sensors.

4.3. Constant air temperature sensor

To enable the measurement of MRT, the sensor (sphere or ellipsoid) is maintained at the same temperature as the surrounding air resulting in zero convection heat loss. The sensor radiant heat loss or gain is compensated by heat or coolth supply to the sensor. The MRT is calculated as follows:

$$\overline{T}_r^4 = T_s^4 + \frac{P_s}{\sigma \varepsilon_S} \tag{13}$$

There are only a few publications on both the two-sphere radiometer and the constant air temperature sensor. Measurement uncertainties from using these devices are among others: the shape and emissivity of the device is only approximation of the subject shape and emissivity, different spatial position of the sphere or ellipsoid, long response time, etc. [17].

5. Mean radiant temperature consideration in thermal comfort standards

Fanger's model of thermal comfort [15] has become the basis of the Thermal Comfort Standards that exist today (for instance [20,21]). This model is usually applied in buildings that are air-conditioned. In the heat balance model, all the six parameters appear in the equation and therefore their impacts can be easily quantified. Fig. 3 [15] shows the rate at which the air temperature must change, δt_a, to offset a change in MRT, δt_r, as a function of clothing resistance for different air velocities for three different

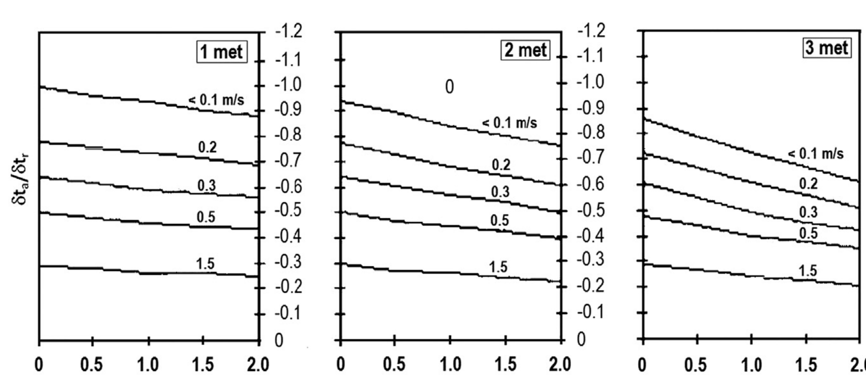

Fig. 3. $\delta t_a/\delta t_r$ as a function of the thermal resistance of clothing with relative velocity as parameter (RH=50%) [15].

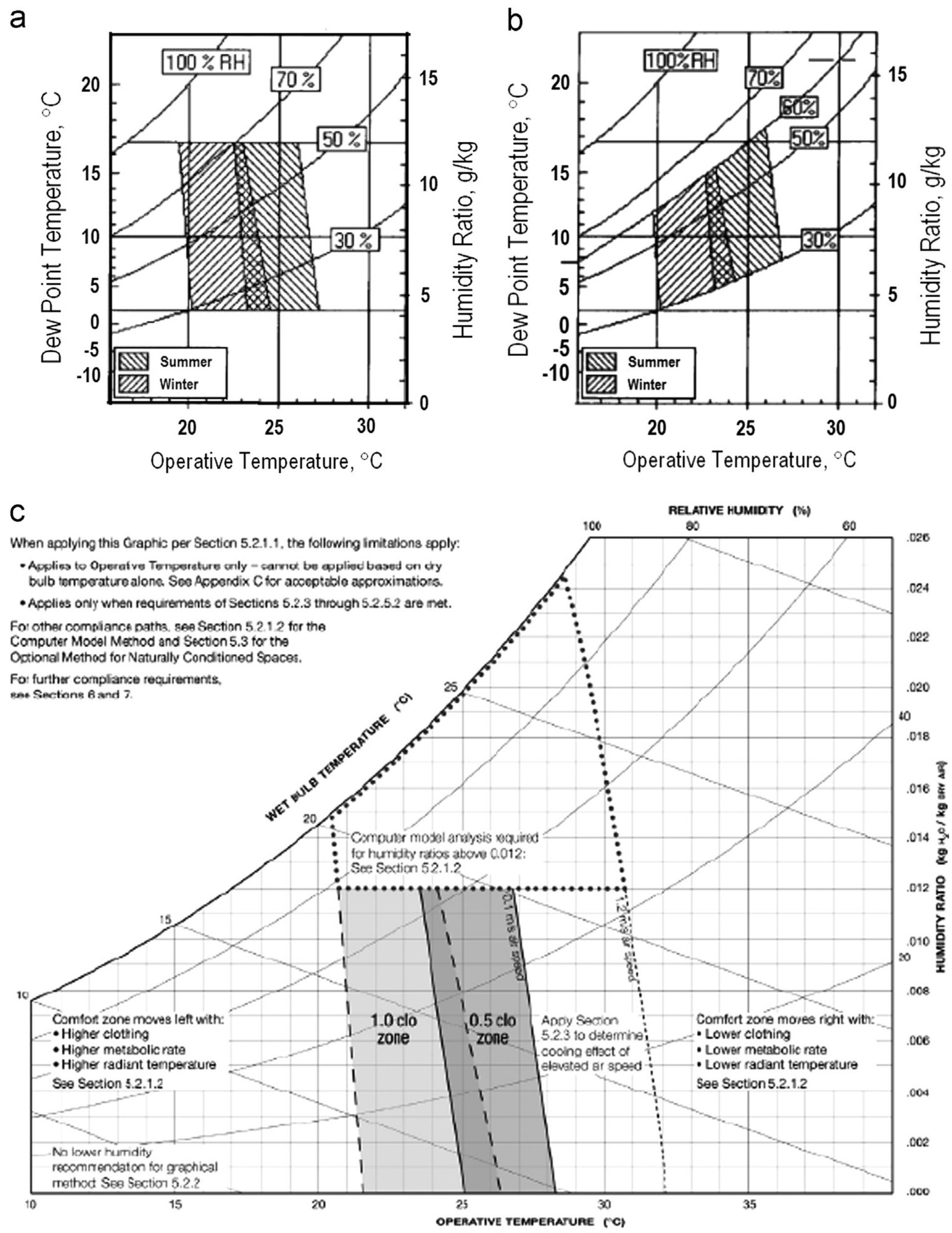

Fig. 4. Acceptable ranges of operative temperature and humidity ratio for persons clothed in typical summer and winter clothing, at light, mainly sedentary activity ($\leq$ 1.2 met). **(a)** ASHRAE Standard 55-1981 [45], (b) ASHRAE Standard 55-1990 [49] and **(c)** ASHRAE Standard 55-2010 [21].

levels of activity of the subject expressed in three different metabolic rates (1 met, 2 met and 3 met).

While Fig. 3 also appears in standards such as [21], people tend to overlook it and focus instead on the well-known thermal comfort charts as shown in Fig. 4 which help to quantify quickly the effects of each parameter for various conditions. However, when the MRT is different from the air temperature, there can be a problem in identifying how the shape and location of the 'comfort zone' on the comfort chart should be varied, and then each particular case will be different [2].

Upon looking into detail, the 'operative temperature' label of the charts of Fig. 4 is not appropriate and can be misleading. It is inappropriate because the operative temperature is "not a property of a mixture of air and water vapour which can be located on a psychrometric chart" ([2], p. 11). It can be misleading because it can be interpreted as if radiant effect exists for the conditions depicted in the charts. In fact, in constructing the charts, the simplifying assumption has been made that mean radiant temperature equals air temperature [15].

The ASHRAE Standard 55 [21] and European Standard EN 15251 [50] include models of adaptive thermal comfort for use in naturally ventilated buildings, which were developed from the ASHRAE RP-884 database [51] and the EU project Smart Controls and Thermal Comfort (SCATs) [52,53]. While being different from each other and acknowledging the roles of the six thermal comfort parameters appearing in the heat balance model, the adaptive approach to thermal comfort comes up with a much simplified linear model which relates the indoor comfort (operative) temperature to the outdoor temperature [3,54]. The linear relationship can be drawn because it is assumed that occupants in naturally ventilated spaces have adapted to the outdoor conditions by wearing appropriate clothing and conducting sedentary activities

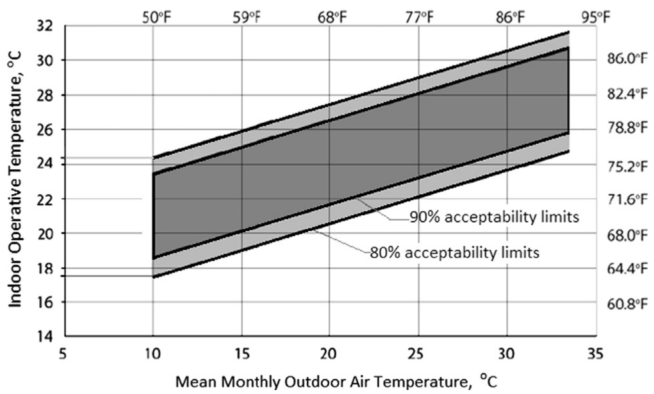

Fig. 5. Acceptable indoor operative temperature as in [21], intended for use in naturally ventilated buildings. The 80% acceptability limits may be used for typical applications and should be used when other information is not available. The 90% acceptability limits may be used when a higher standard of thermal comfort is desired [21].

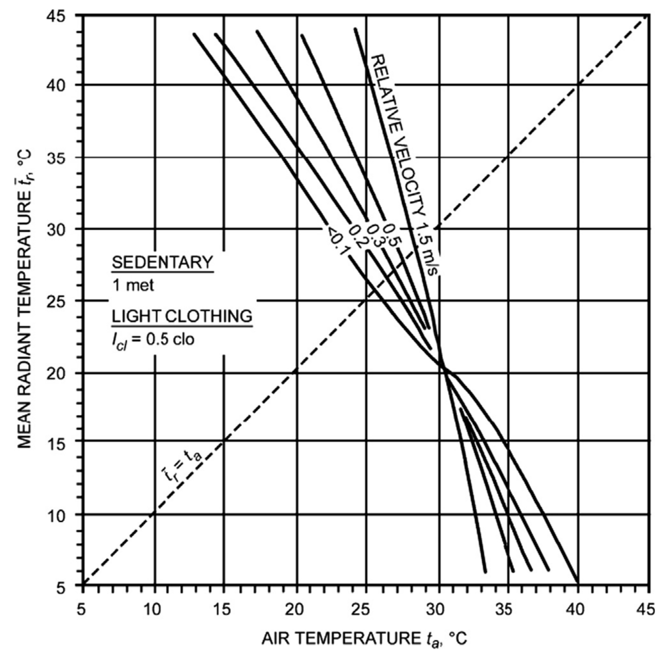

Fig. 6. The effect of air relative velocity on the optimum mean radiant temperatures at various air temperatures and air velocity [15].

and that they control the indoor environment by operating the windows and other ventilation strategies [55,56]. The adaptive models do not clearly express the effect of radiation in its equations as it is assumed that the occupants will adapt or use adaptive strategies in order to be comfortable (for example, by moving from the heat source). The comfort temperature band is determined based on the relationship between the prevailing mean outdoor temperature and indoor operative temperature (Fig. 5), which implies that the effect of mean radiant temperature has been taken into account. Whether or not this is always the case, is not clear. Other adjustable thermal comfort factors which are crucial in the heat balance model are also not clearly expressed [57].

Fanger and Toftum argued that the adaptive approach will be incapable of dealing with "buildings of new types in the future where the occupants may wear different clothing and change their activity pattern" ([58], p. 534). While this critical observation is limited only to *building types* and *clothing*, it is also relevant to the thermal radiation which is not clearly expressed in the adaptive model equation. Halawa and van Hoof [57] also pointed to the limitation of this approach particularly when applied in buildings employing novel techniques to provide thermal comfort such as using radiant cooling or heating.

Referring back to Fig. 2 it can be argued that the subjects' sensitivity to the asymmetry increases when such an asymmetry is imposed to the subjects while they are also exposed to conditions that give a warmer than neutral sensation (in case of radiant heating asymmetry) or a colder than neutral sensation (in case of radiant cooling asymmetry). In other words, the higher the initial PPD (Predicted Percentage of Dissatisfied) value, the less Δt_{pr} may be tolerated. This is very important to highlight given the fact that current building designs show a trend of using large glass windows, or heating and cooling panels in the ceiling. Ignoring the effect of radiant asymmetry in such a situation can be problematic, particularly, when the adaptive approach advocates higher room temperature settings in summer based on the assumption that occupants adapt to higher outdoor temperatures.

Fig. 6 shows how comfort is affected by the MRT and relative air velocity, situations which are specifically not covered in Figs. 4 and 5. As the MRT increases, the air temperature must decrease in order to maintain equal comfort. The positions and slopes of the equal comfort lines in Fig. 6 depend on the local velocity of the air relative to the subject. As the velocity increases, the reduction in air temperature required to offset an increase in the MRT decreases. This is shown more clearly in Fig. 3, where the rate at which the air temperature must change, δt_a, offsets a change in MRT, δt_r is shown as a function of clothing resistance for different air velocities for three different levels of activity of the subject.

In short, the existing comfort standards, which are generally interpreted to being the main guidance for building design, do not clearly factor in the effect of mean radiant temperature.

6. Radiant effect consideration in air-conditioning controls

Taking mean radiation temperature into consideration has not yet been widely adopted by the air-conditioning industry, partly due to the lack of reliable and affordable devices for measuring some of the quantities (see Section 4) and partly due to the complicated nature of any control systems associated with their implementation.

Whitmer [59] analysed the comfort equation [15] and by using several optimisation techniques set the total minimum energy load required, subject to the thermal comfort constraints, but assumed mean radiant temperature equal to air temperature. Kaya [60] and Kaya et al. [61] proposed a new control strategy for HVAC systems based on Fanger's criteria, but also ignored the effect of mean radiant temperature.

Halawa [2], and Halawa and Marquand [62] developed a control strategy whereby the operative temperature becomes the controlled variable in a variable air volume (VAV) air-conditioning system. An algorithm for computing the optimum air temperature set point based on the Fanger/ASHRAE comfort criteria was developed. In that study the effect of mean radiant temperature on VAV system was discussed. The study indicated that switching the control from air temperature (dry bulb temperature) to operative temperature (which accounts for mean radiant temperature) may lead to the need for considering readjustment of the maximum heating/cooling load capacities of the selected air conditioning system. This is due to the fact that as $\bar{t}_r$ becomes higher or lower than t_a there will be a need to adjust the air temperature by varying the flow rate of supply air to keep the operative temperature, t_o close to its set point. However, according to the study, this may not necessarily be the case because the radiant effect is also considered in the extreme cooling load estimation using the conventional methods.

Jain et al. [4] addressed the same problem of using a conventional air temperature-based thermostat control in a highly glazed building. In a conventional control system, the operation of the heating and cooling equipment is set based on the room air temperature. They argued that this could be problematic especially in a highly glazed building because of higher radiant temperature of the window panes, hence higher room operative temperature. Further, they found that the cooling energy consumption could be underestimated by more than 50% when an air temperature-based thermostat control was used. Earlier, Niu and Burnett [63] also suggested that an operative temperature-based thermostat control should be used in building energy simulation studies so as to not underestimate the building's energy use. These two findings are consistent with the earlier work [2,62].

7. Methods of thermal radiant comfort delivery

While there have been a number of issues relating to the understanding, measurement, as well as considerations of mean radiant temperature and radiant asymmetry in practice, there have been recent developments in delivering thermal comfort through thermal radiant concepts. These include radiant slabs and panels for cooling and heating. This Section will briefly discuss the various emerging radiant systems, modelling and control, their effectiveness and practical limitations.

Radiant cooling and heating provide comfort mainly[2] through controlled temperatures of radiating surfaces (floors, walls or ceilings). Radiant systems can be broadly classified into two categories: (1) in-floor radiant system, also known as thermally activated building systems (TABS) [65], and (2) radiant panel system. In terms of heat and coolth delivery, the in-floor radiant systems are more inertial due to generally high thermal mass of floor structure.

7.1. Radiant cooling systems

A radiant cooling system removes some room heat gain through sensible heat exchange between the warm air and the actively cooled surfaces but will not result in significant decrease in the air temperature [66]. It is the direct radiant heat exchange between the cold surfaces and the occupants that creates the cool sensation perceived by the occupants.

7.1.1. In-floor (slab) radiant cooling system

In-floor radiant cooling system consists of piping embedded into buildings structure, usually slabs, through which cold fluid (normally water—hence called hydronic radiant systems or HRS [67]) is passed to remove the heat from the slabs which in turn receives heat from the occupied zone.

Chilled water is passed through pipes embedded in the slabs, which gradually cool the layer(s) of the slab until the coolth reaches the floor surface. Convective heat exchange occurs between the floor surface and the air, whilst radiant heat exchange occurs between the floor surface and all other surfaces of the zone, including walls, ceilings, furniture and body and cloth surfaces of occupants. This system, however, only addresses the sensible load of the space whereas the latent load is addressed by a separate ventilation system.

An in-floor system has a relatively low sensible cooling capacity, i.e., 77 W m^{-2} active surface with the typical surface temperatures of 18–24 °C for the entire range of cooling to heating [66].

[2] According to [62], the system is categorised as a radiant system when more than half of the heat transfer occurs through thermal radiation mode.

7.1.2. Radiant panel cooling systems

A radiant panel system consists of panel which functions as casing for housing the cooling coil placed inside the panel. This results in panel surface being cooler than the surrounding air and the room occupants. This system is also called passive chilled beam [66,68]. Moore et al. [66] classified this system as a 'radiant panel' whilst according to Roth et al. [68] this system transfers heat mainly by convection. No quantitative data was used to justify their respective classification.

7.2. Radiant heating systems

7.2.1. In-floor radiant heating system

In-floor radiant heating system is similar to that of in-floor radiant cooling systems in terms of its construction; however, the heat sources can be based on air, electric or hot water. This system is the reverse of the in-floor radiant cooling system; the heat source is passed to transfer heat to the slabs which then transfer heat to the occupied zone. Air systems can be powered by solar air heating systems and for the residential application this generally requires thermal storage for night heating. An electric system requires electric wires to be embedded into the slab, or alternatively, electric mats can be layered onto subfloor which is further covered with floor material such as tile. In-floor radiant heating system using water as fluid is similar to that of in-floor radiant cooling systems; hot water passes through pipes embedded in the slab and heat transfers through slab to the floor surface.

7.2.2. Radiant panel heating systems

In this system, the heat is provided by electric wires or hot water passing through tubing enclosed in a panel. The panels can be placed under the ceiling, which will warm the upper part of occupants' bodies, or integrated in the wall (or positioned vertically) which creates warm effects over the whole body.

7.3. Effectiveness of in-floor versus radiant panel systems

Olesen [69] summarised the characteristics of and common specifications for radiant floor heating. Due to generally higher view factor of the floor to occupants in the case of floor cooling/heating, radiant cooling/heating delivery is more effective using floor slab than ceiling. According to Olesen [69], the effect of one degree change in floor temperature is 2.5 times the effect of ceiling temperature on the mean radiant temperature (and operative temperature). However, ASHRAE [64] presented a different view on this. Since floors are generally covered with furnishings or carpet, the effectiveness of radiant floor heating can be significantly compromised. Radiant floor system is, however, still more effective in the case of rooms with high ceilings and bare floors (or floor with little covering) and, in the case of radiant cooling, when direct solar gain is to be removed.

According to Olesen [69] citing earlier work by Lebrun and Marret [70,71], radiant floor heating exhibits a uniform vertical air temperature distribution than other systems that are more convective in nature. Also, as Olesen [69] reported, an operative temperature of 22 °C at the height of 1.1 m above the floor was attained at 1 K to 1.5 K with the radiant floor heating as opposed to other systems.

The downdraught effects can be perceived near tall windows and high U-value resulting in local discomfort to occupants near the windows, especially when the air velocity is higher than 0.18 m s^{-1}. According to Olesen [69], this can be mitigated by designing the floor heating system with higher surface temperature near or at the wall/window.

In a study, Fang et al. [72] found that, at a constant pollutant level, the perceived air quality is strongly influenced by temperature and humidity, i.e., the higher the temperature and humidity the lower the perceived air quality. Therefore, floor heating offers better perceived air quality due to resulting air temperature (and humidity) lower than other systems.

7.4. Radiant system design, modelling and control

Energy efficiency and thermal comfort attainable from a radiant system will however very much depend on the system design and the control strategy developed for the system. Numerical simulation work carried out by Henze et al. [73] compared the performance of ventilation assisted TABS with a variable air volume (VAV) system in an office building located in Omaha, Nebraska, through simulation. The building has four floors with an area of 600 m^2 (30 m × 30 m) per floor and is equipped with mechanically actuated external shading devices as solar gain control. The windows occupied about 22% of the building envelope including roof. A geothermal heat pump powered the TABS to provide heating whilst a geothermal heat exchanger provided the cooling. No additional vapour compression cycle was used. The study concluded that supply air temperature of the VAV systems, temperature set points of the active layer and reset schedules of the TABS affect the overall system performance significantly and stressed the importance of the optimal matching of the two subsystems (TABS and VAV). In the study, the set points selected were supply air temperatures "that lead to the largest possible emphasis on the TABS system without causing discomfort". The contribution of TABS to heating was found to be minor which implied the need for proper configuration of ventilation system to suit the TABS system. The study, however, found higher annual cooling energy demand for the case of the coupled system compared to that for the pure VAV system. This was attributed to lower room operative temperatures during occupied period implying better comfort. During summer periods, the room MRT in the case of the TABS is on average 2 K lower, reflected in lower PMV of 0.56 compared with 0.75 in the case of the VAV.

Using a numerical study conducted on a typical office building in Zürich, Switzerland, Lehmann et al. [74] analysed various factors that affect the energetic performance of TABS. They found that the hydronic circuit topology significantly impacts the energetic performance of such a system. Energy savings of around 15–25 kW h m^{-2} a, or equivalent to 20–30% of heating and cooling demand can be attained using separate return pipe from each zone compared to employing a common return pipe. This is because in the case of common return pipe, water returning from each zone mixes, resulting in energy losses. Lehman et al. [74] also found that the use of pulse with modulation (PWM) control strategy, which allows for intermittent operation of the hydronic circulating pumps, resulted in halving the energy required for the operation of water circulation pumps compared to the use of base control strategy (BCS). In conclusion, the study indicated that significant energy savings could be attained using adapted system topologies and appropriate control solutions for TABS.

8. Discussion

This critical overview shows once more that the radiant thermal field is an essential parameter of thermal comfort. In summer, the radiant temperature imposes an additional cooling load to an air-conditioned space in order to keep the occupants thermally comfortable. In such a situation, the use of air temperature as the sole controlled parameter (in a control system) cannot be relied upon as it can underestimate the cooling requirement of a space.

Radiant effects commonly sensed by occupants of offices occur normally in the form of radiation asymmetry whilst the MRT is the "by-product". Recommended levels of acceptability for 3 classes set for local discomfort caused by radiant temperature asymmetry have been established; however, these values are based on a study involving mainly younger participants engaged in sedentary activity; as such it cannot be taken as a general guide.

Despite the importance of considering thermal radiation effects, existing graphical representations of thermal comfort in standards such as ASHRAE 55-2010 [21] do not clearly show the effect of radiant temperature and radiant asymmetry; they are often neglected due to their complexity and practical limitations. Likewise, in the adaptive model of thermal comfort for naturally ventilated buildings, the mean radiant temperature is not directly expressed. Instead, it is implicitly expressed within the comfort equation itself through the use of operative temperature as the indoor comfort temperature. It is not clear how the adaptive approach arrived to this formulation. Ignoring the effect of radiant asymmetry can be problematic when the adaptive model may suggest higher acceptable room temperatures in summer based on the assumption that occupants will always use adaptive strategies to attain comfort. Here, assumption-based modelling and occupant-related practice may lead to conflicting situations.

The presence of a strong thermal radiant field can also become a serious issue in the context of productivity. In many offices, due to limited space or for other reasons, two people or more can occupy the same relatively small room. Imagine the situation where the one sitting near the wall or window with strong radiant field is the one who prefers to be on the cooler side of thermal sensation, and the one sitting near the inner wall prefers the warmer side of thermal sensation. In such a situation, conflicting thermal preferences due to the presence of radiant asymmetry can create what we term *thermal comfort tension*. To the best of the authors' knowledge this phenomenon has not yet been addressed in a qualitative and quantitative manner by relevant authorities such as those related to occupational health and safety.

The abovementioned observations lead to the questioning of the usefulness of substituting the air temperature sensor with an operative temperature sensor as suggested by sources [2,4,63], as discussed in the previous section. While the investigations advocate the use of the operative temperature sensor for properly accounting for the impact of the thermal radiant field, it may not result in the desired outcome. Addressing the impact of the thermal radiation field through the use of the operative temperature sensor may mean lowering the air temperature in the whole space, which can result in the occupants near the radiant field source feeling 'cooler' and more comfortable, but those far from it may feel 'colder' or even 'too cold' and become less comfortable.

In such a situation, one possible solution is minimising the radiant field impact through external or internal shading or any equivalent devices. This, however, may not totally resolve the issue, especially in the case of buildings with manually-operated window shades. A review of this type of shading device by O'Brien et al. [75] reveals that the majority of office building occupants operate the window shades based on long-term solar radiation intensity and solar geometry trends rather than on short term events. In addition, the occupants operate their shades not necessarily due to thermal comfort demand but mainly to improve visual condition, particularly, to reduce glare. On the other spectrum, applying fixed external window shades is also generally not desirable in buildings that require space heating, because the shading device will reduce the amount of solar radiation coming into the building during the heating season. Further, the capital, construction and maintenance costs of such device often discourage building owners to install them.

Currently, reliable and affordable mean radiant temperature and radiation asymmetry sensors are virtually non-existent. In such a situation, minimising the impacts of thermal radiation field on occupied space may only be realised during the design stage. In fact, due to the nature of delivering comfort in the conventional mechanical ventilation system – in which air with at a pre-set temperature and mass flow rate is supplied to the room to address the cooling load – the introduction of sensor and associated control strategy that takes into account thermal radiant field is not advisable. This is because such a 'solution' will likely to create another 'problem' in the form of thermal comfort tension among the room occupants.

Another possible solution to minimise the impacts of thermal radiation field in an occupied space is the creation of perimeter 'zones' that are likely exposed to the thermal radiant field. For such a zone, operative temperature control strategy discussed previously can be applied. However, while this can be done during the design of a new building, from economic and architectural/design stand points this may not always be practicable. The general trend in building design now, particularly offices, seems to be heading towards minimizing the presence of walls or partitions. Even if this is possible, the lack of technically viable and affordable devices makes such an option unattainable (see Section 4). In buildings such as hospitals, however, this solution may be necessary and is possible because different zones or rooms are usually separated by walls for obvious reasons. Khodakarani and Nasrollah [76] and Khodakarani and Knight [77] recommended that, as different occupant groups in hospitals have different comfort condition requirements particularly for the healing process of patients, different radiant temperatures shall be provided to accommodate these differences.

9. Conclusions

The paper has presented a critical overview of the impacts of the presence of thermal radiant field on thermal comfort, control and energy consumption of buildings. The main conclusions are:

- The thermal radiant field in the form of radiant asymmetry and mean radiant temperature in many situations are significant and must be treated as an important thermal comfort parameter.
- Findings from previous research on the thermal radiant field have been mainly based on experimental data involving a small number of participants only, while the results are not consistent. Some studies support existing models and standards, whereas others do not, and this leads to uncertainties or even conflicting outcomes.
- In real situations, radiant asymmetry and the mean radiant temperature are inseparable, and, therefore, must be treated as such in the design for thermal comfort.
- Failure to address the impacts of the presence of the thermal radiant field can lead to *thermal comfort tension* in a zone where occupants of opposing thermal sensation preferences are seated or placed in a 'wrong' position.
- Existing thermal comfort standards have not adequately addressed issues arising from the strong presence of the thermal radiant field.
- The use of the operative temperature as a parameter in the simplified ASHRAE Comfort chart can mislead the audience. Operative temperature is not a property of air which can be located on the psychrometric chart. The ASHRAE comfort chart itself is based on the assumption that mean radiant temperature is equal to air temperature. Any condition in which mean radiant temperature is different from the air temperature can hardly be identified on the chart and has to be identified through the heat balance equation from which the simplified graph originated. The same can be said of the adaptive comfort chart based on the simplified linear adaptive model which "takes into account" the mean radiant temperature through its adoption of the operative temperature as the dependent variable.
- Any future attempt to address the impacts of thermal radiant field through building design can only be aided by a comprehensive thermal comfort model which takes this factor into account. Currently, the heat balance model includes only one "aspect" of the thermal radiant field (i.e., mean radiant temperature) in the equation, and, therefore, the model needs improvement. On the other hand, due to its simplification in dealing with mean radiant temperature, the adaptive model cannot be used as scientific aid for this purpose without potential errors.
- At present there are no technically viable and affordable devices (sensors) that can be used in the control strategies to address the impacts of the thermal radiant field. Even if such a device exists, the introduction of such a control strategy may be impracticable from economic and architecture/design considerations. In addition, thermal comfort tension situation may not be resolved by such a strategy.
- While applying window shadings whether internal or external to the building can help reduce the radiant field impact, it also poses other problems. Building occupants often operate them in order to reduce glare instead of minimizing the radiant field impact. In buildings where heating is necessary, having window shading, particularly if it is fixed, will reduce solar heat gain during heating periods. The capital, construction and maintenance costs of such devices often prevent building owners from implementing them.

  There have been increased research and development activities on systems which deliver thermal comfort based on thermal radiant concepts. These include radiant slabs and panels for cooling and heating. These systems generally need to be coupled with conventional systems to address both sensible and latent loads. Whilst energy savings and improved comfort potentials using these systems have been identified, further work needs to be carried out in terms of system design, configuration, and control.

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