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Thermal comfort: A review paper

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ABSTRACT

This paper presents a literature review of thermal comfort. Both rational and adaptive thermal comfort approaches are presented. An overview of the human body thermoregulatory system as well as the mathematical modelling of heat exchanged between human body and its environment in the situations of both awaked and sleeping people is presented.

Contents

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C	convective heat loss (W/m^2)
C_p	specific heat (J/kg K)
C_{res}	sensible heat loss due to respiration (W/m^2)
E	evaporative heat loss (W/m^2)
F_{cl}	clothing area factor
h	heat transfer coefficient (W/m^2 K)
L_R	Lewis ratio (K/kPa)
m	body mass (kg)
M	metabolic heat production (W/m^2)
p	pressure (kPa)
q	heat flow (W/m^2)
R	radiative heat loss (W/m^2)
R_a	thermal resistance of air layer (m^2 °C/W)
R_{cl}	thermal resistance of clothing (m^2 K/W)
$R_{e,cl}$	evaporative resistance of clothing (m^2 kPa/W)
$R_{e,t}$	total evaporative resistance (m^2 kPa/W)
R_t	total resistance of a bedding system including the air layer around a covered body (m^2 °C/W)
S	heat storage (W/m^2)
t	time (s)
T	temperature (K or °C)
v	air velocity (m/s)
w	skin wettedness
W	external work (W/m^2)

Greek letters

α_{sk}	fraction of total body mass concentrated in skin compartment

Subscripts

a	water vapour in ambient air
a,s	water vapour in saturated air at ambient temperature
b	body tissue
c	convective
cl	clothing
cr	core compartment
dif	moisture diffusion through skin
e	evaporative
o	operative
r	radiant
res	respiration
rew,req	regulatory sweating required for comfort
rsw	regulatory sweating
sk	skin compartment
sk,req	skin required for comfort
sk,s	water vapour in saturated air at skin temperature

1. Introduction

Thermal comfort has been defined by Hensen as "a state in which there are no driving impulses to correct the environment by the behaviour" [1]. The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) defined it as "the condition of the mind in which satisfaction is expressed with the thermal environment" [2]. As such, it will be influenced by personal differences in mood, culture and other individual, organizational and social factors. Based on the above definitions, comfort is not a state condition, but rather a state of mind. The definition of thermal comfort leaves open as to what is meant by condition of mind or satisfaction, but it correctly emphasizes that the judgment of comfort is a cognitive process involving many inputs influenced by physical, physiological, psychological, and other factors [3].

Thermal sensations are different among people even in the same environment. Even though the sensors render the same results regardless to the geographical position where a measurement is being taken, this is not the case for persons. Indeed, persons staying in very similar spaces, subjected to the same climate, and belonging to a common culture, issue very different opinions on thermal comfort due to the combination of a large number of factors that affect the perception of human beings. Subjects' diagnosis is therefore an indispensable tool to achieve an overall evaluation of the study parameters [4]. Conventionally, thermal discomfort is treated as a subjective condition while thermal sensation is an objective sensation [1]. Satisfaction with the thermal environment is a complex subjective response to several interacting and less tangible variables [5]. In other words, there is really no absolute standard for thermal comfort. In general, comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized. Comfort also depends on behavioural actions such as altering clothing, altering activity, changing posture or location, changing the thermostat setting, opening a window, complaining, or leaving a space. In 1962, Macpherson defined the following six factors as those affecting thermal sensation: four physical variables (air temperature, air velocity, relative humidity, mean radiant temperature), and two personal variables (clothing insulation and activity level, i.e. metabolic rate) [3]. Thermal comfort standards determine the energy consumption by a building's environmental systems; therefore, they play an important role in building sustainability [6]. This energy often involves the combustion of fossil fuels, contributing to carbon dioxide emissions and climate change [7]. Thermal comfort is also a key parameter for a healthy and productive workplace [8,9].

With the urgent need to reduce the economic and environmental cost of energy consumption, investigations covering many aspects related to thermal comfort in indoor environments have attracted authors for decades. These include establishing models [10,11] and indices [12], carrying out experiments in climate chambers [10,13] and field surveys [3,14], establishing thermal comfort standards and evaluation methods [15,16], etc. The most important findings are now the basis of national and international standards, e.g. [17,18]. They focused on correlations for thermal comfort criteria or on health issues like the Sick-Building-Syndrome [9]. To determine appropriate thermal conditions, practitioners refer to standards. The standards define temperature ranges that should result in thermal satisfaction for at least 80% of occupants in a space [19]. The international comfort standards such as ASHRAE standards and the International Standards Organization (ISO) are almost exclusively based on theoretical analyses of human heat exchange performed in mid-latitude climatic regions in North America and northern Europe [5,17]. They were based primarily on mathematical models developed by Fanger on the basis of studies from special climate-controlled chamber experiments. Moreover, these standards are suitable for static, uniformly thermal conditions and are based on the hypothesis that regardless of race, age and sex; human beings are thought to feel comfortable in a narrow, well-defined range of thermal conditions [2,14].

De Dear and Brager [20] noted that "current thermal comfort standards and the models underpinning them purport to be equally applicable across all types of building, ventilation, occupancy pattern and climate zone". The thermal comfort standards prescribed by ISO 7730 are the first that have been used on a world-wide basis [21].

Although there is much documented material worldwide concerning human thermal comfort from the physiological, adaptive and social convention paradigms, throughout sub-Saharan Africa and particularly in the tropical regions, there have been few literature reports on occupants' comfort and residential thermal environment. Moreover, the standards are almost based on experiments across a variety of climate zones including temperate, hot-humid and cold. Different climatic regions, such as the tropics, may require different levels of comfort parameter mandated in the standards. As a part of a project aiming to study thermal comfort in the tropical Sub-Saharan Africa regions, the objective of this work is to review the present state of thermal comfort approaches analysis, physiological basis of thermal comfort and thermal modelling. Such studies could enable the development and recommendations of different levels of tropical comfort parameters mandated in the standards.

2. Thermal comfort approaches

At present, two different approaches for the definition of thermal comfort coexist, each one with its potentialities and limits: the rational or heat-balance approach and the adaptive approach [22]. The rational approach uses data from climate chamber studies to support its theory, best characterized by the works of Fanger while the adaptive approach uses data from field studies of people in building [7].

2.1. The rational or heat-balance approach

Steady-state experiments showed that, cold discomfort is strongly related to the mean skin temperature and that warmth discomfort is strongly related to the skin wettedness caused by sweat secretion. Dissatisfaction may be caused by the body as a whole being too warm or cold, or by unwanted heating or cooling of a particular part of the body (local discomfort) [1]. These relations are the basis for methods like, Fanger's [10] comfort model that incorporates the six factors mentioned by Macpherson, and the two-node model of Gagge et al. [12]. In an evaluation by Doherty and Arens [22], it was shown that these models are accurate for humans involved in near-sedentary activity and steady-state conditions.

The heat-balance approach is based on Fanger's experiments [10] in controlled climate chamber on 1296 young Danish students, using a steady-state heat transfer model. In these studies, participants were dressed in standardised clothing and completed standardised activities, while exposed to different thermal environments. In some studies the thermal conditions were chosen, and participants recorded how hot or cold they felt, using the seven-point ASHRAE thermal sensation scale ranging from cold (-3) to hot (+3) with neutral (0) in the middle as follows: (i) ± 1: slightly warm (+) or cool ($-$); (ii) ± 2: warm (+) or cool ($-$); (iii) ± 3: hot (+) or cold ($-$); (iv) 0: neutral (neither cool nor warm) [14]. In other studies, participants controlled the thermal environment themselves, adjusting the temperature until they felt thermally 'neutral' [19].

Fanger's model combines the theories of heat balance with the physiology of thermoregulation to determine a range of comfort temperatures which occupants of buildings will find comfortable. According to these theories, the human body employs physiological processes (e.g. sweating, shivering, regulating blood flow to the skin) in order to maintain a balance between the heat produced by metabolism and the heat lost from the body. Maintaining this heat balance is the first condition for achieving a neutral thermal sensation [19]. However, Fanger [10] noted that "man's thermoregulatory system is quite effective and will therefore create heat balance within wide limits of the environmental variables, even if comfort does not exist".

To be able to predict conditions where thermal neutrality would occur, Fanger [24] investigated the body's physiological processes when it is close to neutral. He determined that the only physiological processes influencing heat balance in this context were sweat rate and mean skin temperature, and that these processes were a function of activity level. He used data from a study on 183 college-age participants exposed to different thermal conditions while wearing standardised clothing to develop a linear relationship between activity level and sweat rate. He also conducted a study using 20 college-age participants who wore standardised clothing and took part in climate chamber tests at four different activity levels (sedentary, low, medium and high), to derive a linear relationship between activity level and mean skin temperature. After substituting these two linear relationships into heat balance equations, a 'comfort equation' was obtained. The comfort equation predicts conditions where occupants will feel thermally neutral.

That comfort equation was expanded [10] using data from 1296 participants. The resulting equation described thermal comfort as the imbalance between the actual heat flow from the body in a given thermal environment and the heat flow required for optimum (i.e. neutral) comfort for a given activity. This expanded equation related thermal conditions to the seven-point ASHRAE thermal sensation scale, and became known as the "Predicted Mean Vote" (PMV) index. The PMV was then incorporated into the "Predicted Percentage of Dissatisfied" (PPD) index. Fanger's PMV-PPD model on thermal comfort has been a path breaking contribution to the theory of thermal comfort and to the evaluation of indoor thermal environments in buildings. It is widely used and accepted for design and field assessment of thermal comfort [3].

In addition to Fanger's PMV-PPD model, a two-node model also known as the Pierce two-node model developed by Gagge et al. [12] (at the J.B. Pierce Foundation Laboratory, Yale University) was based on the heat balance equation developed by Stolwijk and Hardy [26], and Gagge and Nishi [27]. That comfort model used a two-compartment body structure, dividing a body into two concentric cylinders. The inner cylinder for the body core whose temperature T_{cr} is 37.1 °C, and the outer one for the skin layer whose temperature T_{sk} is 33.1 °C [3].

The PMV-PPD model is useful only for predicting steady-state comfort responses while a two-node model can be used to predict physiological responses or responses to transient situations [3].

There have been extensive studies to evaluate thermal comfort using test chambers; e.g. Fanger [10] in Denmark, Tanabe et al. [28] in Japan, Chung and Tong [29] in Hong Kong, De Dear and Leow [30] in Singapore, etc.

2.1.1. The predicted mean vote (PMV)

The PMV index suggested by Fanger predicts the mean response of a large group of people according to the ASHRAE thermal sensation scale. Subjects exposed to the climate chambers are asked to give their opinions according to the ASHRAE seven-point scale of thermal sensation. A mean vote (MV) is obtained for a given condition by finding the mean value of the feeling given by all the subjects for that condition. Fanger related PMV to the imbalance between the actual heat flow from a human body in a given environment and the heat flow required for optimum comfort at a specified activity by the following equation [3]:

$$PMV = [0.303\exp(-0.036M) + 0.028]L = \alpha L \quad (1)$$

where L is the thermal load on the body, defined as the difference between internal heat production and heat loss to the environment for a person hypothetically kept at comfort values of T_{sk} and E_{rsw} at the activity level, and α the sensitivity coefficient.

The Institute for Environmental Research of the State University of Kansas, under ASHRAE contract, has conducted extensive research on the subject of thermal comfort in sedentary regime. The purpose of this investigation was to obtain a model to express the PMV in terms of parameters easily sampled in an environment. The results have yielded to an expression of the form [31]:

$$PMV = aT + bP_v - c \quad (2)$$

where P_v is the pressure of water vapour in ambient air and T the temperature. Coefficients a, b and c are given in Table 1. With these criteria it has been given a comfort zone that, on average, is close to conditions of 26 °C and 50% relative humidity. This study was undergone with subjects to a sedentary metabolic activity, dressed with normal clothes and with a thermal resistance of approximately 0.6 clo, and with exposure to the indoor ambiences of 3 h.

2.1.2. The predicted percentage of dissatisfied (PPD)

The PPD predicts the percentage of the people who felt more than slightly warm or slightly cold (i.e. the percentage of the people who inclined to complain about the environment). Using the seven-point scale of thermal sensation (−3 to +3), Fanger [32] postulated: are declared uncomfortable all those who responded ±2 and ±3. Those who responded ±1 and 0 are declared comfortable. The percentages of subjects who responded ±2 and ±3 are determined for each class of PMV; that variable has been called PPD. The relationship between PPD and PMV is given by [33]:

$$PPD = 100 - 95\exp[-(0.03353PMV^4 + 0.2179PMV^2)] \quad (3)$$

The merit of this relation is that, it reveals a perfect symmetry with respect to thermal neutrality (PMV = 0). It can be seen (Fig. 1) that, even when the PMV index is 0, there are some individual cases of dissatisfaction with the level of temperature, although all are dressed in a similar way and that the level of activity is the same. This is due to some differences of approach in the evaluation of thermal comfort from one person to another. It is shown that at PMV = 0, a minimum rate of dissatisfied of 5% exists [34].

Thermal comfort standards use the PMV model to recommend acceptable thermal comfort conditions. The recommendations made by ASHRAE Standard 55 are shown in Table 2. These conditions were assumed for a relative humidity of 50%, a mean relative velocity lower than 0.15 m/s, a mean radiant temperature equal to air temperature and a metabolic rate of 1.2 met. Clothing insulation was defined as 0.9 clo in winter and 0.5 clo in summer [31].

Depending on the ranges PPD and PMV admissible, three kinds of comfort zones can be accessed as reflected in Table 3 [31].

However, the laboratory studies offer static and consistent conditions for measurement not possible in the field studies. It is now widely accepted that the previously used climate chambers fail to provide the participating humans with so-called "experiential realism" in determining thermal comfort [35], since "real" people live in changeable, inconsistent environments, which may cause concerns when the standards are applied to residents living in real-world situations [14]. McIntyre [36] presented a comparison of Fanger's climate chamber work with field studies reviewed by Humphreys [21], suspecting that certain intervening variables that occur in the "real" world might not be reproducible in the climatic chamber. Oseland [25] reported on significant discrepancies occurring between predicted mean votes (PMV) and actual mean votes (AMV) values obtained in offices and homes as compared with climate chamber studies, attributing the differences to

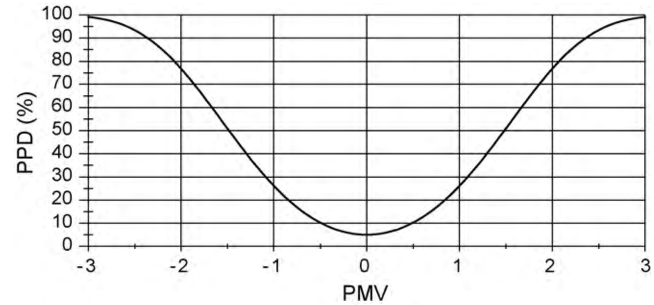

Fig. 1. Relationship PMV versus PPD.

Table 1
Values of the coefficients a, b and c as a function of spent time and the sex of the subject [31].

Time/sex	a	b	c
1 h/man	0.220	0.233	6.673
Woman	0.272	0.248	7.245
Both	0.245	0.248	6.475
2 h/man	0.221	0.270	6.024
Woman	0.283	0.210	7.694
Both	0.252	0.240	6.859
3 h/man	0.212	0.293	5.949
Woman	0.275	0.255	8.620
Both	0.243	0.278	8.802

Table 2
ASHRAE Standard recommendations [31].

	Operative temperature	Acceptable range
Summer	22 °C	20–23 °C
Winter	24.5 °C	23–26 °C

Table 3
Predicted percentage of dissatisfied (PPD) based on the predicted mean vote (PMV) [31].

Comfort	PPD	Range of PMV
1	<6	−0.2 < PMV < 0.2
2	<10	−0.5 < PMV < 0.5
3	<15	−0.7 < PMV < 0.7

contextual and adaptation effects as follows: "since the development of the PMV equation many field studies have shown differences between the occupants' reported TS [thermal sensation] and those predicted by PMV and the corresponding neutral temperatures". Thus, the field studies closer to the "real" world may be preferable to climate chamber ones [37].

2.2. Adaptive approach

Adaptive approach derives from field studies, having the purpose of analysing the real acceptability of thermal environment, which strongly depends on the context, the behaviour of occupants and their expectations. The adjustments have been summarized by De Dear [23] in three categories: behaviour adaptation, physiological adaptation and psychological adaptation. In recent years, different authors have encouraged field studies in addition to laboratory experiments, in order to get more reliable information about the actual workplace comfort and the relevant (interacting) parameters. Field studies also allow for analyses of other factors than those that can be simulated in chambers, as the subjects provide responses in their everyday habitats, wearing their everyday clothing and behaving without any additional restrictions [38]. The subjectivity in thermal experience and the interpretations flowing from a very complex interaction between the occupants and their environment has been the focus of a great deal of study and provides the theoretical underpinning to the adaptive approach to thermal comfort studies [39]. Several adaptive studies are found in the literature, these include: (i) thermal comfort models and techniques, (ii) comparative studies between traditional and modern living spaces, (iii) building performance assessing methods, (iv) low energy consumption systems, (v) comparative studies with regard to sex, (vi) effects of indoor climates on thermal perceptions, (vii) thermal comfort in classrooms, (viii) adaptive algorithms, (ix) patients' thermal comfort in hospitals, and (x) thermal comfort in outdoor environments, etc.

2.2.1. Thermal comfort models and techniques

Ogbonna and Harris [5] used adaptive thermal comfort paradigm (based on the theory that physiological and adaptive factors play equally central roles in the perception and interpretation of thermal comfort) to provide empirical data about the range of conditions for which occupants in naturally ventilated buildings are comfortable in Jos; a Nigerian city in the tropical savannah region of Africa. From the thermal-comfort and indoor-air-quality points-of-views, such studies would enable the development and recommendation of comfort standards. Jannot [40] used meteorological data of Ouagadougou; western Africa to show that acceptable thermal comfort can be obtained all year round by direct evaporative cooling of outside air. Experimental results obtained by testing a direct evaporative cooler were given and showed that the use of both a direct evaporative cooler and some techniques of passive cooling must be done for the best thermal comfort. Pasupathy et al. [41] analysed efficient and economical technology that can be used to store large amounts of heat or cold in a definite volume. They showed that thermal storage plays an important role in building energy conservation, which is greatly assisted by the incorporation of latent heat storage (LHS) in building products. LHS in a phase change material is very attractive because of its high storage density with small temperature swing. They also showed that increasing the thermal storage capacity of a building can increase human comfort by decreasing the frequency of internal air temperature swings so that the indoor air temperature is closer to the desired temperature for a longer period of time. Chu and Jong [42] proposed a least enthalpy estimator (LEE) that combines the concept of human thermal comfort with the theory of enthalpy to predict the load for a suitable setting pair in order to maintain more precisely the thermal comfort level and save energy in the air conditioning system. Experimental test showed that, the air conditioning control system setting would rapidly and generally respond to the worst case of space thermal condition and finely adjust to the gradual change in the space thermal conditions as determined by the LEE. Thus, the LEE based thermal comfort controller could achieve the requirements of both thermal comfort and energy saving simultaneously. Shukuya [43] discussed how a built environmental control system such as space heating and cooling can be described by the concept of exergy, which quantise what is consumed by any working systems from man-made systems such as heat engines or buildings to biological systems including human body. The use of exergy concept is to deepen the understanding of the built environment and thereby to develop a variety of low-exergy systems for future buildings. Chen [44] presented an overview of the tools used to predict ventilation performance in buildings. The tools reviewed were analytical models, empirical models, small-scale experimental models, full-scale experimental models, multizone network models, zonal models, and computational fluid dynamics (CFD) models. He found that the analytical and empirical models had made minimal contributions to the research literature in the past year. The small- and full-scale experimental models were mainly used to generate data to validate numerical models. The multizone models were improving, and they were the main tool for predicting ventilation performance in an entire building. The zonal models had limited applications and could be replaced by the coarse-grid fluid dynamics models. The CFD models were most popular even though considerable efforts are still to be made to seek more reliable and accurate models or by coupling CFD with other building simulation models. The applications of CFD models were mainly for studying indoor air quality, natural ventilation, and stratified ventilation as they were difficult to be predicted by other models. Beker and Paciuk [45] highlighted the role of contextual variables (local climate, expectations, available control) in thermal adaptation in actual settings and established the baseline data for local standardized thermal and energy calculations. Hwang et al. [34] investigated on thermal comfort in workplaces and residences in Taiwan to clarify two questions in detail: (i) do people in the tropical climate regions demonstrate a correlation between thermal sensation and thermal dissatisfaction the same as the PMV/PPD formula in the ISO7730? and (ii) does the difference in opportunities to choose from a variety of methods to achieve thermal comfort affects thermal perceptions of occupants? A new predicted formula of percentage of dissatisfied (PD) relating to mean thermal sensation votes (TSVs) was proposed for hot and humid regions. Besides an increase in minimum rate of dissatisfied from 5% to 9%, a shift of the TSV with minimum PD to the cool side of sensation scale was suggested by the new proposed formula. They also found that the effectiveness, availability and cost of a thermal adaptation method can affect the interviewee's thermal adaptation behaviour. Yao et al. [6] presented a theoretical adaptive model of thermal comfort based on the Black Box theory, taking into account factors such as culture, climate, social, psychological and behavioural adaptations, which have an impact on the senses used to detect thermal comfort. The model is called the adaptive predicted mean vote (aPMV) model. Fig. 2 shows the thermal comfort mechanism of the adaptive approaches proposed by these authors. The aPMV model explains, by applying the cybernetics concept, the phenomena that the predicted mean vote (PMV) is greater than the actual mean vote (AMV) in free-running buildings, which has been revealed by many researchers in field studies. An adaptive coefficient λ representing the adaptive factors that affect the sense of thermal comfort has been proposed. Zingano [46] analysed data from eighteen meteorological stations evenly distributed throughout Malawi; eastern Africa, to discuss the importance of humidity to thermal comfort temperatures. He presented an indirect method for determining the midpoint of the thermal comfort temperature by

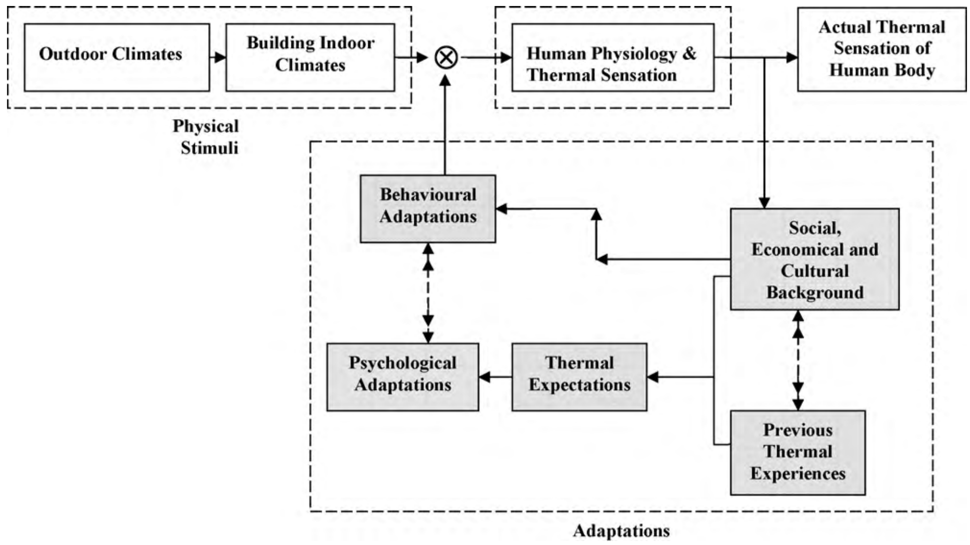

Fig. 2. The thermal comfort adaptive model mechanism.

analysing preferred bath water temperature and found that 24.6 °C can be taken as the midpoint of the comfortable thermal zone in Malawi. On the other hand, Holm and Engelbrecht [47] compared the neutrality temperatures based on the new effective temperature (ET*) with that based on the dry bulb (DB) temperature and provided the motivation why the DB base is preferable, given the relatively favourable South African climate conditions and the ease of calculation. They showed that in South Africa the temperature difference between TnET* (neutrality based on new effective temperature) and TnDBT (neutrality based on dry bulb temperature) is negligible and recommended that the more practical dry bulb-based neutrality temperature (TnDBT) be adopted for naturally ventilated buildings in the format of $TnDBT = 17.6 + 0.31 \times To_{ave}$ with $17.8\,°C < TnDBT < 29.5\,°C$ Where To_{ave} is an average outdoor DBT of the day, month or year. DBT is calculated as the average of maxima and minima.

2.2.2. Comparative studies between traditional and modern living spaces

Ealiwa et al. [37] investigated on thermal comfort within two types of buildings (traditional and modern) in the Ghadames oasis of Libya. They showed that the measurements of PMV in modern air-conditioned buildings provide satisfactory comfort conditions according to ISO 7730. But, equivalent measurements and survey results in traditional buildings indicated that although the PMV based on measurements and ISO 7730 implied discomfort (hot), the occupants expressed their thermal satisfaction with the indoor comfort conditions. Higher standard of thermal comfort were expressed in old buildings than the modern one. On the other hand, Akair and Bánhidi [48] also conducted a thermal comfort field survey in three towns from two climatic zones in Libya. They suggested that in Libya, the thermal comfort temperature can be calculated from one of the expressions: Tc-Griffiths = 0.518To − Avg + 10.35, Tc-Brager = 0.680To − Avg + 6.88 where Tc-Griffiths (°C) is the comfort temperature calculated using the Griffiths method, Tc-Brager (°C) is the comfort temperature found according to De Dear and Brager Method, To-Avg (°C) is the monthly mean outdoor temperature. For buildings equipped with heating and air conditioning systems, a variable indoor temperature has to be taken according to the comfort temperature calculated from the above equation. Such method allows an important energy saving compared with the existing standards.

2.2.3. Building performance assessing methods

Wagner et al.'s [9] investigations on workplace occupant satisfaction in office buildings in Germany revealed that the occupants' control of the indoor climate and moreover the perceived effect of their intervention strongly influence their satisfaction with thermal indoor conditions. They introduced a method for assessing the building performance by occupant surveys calculating the weighted importance of every satisfaction parameter in relation to the general acceptance of the workplace and then ranking the different satisfaction parameters.

2.2.4. Low energy consumption systems

Mui and Chan [49] developed new notions about adaptive comfort temperature (ACT) in buildings in humid sub-tropical regions. An adaptive interface relationship of indoor comfort temperature with outdoor air temperature was found in order to optimise the energy used for cooling air, and to achieve the acceptance of thermal comfort, as determined by physical measurements and subjective surveys. They showed that with the use of the ACT model, the total percentage of energy saving was about 7%. Taylor et al. [8] used the data from a two-storey rammed earth building in Wodonga; Australia to describe an evaluation of the building in terms of measured thermal comfort and energy use. They found that the building was too hot in summer and too cold in winter. Comparison with another office building in the same location showed that the rammed earth building used more energy for heating. They showed that improvements could be made by design and control strategy changes. Han et al. [14] reported on thermal comfort inside residences of three cities in the hot-humid climate of central southern China. Results of this study can be used to design low energy consumption systems for occupant thermal comfort in central southern China. Schiavon and Melikov [35] investigated on the potential saving of cooling energy by elevated air speed which can offset the impact of increased room air temperature on occupants' comfort, as recommended in the present standards (ASHRAE 55 2004, ISO 7730 2005 and EN 15251 2007). Fifty-four cases covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem and Athens), three indoor environment categories I, II and III (according to standard EN 15251 2007) and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated and revealed that the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature. Van Hoof and Hensen [50] discussed two implementations of the adaptive comfort model in terms of usability and energy use for moderate maritime climate zones of the Netherlands. They found that for moderate climate zones, the adaptive model is only applicable during summer months, and can reduce energy for naturally conditioned buildings. However, the adaptive thermal comfort model has very limited application potential for such climates. Additionally, they suggested a temperature parameter with a gradual course to replace the mean monthly outdoor air temperature to avoid step changes in optimum comfort temperatures. Su et al. [51] showed that the energy-utilization coefficient is not suitable to the evaluation of natural ventilation for in which the non-renewable energy is not needed. They also established the thermal comfort model of people under natural ventilation environment, an evaluation method was established based on it, and the natural ventilation system of an office building in China was evaluated. Recently, Kwok and Rajkovich [7] examined adaptation at two scales: the first with an approach examining the broader concept of thermal comfort of the immediate environment and a second parallel approach as longer term building design response to global warming. They argued that both mitigation of greenhouse gases and adaptation to climate change should be added to building codes and standards.

2.2.5. Comparative studies with regard to sex (male, female)

Wang's [52] investigations on the thermal environment and thermal comfort in residential buildings in Harbin, northeast of China showed that males are less sensitive to temperature variations than females; the neutral operative temperature of males was 1.1 °C lower than that of females.

2.2.6. The effects of indoor climates on thermal perceptions

Cena and De Dear [38] discussed the effects of indoor climates on thermal perceptions and adaptive behaviour of office workers during a large field study in Kalgoorlie-Boulder, located in a hot-arid region of Western Australia. They found that for clothing insulation levels of 0.5 clo in summer and 0.7 in winter, thermal neutrality according to the ASHRAE seven-point scale occur at 20.3 °C in winter and 23.3 °C in summer. The effect of hot-dry/cool-dry seasonality on thermal comfort responses of office workers was also significant.

2.2.7. Thermal comfort in classrooms

Corgnati et al. [53] investigated on thermal comfort in Italian classrooms. The surveys were carried out during the mid-season in free running conditions in Turin, located in the North-West of Italy. Their study followed a previous one based on a monitoring campaign performed during the heating season. Responses from these two different configurations were integrated, analysed and compared. A trend characterized by a gradual change in the thermal preference from the heating season to the mid and warm season was showed. Buratti and Ricciardi [54] applied a questionnaire in autumn, winter and spring in classrooms of the Universities of Perugia, Terni and Pavia. They found a linear correlation between the PMV versus the difference between the Equivalent Uniform Temperature and the Comfort Uniform Temperature while a second-degree polynomial relation was obtained between the PPD versus the absolute value of the same difference between temperatures.

2.2.8. Adaptive algorithms

Moujalled et al. [55] surveyed five buildings located in the southeast region of France to develop adaptive algorithms for naturally ventilated buildings from both static and adaptive approach. Kumar and Mahdavi [56] showed that the mathematical models of thermal comfort sometimes fail to accurately describe or predict thermal comfort in workplace settings even when the values of environmental and personal parameters are known, and suggested a critical need to provide a thermal comfort evaluation framework that, in addition to the algorithmic implementation of mathematical thermal comfort prediction models, would make use of the empirical knowledge base accumulated over years from field experiments around the world.

2.2.9. Patients' thermal comfort in hospitals

Hwang et al. [57] investigated on the applicability of the comfort criteria of ASHRAE Standard in hospital environments in Taiwan and found that above half of the measured samples failed to meet the specifications of Standard 55 comfort zone due to improper humidity control. Acceptability votes by patients exceeded the Standard's 80% criterion, regardless of whether the physical conditions were in or out of the comfort zone. Results of chi-square tests revealed that patients' physical strength significantly effected their thermal requirements and the net effect of health yields a marked difference in thermal neutrality and preference, and also in the comfortable temperature range.

2.2.10. Thermal comfort in outdoor spaces

Zambrano et al. [58] presented an evaluation of thermal comfort in four square locations of the condominium Downtown, in the west zone of Rio de Janeiro. They showed the applicability of the Fanger's model in evaluation studies of the thermal comfort in outdoor spaces. Also, from the considered environment conditions they showed that it was possible to establish this influence on the thermal comfort for the locations selected in the square.

However, one should be very careful when interpreting the results of thermal comfort campaigns. The actual standards help but should not be considered as absolute references. The individual state of mind that expresses satisfaction with the thermal environment is too diverse for that when small groups are considered. The ASHRAE standard 55-2004 is right when stating that subjective evaluation is required even though a reference method as given in the standards is needed [59]. The results of field studies indicate that the agreement between the expression of thermal comfort proposed by the standards and sensations people really feel is not good [14]. Various field studies have investigated the preference votes regarding the indoor thermal environment, with respect to conditions of thermal neutrality (the condition in which the subject would prefer neither warmer nor cooler surroundings) among other that of McIntyre. The McIntyre's preference is based on the three-point scale whether the respondent would like a change in the thermal environment. Possible responses are "want warmer" (+1), "want no change" (0), or "want cooler" (-1) [37,60]. McIntyre found that people of warm climates may prefer what they call a "slightly cool" environment and, on the contrary, people of cold climates may prefer what they call a "slightly warm" environment. Recent field studies confirm the same tendency outlined by McIntyre's research [52].

3. Physiological basis of comfort

3.1. Human body: a thermodynamic machine

The Chartered Institute of Building Service Engineers (CIBSE) Guide prescribes that an average person emits 115W thermal energy which is a by-product of the body metabolism from the ingested food [46]. The human body is not exempted from the effects of the second law of thermodynamics; the heat generated from metabolism (i.e. oxidation of food elements) has to be dissipated. Human body can be assimilated to a thermodynamic machine whose efficiency η is given by [46]:

$$\eta = 1 - \frac{T_a}{T_b} \tag{4}$$

where T_a is the ambient temperature (in °C) and T_b is the body temperature (in °C). The three common body index temperatures are 36.6 °C (oral), 37 °C (anal), and 35 °C (skin temperature). While 37 °C is the temperature of the internal organs, the skin temperature is the reference datum for the thermal comfort sensation [46].

3.2. The human dynamic thermoregulatory system

Human body produces heat principally by metabolism, exchanges heat with the environment (mainly by radiation and convection) and loses heat by evaporation of body fluids [1]. 75% of the energy is dissipated by radiation and convection while the balance is dissipated by evaporation [46]. When the body heat cannot be dissipated to the surrounding environment; a condition that occurs when the ambient temperature is higher than the body temperature, then thermal discomfort starts.

During normal rest and exercise the heat transfer processes result in average vital organ temperatures near 37 °C. The body's temperature control system tries to maintain these temperatures when thermal disturbances occur. According to Hensel [60], the human thermoregulatory system is more complicated and incorporates more control principles than any actual technical control system. It behaves mathematically in a highly non-linear manner and contains multiple sensors, multiple feedback loops and multiple outputs. Fig. 3 shows some basic features of the human thermoregulatory system. The controlled variable is an integrated value of internal temperatures (i.e. near the central nervous system and other deep body temperatures) and skin temperatures. The controlled system is influenced by internal (e.g. internal heat generation by exercise) and external (e.g. originating from environmental heat or cold) thermal disturbances. External thermal disturbances are rapidly detected by thermoreceptors in the skin. This enables the thermoregulatory system to act before the disturbances reach the body core. Important in this respect is that the thermoreceptors in the skin respond to temperature as well as to the rate of change of temperature. Autonomic thermoregulation is controlled by the hypothalamus. There are different autonomic control actions such as adjustment of: heat production (e.g. by shivering), internal thermal resistance (by vasomotion; i.e. control of skin blood flow), external thermal resistance (e.g. by control of respiratory dry heat loss), water secretion and evaporation (e.g. by sweating and respiratory evaporative heat loss). The associated temperatures for these autonomic control actions need not necessarily be identical nor constant or dependent on each other [1].

Besides autonomic thermoregulation, there is also behavioural thermoregulation with control actions such as active movement and adjustment of clothing. Behavioural thermoregulation is associated with conscious temperature sensation as well as with thermal comfort or discomfort [1].

A number of models for simulation of the dynamic behaviour of the human thermoregulatory system have been developed in the past. A well-known example is the model of Stolwijk [61] which was later expanded by Gordon [62]. In this model, the human body is divided into a large number of segments (originally 24 and in Gordon's version 140) linked together via the appropriate blood flows. Each segment represents volume, density, heat capacitance, heat conductance, metabolism and blood flow of a certain part of the body [1]. The temperature and rate of change of temperature of each segment is available as an input into the control system, and any effector output from the control system can be applied to any part of the controlled system. The main application field for this kind of model is research on body temperature regulation itself.

4. Mathematical modelling of heat exchanged between human body and its environment

4.1. The DuBois area

The total metabolic rate of work produced within the human body is dissipated to the environment through skin surface. The most useful measure of nude body surface area is given by the

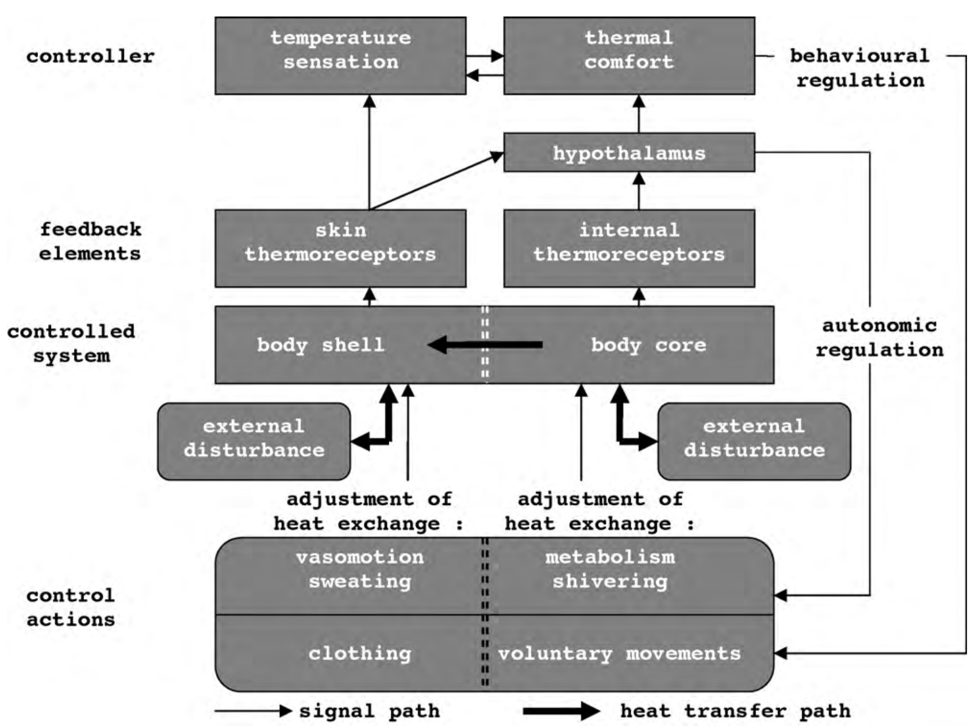

Fig. 3. Diagram of autonomic and behavioural human temperature regulation [1].

following formula known as the DuBois area (A_d) [33]:

$$A_d = 0.2025 m^{0.425} l^{0.725} \quad (5)$$

where m is the body mass and l is the body height.

4.2. Thermal effects participating into the heat exchanges

Six main effects participate into the heat exchanges between human body and its environment: conductive, convective, radiative, moisture, clothing and metabolic effects.

4.2.1. Conductive effect

Even if the human body exchanges heat by conduction (K), only small body surface is concerned. When great body area is in contact for example with furniture (e.g. chair, armchair, sofa, bed, etc.), thermal equilibrium rapidly occurs. The body area in contact with the furniture behaves as an insulating toward the environment. Generally in the steady state, the bodies' temperatures and the conductive effects are neglected; rather they are included into the convective exchanges [33].

4.2.2. Convective effect

The global convective heat flux exchanged (C) between the human body and its environment is [33]:

$$C = h_c(T_a - \bar{T}_{sk}) A_c F_{cl} \quad (6)$$

where h_c is the convective heat transfer coefficient, T_a is the ambient air temperature and $\bar{T}_{sk}$ is the mean skin temperature, A_c is the effective convection body area (almost taken to be equal to the DuBois area A_d), F_{cl} is the clothing area factor (F_{cl} = 1 when there is no clothing insulation and ≈ 0 for high clothing insulation). The convective heat coefficient is given by [33]:

$$h_c = 3.5 + 5.2 v_{ar} \quad \text{for} \quad v_a \le 1\,\text{m/s} \quad (7)$$

$$h_c = 8.7 v_{ar}^{0.6} \quad \text{for} \quad > 1\,\text{m/s} \quad (8)$$

where v_a is the air velocity, v_{ar} is the resultant air velocity taking into account the ambient air velocity and that due to activities and displacements of the person [33]:

$$v_{ar} = v_a + 0.0052(M - 58) \quad (9)$$

M is the metabolic heat production defined in paragraph 4.2.5 with the supplementary condition $M = 200\,\text{W/m}^2$ when M is found greater than 200 W/m^2 so that to limit the second term of Eq. (6) to 0.7.

4.2.3. Radiative effect

The radiative heat lost from the skin (R) is given by [33]:

$$R = h_r(\bar{T}_r - \bar{T}_{sk}) A_r F_{cl} \quad (10)$$

where h_r is the radiative heat transfer coefficient, $\bar{T}_r$ is the mean radiant temperature and $\bar{T}_{sk}$ is the mean skin temperature, A_r is the effective radiation area of the body, F_{cl} is the clothing area factor. The radiative heat transfer coefficient h_r is given by [33]:

$$h_r = 4\sigma\varepsilon_{sk}\left(\frac{\bar{T}_r + \bar{T}_{sk}}{2}\right)^3 \quad (11)$$

where $\sigma = 5.67 \times 10^8\,\text{W/m}^2\,\text{K}^4$ is the Stefan-Boltzmann coefficient and ε_{sk} = 0.97 is the emissivity of the skin. The effective radiation area of the body (A_r) is not easy to be evaluated. It can be written as [33]:

$$A_r = \left(\frac{A_r}{A_d}\right) A_d \quad (12)$$

where the ratio $A_r/A_d = \begin{cases} 0.67 \text{ for a squatting person,} \\ 0.70 \text{ for a sitting person,} \\ 0.77 \text{ for a standing person.} \end{cases}$

The mean radiant temperature ($\bar{T}_r$) can be evaluated using the empirical formula [33]:

$$\bar{T}_r = [(T_g)^4 + 2.5 \times 10^8 v_a{}^{0.6}(T_g - T_a)]^{1/4} \quad (13)$$

where T_g is the inside globe temperature (found in the ISO 7726).

Beshir and Ramsey [63] proposed a more simplified formula as:

$$\bar{T}_r = T_g + 1.8\sqrt{v_a}(T_g - T_a) \quad (14)$$

4.2.4. Moisture effect

Moisture effect on heat transfer is due to the dampness of some organs like lips, eyes or respiratory tract. At that level the transfers are less but perceptible. At the level of the skin, regulatory-sweating creates evaporative heat loss (E) given by [33]:

$$E = h_e(P_{a_{H_2O}} - P_{sk_{H_2O}}) A_e F_{pcl} \quad (15)$$

where h_e ($h_e = kh_c$ with k = 2.2 K/Torr = 16.7 K/kPa) is the evaporative heat transfer coefficient at the surface, $P_{a_{H_2O}}$ is the water vapour pressure in ambient air, $P_{sk_{H_2O}}$ is the water vapour pressure in saturated air at T_{sk}, A_e is the evaporative surface, F_{pcl} is the clothing permeability factor. The evaporative surface can be rewritten as: $A_e = (A_e/A_d)A_d = wA_d$. The ratio $w = A_e/A_d$ is known as the skin wettedness. The skin wettedness is a rationally derived physiological index defined as the ratio of the actual sweating rate to the maximum rate of sweating that would occur if the skin was completely wet, and skin temperature was incorporated into such a model to indicate the sensation of "comfort and discomfort" caused by perspiration. Skin wettedness is important in determining evaporative heat loss. It ranges from about 0.06 caused by the evaporative heat loss due to moisture diffusion through the skin alone (i.e. with no regulatory sweating) for normal conditions, to 1 when theoretically a skin surface is totally wet with perspiration, a condition that occurs rarely in practice. For large values of the possible maximum evaporative heat loss or long exposures to low humidity, the value of w may drop to as low as 0.02, since dehydration of outer skin layers alters their diffusive characteristics [3]. Fig. 4 presents the relationship between the wettedness and thermal constraint [33].

4.2.5. Metabolic effect

The production of the metabolic heat is the reflection of the cellular life that results from the consumption of oxygen (O_2) and rejection of carbon dioxide (CO_2). In the steady state, the quantity of the metabolic heat produced is deduced from the consumption of oxygen, calculated from the rate of ventilated air and the difference of concentration between the inspired and expired air [33].

In normal conditions, when a body is at rest and in nutritional equilibrium, the global respiratory ratio is $m_{CO_2}/m_{O_2} = 0.83$. For that value, the consumption of a litre of oxygen per hour produces approximately 5.57 W. Since a person at rest consumes approximately 0.31 l of oxygen (18.6 l/h), he produces approximately 104 W (58 W/m^2 = 1met for a standard person of 1.8 m^2): this is

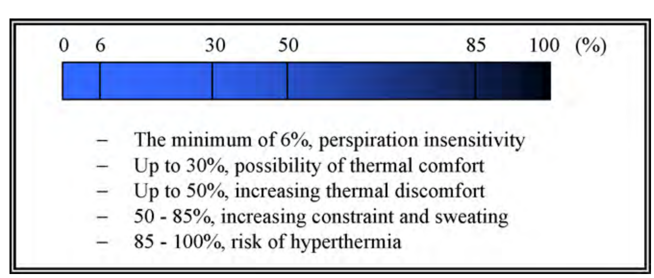

Fig. 4. Wettedness and thermal constraint [33].

Table 4
Metabolic heat of posture [33].

	Posture			
	Sitting	Squatting/ crouching	Standing up	Standing up bent or perched
M_p (W/m^2)	10	20	25	30

the metabolic heat at rest (M_r) (a sitting person, thermoneutrality in clothing, no external influence).

In the condition of activity (in the office, at school, at home), the quantity of the metabolic heat produced is not deduced from the consumption of oxygen (difficult to be evaluated), but from the type of the activity practiced. An approximation consists to use the formula [33]:

$$M = M_b + M_p + M_a \quad (16)$$

where M is the metabolic heat produced, M_b = 45 W/m^2 is the basal or minimal metabolic heat (nude body lying down in the case of thermoneutrality), M_p is the metabolic heat of posture whose values are given in Table 4, M_a is the metabolic heat of activity whose values are given in Table 5.

Metabolism as any other chemical reaction is accelerated with increasing temperature as long as the higher temperature does not lead to the inhibition of the metabolic process. The temperature dependence can be written as [64]:

$$M = 1.1^{\Delta T} M_b \quad (17)$$

where M_b is the basal metabolic heat production rate and ΔT is the temperature increase.

Goto et al. [65] noted that "activity level is probably one of the least well-described parameters of all the parameters that affect thermal sensation, comfort and temperature preferences indoors".

4.2.6. Clothing effect

Clothing is an important factor in achieving thermal comfort at different temperatures.

4.2.6.1. Evaluation of the clothing area factor

Establishing the insulating properties of clothing is a time-consuming and detailed process, that is usually conducted in laboratory experiments devoted to this purpose [19].

4.2.6.1.1. Experimental evaluation

Good results are obtained using a thermal manikin (Fig. 5) which is a flux meter at the human scale and permits to evaluate the heat dissipated. The manikin is constituted of:

- 35 zones, each containing a platinum probe (Pt 100) (≈60 cm) sticked on the resin;
- a heating resistance that permits to have a desired body's temperature;
- a computer connected to the manikin and that permits to regulate the temperature or heat flux.

Table 5
Metabolic heat of activity [33].

	Work (W/m^2)		
	Light	Mean	Heavy
Hands	10–22	22–34	34–46
One arm	25–45	45–65	65–85
Two arms	55–75	75–95	95–115
Body	95–155	155–230	230–330

Fig. 5. Thermal manikin [33].

The heat flux is evaluated for a given standard condition of the nude manikin. This permits to determine the air insulation (I_a). Then, the manikin is clothed with clothes tests, the total insulation (I_t) is therefore evaluated. From these results, the clothing insulation (I_{cl}) is obtained.

4.2.6.1.2. Theoretical approximations

As it is not practical to directly measure clothing insulation in most thermal comfort studies, researchers generally estimate these values, using tables that have been developed from clothing insulation studies [19]. Clothing insulation tables are constructed from laboratory studies, usually using thermal manikins in conditions of still air. The clothing insulation can be evaluated from the ISO 9920 standard using the "clo" of each clothing garment, whose values are obtained from tables [2,18]. The insulation of the clothing ensemble is determined using the Olsen's 1985 summation formula [5]:

$$I_{cl} = \sum_i I_{clu,i} \quad (18)$$

where I_{cl} is the insulation of the entire ensemble (in clo) and $I_{clu,i}$ represents the effective insulation of the clothing garment i.

The clothing area factor taking into account the effects of radiation and convection is evaluated from the formula [33]:

$$F_{cl} = \frac{1}{1 + 0.155(h_c + h_r)I_{cl}} \quad (19)$$

where I_{cl} is in clo (1 clo = 0.155 K m^2/W) and

$$h_c + h_r = \frac{1}{I_a} \quad (20)$$

It can also be evaluated from the formula [33]:

$$F_{cl} = \frac{A_{d,cl}}{A_d} = 1 + 0.31 I_{cl} \quad (21)$$

where A_d is the DuBois surface area of nude body and $A_{d,cl}$ is the surface area of clothed body.

The clothing permeability factor F_{pcl} is also given by [33]:

$$F_{pcl} = \frac{1}{1 + 0.155 h_e (I_{cl}/i_m)} \tag{22}$$

where h_e is the evaporative heat transfer coefficient and i_m the permeability factor of the clothe.

In general, four types of insulation are found in the literature [33]:

- the total insulation; including air and clothing,
- the effective insulation; only clothing is concerned,
- the intrinsic or basic insulation; increasing exchange area is taken into consideration,
- the resultant insulation; the blowing and pumping effects due to human activities as well as the resistance due to sweating are taken into consideration.

4.3. Heat exchange between human body and its environments

Thermal comfort is strongly related to the thermal balance of the body. This balance is influenced by:

- environmental parameters like: air temperature and mean radiant temperature, relative air velocity and relative humidity,
- individual parameters like: activity level or metabolic rate,
- and clothing thermal resistance.

Fanger [10], Hardy [66], Gagge and Nishi [27], and Gagge and Hardy [67] gave quantitative information on calculating the heat exchange between people and their environments. The mathematical descriptions of an energy balance equation and the statements for various terms of the heat exchange used in the heat balance equation are detailed in the ASHRAE Handbook of Fundamentals [68].

4.3.1. Energy balance of human body

The thermal balance begins with two necessary initial conditions to maintain the thermal comfort [31]: (i) A neutral thermal sensation must be obtained from the combination of skin temperature and full body. (ii) In a full body energy balance, the amount of heat produced by metabolism must be equal to that lost to the atmosphere (steady state).

A human body can be considered as consisting of two concentric thermal compartments: the skin and the core [26]. Fig. 6 shows the thermal interaction between a human body and its environment. The total metabolic rate of work (M) produced within the body is the metabolic rate required for the person's activity plus the metabolic rate required for shivering. A portion of the body's energy production may be expended as the external work done by muscles (W). The net heat production in the human body ($M - W$) is either stored (S), causing the body's temperature to rise, or dissipated to the environment through skin surface (q_{sk}) and respiratory tract (q_{res}). Therefore, the heat balance for a human body is [3]:

$$\begin{aligned} M - W &= q_{sk} + q_{res} + S \\ &= (C + R + E_{sk}) + (C_{res} + E_{res}) + (S_{sk} + S_{cr}) \end{aligned} \tag{23}$$

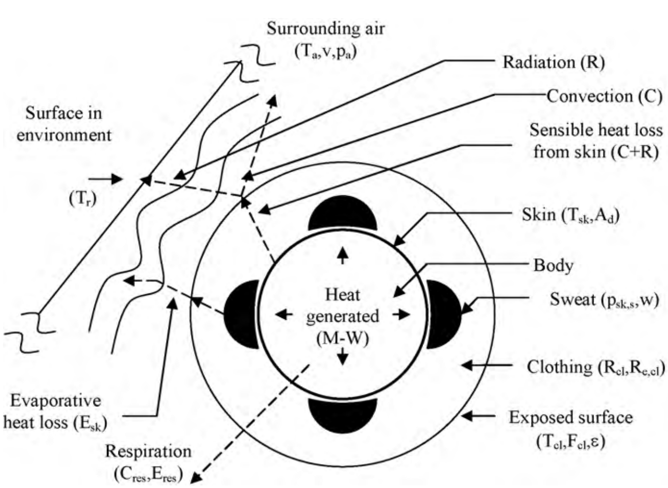

Fig. 6. The thermal interaction between a human body and its environment [3].

The rate of heat storage in the body can be written separately for each compartment in terms of thermal capacity and change rate of temperature in each compartment as follows [3]:

$$S_{cr} = \frac{(1-\alpha_{sk}) m c_{p,b}}{A_d} \frac{dT_{cr}}{dt}, \qquad S_{sk} = \frac{\alpha_{sk} m c_{p,b}}{A_d} \frac{dT_{sk}}{dt} \tag{24}$$

The thermal balance is totally accepted and followed by ISO 7730 for the study of the comfort conditions, regardless of the climatic region.

4.3.2. Thermal exchanges between a human body and its environment

4.3.2.1. Sensible heat loss from the skin

Sensible heat exchange from skin surface to a surrounding environment must pass through clothing. Both convective and radiative heat losses from the outer surface of a clothed body can be expressed in terms of a heat transfer coefficient and the difference between the mean temperature of the outer surface of the clothed body and an appropriate environmental temperature [3]:

$$C = F_{cl} h_c (T_{cl} - T_a), \qquad R = F_{cl} h_r (T_{cl} - \bar{T}) \tag{25}$$

The coefficients h_c and h_r are both evaluated at the clothing surface. Eqs. (24) and (25) are commonly combined to describe the total sensible heat exchange by these two heat exchange mechanisms in terms of an operative temperature T_O ("the uniform temperature of an imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual non-uniform environment" [19]), and a combined heat transfer coefficient h as [3]:

$$C + R = F_{cl} h_c (T_{cl} - T_O) \tag{26}$$

where $T_O = (h_r \bar{T}_r + h_c T_a)/(h_r + h_c)$ with $h = h_r + h_c$. Based on the above relation, the operative temperature T_O, can be defined as the average of the mean radiant and the ambient air temperatures, weighted by their respective heat transfer coefficients.

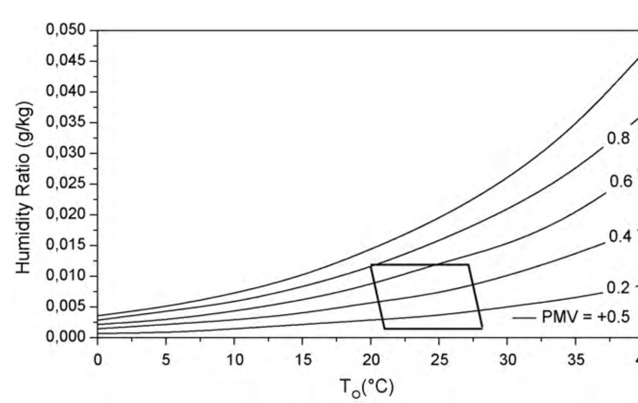

Fig. 7. Comfort zone [31].

On other cases, for occupants engaged in near sedentary physical activity (with metabolic rates between 1.0 met and 1.3 met), not in direct sunlight, and not exposed to air velocities greater than 0.20 m/s the relationship can be approximated with acceptable accuracy by [31]:

$$T_O = \frac{\bar{T}_r + T_a}{2} \tag{27}$$

It can be defined as a comfort zone for some given values of humidity, air speed, metabolic rate and insulation produced by clothing, in terms of operating temperature or in terms of the combination of air temperature and the average radiant temperature. This area is represented in Fig. 7 for air speeds not greater than 0.20 m/s [31].

The actual transport of sensible heat passing through clothing involves conduction, convection, and radiation. It is usually convenient to combine these into a single thermal resistance of clothing R_{cl} [3]:

$$C + R = \frac{\bar{T}_{sk} - T_{cl}}{R_{cl}} \tag{28}$$

Combining Eqs. (26) and (28) to eliminate T_{cl} one can obtain [3]:

$$C + R = \frac{\bar{T}_{sk} - T_O}{R_{cl} + 1/(F_{cl}h)} \tag{29}$$

4.3.2.2. Evaporative heat loss from the skin

Evaporative heat loss from skin E_{sk}, depends on the amount of moisture on skin and the difference between the water vapour pressure at skin surface and that in the ambient environment [3]:

$$E_{sk} = \frac{w(p_{sk,s} - p_a)}{R_{e,cl} + 1/(F_{cl}h_e)} \tag{30}$$

Evaporative heat loss from skin is a combination of the evaporation of sweat secreted due to thermoregulatory control mechanisms E_{rsw}, and the natural diffusion of water through skin E_{dif} [3]:

$$E_{sk} = E_{rsw} + E_{dif} \tag{31}$$

Theoretically, the maximum possible evaporative heat loss from a skin surface, E_{max}, occurs when the skin surface is completely wet (i.e. the skin wettedness, w is equal to 1). The skin wettedness can therefore be considered as the ratio of the actual evaporative heat loss to the maximum possible evaporative heat loss, E_{max} [3]:

$$\frac{E_{sk}}{E_{\max}} \tag{32}$$

4.3.2.3. Respiratory losses

Respiratory heat loss q_{res}, is often expressed in terms of sensible heat loss C_{res}, and latent heat loss E_{res}. Sensible loss (C_{res}) and latent loss (E_{res}) due to respiration are relatively small and can be estimated, respectively, by the following equations [69]:

$$C_{res} = 0.0014M(34 - T_a), \qquad E_{res} = 0.0173M(5.87 - p_a) \tag{33}$$

Table 6
Methods to calculate general thermal comfort indexes [31].

Method 1	Air velocity	Air temperature	Mean radiant temperature	Humidity
	Measure	Measure	Calculate	Measure
Method 2	Air velocity Measure	Operative temperature Measure		Humidity Measure
Method 3	Equivalent temperature Measure			Humidity Measure
Method 4	Air velocity Measure	Effective temperature Calculate		

4.4. Methods to calculate general thermal comfort indexes

After studying the equations that define the heat balance of a person, we can deduce the need of sampling the instantaneous evolution of operative temperature, air velocity and relative humidity. To collect the thermal comfort data, one can employ transducers as that employed by the thermal comfort module of Innova Airtech 1221 [31]. The parameters that must be measured directly or calculated are indicated in Table 6. The term "Equivalent Temperature", which is often used instead of Dry Heat Loss. This equivalent temperature can be calculated from the dry heat loss and, by definition is the uniform temperature of a radiant black enclosure with zero air velocity in which an occupant would have the same dry heat loss as the actual non-uniform environment [31].

5. Thermal comfort for sleeping environments

From its definition, comfort is not a state condition, but rather a state of mind. Therefore, "thermal comfort" does not make too much sense for people during sleep. Based on the fact that a person sleeping in an air conditioned environment can be considered as being in a steady state and close to thermally neutral, Lin and Deng [3] introduced modifications to Fanger's comfort model to develop a comfort equation applicable to sleeping thermal environments.

5.1. Assumptions and modifications adopted for sleeping environments

Above equations are normally applicable to sedentary or near sedentary physical activity levels, e.g. typical office work. Assumptions and modifications are needed if these equations are to become applicable to sleeping environments. For a sleeping person in a reclining posture with a specific bedding system which consists of a bed and mattress, bedding and sleepwear, it is assumed that the sleeping person is immobile during the whole period of sleep, therefore [3]:

$$M = 40\ \mathrm{W/m^2}, \qquad W = 0\ \mathrm{W/m^2} \tag{34}$$

For a bedding system rather than clothing, the intrinsic clothing resistance R_{cl}, in Eq. (28), cannot be determined because the clothing area factor, F_{cl}, is meaningless when a body is lying on a bed. Therefore, Eq. (28) can be rearranged in terms of the total thermal resistance (R_t) provided by a bed, pillow, bedding, sleepwear and the air layer surrounding a human body so that the intrinsic clothing resistance R_{cl}, and the clothing area factor F_{cl}, may be substituted by R_t, as follows [3]:

$$R_t = R_{cl} + \frac{1}{hF_{cl}} = R_{cl} + \frac{R_a}{F_{cl}} \tag{35}$$

$$C + R = \frac{\bar{T}_{sk} - T_O}{R_t} \tag{36}$$

Similarly, Eq. (30) can be rewritten to become [3]:

$$E_{sk} = w\frac{p_{sk,s} - p_a}{R_{e,t}} \tag{37}$$

According to the Lewis relation [3]:

$$i_m L_R = \frac{R_t}{R_{e,t}} \tag{38}$$

Combining Eqs. (37) and (38) to eliminate $R_{e,t}$ lead to [3]:

$$E_{sk} = i_m L_R w(p_{sk,s} - p_a) = \frac{R_t}{R_{e,t}} \tag{39}$$

The Lewis ratio (L_R) is approximately equals to 16.5 K/kPa at typical indoor conditions [3]. Since the purpose of the thermoregulatory system in a human body is to maintain an essentially constant internal body temperature, it can be assumed that for long exposure (for periods not less than 15 min as specified in ASHRAE Standard 55) to a constant sleeping thermal environment with a constant $(M - W)$, a heat balance will exist for the human body (steady state condition). In other words, there will be no significant heat storage within the body. Therefore, Eq. (23) can be changed to [3]:

$$S_{cr} = \frac{(1-\alpha_{sk})mc_{p,b}}{A_d}\frac{dT_{cr}}{dt} = 0, \qquad S_{sk} = \frac{\alpha_{sk}mc_{p,b}}{A_d}\frac{dT_{sk}}{dt} = 0 \tag{40}$$

Based on all the assumptions and modifications introduced above for sleeping environments, Eq. (23) can be rewritten to become [3]:

$$40 = \frac{\bar{T}_{sk} - T_O}{R_t} + \frac{i_m L_R w(p_{sk,s} - p_a)}{R_t} + 0.056(34 - T_a) + 0.692(5.87 - p_a) \tag{41}$$

5.2. Conditions for thermal comfort in sleeping environments

A basic condition for thermal comfort in sleeping environments is that thermal neutrality is achieved during sleep [3]. Obviously the first requirement for thermal comfort in sleeping environments is that the heat balance Eq. (41) be satisfied. However, heat balance alone is not sufficient to achieve thermal comfort [3]. In a wide range of environmental conditions where heat balance can be obtained, thermal comfort may be achieved only within a narrow range of the conditions. The following linear regression equations indicate values of T_{sk} and E_{rsw} that provide thermal comfort, which were proposed as the second and third conditions for optimal thermal comfort by Fanger [10]:

$$T_{sk,req} = 35.7 - 0.0275(M - W), \tag{42}$$

$$E_{rsw,req} = 0.42(M - W - 58.15) \tag{43}$$

It can be seen from the two equations that in a state of physiological thermal neutrality during sedentary (M = 58.15 W/m^2, W = 0), the mean skin temperature is around 34.1 °C and there is no regulation of body temperature by sweating (i.e. sweating does not occur). The skin temperature necessary for comfort falls, and moderate sweating takes place at a higher activity level. However, in a state of thermal neutrality for a sleeping person whose activity level is lower than sedentary (M = 40 W/m^2, W = 0), the mean skin temperature would increase and sweating would either not occur ($E_{rsw,req}$ is meaningless when less than zero). Therefore, the second and third conditions for thermal comfort in a sleeping environment may be changed to:

$$\bar{T}_{sk,req} = 35.7 - 0.0275(M - W) = 34.6\,^\circ\mathrm{C} \tag{44}$$

$$E_{rsw,req} = 0 \tag{45}$$

With no regulatory sweating for normal conditions, the skin wettedness (w) equals to 0.06, caused by E_{dif} alone [12]:

$$w = 0.06 \tag{46}$$

5.3. Comfort equation for sleeping environments

Combining the three conditions, i.e. Eqs. (41), (44) and (46) for thermal comfort in a sleeping environment to obtain [3]:

$$40 = \frac{34.6 - T_O}{R_t} + \frac{0.06 i_m L_R(p_{sk,s} - p_a)}{R_t} + 0.056(34 - T_a) + 0.692(5.87 - p_a) \tag{47}$$

In order to solve Eq. (47), some of its parameters such as the heat transfer coefficients h_r and, h_c, and the permeation efficiency i_m, need to be determined. ASHRAE Handbook of Fundamentals [68] provides the necessary data and methods used to calculate these parameters. The radiant heat transfer coefficient h_r is nearly constant for typical indoor temperatures and a value of 4.7 W/m^2 K is sufficient for most calculations. The convective heat coefficient h_c, for reclining persons can be calculated by [69]:

$$h_c = 2.7 + 8.7v^{0.67} \quad \text{for } 0.15 < v < 1.5 \tag{48}$$

$$h_c = 5.1 \quad \text{for } 0 < v < 0.15 \tag{49}$$

Quantitative values of h_c are important, not only in estimating convection loss, but in evaluating operative temperature T_o. An experimental study by McCullough et al. [70] has suggested that the permeation efficiency i_m of ensembles worn indoors generally fell in the range of 0.3–0.5, and that assuming i_m = 0.38 is reasonably accurate for most applications although i_m for a given clothing ensemble is a function of the environment as well as the clothing properties. Since the properties of bedding are likely to be similar to that of clothing, such a value (i_m = 0.38) can also be adopted for a bedding system.

According to the properties of saturated water/steam, the water vapour partial pressure in saturated air $p_{sk,s}$, when T_{sk} = 34.6 °C is [3]:

$$p_{sk;s} = 5.52\,\mathrm{kPa} \tag{50}$$

Using above equations and i_m = 0.38, L_R = 16.5 K/kPa, h_r = 4.7 W/m^2 K, a comfort equation for sleeping environments, which combines both environmental and personal variables to produce a thermal neutral sensation, may be derived from Eq. (47) as follows [3]:

$$40 = \frac{1}{R_t}\left[\left(34.6 - \frac{4.7\bar{T}_r + h_c T_a}{4.7 + h_c}\right) + 0.3762(5.52 - p_a)\right] + 0.056(34 - T_a) + 0.692(5.87 - p_a) \tag{51}$$

R_t is the total thermal resistance for a bedding system. Obviously the satisfaction of the comfort equation (51) means that the three comfort conditions are met at the same time since it combines the three equations for thermal comfort in sleeping environments. There are five variables, R_t, $\bar{T}_r$, T_a, p_a, and h_c in Eq. (51). The convective heat transfer coefficient h_c, is the function of air velocity v. Therefore, four variables (i.e. $\bar{T}_r$, T_a, p_a, and v) are thermal environmental variables. The total thermal resistance R_t, is the function of a number of variables such as bedding, sleepwear, bed and mattress, the percentage coverage of body surface area by bedding and bed, air velocity, direction of airflow, and posture, etc.

5.4. PMV and PPD for sleeping environments

Fanger's model is also applicable to sleeping environments although the relatively low activity level (sleep) was not included in the experiments. PMV for a sleeping environment can be evaluated by [3]:

$$PMV = 0.0998\left\{40 - \frac{1}{R_t}\left[\left(34.6 - \frac{4.7\bar{T}_r + h_c.T_a}{4.7 + h_c}\right) + 0.3762(5.52 - p_a)\right]\right\} - 0.0998[0.056(34 - T_a) + 0.692(5.87 - p_a)] \quad (52)$$

The PPD for a sleeping environment can then be determined from Eq. (3).

6. Conclusion

This paper presents a literature reviews on thermal comfort, both rational and adaptive approaches are presented. In has been seen that each approach has its potentialities and limits. With people involved in near-sedentary activity and steady-state conditions, rational approach easily and reasonably produces accurate predictions of occupant thermal sensation, but it is not always a good predictor of actual thermal sensation, particularly in field study settings. Adaptive approach allows for analyses of other factors than those that can be simulated in chambers, as the subjects provide responses in their everyday habitats, wearing their everyday clothing and behaving without any additional restrictions; field studies are therefore to be encouraged in addition to laboratories experiments. However, one should be very careful when interpreting the results of thermal comfort campaigns. The actual standards help but should not be considered as absolute references. In fact, the individual state of mind that expresses satisfaction with the thermal environment is too diverse for that when small groups are considered.

An overview of the human body thermoregulatory system is also presented as well as the mathematical modelling of heat exchanged between human body and its environment in the situations of awaked and sleeping people. It could be noted from Fanger's studies that man's thermoregulatory system is quite effective and creates heat balance within wide limits of the environmental variables, even if comfort does not exist. The only physiological processes influencing heat balance in this context are sweat rate and mean skin temperature, these processes are a function of activity level. Fanger's model, primarily developed for indoor environments can also be used for outdoor spaces and the mathematical model of thermal exchanges between human body and its environment in the situation of awaked people can also be used in the situation of sleep, provided some modifications.

But, even there is much documented material worldwide concerning human thermal comfort from the physiological, adaptive and social convention paradigms, throughout sub-Saharan Africa and particularly in tropical regions, there have been few literature reports on occupants' comfort and residential thermal environments. ASHRAE's series of state-of-the-art, fully compatible field experiments across a variety of climate zones including temperate, hot-humid and cold exist, but different climatic regions such as the tropics, may require different levels of comfort parameters mandated in the standards.

As a part of a project whose overall objective is to study thermal comfort in tropical sub-Saharan Africa regions, this literature review is a contribution for the purpose. Such studies would enable the development and recommendations of comfort standards.

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