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THERMAL COMFORT SIMULATION FOR NON UNIFORM BOUNDARY CONDITIONS

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Abstract. *The objective of the present work is to simulate thermal comfort for humans exposed to dynamic and non-uniform boundary conditions according to a scale similar to that established by ASRHAE (1997). Fiala's (1998) model equations and properties for passive and active systems were used and the model was integrated numerically using the finite difference method. According to the Fiala model the human body is compartmentalized in ten different parts, each consisting of up to seven tissues (brain, bone, lung, core, muscle, fat, skin). The parts were assigned specified radiation and convection coefficients, and were thermally coupled by perfusion.*
The results are organized in terms of three case studies. The first case study concerns thermal comfort under two metabolic rates in rooms with a positive displacement air conditioning system. For the same room temperature, two different temperature and humidity gradients, and uniform air temperature were compared. The second case simulated thermal comfort of passengers in Rio de Janeiro subway trains. The passenger was initially exposed to 35^o C air temperature and 55% relative humidity, followed by exposure to two different air temperature and humidity conditions for the next 30 minutes. The third case corresponds to thermal comfort of workers under high metabolic rates (3.5 met) in a hangar of the Macapá thermoelectric unit. Indoor air temperature was around 40^o C, 30% relative humidity. Workers were initially acclimated to outdoor air conditions (~30^o C, 78%). The simulation considered three air velocities and 30 minutes exposure.
In all cases, the extended multi-compartment model proved to give reasonable predictions for human thermal comfort sensations, as well as providing guidance for improved air conditioning design.

Keywords: *thermal comfort, indoor air distribution, dynamic thermal conditions.*

1. Introduction

Considering the human body as a thermodynamic system, production of mechanical work is associated with heat generation owing to exothermic metabolic reactions. During equilibrium (homeostasis), the body core temperature is maintained around 37^{o}C by the autonomous thermoregulatory system. Thermoregulation balances the amount of heat produced and lost by modifying the heat transfer through the skin. This is an essential but not unique condition for thermal comfort (Butera, 1998). Since thermal comfort and productivity are directly related, it is desired that the indoor climate is adjusted to certain optimum conditions. Even when the indoor air temperature and humidity are set to the standard appropriate values (~24^{o}C and 50% relative humidity), not necessarily all occupants feel comfortable. There are additional factors that affect the occupants' feeling of comfort, for example, forced air distribution (Athayde, 2001).

Some thermal comfort models were developed in order to better correlate our sensations with the properties of ambient space. The two most frequently used are the Fanger's model, and Gagge's two-node model, both briefly presented in ASHRAE (1997).

The first one calculates the energy balance considering heat production (or metabolic rate), heat loss by perspiration, heat loss by respiration, conduction through clothing, and heat loss by radiation and convection from outer surface (cloth or skin) to air. The body is considered as a single unit, which is in thermal equilibrium with the surrounding air (steady state). As a measure for thermal sensation, Fanger (1972) related his thermal comfort equation to an index. The Predicted Mean Vote (PMV) ranges from – 3 to + 3 according to the ASHRAE scale:

-3	-2	-1	0	+1	+2	+3
Cold	Cool	Slightly cool	Neutral	Slightly warm	Warm	Hot

The Gagge (two-node) model considers the body as consisting of two compartments: skin (about 10% of the body mass) and core. The temperature is assumed to be uniform in each of those compartments. The unsteady response of the system is considered simulating thermoregulatory responses (ASHRAE, 1997). Thermoregulation entails the following four responses depending on the exposure. When first exposed to cold ambient air, the skin thermal resistance is increased by vasoconstriction diminishing blood perfusion. If this is not sufficient to combat heat loss, the metabolic rate of the body is increased by elevating muscular tension, thus activating shivering. On the other hand, when the ambient is hot, vasodilatation occurs expanding peripheral blood vessels and diminishing skin thermal resistance. If this is not sufficient to balance the metabolic rate, the skin starts to sweat and this serves to increase heat loss via evaporation (Butera, 1998).

Traditional thermal comfort models consider exposure to uniform ambient conditions. The recent technological development in terms of central air-conditioning of spaces such as large offices, theaters and public buildings, is demanding a more careful analysis of the dynamic and psychrometric conditions of the circulating air in thermal controlled ambient.

A new trend is underfloor air distribution, which improves air quality and reconfiguring flexibility, provides individual adjustments and reduces energy costs (Bauman and Webster, 2001). Positive displacement flow works similarly, and has the same advantages. The air is insufflated in higher temperatures and lower velocities, thus reducing turbulence, compared to air supplied at ceiling level. The air temperature is more uniform when the colder air is supplied form the ceiling, being mixed due to buoyancy effects. System designers are increasingly concerned about the temperature gradients that space occupants will be exposed to (Athayde, 2001; Bauman and Webster, 2001).

An extended model that can be used for both dynamic situations and non-uniform boundary conditions is Fiala's (1998) model. According to this model, the body is composed of ten different parts (in fact, fifteen parts when doubling the units for hands, arms, shoulders, legs and feet), modeled by spheres or cylinders. Each part is multi-layered in terms of distinct tissues, and can be exposed to up to three boundary conditions. The boundary conditions include air temperature, air humidity, radiative and convective heat exchange, and clothing characteristics. The different parts are coupled by blood perfusion. Physiological responses are accommodated in terms of fitted functions of core and skin temperature. The model was initially applied for the "average human" (Fiala, Lomas and Stohrer, 1999).

There is a large variability among humans in terms of height, body mass, metabolic rate, clothing and habits. The gender, age and health also interfere with body metabolism and balance. Additionally, it is hard for the human being to objectively separate thermal sensation vote from other comfort votes, such as noise, ergonometric condition or other external influences. However, it is necessary to group average anthropometric and physical conditioning characteristics in broad categories in order to put the thermal comfort models to practical use.

2. Model

Fiala's model (Fiala, 1998) was deemed as appropriate to accommodate unsteady and non-uniform ambient thermal conditions. It is briefly described in this section along with a few modifications that were made.

Physiological responses are considered with respect to basal values (set-points, denoted by the subscript "0"), for specified ideal comfort conditions. These basal values were obtained using 30°C air temperature, 50% relative humidity, 0.05 m/s air velocity, resting state (0.8 met) and no clothes. Deviations in core and skin temperatures from these values activate vasodilatation, vasoconstriction, sweating or shivering. The metabolic rate, clothing, air temperature and humidity are the input parameters of the model. As output, we get the estimated Dynamic Thermal Sensation (DTS) vote, a function of the physiological responses, which is defined similarly to ASHRAE's scale.

The heat transfer problem is transient but it is limited to one dimension, since heat flux in radial direction is expected to be greater than in the axial or azimuthal directions. The governing equation is quite similar to Pennes "bioheat equation":

$$\rho \cdot c \cdot \frac{\partial T}{\partial t} = k\left(\frac{\partial^2 T}{\partial r^2} + \frac{\omega}{r}\frac{\partial T}{\partial r}\right) + q_m + \rho_{bl} w_{bl} c_{bl} (T_{bla} - T)$$

where:
r – radius [m]
ρ - density [kg/m^3]
k – thermal conductivity [W/(m K)]
T – temperature [°C]
c – specific heat [J/(kg K)]
subscript bl - blood property
ω =1 for cylindrical coordinates and ω = 2 spherical coordinates.

The metabolic heat production q_m [W/m^3] is a function of the basal local values of temperature and metabolic rate, deviations due to additional work, and shivering. According to the input value for physical activity, our metabolic rate and its spatial distribution differs from the basal value. The metabolic rate can be distributed in two ways, depending on simulating either seated or standing person, with more heat production for superior or inferior limbs, respectively. The heat lost by respiration is reduced by different fractions from metabolic rates for the muscle tissues of face and neck, and lung. The heat loss by respiration is estimated using the hole body metabolic rate, air temperature and air humidity values.

The perfusion rate w_{bl} is a function of the basal perfusion rate $w_{bl,0}$, the skin temperature of the part, hypothalamus temperature and global skin temperature. When w_{bl} increases, the heat transfer between blood and tissues rises.

The temperature T_{bla} corresponds to the arterial blood temperature defined in Pennes' model. In the extended model, blood temperature is not assumed to be the same for the whole body. There's a different T_{bla} value for each part of the body, calculated using all blood perfusion rates and local temperature values of the body, and separate heat exchange coefficient for each part.

The heat exchanged with the surrounding air due to convective heat transfer for each body section is a function of the clothing, local skin temperature, air temperature and air velocity. The radiative heat transfer depends on the clothing, skin temperature, surrounding surfaces temperature and surrounding heat sources. The sensible heat exchange due to perspiration and transpiration depends on skin temperature, clothing and air relative humidity as per ASHRAE (1997) formulation.

3. Numerical Model

Since the model system of equations did not have analytical solution, it was necessary to integrate it numerically. A finite-difference spatial discretization, using the Crank–Nicholson scheme was chosen because it is numerically stable, the differential equations have one spatial coordinate and constant coefficients. The discretized equations were derived using *Mathematica 3.0* (Wolfram Research) and the numerical solution was obtained using *Fortran 6.1*. Ten matrices were created, one for each body compartment containing all the nodal points. The detailed discretized equations for generic, central, interface and boundary points are given in Falkenberg (2001), together with discretization characteristics. Material properties and generic coefficients were borrowed from Fiala (1998). Since all parameters are coupled and the problem is non-linear, it was solved by fixed point iteration for each time instant until convergence was achieved, before the next time step could be evaluated. The model was coded to give as output the amounts of each mode of heat loss (evaporative, convective and radiative, and by respiration), mean skin temperature, hypothalamus temperature and physiological response coefficients.

4. Case studies

4.1. Positive displacement air-conditioning systems

When positive displacement or underfloor air distribution is used, the air surrounding the occupant is not completely mixed, but contains temperature and humidity gradients. This type of air-conditioning system is expected to improve indoor air quality with reduced cost (Bauman and Webster, 2001), but its thermal comfort characteristics remain to be studied. Usually, for office activities, adults metabolic rates vary from 1 met (reading, seated) to 1.2 met (filing, seated) (ASHRAE, 1997). The unit "met" corresponds to 58.2 W/m^2, and the average human has a total skin area equal to 1.86 m^2. In other words, an office worker when seated and occupied with an office activity that demands low energy consumption, needs to eliminate about 58.2 x 1.86 = 108 W to achieve thermal equilibrium, and 58.2 x 1.2 x 1.86 = 130 W when the task demands a little bit more effort.

The clothing items considered were long sleeved shirt, trousers, socks and shoes. The chair thermal resistance is also considered. The total thermal resistance was 0.7 clo (1 clo = 0.155 m^2 K/W), calculated according to Fiala (1998).

To compare expected comfort sensations due to different indoor air conditions, three air distributions were considered:

(a) uniform air temperature and distribution: 25^{o}C, 50% RH (RH = relative humidity)
(b) temperature and humidity gradient: 22^{o}C, 67% RH (feet) to 26^{o}C, 55% RH (at head)
(c) temperature and humidity gradient: 18^{o}C, 80% RH (feet) to 26^{o}C, 55% RH (at head)

Item (a**)** corresponds to ceiling air distribution, and the other two correspond to underfloor or positive displacement distribution. For each temperature distribution, two metabolic rates (1.0 met and 1.2 met) and three air velocities 0.05 m/s, 0.10 m/s and 0.15 m/s were tested. Results are presented in Figures 1 - 6. The Predicted Mean Vote (PMV), calculated according to ASHRAE (1997), is also presented. Another set of calculations is performed considering uniform convective (3.1 W/(m^2 ^{o}C)) and radiative (4.6 W/(m^2 ^{o}C)) heat transfer coefficients.

The respiratory heat losses in all cases did not depend on the air distribution, but on the metabolic rate: 12 W for 1.0 met, and 15 W for 1.2 met. This result is justified owing to similar air temperature and humidity for face exposure in all three distributions. The percentage of latent heat loss over total heat loss, when simulating 1.2 met activity, was ~5% lower with 4^o C air temperature gradient than with uniform air distribution for all three air velocities. The latent heat rate was from 6% (0.15 m/s air velocity) to 12% (0.05 m/s air velocity) lower for the 8^o C gradient. Reducing heat loss is desired, mainly in humid areas, which reduces the energetic costs of the air-conditioning systems. In all cases, the initial condition was at the basal temperature, justifying the initial discomfort. As expected, uniform heat transfer coefficients resulted in quite different DTS values than using coefficients calculated with iterative procedure for each body part.

The comfort zone for ASHRAE scale is – 0.5 to + 0.5 (ASHRAE, 1997). For 1.2 met activities, the DTS vote is expected to be in the comfort range, except for higher air velocity and temperature gradients. For low activity, the comfort zone is achieved only for low velocity and temperature gradient.

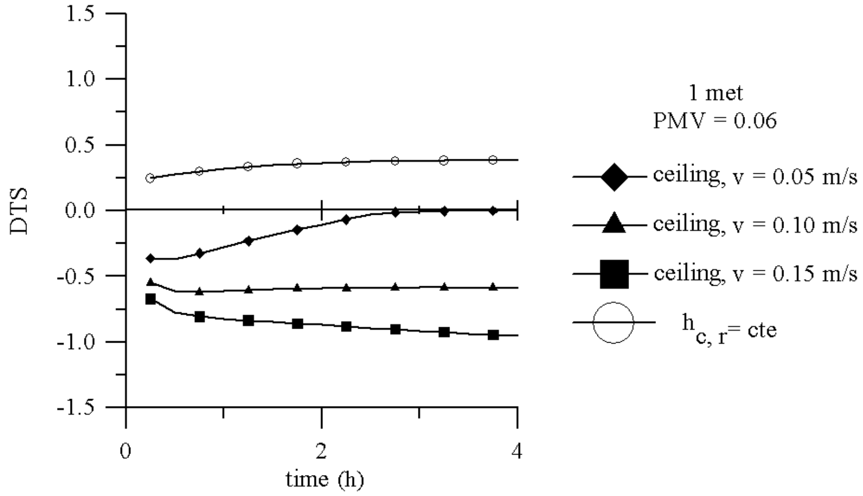

Figure 1. Dynamic thermal sensation for individuals occupied with 1 met metabolic rate activity subjected to uniform air and temperature distribution.

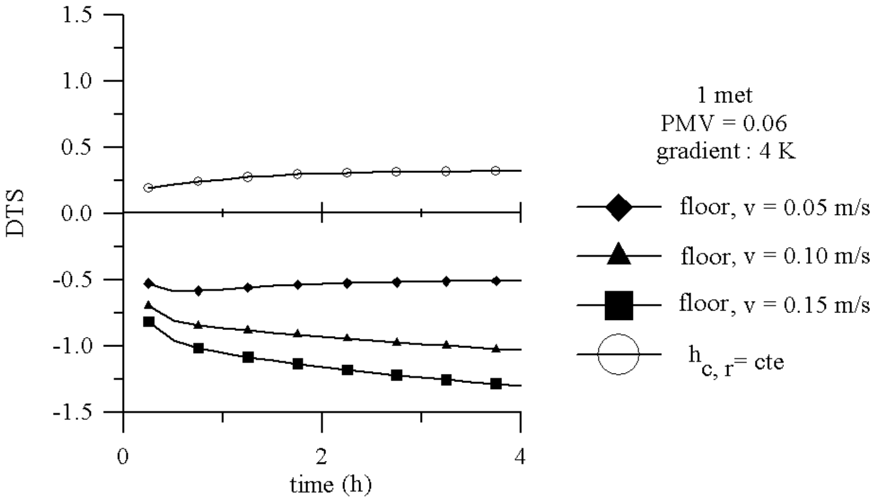

Figure 2. Dynamic thermal sensation for individuals occupied with 1 met metabolic rate activity subjected to non-uniform air and temperature distribution (12%RH range and 4^o C air temperature gradient).

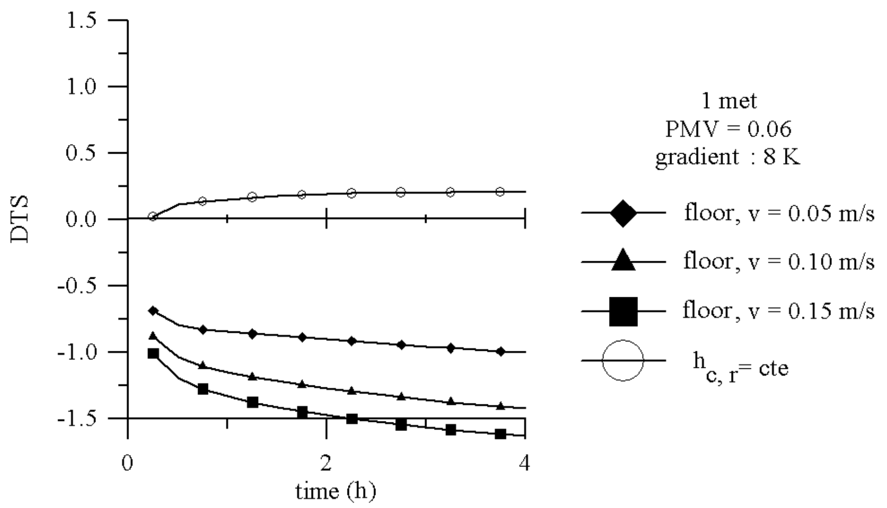

Figure 3. Dynamic thermal sensation for individuals occupied with 1 met metabolic rate activity subjected to non-uniform air and temperature distribution (25%RH range and 8^{o} C air temperature gradient).

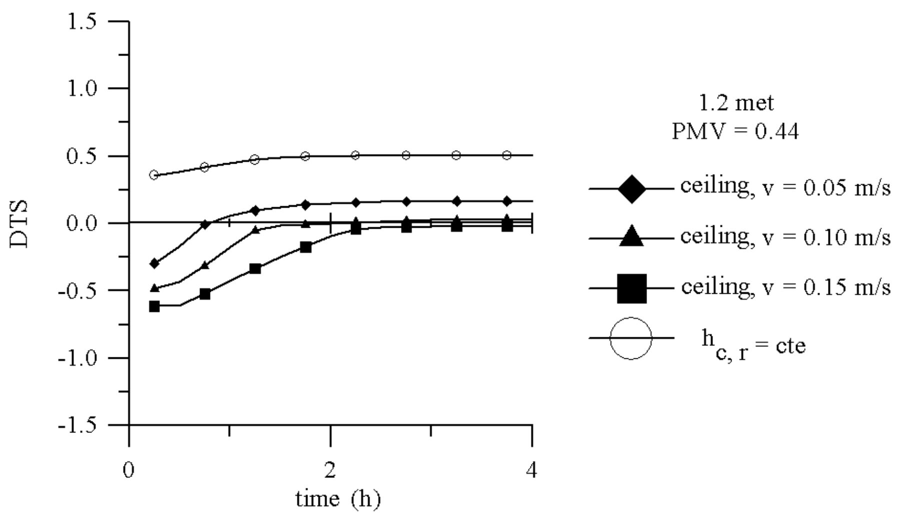

Figure 4. Dynamic thermal sensation for individuals occupied with 1.2 met metabolic rate activity subjected to uniform air and temperature distribution.

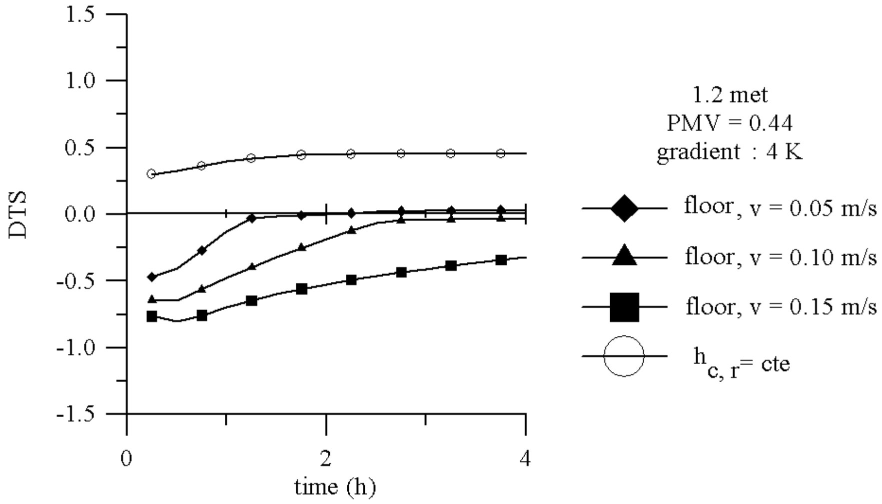

Figure 5. Dynamic thermal sensation for individuals occupied with 1.2 met metabolic rate activity subjected to non-uniform air and temperature distribution (12%RH range and 4^o C air temperature gradient).

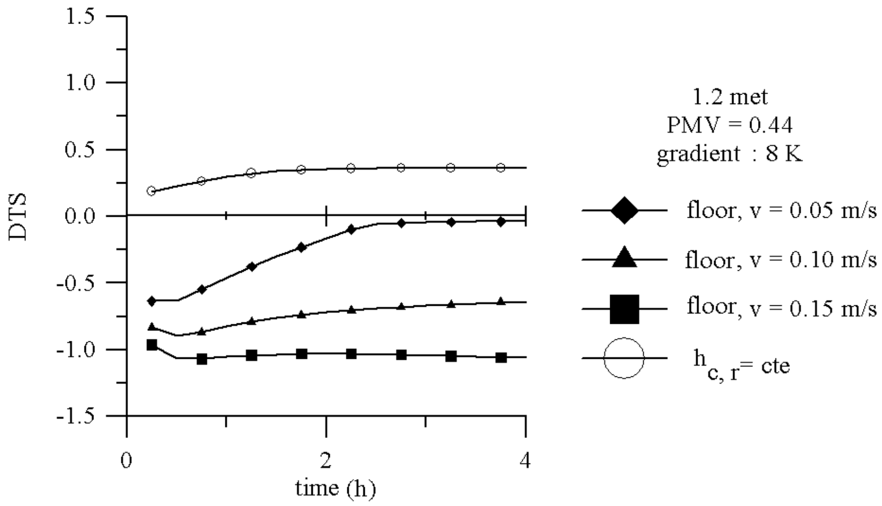

Figure 6. Dynamic thermal sensation for individuals occupied with 1.2 met metabolic rate activity subjected to non-uniform air and temperature distribution (25%RH range and 8^o C air temperature gradient).

According to the results obtained, underfloor air distribution can provide a satisfactory compromise between cost and air quality (as Bauman and Webster (2001) concluded), provided that the architect and engineer can control the air velocity and thermal gradient to which the occupant will be exposed.

4.2. Dynamic thermal sensation in subway trains

Another research project which focused on Rio de Janeiro train passengers revealed discomfort complains for line #2 during summer time. This line has underground and also surface segments, the latter exposed do solar radiation. Air comfort predictions were requested.

The practice of the train company is to apply the UIC code (Union Internationale des Chemins de Fer – International Union of Railways). "The UIC was founded in 1922, in the wake of the intergovernmental conferences of Portorosa and Genoa, with the aim of creating uniform conditions for the establishment and operation of railways" (http://www.uic.asso.fr/s_apropos/apropos/presentation_en.html?changeLang=_em, 12/01/2001). The UIC code relates internal and external temperatures to identify the comfort temperature. Another prediction was obtained using the software ASHRAE Thermal Comfort model.

To compare both predictions, an expected sequence of exposures was examined. The passenger was considered coming from an air-conditioned environment of 25° C, 55% RH. Subsequently, he is exposed to 35.3° C convective air temperature, air velocity 0.1 m/s, 32.3° C radiant temperature, 55% RH, 1.2 met for 30 minutes. The following exposure corresponds to the passenger standing in the railway car for 30 minutes, which corresponds to the complete itinerary. The UIC code suggested indoor conditions at 26° C, 50% RH for comfort. The ASHRAE software suggested 24° C, 55% RH. The radiant temperature was considered to be 2° C above the convective one. The same clothing with the previous case study was considered, but it was adjusted for standing person, which corresponds to 0.6 clo (Falkenberg, 2001).

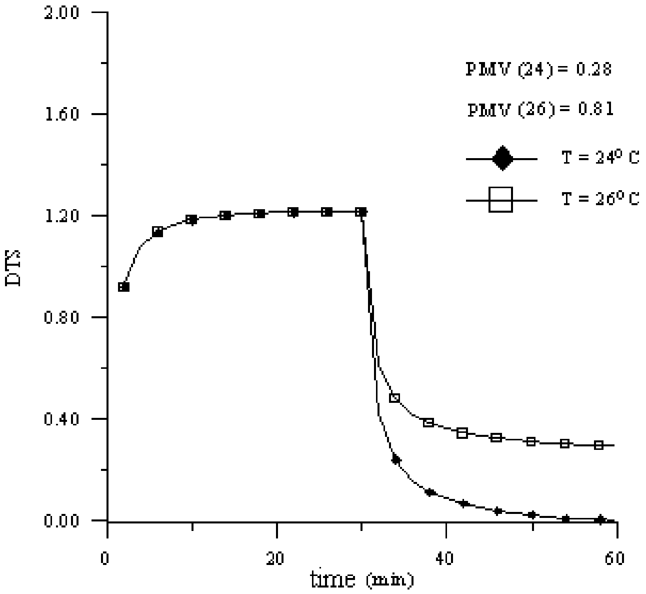

Figure 7. Dynamic thermal sensation for individuals occupied with 1.2 met metabolic rate activity subjected to uniform air and temperature distribution, before entering the subway train (0 to 30 minutes) and along the itinerary (30 to 60 minutes).

The simulation (graphic presented on Figure 7) shows the dynamic adaptation to both indoor conditions. The best result for DTS was obtained when the indoor air is at 24° C, 55% RH. The total sensible heat loss (from both perspiration and transpiration) for this case was 23W, while for 26° C, 50% RH it was 43W, after ten minutes inside the train. The acceptable comfort zone was achieved in both cases in the first few minutes.

The extended model allowed us to compare both thermal conditions concerning to comfort vote and energy cost. It accommodated various exposure scenarios, providing richer information about the user sensations than usual models.

4.2. Thermoelectric machinery hangar

The third case study relates to physically demanding working conditions in a thermoelectric machinery hangar. Due to the motors, the hangar has powerful heat sources, and because of the way it was built, natural ventilation was not enough to provide cooling. From time to time, workers had to practice preventive maintenance. This activity requires high metabolic rate and causes discomfort when performed in hot areas.

Before modifying the building ventilation, it was questioned whether simply increasing air velocity would improve working conditions. The considered thermoelectric plant is situated in Macapá, a very hot and humid city. To simulate the worker exposures, the subject was first considered to be exposed for ten minutes to 29.5 ° C convective air temperature, 31.5 ° C radiant air temperature, 78% RH, 1.2 met, 0.6 clo. Subsequently and for thirty minutes, the metabolic rate was increased to 3.5 met, the air convective temperature was raised to 40.0 ° C, radiant temperature to 38.7 ° C, and RH changed to 30%. Three different air velocities were tested: 0.5 m/s, 1.0 m/s and 2.0 m/s. None of them resulted in comfort improvements (see Figure 8), leading to the conclusion that other interventions were necessary to improve comfort.

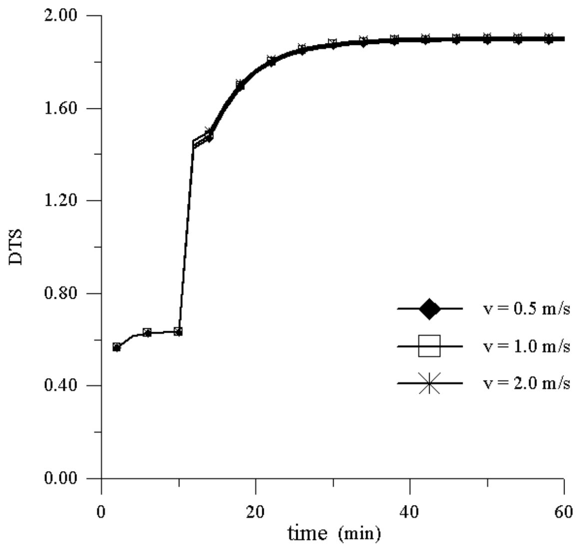

Figure 8. Dynamic thermal sensation for thermoelectric machinery hangar workers.

5. Conclusions

Overall the model predictions for the three case studies are reasonable. The extended model has been proven provide an important tool in the systematic study of dynamic and non-uniform exposures, allowing at the same time realistic inputs. The output information is not only valuable in predicting the dynamic thermal sensation of the occupant, but also in redesigning the air-conditioning system.

The numerical model was exhaustively tested, but its ultimate usefulness can only be assured in comparison to experimental data. More information on clothing thermal resistance and on local thermal resistance values is also necessary. The model was built for a generic population. Future work would allow it to be individualized, by accommodating varying body mass, height and other anthropometrics, thus creating a predictive tool for more specific populations (Havenith, 2001).

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