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A review of general and local thermal comfort models for controlling indoor ambiences

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1. Introduction

General thermal comfort is defined by certain thermal conditions that, on average, affect the environment in order to ensure comfort from its broader view. This expression is related with the general condition of an environment, but in each zone we can find parameters out of the mean value. As a result, it is necessary to study the localized effect of each thermal comfort variables over the human thermoregulation, to obtain an adequate thermal comfort. However, it is possible to improve indoor ambiences through relevant building structural modifications, particularly thermal inertia, air conditioning facilities and human habits.

In this chapter, a research about the principal works on general and local thermal comfort, to define the better models employed as control algorithm in Heating Ventilation and Air Conditioning Systems (HVAC) to improve energy saving, material conservancy and work risk prevention, was conducted.

2. Earlier Research Works

When we try to comprehend general thermal comfort, it is common to analyse Fanger's PMV model; this model is based on thermoregulation and heat balance theories. According to these theories, the human body employs physiological processes in order to maintain a balance between the heat produced by metabolism and heat lost from the body.

In 1967, Fanger investigated the body's physiological processes, when it is close to neutral to define the actual comfort equation. Investigations (Fanger, 2003) began with the determination that the only physiological processes influencing heat balance were sweat rate and mean skin temperature as a function of activity level. Later, he used data from the study by McNall et al. (1967), to derive a linear relationship between activity levels and sweat rate, and conducted a study to derive a linear relationship between activity level and mean skin temperature. These two linear relationships were substituted into heat balance equations to create a comfort equation and describe all combinations of the six PMV input variables that result in a neutral thermal sensation.

Once an initial comfort equation was obtained, it was validated against studies by Nevins et al. (1966) and McNall et al. (1967), in which participants rated their thermal sensation in response to specified thermal environments. To consider situations where subjects do not

feel neutral, the comfort equation was corrected by combining data from Nevins et al. (1966), McNall et al. (1967) and his own studies (Fanger, 1970). 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 comfort (i.e. neutral) for a given activity. This expanded equation related thermal conditions to the seven-point ASHRAE thermal sensation scale, the PMV index. Fanger (1970) also developed a related index, the Predicted Percentage Dissatisfied (PPD). This index is calculated from PMV and predicts the percentage of people who are likely to be dissatisfied with a given thermal environment.

Thermal comfort standards use the PMV model to recommend acceptable thermal comfort conditions. The recommendations made by ASHRAE 2004, ISO 7730:2005 and ISO 7726:2002 are seen in Table 1. These thermal conditions should ensure that at least 90% of occupants feel thermally satisfied.

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

Table 1. ASHRAE standard recommendations.

When the general thermal comfort condition was defined by Scientifics, it developed research works to define the local comfort conditions related with air velocity, temperature and asymmetric radiation. In 1956, when Kerka and Humphreys began their studies on indoor environment, the first serious studies on local thermal comfort background began. However, man has had a special interest in controlling indoor environments. In these studies, they init to use panels to assess the intensity of smell of three different fumes and smoke to snuff. The main findings reveal that the intensity of the odour goes down slightly with a slight increase in atmospheric humidity. Another finding indicates that, in the presence of smoke snuff, the intensity of the odour goes down with increasing temperature for a constant partial vapour pressure.

In 1974, Cain explored the adaptation of man to four air components and to different concentrations over a period of time. The main conclusions revealed that there was no significant difference between pollutants. In 1979, Woods confirmed the results of Kerka and concluded that smell perception of odour intensity is linearly correlated with the enthalpy of air. In 1983, Cain et al. studied the impact of temperature and humidity on the perception of air quality. They concluded that the combination of high temperatures (more than 25.5°C) and relative humidity (more than 70%) exacerbate odour problems. Six years later, in 1989, Berglund and Cain discussed the adaptation of pollutants over time for different humidities. This study concluded that air acceptability, for different ranges of humidity at 24°C, is stable during the first hour. The subjective assessment of air quality was mainly influenced by temperature conditions and relative humidity and, second, by the polluted air. The linear effect of acceptance is more influenced by temperature than by relative humidity. In 1992, Gunnarsen et al. studied the possibility of adapting the perception of odour intensity; this adaptation was confirmed after a certain time interval. In 1996, Knudsen et al. carried out research into the air before accepting a full body and facial exposure. The problem with this test is that the process is carried out at a constant temperature equal to 22°C and the relative humidity is not controlled.

In 1998, Fang and co-workers carried out an initial experiment in a chamber, with clean air heated to 18ºC and 30% relative humidity (see Fig. 1). In this experiment, 40 subjects without specific training were subjected to the conditions in these chambers (Fig. 1). As a precaution, they were warned not to use strong perfumes before the experiment. The subjects underwent a facial exposure and questioned about their first impression of the air quality inside the chamber. In this case, we consider the existence of clean air where there are no significant sources of pollution and the air has not been renewed with outdoor air. From these studies, it was concluded that there is a linear relationship between the acceptability and enthalpy of the air. At high temperature levels and humidity, the perception of air quality appears more influenced by these variables than by the air pollutants. These findings need further validation which involves the development of more experiences.

In a second experiment, Fang and co-workers carried out a study of the initial acceptability and subsequent developments. They used clean air and whole body exposure to different levels of temperature and humidity. This experiment was divided into two sets: one aimed at defining the feeling of comfort and the other at defining the perception of smell.
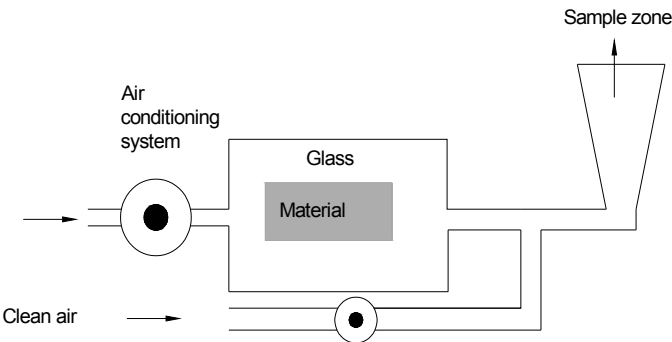

Fig. 1. House heated Climpaq designed by Albrectsen in 1988.

For these experiments, a system was developed based on two stainless steel chambers (3.60 x 2.50 x 2.55 m), independent and united by a door that allowed a camera to pass from one to the other; the individual who performs the test may turn to the second chamber at each stage of the experiment. The camera was subjected to a new odour level, temperature and/or humidity. (Fig. 2 reveals the shape of the chamber.) The experiment focused on conducting a survey on 36 students (26 males and 10 females) who had not been trained in issues of indoor environments. All were nearly 25 years old and had their whole body exposed in the chamber. The scale of values, employed during the survey, is seen in Fig. 3.
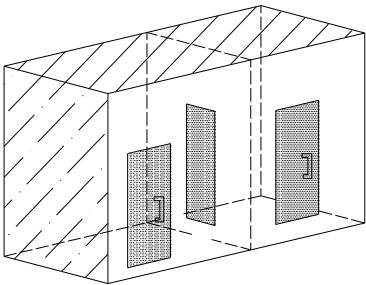

Fig. 2. New experimental chamber.

In these chambers, different temperatures and humidity within the ranges 18–28ºC and 30–70%, respectively, remained constant. The number of air changes in both chambers was the same and equal to 420 l/s. The existing pollutants came from the chamber or from the air renovation system.

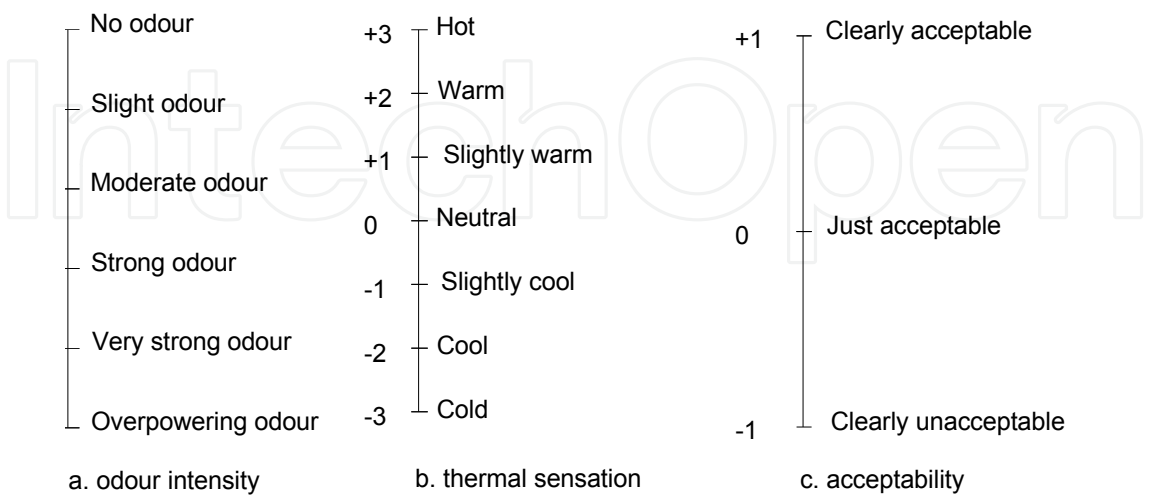| No odour | +3 | Hot | +1 | Clearly acceptable |
Slight odour	+2	Warm		
Moderate odour	+1	Slightly warm		
	0	Neutral	0	Just acceptable
Strong odour	-1	Slightly cool		
Very strong odour	-2	Cool		
Overpowering odour	-3	Cold	-1	Clearly unacceptable
a. odour intensity		b. thermal sensation		c. acceptability

Fig. 3. Used survey.

Every 20 minutes, existing conditions were varied which prompted the individual to change camera. The questionnaires were filled in every 2.5, 5, 10, 15 and 20 minutes. Through the process, the subjects could adapt their clothing to the environment around them to achieve thermal neutrality.

During the second round of experiments, individuals were submitted to the same procedure as the earlier one. In this case, a contaminated source, particularly PVC, was introduced and air renovation descended to 200 l/s. The pollutants were hidden in the camera and individuals were introduced in groups of six to answer the survey. The findings from the first experiment indicated that, depending on the temperature and relative humidity in the new chamber, there was a sudden jump in the alarm. The alarm, after 20 minutes, does not depend on the conditions of initial temperature and relative humidity.
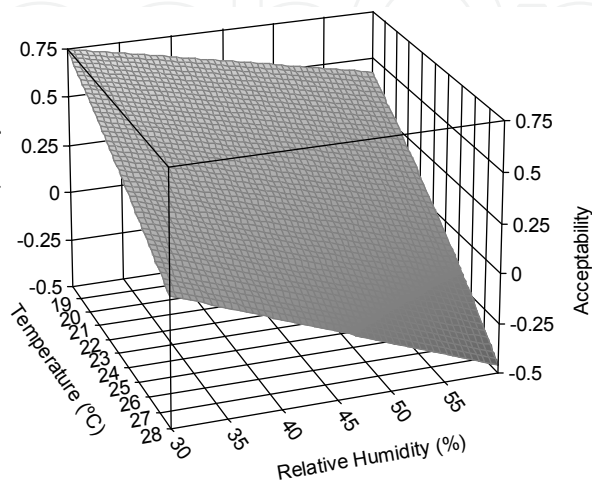

Fig. 4. Influence of temperature and relative humidity on the acceptability.

The results reveal that there is an increasing acceptability with the drop in temperature and relative humidity, and that cooling of the mucous membranes is essential to perceive the air as acceptable because it demonstrates the influence of the air enthalpy. The results indicated that, for a whole body exposure, there is a linear relationship of the acceptability with the enthalpy (for clean air as polluted, see Fig. 4). In conclusion, there is no difference between the initial acceptability and acceptability after 20 minutes of exposure. It also follows that the acceptability is independent of the environment conditions that surrounds the individual, before entering the camera.

The results of tests on odours indicate that the intensity of the odour varies little with temperature and relative humidity, and that there is some adjustment to smell after about 20 minutes. The studies by Berglund and Cain (1989) were proved in the absence of adaptation of acceptability in time. It also checks the result of Gunnarsen (1990), when it confirmed adaptation to the smell inside after a little while.

3. Results on General Thermal Comfort Models

3.1. P.O. Fanger model
Thermal comfort models were obtained from different bibliographic references (ISO and ASHRAE Standards), to determine which are more interesting.

The main object of heating, ventilation and air conditioning is to provide comfort to the occupants by removing or adding heat and humidity of the occupied space (ISO 7730:2005). Correspondingly, the main object of the study on the thermal comfort conditions is generally able to determine the conditions for achieving human internal thermal neutrality with minimal power consumption. To do this, the need to study a human body's response to certain environmental conditions arises.

It is considered a comfortable environment where there is no thermal perturbation, namely that the individual does not feel too cold or hot. This is achieved when the brain interprets the signals as two opposing forces, where the sensations of cold work in one direction and heat in the other. If the signals received in both directions are of the same magnitude, the resulting feeling is neutral. A person in thermal neutrality and completely relaxed is in a special situation, where the cold or heat sensors are not activated. To define the thermal comfort conditions of a climate, it must be given some characteristic parameters of the environment and its occupants. These parameters allow comparisons between the different environments of the study. Only after a thorough research, the thermal comfort and indoor air quality be judged the quality of the thermal environment and, consequently, the efficiency of the HVAC systems. Now, it can be revealed as the most important parameters in the design of the facilities of the air-conditioning systems.

To determine the thermal comfort rates of an environment, it can be found in two methods. One based on the study of thermal balance of the human body (Fiala et al., 2001) and the other based in empirical equations. This last method employs equations that define the same comfort rates with greater simplicity than the first. Another advantage is that they are expressed in terms of parameters much more easily in the sample for longer periods and, therefore, relate to the environment quality with energy savings.

The thermal balance is totally accepted and followed by ISO 7730:2005 for the study of comfort conditions, regardless of the climatic region. The thermal balance begins with two necessary initial conditions to maintain thermal comfort:

1) It must be obtained in a neutral thermal sensation from the combination of skin temperature and full body.

2) In a full body energy balance, the amount of heat produced by the metabolism must be equal to that lost to the atmosphere (steady state). Equation 3 was obtained by applying the above principles.

The rate of heat storage in the body was considered as two nodes (skin and core). The comfort equation can be obtained by setting the heat balance in thermally comfortable conditions for an individual. Based on these parameters, it can be established that the indices generally used to define a thermal environment (Equation 1) predicts the mean vote and 2 percent dissatisfaction.

$$PMV = \left(0.303 \cdot e^{-0.036 \cdot M} + 0.028\right) \cdot L \tag{1}$$

$$PPD = 100 - 95 \cdot e^{-\left(0.03353 PMV^4 + 0.2179 PMV^2\right)} \tag{2}$$

$$M - W = q_{sk} + q_{res} + S \tag{3}$$

$$M - W = (C + R + E_{sk}) + (C_{res} + E_{res}) + (S_{sk} + S_{cr}) \tag{4}$$

Where:

M — rate of metabolic heat production (W/m²)
W — rate of mechanical work accomplished (W/m²)
q_{sk} — total rate of heat loss from skin (W/m²)
q_{res} — total rate of heat loss through respiration (W/m²)
C+R — sensible heat loss from skin (W/m²)
C_{res} — rate of convective heat loss from respiration (W/m²)
E_{res} — rate of evaporative heat loss from respiration (W/m²)
S_{sk} — rate of heat storage in skin compartment (W/m²)
S_{cr} — rate of heat storage in core compartment (W/m²)

PMV scale is a computational model for the evaluation of generic comfort conditions and predictions of its limits. It is constituted by seven thermal sensation points ranging from 3 (cold) to +3 (hot), where 0 represents the neutral thermal sensation.

To predict the number of persons who are dissatisfied in a given thermal environment, the PPD index is used. In this index, individuals who vote –3, –2, –1, 1, +2 and +3 on the PMV scale are considered thermally unsatisfied. Its evolution, as a function of PMV, is reflected in Fig. 5.

For a PMV value between –0.85 and +0.85, the percentage of dissatisfied (PPD) is 20 and the assumption of a stricter PPD of 10% corresponds to a PMV between –0.5 and +0.5.

As a result, it can be three kinds of comfort zones, depending on the admissible ranges PPD and PMV (Table 2).
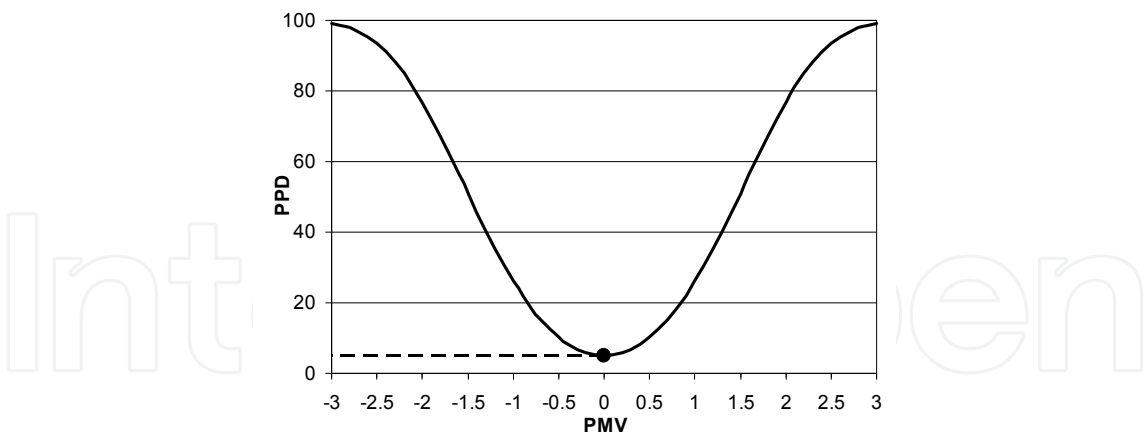
Fig. 5. Evolution of PPD on the basis of PMV.

Comfort	PPD	Range del PMV
A	<6	–0.2 < PMV < 0.2
B	<10	–0.5 < PMV < 0.5
C	<15	–0.7 < PMV < 0.7

Table 2. Predicted percentage of dissatisfied (PPD) based on the predicted mean vote (PMV).

One must remember that the evaporative heat loss from skin Esk depends on the amount of moisture on the skin, and the difference between the water vapour pressure on the skin and in the ambient environment. Finally, in the case of office workers, external work W can be considered zero. To deduce the comfort equation, the comfortable temperature of the skin and the sweat production equation with the full body thermal balance was combined (Stanton et al., 2005). This equation describes the relationship between measures of physical parameters and thermal sensation experienced by a person in an indoor environment. The comfort equation is an operational tool where physical parameters can be used to assess the thermal comfort conditions of an indoor environment. However, the comfort equation, obtained by Fanger, is too complicated to be solved through manual procedures.

On the sample of the thermal conditions of an interior environment, the human body does not feel the temperature of the compound; he feels the losses that occur with the thermal environment. Therefore, the parameters to be measured are those which affect the loss of heat: air temperature (ta), average temperature radiant ($\overline{t_r}$), relative humidity of the air (RH) and air velocity (v).

1. Metabolic rate (met): is the amount of energy emitted by an individual as a function of the level of muscle activity. Traditionally, metabolism has been estimated at met (1 met=58.15 W/m^2 surface of the body).

2. Cloth insulation (clo): is the unit used to measure the insulation of the clothing produced by clo, but the unit more technical and frequent use is m^{2}ºC/W (1 clo=155 m^{2}ºC/W). The scale is such that a naked person has a value of 0.0 clo and the typical street garment has 1.0 clo. The value of the clo, for people dressed, can be calculated as an addendum to the clo of each garment.

3. Mean radiant temperature: defines the radiant temperature of man, $\bar{t}_r$, as a uniform temperature in an imaginary black enclosure, in which a person would experience the same losses by radiation than in the real compound.

4. Operative Temperature: is the temperature in the walls and air of an equivalent compound that experiments the same heat transfer to the atmosphere by convection and radiation than in an enclosure where these temperatures are different.

5. Relative humidity: is defined as the relationship between the partial vapour pressures of water vapour in moist air and vapour pressure under saturated conditions. Often, it has been considered that the relative humidity of the interior environment is of little importance in the design of air conditioning elements. But now, the effect has become apparent on the comfort (ASHRAE; Fanger, 1970; Wargocki et al., 1999), perception of indoor air quality (Fang et al., 1998), health of the occupants (Molina, 2000) and energy consumption (Simonson, 2001).

6. Air velocity: No established clear link between air velocity and thermal comfort. For this reason, ASHRAE confirmed an air speed rise to a higher air temperature, but maintaining conditions within the comfort zone. In this, a series of curves of allowed temperature can be found for a given air speed, which is equivalent to those that produce the same heat loss through the skin.

After studying the equations that define the heat balance of a person, we can deduce the need of the sample for the instantaneous evolution of operative temperature, air velocity and relative humidity. To facilitate this procedure, it was summarised that the parameters must be measured directly or calculated (Table 3).

In Table 3, we found 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.

Method 1	Air velocity	Air temperature (t_a)	Mean radiant temperature ($\bar{t}_r$)	Humidity (w)
	Measure	Measure	Calculate	Measure
Method 2	Air velocity	Operative temperature (t_o)		Humidity (w)
	Measure	Measure		Measure
Method 3	Equivalent temperature (t_{eq})			Humidity (w)
	Measure			Measure
Method 4	Air velocity	Effective temperature (ET*)		
	Measure	Calculate		

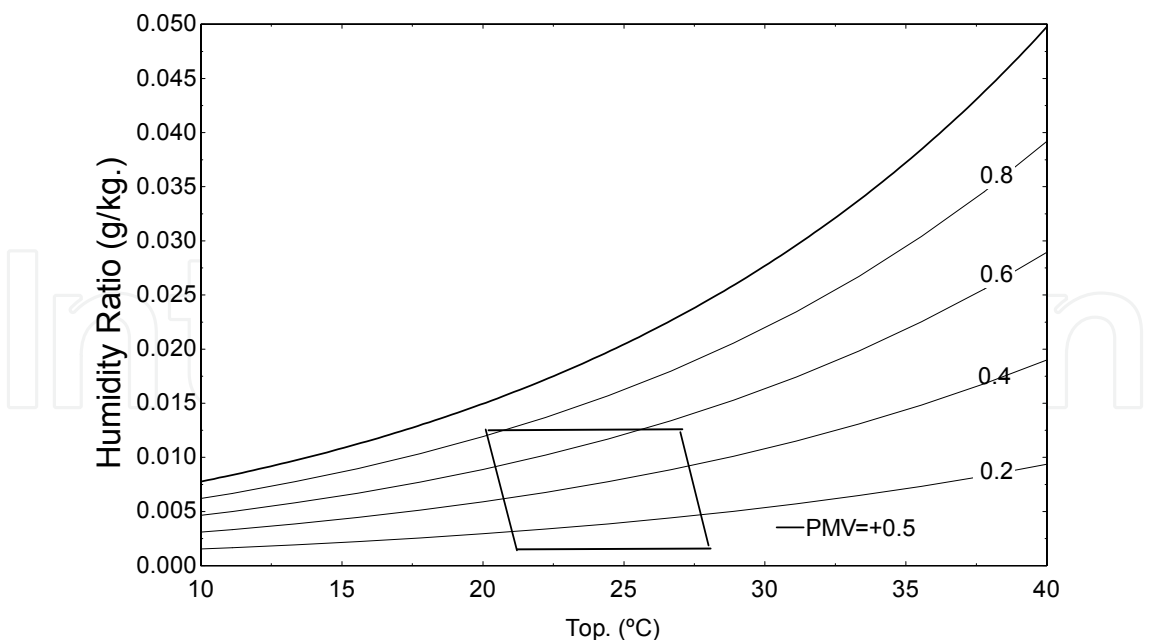Table 3. Methods to calculate general thermal comfort indexes.

Fig. 6. Comfort zone.

Finally, 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 the combination of air temperature and average radiant temperature. For air speeds not greater than 0.20 m/s, see Fig. 6.

3.2. Alternative PMV models

Among the thermal environment indices, the principal is the PMV. The conclusion on the work done by Oseland, subsequently reflected by ASHRAE, is that the PMV can be used to predict the neutral temperature with a margin of error of 1.4°C compared with the neutral temperature defined by the equation of thermal sensation. This thermal sensation expresses an equivalent index to the PMV. Its principal difference is that thermal sensation is obtained by regression of a survey to different individuals located in an environment. This survey presents a scale (Table 4).

An example, of a thermal sensation model that takes into account the effect of the clo, has been developed by Berglund (Equation 5).

Tsens	Thermal sensation
3	Warm
2	Heat
1	Soft
0	Neutral
−1	Soft freshness
−2	Freshness
−3	Cold

Table 4. Thermal sensation values.

$$T_{sens} = 0.305 \cdot T + 0.996 \cdot clo - 8.08$$

(5)

When Brager studied office buildings in San Francisco during the winter, he revealed that the PMV was found to be lower (colder) than the obtained thermal sensation. That defined a neutral temperature of 24.8°C, which was 2.4°C above the estimated value. After considering various UK offices with mechanical ventilation, it was demonstrated that the PMV differed by 0.5 points with the thermal sensation, which is equivalent to 1.5°C differences.

In Australia, Dear and Auliciems (1985), found a difference of 0.5–3.2° C between the neutral temperatures, estimated by surveys and determined by the PMV model of Fanger. Subsequently, Dear conducted a study in 12 Australian office buildings; it was defined that a temperature difference, between neutral temperatures proposed by surveys and PMV, was determined to about 1°C. Dear et al. extended their studies to those made by Brager. They returned to analyze and correct the data by the seat isolation. Again, discrepancies were found between the neutral temperature, based on the value obtained through surveys, and the value predicted by the equations. As a result, it seems that for real conditions, the thermal sensation of neutrality is in line with a deviation of the order of 0.2–3.3°C and an average of 1.4°C of the thermal neutrality conditions. The error was attributed to a PMV erroneous definition of the metabolic activity and the index of clo, or unable to take into account the isolation of the seat.

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. As a result, an investigation of 1,600 school-age students revealed statistics correlations between the level of comfort, temperature, humidity, sex and exposure duration. Groups consisting of 5 men and 5 women were exposed to a range of temperatures between 15.6 and 36.7°C, with increases of 1.1°C at 8 different relative humidifies of 15, 25, 34, 45, 55, 65, 75 and 85% and for air speeds of lower than 0.17 m/s. During a study period of 3 hours and in intervals of half hours, subjects reported their thermal sensations on a ballot paper with 7 categories ranging between –3 and 3 (Table 4). These categories show a thermal sensation that varies between cold and warm, passing 0 that indicates thermal neutrality. The results have yielded to an expression of the form (Equation 6).

$$PMV = a \cdot t + b \cdot p_v - c$$

(6)

Time/sex	A	B	C
1 hour/man	0.220	0.233	5.673
Woman	0.272	0.248	7.245
Both	0.245	0.248	6.475
2 hours/man	0.221	0.270	6.024
Woman	0.283	0.210	7.694
Both	0.252	0.240	6.859
3 hours/man	0.212	0.293	5.949
Woman	0.275	0.255	8.620
Both	0.243	0.278	6.802

Table 5. The coefficients a, b and c are a function of spent time and the sex of the subject.

By using this equation and taking into account sex and exposure time to the indoor environment, it should be used as constants (Table 5).

With these criteria, a comfort zone is, on average, close to conditions of 26ºC and 50% relative humidity. The study subjects have undergone a sedentary metabolic activity, dressed in normal clothes and with a thermal resistance of approximately 0.6 clo. Its exposure to the indoor ambiences was for 3 hours.

4. Results on Local Thermal Comfort Models

For an indoor air quality study, there are a number of empirical equations used by some authors over the last few years (Simonson et al., 2001). Indices, such as the percentage of dissatisfaction with local thermal comfort, thermal sensation and indoor air acceptability, are determined in terms of some simple parameter measures, such as dry bulb temperature and relative humidity. For instance, the humidity ratio and relative humidity are the most important parameters to compare the effect of moisture in the environment, whereas temperature and enthalpy reflect the thermal energy of each psychometric process. Simonson revealed that moisture had a small effect on thermal comfort, but a lot more on the local thermal comfort. The current regulations (ISO 7730, ASHRAE and DIN 1946) do not coincide with the exact value of moisture in the environment for some conditions, but concludes that a very high or very low relative humidity worsens comfort conditions.

The agreement chosen by ANSI/ASHRAE and ISO 7730 to establish the comfort boundary conditions was about 10% of dissatisfaction. Other authors believe that the local thermal comfort is primarily a function of not only the thermal gradient at different altitudes and air speeds, but may be also owing to the presence of sweat on the skin or inadequate mucous membrane refrigeration. To meet the local thermal comfort produced by the interior air conditions, Toftum et al. (1998a, b) studied the response of 38 individuals who were provided with clean air in a closed environment. The air temperature conditions ranged between 20 and 29ºC and the humidity ratio between 6 and 19 g/kg, as from 20ºC and 45% RH to 29ºC and 70% RH. Individuals assessed the ambient air with three or four puffs, and thus the equation for the percentage of local dissatisfaction was developed (Equation 7). ASHRAE recommends keeping the percentage of local dissatisfaction below 15% and the percentage of general thermal comfort dissatisfaction below 10. This PD tends to decrease when the temperature decreases and, as a result, limited conditions can be employed to define the optimal conditions for energy saving in the air conditioning system.

$$PD = \frac{100}{1 + e^{(-3.58 + 0.18 \cdot (30 - t) + 0.14 \cdot (42.5 - 0.01 p_v))}}$$

(7)

Where:

p_v is the partial vapour pressure (Pa)

4.1. Air velocity models

Air velocity affects sensible heat dissipated by convection and latent heat dissipated by evaporation, because both the convection coefficient and the amount of evaporated water per unit of time depend on it; therefore, the restful feeling becomes affected by air drafts.

Aiming towards energy saving in summer, the ambient air temperature can be kept slightly higher than the optimum and achieve a more pleasant feeling by increasing air velocity. The maximum acceptable air speed is 0.9 m/s.

In winter, the air circulation causes a cold feeling and to keep air temperature above that needed to avoid a feeling of discomfort, with its corresponding energy consumption. In winter, considering that the dry air temperature tends to be in the low band of comfort, air conditions in inhabited areas must be carefully studied, in order to maintain the conditions of wellbeing without wasting energy. It is recommended that the winter air velocity in the inhabited zone should be lower than 0.15 m/s. Localized draft problems are more common in indoor environments, vehicles and aircraft, with air conditioning. Even without a speed-sensitive air, there may be dissatisfaction owing to excessive cooling somewhere in the body.

In principle, there is sensitivity to currents on the nude parts of the body; therefore, only noticeable current flows on the face, hands and lower legs. The amount of heat lost through the skin because of the flow depends on the average speed of air, temperature and turbulence. Owing to the behaviour of the cold sensors on the skin, the degree of discomfort depends not only on the loss of local heat, but also on the influence in temperature fluctuations. For equal thermal losses, there is a greater sense of dissatisfaction with high turbulence in the air flow.

Some studies exhibit the types of fluctuations that cause greater dissatisfaction. These have been obtained from groups of individuals subjected to various air speed frequencies. The oscillations with a frequency of 0.5 Hz are the most uncomfortable, whereas oscillations with a higher frequency of 2 Hz produce less sensitive effects.

According to the ISO 7730:2005, drafts produce an unwanted local cooling in the human body. The flow risk can be expressed as the percentage of annoyed individuals and calculated (Equation 8).

The draft risk model is based on studies of 150 subjects exposed to air temperatures between 20 and 26ºC, with average air speed between 0.05 and 0.4 m/s and turbulence intensities from 0 to 70%. The model is also applicable to low densities of people, with sedentary activity and a neutral thermal sensation over the full body.

The draft risk is lower for non-sedentary activities and for people with neutral thermal sensation conditions. Fig. 7 reveals the relationship between air speed, temperature and the degree of turbulence, for a percentage of dissatisfaction of 10 or 20%. The different curves refer to a percentage of turbulence from 10 to 80.

$$DR = (34 - t)(v - 0.05)^{0.62}(0.37vT_u + 3.14)$$

(8)

Where:

v is the air velocity (m/s)

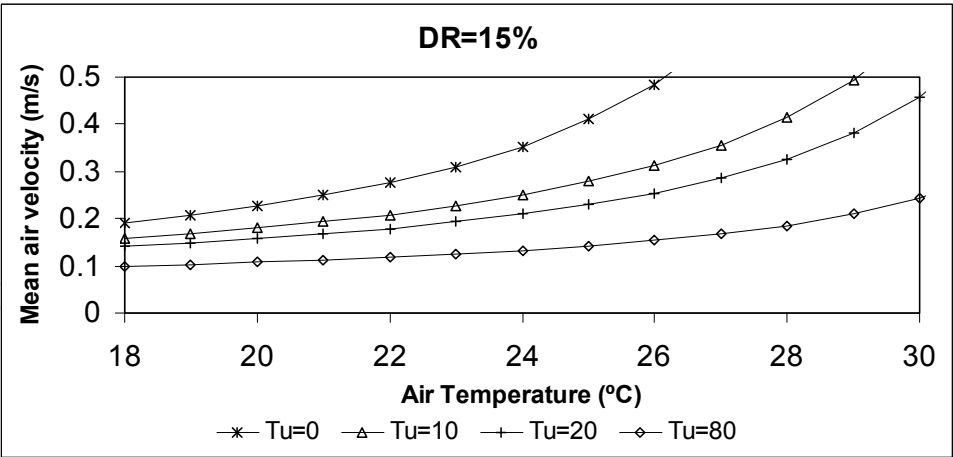t is the air temperature (ºC)

Tu is turbulence intensity (%)

Fig. 7. Average air velocity, depending on temperature and the degree of turbulence thermal environments, for type A, B and C.

4.2. Asymmetric thermal radiation

A person located in front of an intense external heat source, in cold weather, may notice after a certain period of time some dissatisfaction. The reason is the excessive warm front and high cooling on the other side. This uncomfortable situation could be remedied with frequent changes in position to achieve a more uniform heating. This example reveals the uncomfortable conditions owing to a non-uniform radiant heat effect.

To evaluate the non-uniform thermal radiation, the asymmetric thermal radiation parameter ($\bar{t}_r$) is used. This parameter is defined on the basis of the difference between the flat radiation temperature (t_{pr}) of the two opposite sides of a small plane element. The experiences of individuals exposed to variations in asymmetrical radiant temperature, such as the conditions caused by warm roofs and cold windows, produce the greatest impact of dissatisfaction. During earlier experiences, the surface of the enclosure and air temperature was preserved.
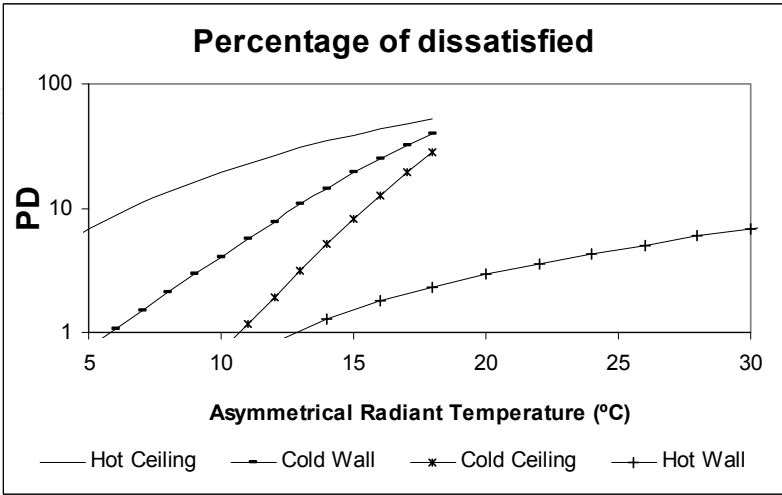

Fig. 8. Percentage of dissatisfied as a function of asymmetrical radiant temperature, produced by a roof or wall cold or hot.

The Parameter can be obtained by two methods: the first is based on the measure in two opposite directions, using a transducer to capture radiation that affects a small plane from the corresponding hemisphere. The second is to obtain temperature measurements from all surfaces of the surroundings and calculating the Δt_{pr}.

Equations 9, 10, 11 and 12 show the employed models for each case. Finally, the curves obtained are reflected in Fig. 8.

Hot ceiling ($\Delta t_{pr} < 23^{\circ}C$)

$$PD = \frac{100}{1 + \exp(2.84 - 0.174 \cdot \Delta t_{pr})} - 5.5 \tag{9}$$

Cold wall ($\Delta t_{pr} < 15^{\circ}C$)

$$PD = \frac{100}{1 + \exp(6.61 - 0.345 \cdot \Delta t_{pr})} \tag{10}$$

Cold ceiling ($\Delta t_{pr} < 15^{\circ}C$)

$$PD = \frac{100}{1 + \exp(9.93 - 0.50 \cdot \Delta t_{pr})} \tag{11}$$

Hot wall ($\Delta t_{pr} < 35^{\circ}C$)

$$PD = \frac{100}{1 + \exp(3.72 - 0.052 \cdot \Delta t_{pr})} - 3.5 \tag{12}$$

Where:

Δt_{pr} is the flat radiation temperature ($^{\circ}C$).

4.3. Vertical temperature difference

In general, there is an unsatisfied sensation with heat around the head and cold around the feet, regardless of whether the cause is convection or radiation. We can express the vertical temperature difference of the air existing at the ankle and neck height, respectively. Experiments on people's neutral thermal conditions have been conducted.

Based on these results, a temperature difference between head and feet of 3ºC produces a dissatisfaction of 5%. The curve obtained is reflected in Fig. 9. For a person in a sedentary activity, ISO 7730 is the acceptable value of 3ºC. The corresponding model is revealed in Equation 13.

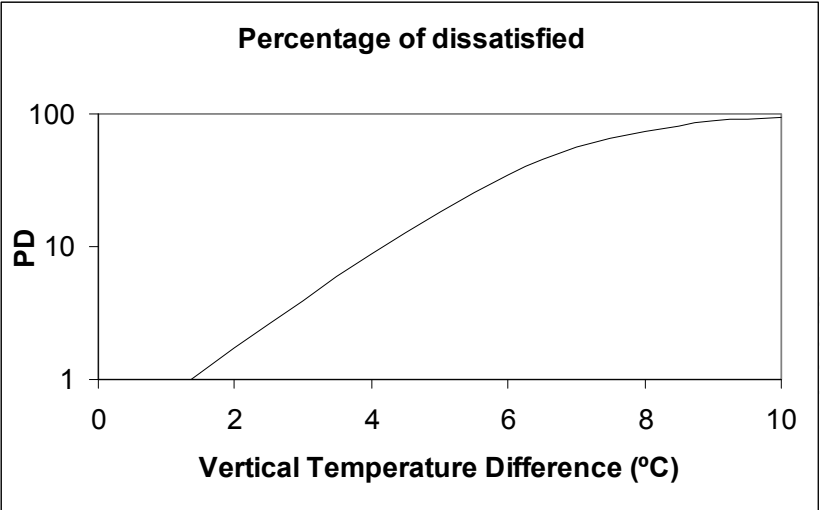$$PD = \frac{100}{1 + \exp(5.76 - 0.856 \cdot \Delta t)} \tag{13}$$

Fig. 9. Percentage of dissatisfied, depending on the vertical temperature difference.

4.4. Soil temperature

Direct contact between the feet and ground may cause local dissatisfaction, owing to a temperature which is either too high or low. Heat losses are dependent on other parameters, such as conductivity, heat capacity of the ground material and insulation capacity of the entire foot–footwear. ISO 7730 standard provides levels of comfort in sedentary activities for a 10% dissatisfied.

This leads to acceptable ground temperatures of between 19 and 29ºC. Studies have designated obtaining the curve (Fig. 10), and Equation 14 reflects the model of the percentage of dissatisfaction for different floor temperatures.

$$PD = 100 - 94 \cdot \exp(-1.387 + 0.118 \cdot t_f - 0.0025 \cdot t_f^2)$$

(14)

Where:

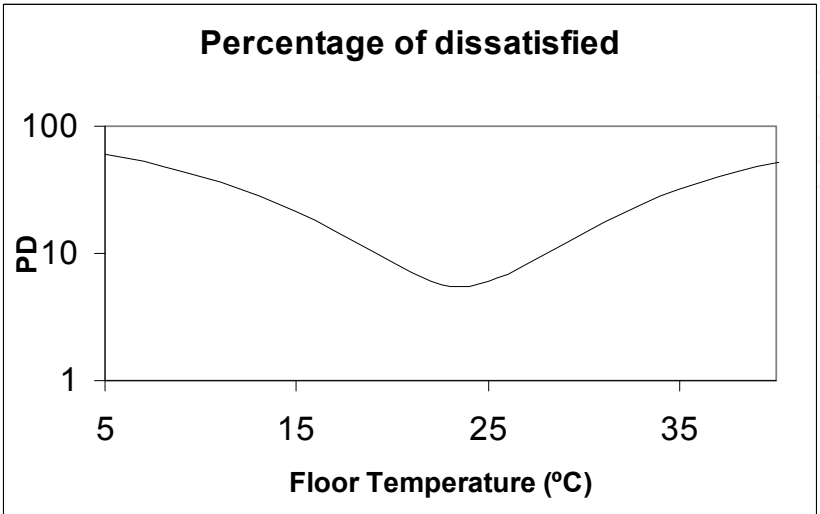t_f is the floor temperature (ºC).

Fig. 10. Percentage of dissatisfied, depending on the temperature of the floor.

5. Conclusions and Future Research Works

Given the varied activities of international involvement in indoor environments, it was necessary for an intense research report about thermal comfort models, based on results of scientific research and actual ISO and ASHRAE Standards. From this research, it was concluded that, apart from the thermal comfort models, there are many more theoretical models, both deterministic and empirical. As a result, some empirical models (Equation 15) present an interesting application to building design and/or environmental engineering owing to its easy resolution. Furthermore, these models present a nearly similar prediction of thermal comfort than Fanger's model, if they are applied considering its respective conditions of special interest for engineering application. Regardless, Fanger's thermal comfort model presents an in-depth analysis that relates variables that act in the thermal sensation. As a result, this model is the principal tool to be employed as reference for future research (Orosa et al., 2009a, b) about indoor parameters on thermal comfort and indoor air quality.

$$PMV = a \cdot t + b \cdot p_v - c \tag{15}$$

However, different parameters can alter general thermal comfort in localized zones of the indoor environment, such as air velocity models, asymmetric thermal radiation, vertical temperature difference, soil temperature and humidity conditions.

All these variables are related with the local thermal discomfort by the percentage of dissatisfied that are expected to be found in this environment (PD). The result of the effect of relative humidity on local thermal comfort, in particular, is of special interest (Equation 16).

$$PD = \frac{100}{1 + e^{(-3.58+0.18 \cdot (30-t)+0.14 \cdot (42.5-0.01 p_v))}} \tag{16}$$

Finally, an important conclusion for this review is that it is possible to save energy if you lower the number of air changes, temperature and relative humidity (Orosa et al., 2008a, b, 2009c, d). These discussions, to maintain the PD with the corresponding energy savings, are ongoing. Cold, very dry air with high pollution causes the same number of dissatisfaction than clean, mild and more humid air. Of interest is that if there is a slight drop in temperature and relative humidity, pollutants emitted by each of the materials (Fang, 1996) will be reduced. However, field tests are recommended by the researchers, so that they can perform characterization of environments according to their varying temperature and relative humidity. This may start the validation of models that simulate these processes by computer and implement HVAC systems to reach better comfort conditions and, at the same time, other objectives, such as energy saving, materials conservancy or work risk prevention in industrial ambiences (Orosa et al., 2008c).

6. Acknowledgements

7. References

ASHRAE 55-2004. (2004). Thermal Environmental Conditions for Human Occupancy. *ASHRAE Standard.*

Berglund, L.; Cain, W.S. (1989). Perceived air quality and the thermal environment. In: *Proceedings of IAQ '89: The Human Equation: Health and Comfort*, San Diego, pp. 93–99.

Cain, W.S.; Leaderer, B.P.; Isseroff, R.; Berglund, L.G.; Huey, R.J.; Lipsitt, E.D.; Perlman, D. (1983). Ventilation requirements in buildings– I. Control of occupancy odour and tobacco smoke odour, *Atmospheric Environment*, 17, pp.1183–1197.

Cain, WS. (1974). Perception of odor intensity and the time-course of olfactory adaptatio. *ASHRAE Trans* 80, pp.53–75.

Charles, K.E. (2003). Fanger's Thermal Comfort and Draught Models. IRC-RR-162. Http://irc.nrc-cnrc.gc.ca/ircpubs. (Accessed July 2009)

Fanger, P.O.; (1970). Thermal comfort. Analysis and applications in environmental engineering. McGrawHill. ISBN:0-07-019915-9

Fang, L.; Clausen, G.; Fanger, P.O. (1998). Impact of Temperature and Humidity on Perception of Indoor Air Quality During Immediate and Longer Whole-Body Exposures. *Indoor Air.* Vol. 8, Issue 4. pp.276-284.

Fiala, D.; Lomas, K.J.; Stohrer, M. (2001). Computer prediction of human thermoregulatory and temperature responses to a wide range of environmental conditions. *Int. J. Biometeorol.* 45, 143-159.

Gunnarsen, L.; Fanger, P.O. (1992). Adaptation to indoor air pollution. *Environment International.* 18, pp. 43–54.

ISO 7730:2005. (2005). Ergonomics of the thermal environment -- Analytical determination and interpretation of thermal comfort using calculation of the PMV and PPD indices and local thermal comfort criteria.

ISO 7726:2002. (2002). Ergonomics of the thermal environment - Instruments for measuring physical quantities.

Knudsen, H.N.; Kjaer, U.D.; Nielsen, P.A. (1996). Characterisation of emissions from building products: long term sensory evaluation, the impact of concentration and air velocity. In: *Proceedings of Indoor Air '96*, Nagoya. International Conference on Indoor Air Quality and Climate, Vol. 3, pp. 551–556.

McNall, Jr; P.E., Jaax; J., Rohles, F. H.; Nevins, R. G.; Springer, W. (1967). Thermal comfort (and thermally neutral) conditions for three levels of activity. *ASHRAE Transactions,* 73.

Nevins, R.G.; Rohles, F. H.; Springer, W.; Feyerherm, A. M. (1966). A temperature-humidity chart for thermal comfort of seated persons. *ASHRAE Transactions*, 72(1), 283-295.

Molina M. (2000). Impacto de la temperatura y la humedad sobre la salud y el confort térmico, climatización de ambientes interiores (Tesis doctoral) . Universidad de A Coruña.

Orosa, J.A.; García-Bustelo, E. J. (2009) (a) Ashrae Standard Application in Humid Climate Ambiences". *European Journal of Scientific Research.* 27 , 1, pp.128-139.

Orosa, J.A.; Carpente, T. (2009) (b). Thermal Inertia Effect in Old Buildings. *European Journal of Scientific Research.* .27 ,2, pp.228-233.

Orosa, J.A.; Oliveira, A.C. (2009) (c). Energy saving with passive climate control methods in Spanish office buildings. *Energy and Buildings*, 41, 8, pp. 823-828.

Orosa, J.A.; Oliveira, A.C. (2009) (d). Hourly indoor thermal comfort and air quality acceptance with passive climate control methods. *Renewable Energy*, In Press, Corrected Proof, Available online 31 May.

Orosa, J.A.; Baaliña, A. (2008) (a). Passive climate control in Spanish office buildings for long periods of time. *Building and Environment*. doi:10.1016/j.buildenv.2007.12.001

Orosa, JA; Baaliña, A. (2008) (b). Improving PAQ and comfort conditions in Spanish office buildings with passive climate control. *Building and Environment*, doi:10.1016/j.buildenv.2008.04.013

Orosa, J.A., 2008 (c) University of A Coruña. Procedimiento de obtención de las condiciones de temperatura y humedad relativa de ambientes interiores para la optimización del confort térmico y el ahorro energético en la climatización. Patent number: P200801036.

Simonson, C.J.; Salonvaara, M.; Ojanen, T. (2001). Improving Indoor Climate and Comfort with Wooden Structures. Technical research centre of Finland. Espoo 2001.

Stanton, N.; Brookhuis, K.; Hedge, A.; Salas, E.; Hendrick, H.W. (2005). *Handbook of Human Factors and Ergonomics Methods*. CRC Press, 2005. ISBN 0415287006, 9780415287005.

Toftum, J.; Jorgensen, A.S.; Fanger, P.O. (1998). Upper limits for indoor air humidity to avoid uncomfortably humid skin. *Energy and Buildings*. 28, pp. 1-13.

Toftum, J.; Jorgensen, A.S.; Fanger, P.O. (1998). Upper limits of air humidity for preventing warm respiratory discomfort. *Energy and Buildings*. 28, pp.15-23.

Wargocki, P.; Wyon, D.P.; Baik, Y.K.; Clausen, G.; Fanger, P.O. (1999). Perceived air quality, Sick Building Syndrome (SBS) symptoms and productivity in an office with two different pollution loads. *Indoor Air*, 9, 165–179.

Woods, J.E. (1979). Ventilation, health & energy consumption: a status report, *ASHRAE Journal*, July, pp.23–27.

Air Quality

Edited by Ashok Kumar

Air pollution is about five decades or so old field and continues to be a global concern. Therefore, the governments around the world are involved in managing air quality in their countries for the welfare of their citizens. The management of air pollution involves understanding air pollution sources, monitoring of contaminants, modeling air quality, performing laboratory experiments, the use of satellite images for quantifying air quality levels, indoor air pollution, and elimination of contaminants through control. Research activities are being performed on every aspect of air pollution throughout the world, in order to respond to public concerns. The book is grouped in five different sections. Some topics are more detailed than others. The readers should be aware that multi-authored books have difficulty maintaining consistency. A reader will find, however, that each chapter is intellectually stimulating. Our goal was to provide current information and present a reasonable analysis of air quality data compiled by knowledgeable professionals in the field of air pollution.

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