Chapter 2
Indoor Air Standards and Models

2.1 Indoor Air Standards

2.1.1 Introduction

This chapter aims to reveal the main standards applicable for indoor environments at international level. Specifically, the mean values and limiting conditions and the models for indoor environments are revealed as the main tools to be employed in real case studies.

Following a methodic enumeration of standards, they are classified into general thermal comfort, local thermal comfort and indoor air quality (IAQ) standards.

2.1.2 General Thermal Comfort Standards

In this section, the main standards of the American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) and International Organization for Standardization (ISO) for general thermal comfort are described. At the same time, different sections divide thermal comfort in moderate environments and work risk prevention in extreme environments. Their contents are described in the ASHRAE 55 standards.

2.1.2.1 Moderate Indoor Environments

As described earlier, the main ISO [1, 2], NTP [3] and ASHRAE [4, 5] standards that may be of interest to the ergonomics or thermal comfort researcher are enumerated in the following sections. Note that ISO 7730 describes the indices and sampling procedures, in accordance with the previous chapter.

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Table 2.1 International and national standards for general thermal comfort in moderate environments

ISO Standards	ISO 11399:1995 Ergonomics of the thermal environment—principles and application of relevant international standards
	ISO 7730:2005 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 9920:2007 Ergonomics of the thermal environment—estimation of thermal insulation and water vapour resistance of a clothing ensemble
	ISO 8996:2004 Ergonomics of the thermal environment—determination of metabolic rate
	ISO 7726:1998 Ergonomics of the thermal environment—instruments for measuring physical quantities
	ISO 10551:1995 Ergonomics of the thermal environment—assessment of the influence of the thermal environment using subjective judgment scales
ASHRAE Standard	ANSI/ASHRAE 55-2004 Thermal environmental conditions for human occupancy
	NTP 74: Thermal comfort
Spanish Thermal Comfort Standards	NTP 501: Thermal environment: local thermal discomfort
	NTP 242: Ergonomics: ergonomic office workplaces analysis
	NTP 503: Acoustic comfort: noise in offices
	NTP 358: Odours: a factor of indoor air quality and comfort

The first standard is ISO 11399. On the other hand, the main predicted mean vote (PMV) and predicted percentage of dissatisfied (PPD) indices are described in ISO 7730, in accordance with the heat that must be transferred from the whole body into real environments, as described in ISO 9920 and ISO 8996 standards. Furthermore, the general measuring methodology is described in ISO 7726 standard. We find that ASHRAE 55 is equivalent to the contents described in the ISO standards.

Once the models are developed, they must be tested with respect to the real questionnaires, as reflected in ISO 10551 standard.

It is interesting to define the application of these general standards to particular conditions. It is the case of the national standards developed by the Spanish ministry of work. Its standards are entitled NTP (technical standards for work risk prevention).

Within the NTP standards, standards 74 and 501 are of special interest, related to the general and local thermal comfort under working conditions. In some working areas, such as office buildings, an in-depth analysis was made, reflected by the standard 242.

Some new parameters to be considered at the time of evaluating IAQ and local thermal comfort, such as noise and odours, were developed (503 and 358) (Table 2.1).

Table 2.2 International and national standards for general thermal comfort in extreme environments

	Standard title
ISO	ISO 9886:2004 Ergonomics—evaluation of thermal strain by physiological measurements
	ISO 7933:2004 Ergonomics of the thermal environment—analytical determination and interpretation of heat stress using calculation of the predicted heat strain
	ISO 11079:2007 Ergonomics of the thermal environment—determination and interpretation of cold stress when using required clothing insulation (IREQ) and local cooling effects
NTP	NTP 387: Working conditions analysis: the ergonomic workplace analysis
	NTP 322: Estimation of the heat stress: WBGT
	NTP 350: Heat stress evaluation. Required sweating index
	NTP 462: Cold stress: occupational exposures evaluation
	NTP 18: Heat stress evaluation of severe exposures
	NTP 279: Thermal environmental and dehydration
	NTP 534: Mental workload: factors
	NTP 445: Mental workload: fatigue
	NTP 179: Mental workload: definition and measurement
	NTP 575: Mental workload: indicators
	NTP 445: Mental workload: fatigue
	NTP 275: Mental workload in health care workers: an assessment checklist
	NTP 177: Physical work load: definition and measurement
	NTP 295: Physical work load evaluation by continuous register of heart rate
	NTP 405: Human factor and accident rates. Social aspects

As for general indoor moderate environments found in ISO and NTP standards, there is an adaptation of these standards to extreme indoor environments (Table 2.2). Notably, standard ISO 9886 shows the measuring methodology to detect thermal strain. Heat and cold stress determination and interpretation are described in ISO standards 7933 and 11079.

As seen in Table 2.2, there are a lot of standards at the national level to define and analyse this situation. The first general standard is the NTP 387 that describes the measuring methodology at the workplace. After that, indices for heat and cold stress evaluation are described in the summer standards, such as NTP 322, 350, 462 and 18.

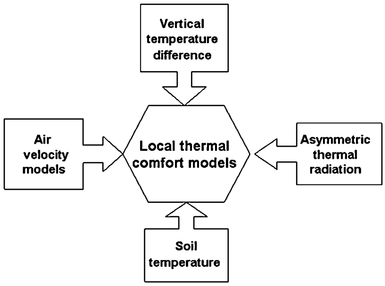
Notably, allegation and interpretation of mental work load is described in standards NTP 279, 534, 445, 179, 575, 475 and 275, and allegation and interpretation of physical work load in standards NTP 177 and 295.

Finally, new indices that show the percentage of accidents are described in standard NTP 405 (Table 2.2).

2.1.3 Local Thermal Comfort Standards

ASHRAE 55 and ISO 7730 are the only standards that define the local thermal comfort in an indoor environment, and its main indices and measuring procedures are described in this section.

Fig. 2.1 Local thermal comfort parameters



Most of the information that affects local thermal comfort is revealed in the models' section, and here we only show the main parameters. The first step is to define local thermal comfort as that defined in some particular body zones, due to: air velocity, asymmetric thermal radiation, vertical temperature difference and soil temperature (Fig. 2.1).

2.1.3.1 Air Velocity Models

Air velocity is related to sensible heat released by convection and latent heat released by evaporation and, hence, the feeling of thermal comfort is influenced by draft. For example, one bares a higher indoor air temperature when indoor air velocity is increased. So, during the summer season, ventilation helps to reduce cooling energy consumption. The opposite effect is obtained during the winter, when a higher air velocity also implies a higher heating energy consumption. To control these effects, there are two air velocity limits: indoor air velocity must never be over 0.9 m/s during the summer season and below 0.15 m/s during the winter.

Another parameter to be considered, to define local thermal comfort, is the influence of temperature fluctuation. For example, in two different conditions we experience the same thermal loss, feeling higher dissatisfaction when the air temperature experiments clear changes with time, called air turbulence.

2.1.3.2 Asymmetric Thermal Radiation

Asymmetric thermal radiation happens when an occupant of an indoor environment is exposed to a heat source on one side of his body during long periods of time, and experiences a certain degree of dissatisfaction. This happens, for example, in an indoor ambience with a warm roof and cold windows.

2.1.3.3 Vertical Temperature Difference

The vertical temperature difference can be defined as the temperature difference that exists between the ankle and neck.

2.1.3.4 Soil Temperature

Soil temperature is related to the difference in temperature between feet and ground temperature, depending on variables such as conductivity and the heat capacity of ground materials.

2.1.4 Indoor Air Quality Standards

Various international organizations, such as the World Health Organization (WHO) and the International Council of Building Research, the ASHRAE, some countries, such as Sweden (the Swedish Council of Building Research), the United States, Canada and Australia, have developed guidelines and standards of exposure to indoor air pollutants.

The air conditioning system has to ensure that the air contains acceptable low concentrations of pollutants. Hence, it must be properly designed and maintained to reduce pollutants to acceptable levels by dilution with clean air or elimination of foreign particles by filtration.

According to ASHRAE, an acceptable indoor air is one in which there are no known contaminants in harmful concentrations, as determined by the competent authorities and a substantial majority (80% or more) of staff are not exposed to dissatisfaction. Obviously, the definition is vague, not only with regard to acceptable levels, but also to the concept of dissatisfaction.

There are no standards to regulate the presence of microorganisms in the environment. The Committee for Bioaerosols of the American Conference of Governmental Industrial Hygienists (ACGIH) has recently published a guide for the assessment of bioaerosols in an indoor environment that can be used as a starting point.

For those chemicals that do not have a reference value, it is acceptable (ASHRAE 62 [5, 6]) that a concentration of 1/10 TLV does not produce a significant increase in the number of complaints from members of a group of industrial workers.

Table 2.3 lists the maximum concentrations of pollutants that may be present in an outdoor air and also a minimum that can be used for ventilation in an enclosed building. Information for common air pollutant internal maximum exposure limits of the Occupational Safety and Health administration (OSHA) and the ACGIH in an industrial environment is included in Table 2.4.

Table 2.3 Reference values of external air quality by US EPA, Environmental Protection Agency

Contaminant	Long exposure			Short exposure		
	Mean concentration			Mean concentration		
	μg/m^3	ppm	Time	μg/m^3	ppm	Time (h)
SO_2	80	0.03	1 year	365	0.14	24
CO_2	–	–	–	40,000	35	1
				10,000	9	8
N_2	100	0.053	1 year	–	–	–
O_3	–	–	–	235	0.12	1
Pb	1.5	–	3 months	–	–	–
Particulates	75	–	1 year	260	–	24
Radon	0.2		Pico curies/l			

Table 2.4 Reference values and concentrations recommended for some industrial pollutants by OSHA and ACGIH

Contaminant	Concentration	Exposure time	Origin
Asbestos	0.2–2.0 libres/cm^3	8 h	TLV-TWA
SO_2	2 ppm	8 h	PEL-TWA
	5 ppm	15 min	PEL-STEL
	10,000 ppm	8 h	PEL-TWA
CO_2	5,000 ppm	8 h	TLV-TWA
	30,000 ppm	15 min	PEL-STEL
	1 ppm	15 min	PEL-STEL
NO_2	3 ppm	8 h	TLV-TWA
	5 ppm	15 min	PEL-STEL
Formaldehydes	1 ppm	8 h	PEL-TWA
	2 ppm	15 min	TLV-STEL
CO	35 ppm	8 h	PEL-TWA
	200 ppm	15 min	PEL-TECHO
	50 ppm	8 h	TLV-TWA
	400 ppm	15 min	TLV-STEL
O_3	0.1 ppm	8 h	PEL-TWA
	0.2 ppm	15 min	PEL-STEL
Pb	0.005 mg/m^3	8 h	PEL-TWA
	0.15 mg/m^3	8 h	TLV-TWA

PEL permissible exposure limit; *TLV* threshold limit value; *TWA* time weighted average; *STEL* short term exposure limit

As seen in the previous sections, the actual standards related to IAQ are numbered (Table 2.5). To control IAQ, different standards were developed.

The first standards show the vocabulary and units for the measuring process, as described in ISO 4225 and 4226. Once the basic concepts are developed, the design process is described in standards 16813 and 16814.

After the design process, the measuring process in real indoor environments must be developed, as described in standards ISO 16000-1, 16000-5 and 16000-8.

We also need information on how to measure temperature, relative humidity and how to relate it with mould and its corresponding health effects, as described in standards ISO 8756, ISO 16000-16 and 7708.

As in general and local thermal comfort, the contents of ISO standards are reflected in the corresponding ASHRAE standard. However, it is the ANSI/ASHRAE 62.2-2004 that shows these contents.

The NTP describes the same contents as the ISO standards and adapt these to the particular conditions of working environments. In this standard, we find the general concepts shown in NTP 243 and different procedures for the characterisation of IAQ 431. Within this characterisation, we find the detection of fungi and microbiological hazards, as described in standards NTP 488, 299, 335 and 313, among others.

On the other hand, to evaluate the exchange of indoor environments, we have standards NTP 549 and 345.

Finally, to detect and control sick building syndrome (SBS), new standards were developed as NTP 288, 289, 290 and 380; these standards show the new criteria to develop future ventilation standards, as described in NTP 343 (Table 2.5).

2.2 Models

2.2.1 Introduction

According to ISO 7730 standard [1], thermal comfort is defined as the mental condition that expresses satisfaction with the surrounding environment. Even though it is easy to understand, it is at the same time difficult to define by equations. Equations of thermal comfort can be divided into local and general thermal comfort. General thermal comfort aims to define the mean parameters that a thermal environment must depict, so that a maximum number of persons experience a neutral thermal sensation.

The neutral thermal sensation is related to the fact that there are no hot or cold sensors activated, and hence that there is no heat release, or received from internal or external heat sources. In this regard, when defining general thermal comfort we must consider that, under the same weather conditions, there are other hot and cold sources that must be considered for different indoor ambiences. Hence, we must define thermal comfort for each specific indoor condition.

Table 2.5 Indoor air quality standards

ISO standard	ISO 4225:1994 Air quality—general aspects—vocabulary
	ISO 4226:2007 Air quality—general aspects—units of measurement
	ISO 16813:2006 Building environment design—indoor environment—general principles
	ISO 16814:2008 Building environment design—indoor air quality—methods of expressing the quality of indoor air for human occupancy
	ISO 16000-1:2004 Indoor air Part 1: general aspects of sampling strategy
	ISO 16000-5:2007 Indoor air Part 5: sampling strategy for volatile organic compounds (VOCs)
	ISO 16000-8:2007 Indoor air Part 8: determination of local mean ages of air in buildings for characterizing ventilation conditions
	ISO 8756:1994 Air quality—handling of temperature, pressure and humidity data
	ISO 16000-16:2008 Indoor air Part 16: detection and enumeration of moulds—Sampling by filtration
	ISO 7708:1995 Air quality—particle size fraction definitions for health-related sampling
ASHRAE Standard	ANSI/ASHRAE 62.1-2004 Ventilation for acceptable indoor air quality
	ANSI/ASHRAE 62.2-2004 Ventilation and acceptable indoor air quality in low-rise residential buildings
NTP [3]	NTP 243: Indoor air quality
	NTP 431: Characterisation of indoor air quality
	NTP 488: Indoor air quality: identification of fungi
	NTP 299: Method for airborne bacteria and fungi counting
	NTP 335: Indoor air quality: pollen grains and fungi spores evaluation
	NTP 313: Indoor air quality: microbiological hazards in air conditioning and ventilation systems
	NTP 347: Chemical contamination: Air concentration assessment
	NTP 315: Air quality: indoor low concentration gases
	NTP 521: Indoor air quality: emissions from building materials and cleaning products
	NTP 549: Carbon dioxide in evaluating indoor air quality
	NTP 345: Ventilation assessment using tracer gases
	NTP 289: Sick-building syndrome: risk factors
	NTP 288: Sick-building syndrome and building related diseases: bioareosol involvement
	NTP 290: Sick-building syndrome: questionnaire for its detection
	NTP 380: Sick-building syndrome: simplified questionnaire
	NTP 343: New criterion for future indoors ventilation standards

2.2.2 *Moist Air Models*

This model is described by ASHRAE [6] and subsequently implemented by Simonson [7] to determine the influence of coatings on indoor environments.

The basic equations that relate temperature and relative humidity are well known and are based on the working assumptions of the moist air model. They consider that air is a mixture of ideal gases that can be defined as dry air and water vapour. This water vapour presents an enthalpy equal to the enthalpy of saturated steam at the same temperature, for temperatures ranging from −10 to 50°C.

The relative humidity is defined after the relationship between partial pressure of water vapour in the air (P_v) and partial pressure of water vapour in the saturated air (P_{vsat}). Therefore, relative humidity is expressed as:

$$\mathrm{RH} = \frac{P_V}{P_{VSAT}} \tag{2.2.2.1}$$

Furthermore, the partial water vapour pressure in the saturated condition is a function of temperature (T):

$$P_{vsat} = f(T) = e^F \tag{2.2.2.2}$$

where the value of F is defined by ASHRAE:

$$27\,\mathrm{K} < \mathrm{T} < 273\,\mathrm{K} \quad F = \frac{\mathrm{C}_1}{T} + \mathrm{C}_2 + \mathrm{C}_3 T + \mathrm{C}_4 T_2 + \mathrm{C}_5 T_3 + \mathrm{C}_6 T_4 + \mathrm{C}_7 \ln \mathrm{T}$$

$$273\,\mathrm{K} < \mathrm{T} < 473\,\mathrm{K} \quad F = \frac{\mathrm{C}_8}{T} + \mathrm{C}_9 + \mathrm{C}_{10} T + \mathrm{C}_{11} T^2 + \mathrm{C}_{12} T^3 + \mathrm{C}_{13} \ln \mathrm{T}$$

The values of the constants are:

$C_1 = -5674.5359 C_2 = 6.3925247$	$C_3 = -9.677843 \times 10^{-3}$	
$C_4 = 6.22115701 \times 10^{-7}$	$C_5 = 2.0747825 \times 10^{-9}$	$C_6 = -9.484024 \times 10^{-13}$
$C_7 = 4.1635019$	$C_8 = -5800.2206$	$C_9 = 1.3914993$
$C_{10} = -4.8640239 \times 10^{-2}$	$C_{11} = 4.1764768 \times 10^{-5}$	$C_{12} = -1.4452093 \times 10^{-8}$
$C_{13} = 6.5459673$		

This equation shows that when temperature drops, the pressure of saturated vapour also drops.

Another variable widely used is absolute humidity (w), defined as the ratio between vapour and dry air mass. Absolute humidity is calculated from the relationship between the partial pressure of water vapour and of the air (P_a):

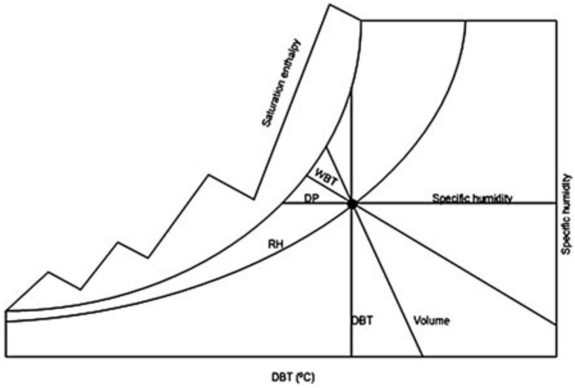
$$w = 0.62198 \frac{p_v}{p_a} \tag{2.2.2.3}$$

This equation describes the relationship between temperature, humidity ratio and relative humidity, which can be graphically expressed through a psychrometric chart.

Finally, the ideal gas law shows that moist air enthalpy (h) represents the sum of the energy of its components (dry air and water vapour). If the temperature and humidity increase, the enthalpy of the air increases (see also Fig. 2.2):

$$h = c_{p_a} t + w(c_{p_w} t + L_o) \tag{2.2.2.4}$$

Fig. 2.2 Psychometric chart



This equation has been used in a simplified form by Simonson, expressed as

$$h = t + w(2501.6 + 1.865t) \tag{2.2.2.5}$$

2.2.3 General Thermal Comfort Models

Once the concept of general thermal comfort is defined, it is the moment to define its two main indices: PMV and PPD.

PMV provides information about the thermal sensation of occupants in an indoor environment, experienced on a scale of seven points. This scale goes from minus three to plus three, passing through zero, which represents the neutral thermal sensation.

PPD is related to the PMV model, owing to the fact that there are differences in the perception of thermal comfort between persons. At the same time, it is related to individual habits, such as food, different clothes and styles and, in general, all the differences between individuals in their daily lives.

As a consequence of the earlier observation, PPD depicts a scale that goes from 0 to 100 in accordance with ASHRAE [8] and ISO 7730 [1] standards. The relationship between PPD and PMV is related to the fact that, for a PMV value of ±0.85, we find a PPD value of 20% (Fig. 2.3). Owing to these values, more standards define a thermal comfort limit of ±0.5 of PMV associated with a PPD of 10% (Table 2.6). In Table 2.6, we see that there are three thermal environment classes: A, B and C.

Finally, the general thermal comfort model depicts, in accordance with ASHRAE [8], an air velocity limit of 0.2 m/s. If air velocity is higher than this value, different values of general thermal comfort indices are expected, due to the higher ability to release heat.

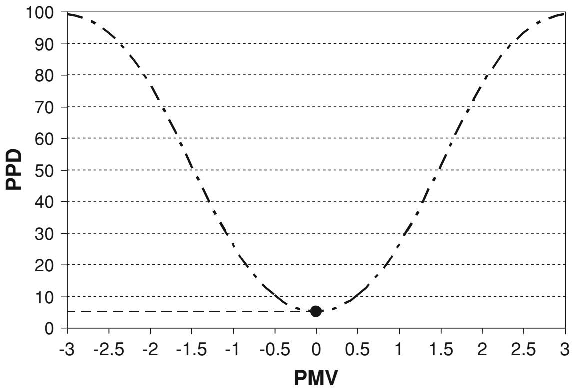


Fig. 2.3 Evolution of PPD on the basis of PMV

Table 2.6 Predicted percentage of dissatisfied based on the predicted mean vote

Comfort	PPD	PMV range
A	<6	$-0.2 < \text{PMV} < 0.2$
B	<10	$-0.5 < \text{PMV} < 0.5$
C	<15	$-0.7 < \text{PMV} < 0.7$

On the other hand, under this air velocity limit, local thermal comfort must be estimated and new parameters, such as the occupants' adaptability after some minutes in an indoor environment, must be considered.

To define general thermal comfort, one needs to estimate metabolic rate and clothes' insulation. The metabolic rate (met) is defined as the amount of energy released as a function of the level of muscular activity. It is defined as 58.15 W/m^2 of body surface, in accordance with the values reflected in the ISO standard.

The clo index is employed to quantify clothes' insulation. One clo is equal to 155 m^2°C/W. For example, a naked person shows a clo value of zero and a person wearing typical street clothes depicts a clo value equal to one.

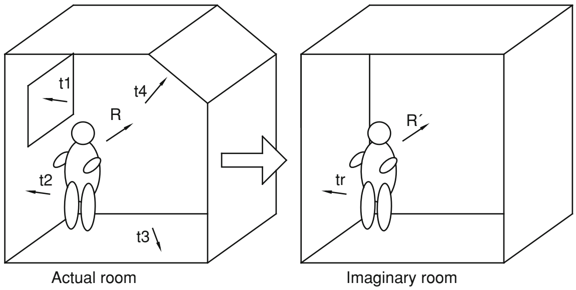
2.2.3.1 Sampled Parameters

To define thermal comfort in an indoor environment, the clo and the met values should be defined, as observed earlier. Once these parameters are estimated, more parameters must be considered, such as mean radiant temperature, operative temperature, relative humidity and air velocity.

Mean radiant temperature ($\bar{t}_r$) is defined as the uniform temperature in an imaginary black enclosure in which a person experiences the same loss by radiation than in the real situation.

The mean radiant temperature is that temperature contained in the walls and air of the compound room, which experiences the same heat transfer to the

Fig. 2.4 Mean radiant temperature



atmosphere by convection and radiation; in the case when such temperatures are different, it is a real environment defined by ASHRAE. This standard proposed different calculation methods of the operative temperature. Some methods employ the equation that defines the sensitive heat loss from the body, per unit of time and surface (Eq. 2.2.3.1.1). In other cases, this parameter can be defined as the arithmetical mean of the mean radiant temperature and air temperature (Eq. 2.2.3.1.2). This simplified equation can be only employed under the following conditions: a metabolic rate from 1.0 to 1.3 met, air velocities not higher than 0.20 m/s and no direct sunlight exposition.

$$t_o = \frac{(h_r \bar{t}_r + h_c t_a)}{(h_r + h_c)} \tag{2.2.3.1.1}$$

where t_a, $\bar{t}_r$ and t_o are the air, mean radiant and operative temperatures.

$$t_o = \frac{(\bar{t}_r + t_a)}{2} \tag{2.2.3.1.2}$$

The importance of operative temperature is based on the fact that it allows to define thermal comfort zones for some previously defined values of relative humidity, air speed, metabolic rate and clothes' insulation. This zone is represented in Fig. 2.4 for air velocities no greater than 0.2 m/s. Furthermore, two zones are typically represented for 0.5 and 1.0 clo, which are related to the hot and cold seasons. These two ranges are represented by Eqs. 2.2.3.1.3 and 2.2.3.1.4.

$$T_{\mathrm{op\,min},I_{\mathrm{cl}}} = [(I_{\mathrm{cl}} - 0.5\mathrm{clo})T_{\mathrm{min},1.0clo} + (1.0\mathrm{clo} - I_{\mathrm{cl}})T_{\mathrm{min},0.5\mathrm{clo}}]/0.5\mathrm{clo} \tag{2.2.3.1.3}$$

$$T_{\mathrm{op\,max},I_{\mathrm{cl}}} = [(I_{\mathrm{cl}} - 0.5\mathrm{clo})T_{\mathrm{max},1.0clo} + (1.0\mathrm{clo} - I_{\mathrm{cl}})T_{\mathrm{max},0.5\mathrm{clo}}]/0.5\mathrm{clo} \tag{2.2.3.1.4}$$

where $T_{\mathrm{max,I_{cl}}}$ is the upper operative temperature limit for clothing insulation I_{cl}, $T_{\mathrm{min,I_{cl}}}$ is the lower operative temperature limit for clothing insulation I_{cl} and I_{cl} is the thermal insulation of the clothing in question (clo).

Recent research has revealed that, to evaluate an indoor environment, great importance must be given to relative humidity. Relative humidity is related to

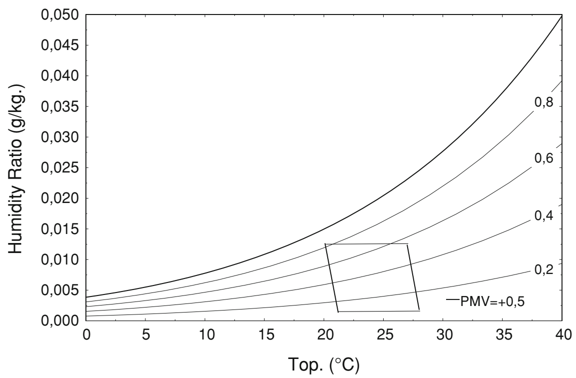


Fig. 2.5 Comfort zone

thermal comfort (ASHRAE [8, 9]; Wargocki in 1999 [10]), perception of IAQ [11], health of occupants [12] and energy consumption [7], as we can see in Fig. 2.5.

Although it is well known that there is a relation between thermal comfort and air velocity, it has not been reflected by in depth studies. To consider this effect, ASHRAE depicts curves of higher temperature supported with different indoor air velocities.

An analysis of the individual thermal balance will now be presented. Thermal comfort is based on the thermal balance of the full body and empirical equations, in accordance with ISO 7730 [1] and two conditions:

(1) The neutral thermal sensation on the skin temperature and full body temperature.
(2) The heat produced by metabolism must be the same as the heat lost to the atmosphere under steady state (Eqs. 2.2.3.1.5, 2.2.3.1.6).

$$M - W = q_{sk} + q_{res} + S \tag{2.2.3.1.5}$$

$$M - W = (C + R + E_{sk}) + (C_{res} + E_{res}) + (S_{sk} + S_{cr}) \tag{2.2.3.1.6}$$

where M is the rate of metabolic heat production (W/m^2), W is the rate of mechanical work accomplished (W/m^2), q_{sk} is the total rate of heat loss from the skin (W/m^2) q_{res} is the total rate of heat loss through respiration (W/m^2), C + R is the sensible heat loss from the skin (W/m^2),
C_{res} is the rate of convective heat loss from respiration (W/m^2),
E_{res} is the rate of evaporative heat loss from respiration (W/m^2),
S_{sk} is the rate of heat storage in the skin compartment (W/m^2),
S_{cr} is the rate of heat storage in the core compartment (W/m^2).

The heat storage in the body is defined by Eqs. 2.2.3.1.7 and 2.2.3.1.8:

$$S_{cr} = \frac{(1-\alpha_{sk})mc_{p,b}}{A_D} \cdot \frac{dt_{cr}}{d\theta} \tag{2.2.3.1.7}$$

$$S_{sk} = \frac{\alpha_{sk}mc_{p,b}}{A_D} \cdot \frac{dt_{sk}}{d\theta} \tag{2.2.3.1.8}$$

where

α_{sk} is the fraction of body mass concentrated in the skin, m is the body mass (kg),
$c_{p,b}$ is the specific heat capacity of the body (kJ/kgK),
A_D is the DuBois surface area (m^2),
t_{cr} is the temperature of the core node (°C),
t_{sk} is the temperature of the skin node (°C),
θ is the time (s).

To calculate thermal comfort indices, the following equations are typically used:

$$\mathrm{PMV} = \left(0.303 \cdot e^{-0.036 \cdot M} + 0.028\right) \cdot L \tag{2.2.3.1.9}$$

$$\mathrm{PPD} = 100 - 95 \cdot e^{-\left(0.03353\,\mathrm{PMV}^4 + 0.2179\,\mathrm{PMV}^2\right)} \tag{2.2.3.1.10}$$

L is the thermal load on the body; L is the difference between the internal heat production and heat loss to the actual environment.

Another parameter to be considered is the evaporative heat loss. This term depends on the amount of moisture on the skin and the difference between water pressure on the skin and in the environment (Eq. 2.2.3.1.11).

$$E_{sk} = \frac{w(p_{sk} - p_a)}{R_e + 1/(f_{cl}h_e)} \tag{2.2.3.1.11}$$

where

w is the wet skin area (m^2),
p_{sk} is the water vapour pressure on the skin (kPa),
P_a is the water vapour pressure in the environment (kPa),
R_e is the evaporative heat transfer resistance on a layer of clothing (m^2 kPa)/W and
h_e is the evaporative heat transfer coefficient (W/m^2 kPa),
f_{cl} is the fraction of body area covered with clothing.

The exchange of heat by respiratory convection and evaporative heat is given by:

$$C_{res} = 0.0014 \cdot M \cdot (34 - t) \tag{2.2.3.1.12}$$

$$E_{res} = 1.72 \cdot 10^{-5} \cdot M \cdot (5867 - P_v) \tag{2.2.3.1.13}$$

Table 2.7 Methods to calculate general thermal comfort indices

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

$$C + R = \frac{(t_{\mathrm{sk}} - t_o)}{R_{\mathrm{cl}} + 1/(f_{\mathrm{cl}} h)} \tag{2.2.3.1.14}$$

where

f_{cl} is the clothing area factor,
R_{cl} is the thermal resistance of clothing (m^2 K)/W,
t_{sk} is the temperature of the skin (°C),
h is the sum of convective and radiative heat transfer coefficients (W/m^2 K).

Equation 2.2.3.1.14 expresses the sensible heat loads from the skin.

The thermal comfort equation can be employed to define thermal comfort in an indoor environment. However, studies show that it is too complicated to be solved with manual procedures. Furthermore, to solve this equation we must consider the need to measure indoor air temperature, relative humidity and air velocity or related parameters (Table 2.7).

In Table 2.7, we find a new parameter called equivalent temperature (t_{eq}). This t_{eq} is defined as the uniform temperature of a radiant black enclosure with zero air velocity, in which an occupant has the same dry heat loss than in the actual non-uniform environment.

2.2.3.2 Alternative General Thermal Comfort Models

As reflected in ASHRAE, there is a certain disagreement between the PMV model and the thermal sensation, as demonstrated by De Dear [13] and Brager and De Dear [14]. This difference, when defined with the neutral temperature, is about 1.4°C. The reason is related to the fact that thermal sensation is obtained from the survey of individuals located in an environment and the PMV method only employs heat and mass transfer.

$$\mathrm{PMV} = at + b_{\mathrm{p}_v} - c \tag{2.2.3.2.1}$$

Table 2.8 The coefficients a, b and c are a function of spent time and the sex of the subject

Time/sex	*a*	*b*	*c*
1 h/man	0.220	0.233	5.673
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	6.802

Specifically, the error in defining the neutral temperature was related to a PMV problem in defining the metabolic rate and the clo value or taking into account the insulation of the seat.

Under ASHRAE contract, the Institute for Environmental Research at the Kansas State University developed a research work to define thermal comfort in sedentary regime. The main objective of this research was to define a model that expressed PMV in terms of easily measurable parameters.

An investigation to 1,600 students revealed a statistical correlation between the level of comfort, temperature, humidity and exposure duration. The research was developed in groups of five men and five women, with a temperature range between 15.6 and 36.7°C and relative humidity between 15 and 85%. Furthermore, the experiments were developed under air speeds below 0.17 m/s. Finally, the adaptation to a thermal environment was considered to happen when the thermal sensation was repeated every half an hour for 3 h.

The result, described in Eq. 2.2.3.2.1, reveals the dependence on three constants, a, b and c, for different periods of exposure (Table 2.8).

From these studies, a comfort zone close to 26°C and 50% was found. However, these two indices employ a scale of seven points that goes from minus three to plus three, with zero as the neutral thermal sensation (Table 2.9).

From these surveys, different regressions could be developed and, consequently, the models obtained depend on the parameters selected to develop the regression—for example, Eq. 2.2.3.2.2 [15].

$$T_{\text{sens}} = 0.305 \cdot T + 0.996 \cdot \text{clo} - 8.08 \tag{2.2.3.2.2}$$

Another option to define general thermal comfort is the operative temperature model. Operative temperature at different activity levels can be defined as a function of operative temperature at sedentary conditions (Eq. 2.2.3.2.3.1). Equation 2.2.3.2.3 can be employed between 1.2 and 3 met and for a minimum operative temperature of 15°C.

$$t_{\text{oac}} = t_{\text{osed}} - 3(1 + \text{clo})(\text{met} - 1.2) \tag{2.2.3.2.3}$$

Table 2.9 Thermal sensation values

T_{sens}	Thermal sensation
3	Warm
2	Heat
1	Slightly hot
0	Neutral
−1	Slightly fresh
−2	Freshness
−3	Cold

Furthermore, there are two equations for winter and summer:

$$\text{Terms of summer}: \; t_{\text{osed}} = 24.5 \pm 1.6^{\circ}C \tag{2.2.3.2.4}$$

$$\text{Terms of winter}: t_{\text{osed}} = 21.8 \pm 1.8^{\circ}C \tag{2.2.3.2.5}$$

These models, applied to define general thermal comfort, are adaptive. Recent research revealed that adaptive models can be employed to define the neutral thermal conditions in indoor environments as a function of outside weather. These models can be employed only when their occupants are in near sedentary activity level (1–1.3 met) and must be able to freely adapt their clothing.

However, a mechanical cooling or heating system can work only on a condition that there must not be any working mechanical ventilation system. Accordingly, windows are the main way to control thermal conditions.

In this regard, Eq. 2.2.3.2.6 was proposed by Nicol and Roaf [16] for naturally ventilated buildings. The other two models proposed by Humphrey [17] were Eqs. 2.2.3.2.7 and 2.2.3.2.8, and three more by Auliciems and de Dear [13] (Eqs. 2.2.3.2.9, 2.2.3.2.10). Finally, ASHRAE proposed Eq. 2.2.3.2.12 as the model to define neutral temperature conditions.

$$T_{\text{n,o}} = 17 + 0.38T_{\text{o}} \tag{2.2.3.2.6}$$

$$T_{\text{n,1}} = 2.6 + 0.831T_{\text{i}} \tag{2.2.3.2.7}$$

$$T_{\text{n,o}} = 11.9 + 0.534T_{\text{o}} \tag{2.2.3.2.8}$$

$$T_{\text{n,i}} = 5.41 + 0.731T_{\text{i}} \tag{2.2.3.2.9}$$

$$T_{\text{n,o}} = 17.6 + 0.31T_{\text{o}} \tag{2.2.3.2.10}$$

$$T_{\text{n,i,o}} = 9.22 + 0.48T_{\text{i}} + 0.14T_{\text{o}} \tag{2.2.3.2.11}$$

ASHRAE:

$$T_{\text{c}} = 17.8 + 0.31T_{\text{o}} \tag{2.2.3.2.12}$$

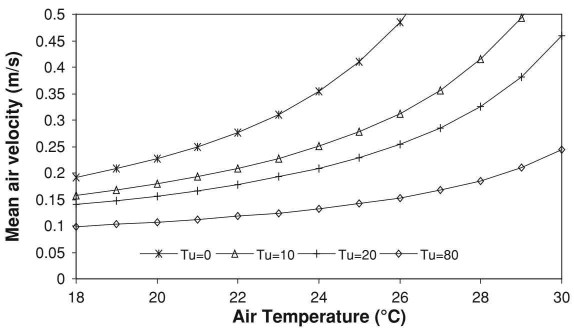


Fig. 2.6 Average air velocity depending on temperature and degree of turbulence of thermal environments for a percentage of dissatisfied persons below 20%

where T_c is the comfort temperature, T_o is the outdoor air temperature, T_i is the mean indoor air temperature, $T_{n,i}$ is the neutral temperature based on mean indoor air temperature and $T_{n,o}$ is the neutral temperature based on mean outdoor air temperature.

2.2.4 Local Thermal Comfort Models

Once the mean conditions that express general thermal comfort are defined, the variables that define thermal comfort in special body zones must be considered: local thermal comfort conditions.

2.2.4.1 Air Velocity Models

The percentage of PD owing to draft is obtained from Eq. 2.2.4.1.1 that is based on a study of 150 subjects. This experiment was developed within a temperature range of 20–26°C, speed ranges between 0.05 and 0.4 m/s and turbulence intensities from 0 to 70%. The draft risk (DR) is given by:

$$\mathrm{DR} = (34 - t)(v - 0.05)^{0.62}(0.37 v T_u + 3.14) \qquad (2.2.4.1.1)$$

From this equation, Fig. 2.6 can be obtained. The figure shows the draft risk of 15% of PD for different indoor air temperature levels and turbulence levels.

2.2.4.2 Asymmetric Thermal Radiation

The effect of asymmetric thermal radiation is now considered. This effect can be obtained by two methods: the first is to measure in two opposite directions, employing a transducer to capture radiation that affects a small plane from the corresponding hemisphere.

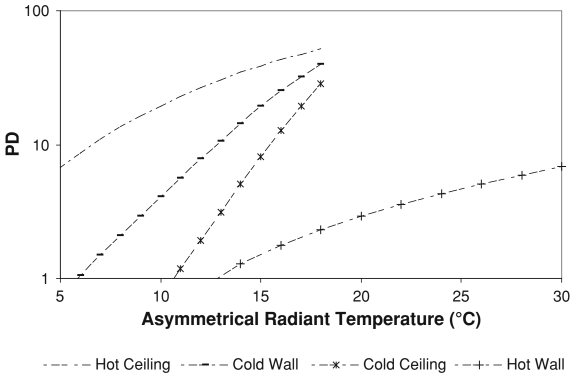


Fig. 2.7 Percentage of PD as a function of asymmetrical radiant temperature, produced by a roof or wall, cold or hot

The second method consists in measuring the temperature of the surrounding surface in indoor environments, and calculate the increment of radiant temperature Δt_{pr}. Equations 2.2.4.2.1–2.2.4.2.4 show the percentage of PD due to hot and cold ceiling, and hot and cold walls.

(a) Hot ceiling $\left(\Delta t_{pr} < 23°C\right)$

$$PD = \frac{100}{1 + \exp(2.84 - 0.174 \cdot \Delta t_{pr})} - 5.5 \tag{2.2.4.2.1}$$

(b) Cold wall $\left(\Delta t_{pr} < 15°C\right)$

$$PD = \frac{100}{1 + \exp(6.61 - 0.345 \cdot \Delta t_{pr})} \tag{2.2.4.2.2}$$

(c) Cold ceiling $\left(\Delta t_{pr} < 15°C\right)$

$$PD = \frac{100}{1 + \exp(9.93 - 0.50 \cdot \Delta t_{pr})} \tag{2.2.4.2.3}$$

(d) Hot wall $\left(\Delta t_{pr} < 35°C\right)$

$$PD = \frac{100}{1 + \exp(3.72 - 0.052 \cdot \Delta t_{pr})} - 3.5 \tag{2.2.4.2.4}$$

Finally, the curves obtained are shown in Fig. 2.7.

2.2.4.3 Vertical Temperature Differences

Recent research works (Eq. 2.2.4.3.1) reflect the general percentage of PD with vertical temperature differences. For different increments of temperature, we find different percentages of PD (Fig. 2.8).

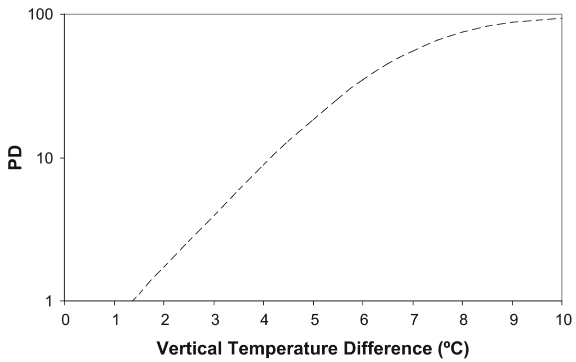


Fig. 2.8 Percentage of PD, depending on the vertical temperature difference

$$PD = \frac{100}{1 + \exp(5.76 - 0.856 \cdot \Delta t)} \tag{2.2.4.3.1}$$

where Δt is the vertical temperature difference (°C).

2.2.4.4 Soil Temperature

Standards show the percentage of PD in accordance with Eq. 2.2.4.4.1 and Fig. 2.9.

$$PD = 100 - 94 \cdot \exp(-1.387 + 0.118 \cdot t_f - 0.0025 \cdot t_f^2) \tag{2.2.4.4.1}$$

where t_f is the floor temperature.

2.2.4.5 Percentage of Dissatisfied Persons with Local Thermal Comfort

New models can define local thermal comfort, thermal sensation and perception of IAQ as a function of moist air temperature and relative humidity, under special conditions and considerations.

The effect of relative humidity, under local thermal comfort, was investigated by Toftum et al. (1998) [18]. In this research, 38 individuals were provided with clean air between 20 and 29°C and relative humidity between 45 and 70%.

The result of the percentage of PD is defined in Eq. 2.2.4.5.1. From this equation, we conclude that the percentage of PD tends to decrease with temperature. In accordance with the ASHRAE standard [5, 7, and 20], this percentage of PD must be below 15%.

$$PD = \frac{100}{1 + e^{(-3.58 + 0.18 \cdot (30 - t) + 0.14 \cdot (42.5 - 0.01 p_v)}} \tag{2.2.4.5.1}$$

These indices and models, used by different researchers in recent works, for example, Simonson et al. [7], were employed to define the effect of internal

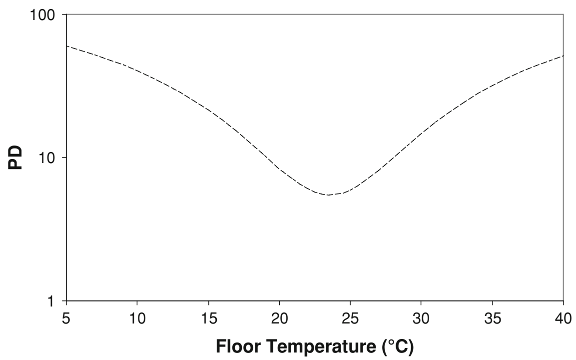


Fig. 2.9 Percentage of PD, depending on floor temperature

coverings on indoor thermal comfort and the perception of IAQ, and also to define better indoor conditions to reduce energy consumption.

2.3 Indoor Air Quality Models

2.3.1 Percentage of Dissatisfied with Indoor Air Quality

Fang et al. [20] obtained the index, appearing in Eq. 2.3.1.1, from the sensory response of subjects exposed to different combinations of temperature and relative humidity. The validity range of temperature and relative humidity is from 18 to 28°C and relative humidity from 30 to 70%.

$$PD_{IAQ} = \frac{\exp(-0.18 - 5.28\,\mathrm{Acc})}{1 + \exp(0.18 - 5.28\,\mathrm{Acc})} \cdot 100 \qquad (2.3.1.1)$$

PD_{IAQ} is a function of another index Acc (Eq. 2.3.1.2), with a scale ranging from +1 (clearly acceptable) to −1 (clearly unacceptable).

$$\mathrm{Acc}_{IAQ} = -0.033\,h + 1.662 \qquad (2.3.1.2)$$

where h is the indoor air enthalpy in kJ/kg.

2.4 Indoor Air Renovation Models

The largest number of complaints about air quality inside a building is within the area of thermal comfort and ventilation. According to the National Institute for Occupational Safety and Health, in more than 50% of the buildings studied, problems are caused by inadequate ventilation.

Thermal comfort is based on a balance between physical activities and clothing on one hand, and relative humidity, temperature, air velocity and radiant temperature on the other hand. ASHRAE have developed standards for confined spaces, which would ensure the comfort level of 90% of the population. In general, the acceptable range of values is relatively narrow, given the relationship that exists between variables. A slight increase in air velocity, for example, can trigger a series of complaints while the temperature remains within acceptable limits.

Similarly, when ventilation is incorrect, as a result of an insufficient supply of fresh air from outside, there may be a source of accumulation of various pollutants to levels that can annoy the occupants. The contribution of outside air should be sufficient enough to dilute the pollutants to levels that are below human perception, and those obviously considered harmful to health.

References since the middle of the eighteenth century recommended a minimum input of fresh air per person, to dilute the concentration of human and bioeffluents to avoid inconveniences due to bad odours.

For 70 years, ASHRAE published several articles recommending an injection of fresh air of at least 34 m^3/h per person to prevent odours and an absolute minimum of 8.5 m^3/h per person to maintain the concentration of carbon dioxide below 2,500 ppm, which is half the average permissible exposure in a work environment. Recently, ASHRAE Standard 62-1989 recommended a minimum of 25.5 m^3/h per person for classrooms, 34 m^3/h for offices and 42.5 m^3/h for hospitals (diseased area). This standard also recommends an increase in the volume when there are problems in the mixture of air breathing zones or unusual sources of pollution. On the other hand, we must remember that the primary purpose of an air conditioning system in an office building is to provide a good level of comfort.

Currently, there is a lot of interest in determining the ventilation rate in indoor environments. In this regard, there have been various models to determine the air change to link it with energy consumption (Cunningham [15]).

Research techniques using a tracer gas to detect flaws in the ventilation conditions are widespread. The tracer gas used in ventilation is often colourless, odourless and inert, and should normally not be present in the atmosphere. This section describes the different models and measuring procedures for air renovation in the indoor environments and their effects on local thermal comfort.

2.4.1 Techniques of Tracer Gas Monitoring

The techniques of tracer gas are the only ones that allow multiple types of quantitative assessments of ventilation, which include measures of air infiltration and renovation, the efficiency of extraction systems (foul air and smoke), as well as the dispersal of pollutants. It can also measure air velocities in ventilation ducts.

Another advantage of tracer gas techniques is the ability to perform actions in occupied environments. This is an effective and accurate method because it takes into account the large effect of occupation to assess the conditions for air renovation and the impact of opening and closing doors and windows. This will also take into account the enormous impact of occupation on the conditions for air renovation. The results of appropriate measures with tracer gas in a ventilation system will provide information on the amount of air introduced in each enclosure, the efficiency of heat recovery units, the amount of air taken from recycling, the "short circuit" effect of air related to outlet and inlet distribution. Both the planning stage and the lack of regular checks can cause a great increase in energy consumption and, in many buildings, one can see the effect of SBS by not taking these factors into account.

The flow of air through a building or room can be assessed by one of the three tracer gas methods: concentration decay method, constant concentration method and constant emission method. These three methods are based on the continuity equation, as seen in Eq. 2.4.1.1, and in its possible simplifications.

$$V\frac{\mathrm{dC}}{d\tau} = F(\tau) + N(\tau)\cdot C_{\mathrm{oa}} - N(\tau)\cdot C(\tau) \quad (2.4.1.1)$$

where

V is the volume of the air in the room (m^3),
C is the tracer gas concentration in the room air (m^3/m^3),
τ is the time (h),
F (τ) is the rate of introduction of tracer gas in the room (m^3/h),
C_{oa} is the concentration of tracer gas in outdoor air (m^3/m^3) and

N (τ) is the air flow through the room (m^3/h):

$$N(\tau) = \frac{F(\tau) - V\frac{\mathrm{dC}}{\mathrm{d}\tau}}{C(\tau) - C_{\mathrm{oa}}} \quad (2.4.1.2)$$

The air change rate, or number of renovations, N, is estimated by dividing the flow rate through the compound by its volume, assuming that $C_{\mathrm{oa}} = 0$.

(a) Concentration decay method
The concentration decay method is the most basic method used to measure air flow during renovations of discrete short periods. In this method, a small amount of tracer gas is completely mixed with the air. To ensure uniformity of the tracer gas concentration at all points in the environment, fans are employed. After this, the tracer gas concentration drops, and the time needed to reach a concentration value close to zero must be measured.
When there are no new issues of tracer gas and the air flow rate entry is constant, the concentration of tracer gas falls exponentially with time. Using the natural logarithm of gas concentrations over time, we get a line where the gradient is the rate of air change N (Eq. 2.4.1.3). If a line cannot be obtained, we must consider that the tracer gas does not mix well with the indoor environment and that the results are not valid.

$$N = \frac{\ln C(0) - \ln C(\tau_1)}{\tau_1} \quad (2.4.1.3)$$

Finally, the equipment needed is a gas monitor, a tracer gas source and fan(s) to homogenise the mix.

(b) Constant emission method
The constant emission method is employed for long periods of continuous measurement of air renovation in simple areas. Tracer gas is emitted at a constant speed during the measuring period. Therefore, if the renovation and the concentration of tracer gas are constant, the number of air changes per hour, N, will be in accordance with:

$$N = \frac{F}{V \cdot C} \tag{2.4.1.4}$$

The tracer gas concentration should be the same in all areas at any given moment.

(c) Constant concentration method
The constant concentration method is used to assess the ongoing renovation in occupied buildings. Using a gas monitor, tracer gas concentrations are measured in each zone. With this information, the dosage of tracer gas is controlled to maintain a constant concentration. A small fan can be used to facilitate the mixture. Air changes are also expressed in Eq. 2.4.1.4. The air renovation is proportional to the tracer gas emission required to maintain constant concentration. This approach has three advantages:

- It allows to get the exact average speed of change during long periods and in situations where the air change varies over time.
- It can be used to assess changes in specific areas.
- It is particularly appropriate to evaluate the continuous outside air infiltration in each individual zone; the exchange of unwanted air between different parts of a building can also be assessed.

2.5 Building Simulation Models

2.5.1 EN ISO 13790 Models

Recently, ISO 13790 proposed some calculation procedures to certify the energy consumption of buildings.

Currently, real energy consumption of buildings, with different levels of permeability of internal coverings, is used to obtain the main constants of the general equation of energy certification of buildings, in accordance with EN ISO 13790.

Particularly, the simplified monthly method of EN ISO 13790 standard will be presented. In this monthly method, the heat demand of the building Q_H is defined for each calculation period through:

$$Q_H = Q_L - \eta Q_G \tag{2.5.1.1}$$

where Q_L is the heat loss of the building, η is the utilisation factor of heat gains and Q_G is the total heat gain. The annual heat demand is the sum of the heat demand over the entire year and it is positive:

$$Q_{Ha} = \sum_{m=1}^{12} Q_{H,m} \tag{2.5.1.2}$$

ISO 13790 standard gives expressions to determine the utilisation factor for heating:

$$\eta_{G,H} = \frac{1 - \gamma_H^{a_H}}{1 + \gamma_H^{a_H}} \tag{2.5.1.3}$$

and for cooling :

$$\eta_{L,C} = \frac{1 - \gamma_C^{a_C}}{1 + \gamma_C^{a_C}} \tag{2.5.1.4}$$

The heat gain and loss ratios for heating and cooling periods, respectively:

$$\gamma_H = \frac{Q_{G,H}}{Q_{L,H}} \tag{2.5.1.5}$$

$$\gamma_C = \frac{Q_{L,C}}{Q_{G,C}} \tag{2.5.1.6}$$

In accordance with [21], the utilisation factor represents the portion of gains (during the heating season) or of losses (during the cooling season) that contribute to the reduction in the heating demand (during the heating season) or cooling demand (during the cooling season).

The non-utilised part of the gains (in winter) or the losses (in summer) depends on the dynamic mismatch between the gains and losses, which may cause overheating (above set-point temperature) in winter or under-cooling in summer. This utilisation factor depends on coefficients a_0 and time constant [22]:

$$a = a_0 + \frac{\tau}{\tau_0} \tag{2.5.1.7}$$

a_0 is a numerical parameter and τ_0 is the reference time constant that depends on the building category. For example, both for heating and cooling in office buildings, the following equation is proposed by the standard [23]:

$$a_H = a_C = 0.8 + \frac{\tau}{70} \tag{2.5.1.8}$$

Therefore, a_H and a_C are linearly correlated to the time constant of the building τ (Eq. 2.5.1.8).

The EN ISO 13790 standard gives default values for the parameters a_0 and τ_0. For example, for the monthly calculation of continuously heated buildings, the values are $a_0 = 1$ and $\tau_0 = 15\,\text{h}$. The value of these parameters can also be provided at national level; therefore, the suitability of the default values of a_0 and τ_0 was studied under Finnish conditions [24]. Other researchers [21] revealed that the differences between the buildings can be accounted by adding a parameter to Eq. 2.5.1.8. For example, Eq. 2.5.1.9 depicts a parameter that depends on the glazed area of the envelope, because highly glazed external envelopes yield a wide-ranging hourly profile of heat losses, leading to a decrease in the internal temperature below the cooling set-point, owing to the fact that this effect is not duly taken into account through the time constant of the building.

$$a_{\mathrm{C}} = 8 - 13\xi + \frac{\tau}{17} \tag{2.5.1.9}$$

where ξ is the ratio between the glazed area of the envelope and conditioned floor area.

It should be noted that, owing to the fact that this formulation was obtained through regional multiple-regression expressions, it is only suitable for Italian national buildings with medium heat capacity (147 $\text{kJ/m}^2\text{K}$), a time constant of 28 h and Italian climatic conditions. Hence, it must be adapted to each climate.

On the other hand, the time constant, referring to a mono-capacitor model of the building, is the time required for the internal minus external temperature difference to decrease in the absence of heat gains, considering a constant external temperature. This parameter, usually expressed in hours, quantifies the change in building internal temperature when submitted to a dynamic solicitation:

$$\tau = \frac{C}{H} \tag{2.5.1.10}$$

where C is the internal heat capacity of the building and H is the total heat loss coefficient of the building caused by transmission and ventilation heat losses.

2.5.2 Heat, Air and Moisture Tools Simulation Models

To solve the balance equations of room models, the balance equations were created from the individual Building Physics Toolbox [25, 26]. Heat, air and moisture (HAM) tools library is a Simulink model upgraded version of H-Tools with a similar structure, and specially constructed for thermal system analysis in building physics.

The library contains blocks for 1D calculation of HAM transfer through the building envelope components and ventilated spaces. The library is a part of IBPT-International Building Physics Toolbox and available for free downloading [27].

This library depicts two main blocks: a building envelope construction (walls and windows) and thermal zone (ventilated spaces), which are enclosed by the building envelope. Component models provide detailed calculations of the hydrothermal state of each subcomponent in the structure, according to the surrounding conditions to which it is exposed.

In Fig. 2.10, we see the main blocks employed for a building simulation, in which we see a block that represents different exterior/interior walls, floor, roof and windows components. These constructions are defined with respect to their physical properties (density of dry material and open porosity), thermal properties (specific heat capacity of the dry material and thermal conductivity) and moisture properties (sorption isotherm, moisture capacity, water vapour permeability and liquid water conductivity), in accordance with the BESTEST structure. Other parameters, for example, internal gains (convective, radiative and moisture gains), air changes and heating/cooling system are considered in the heat and moisture building balance.

The mathematical model employed in these simulations is the result of whole building HAM [28–30] balance, and depends on moisture generated from occupant activities, moisture input or removal through ventilation, and moisture transported and exchanged between indoor air and the envelope [31].

The mathematical model is based on the numerical resolution of the energy and moisture balance through the building. In accordance with the following equations [29], the heat flow depicts a conductive and a convective part:

$$q = q_{\text{conductive}} + q_{\text{convective}} \tag{2.5.2.1}$$

$$q_{\text{conductive}} = -\lambda \frac{\partial T}{\partial x} \tag{2.5.2.2}$$

$$q_{\text{convective}} = m_{\text{a}} \cdot c_{\text{p}_\text{a}} \cdot T + h_{\text{evap.}} \tag{2.5.2.3}$$

where λ is the thermal conductivity (W/mK), T is the temperature (°C), m_{a} is the density of moisture flow rate of dry air (kg/m^{2}s), c_{pa} is the specific heat capacity of the dry air (J/kg K) and $h_{\text{evap.}}$ is the latent heat of evaporation (J/kg).

The moisture flow transfer was separated in liquid and vapour phases (Eqs. 2.5.2.4, 2.5.2.5).

$$m_{\text{l}} = K \cdot \frac{\partial P_{\text{suc}}}{\partial x} \tag{2.5.2.4}$$

where m_{l} is the density of moisture flow rate of vapour phase (kg/m^2\,s), K is the hydraulic conductivity and P_{suc} is the suction pressure (P_{a}).

The vapour phase was divided into diffusion and convection:

$$m_{\text{v}} = -\delta_{\text{p}} \cdot \frac{\partial p}{\partial x} + m_{\text{a}} \cdot x_{\text{a}} \tag{2.5.2.5}$$

where δ_{p} is the moisture permeability (s) and

x_{a} is the water vapour content (kg/kg).

The mass airflow through the structure driven by air pressure differences is described by:

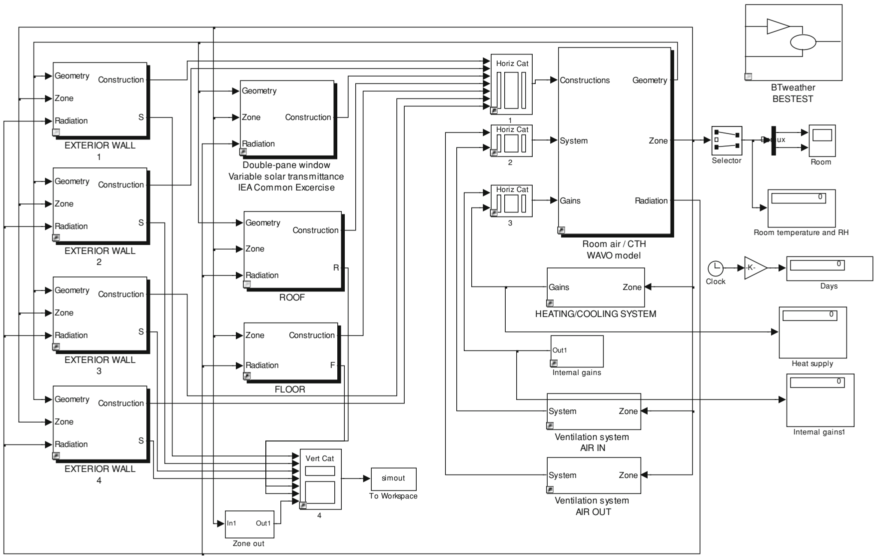


Fig. 2.10 MATLAB blocks for building simulations

$$m_a = r_a \cdot \rho_a \tag{2.5.2.6}$$

where r_a is the density of the air flow rate (m^3/m^2s) and
ρ_a is the density of the material (kg/m^3).
The final energy and moisture balance is revealed through:

$$-\frac{\partial}{\partial x}q = c \cdot \rho_0 \cdot \frac{\partial T}{\partial t} \tag{2.5.2.7}$$

$$-\frac{\delta}{\partial x}m = \frac{\partial w}{\partial t} \tag{2.5.2.8}$$

where ρ_o is the density of the dry material (kg/m^3),
c is the specific heat capacity of the material (J/kg K),
w is the moisture content mass by volume (kg/m^3),
t is the time (s) and
x is the space coordinate (m).

Finally, the numerical model, based on a control volume method, lumps the thermal capacity C in the middle of the total thickness d/2 and, consequently, the thermal resistances for one half are:

$$R = \frac{d/2}{\lambda} \tag{2.5.2.9}$$

$$R_{\mathrm{p}} = \frac{d/2}{\delta_{\mathrm{p}}} \tag{2.5.2.10}$$

$$R_{\mathrm{suc}} = \frac{d/2}{K_{\mathrm{suc}}} \tag{2.5.2.11}$$

The obtained heat and moisture balance equations are:

$$\begin{aligned}\frac{T_i^{n+1}}{\Delta t} = \frac{1}{C^n} \cdot \left\{ \left[\frac{(T_{i-1} - T_i)}{R_{i-1} + R_i} + \frac{(T_{i+1} - T_i)}{R_{i+1} + R_i} \right] - h_{\mathrm{evap}} \cdot \left[\frac{(p_{i-1} - p_i)}{R_{p,i-1} - R_{p,i}} + \frac{(p_{i+1} - p_i)}{R_{p,i+1} - R_{p,i}} \right] \right\} \dots \\ + \left\{ \begin{array}{l} m_a \cdot c_{pa} \cdot (T_{i-1} - T_i)^n, \; m_a > 0 \\ m_a \cdot c_{\mathrm{pa}} \cdot (T_1 - T_{i+1})^n, \; m_a < 0 \end{array} \right\}\end{aligned} \tag{2.5.2.12}$$

$$\begin{aligned}\frac{w_i^{n+1} - w_i^n}{\Delta t} = \frac{1}{d} \cdot \left\{ \left[\frac{(p_{i-1} - p_i)}{R_{p,i-1} + R_{p,i}} + \frac{(p_{i+1} - p_i)}{R_{p,i+1} + R_{p,i}} \right] - \left[\frac{(p_{\mathrm{suc},i-1} - p_{\mathrm{isuc},i})}{R_{\mathrm{suc},i-1} - R_{\mathrm{suc},i}} + \frac{(p_{\mathrm{suc},i+1} - p_{\mathrm{suc},i})}{R_{\mathrm{suc},i+1} - R_{\mathrm{suc},i}} \right] \right\} \dots \\ + \left\{ \begin{array}{l} 6.21.10^{-6} \cdot m_a \cdot (p_{i-1} - p_i)^n, \; m_a > 0 \\ 6.21.10^{-6} \cdot m_a \cdot (p_i - p_{i+1})^n, \; m_a < 0 \end{array} \right\}\end{aligned} \tag{2.5.2.13}$$

where i is the objective node, $i - 1$ and $i + 1$ are the preceding and following nodes and n and $n + 1$ the previous and corresponding time steps.

2.5.3 Building Time Constant Models

As seen before, the time constant is normally found from a slow cooling period with a constant low outdoor temperature such as heat capacity/heat loss factor [32]. This method is based on a seasonal steady state energy balance on the building as a whole or on a particular building zone. The thermal inertia is introduced in terms of the utilisation factor that shows the part of energy gains (solar irradiation and others) that can be stored in the building materials to be transmitted into the zone when needed.

The utilisation factor η is a function of the periodic time constant of the building and the ratio $Q_{\mathrm{gain}}/Q_{\mathrm{loss}}$. The time constant is defined in the standard, as it was showed earlier in Eqs. 2.5.1.1 and 2.5.1.10.

As [32] recommended, when we want to work in a more precise way, the logarithm of the temperature difference of indoors/outdoors is taken and matched to a straight line by the method of least squares. The time constant is the inverse of the coefficient for the independent variable (time) given by this curve fit.

2.5.4 Material Properties Models

2.5.4.1 Moisture Storage Capacity

Real moisture storage capacity, when the indoor air relative humidity changes, is a new parameter that must be defined by researchers in order to obtain the adequate building materials' behavior prediction. In this sense, different authors [7, 33] have developed new methods centred in Eq. 2.5.4.1.1.

$$\mathrm{Cm} = (u_{60\%\,\mathrm{Rh}} - u_{40\%\,\mathrm{RH}})\frac{\rho V 1000}{20} \tag{2.5.4.1.1}$$

where;

C_m is the moisture capacity (g/%RH),
u is the moisture content (kg/kg),
ρ is the density of the material (k/m^3) and
V is the volume of the material (m^3).

On the other hand, the moisture diffusivity can be calculated in an analogous manner to thermal diffusivity by Eq. 2.5.4.1.2.

$$\alpha_m = \frac{\mathrm{kd}}{\mathrm{Cm}/(1000V)\ 100/P} \tag{2.5.4.1.2}$$

where;

P_{vsat} is the saturation pressure for water vapour at 22°C(Pa),
Kd is the water vapour permeability (kg/(s m Pa)),
Cm is the moisture capacity (g/%RH) and
α_m is the moisture diffusivity (m^2/s).

Finally, we must consider that the definition of moisture diffusivity from Eq. 2.5.4.1.3 neglects the moisture storage in the air within the porous material [35, 36]. This effect is negligible for most hygroscopic materials.

$$\alpha_m = \frac{\mathrm{kd}}{\mathrm{Cm}/(1000V)\ 100/P_{\mathrm{vsat}}} \tag{2.5.4.1.3}$$

where;

P_{vsat} is the saturation pressure for water vapour at 22°C(Pa),
k_d is the water vapour permeability (kg/(s m Pa)),
c_m is the moisture capacity (g/%RH) and
α_m is the moisture diffusivity (m^2/s).

Based on these methods, researchers showed that concrete moisture buffering influenced the indoor air conditions with a moisture capacity that is half of the wooden structures and more results will be showed in our case studies.

2.5.4.2 Vapour-Driving Potential

As a result of the vapour-driving potential, the second term of the model of Eq. 2.5.4.2.1 includes another moisture source of indoor air. The reasons to analyse indoor air based on the partial vapour pressure were explained by [37]. They concluded that the driving potential for vapour transport is the difference in vapour pressure on the surfaces (Eq. 2.5.4.2.2).

They also concluded that as the moisture is transported through a finite volume in a medium, and the amount of moisture retained by the volume is altered during any transient stage of the transport process. The basic reason for this is a change in local temperature or vapour pressure.

There is even doubt whether the vapour pressure is the driving force for diffusion through walls when there is no air pressure difference. In the building literature, it is often assumed that vapour pressure is the defining variable when discussing water movement in walls; but there is evidence that it is the humidity difference that drives diffusion through absorbent materials.

For these reasons, vapour partial pressure has been used for determining an index that allows comparison of the effect of coverings by means of equations defined by ASHRAE. The uncertainty of the calculated vapour pressure was ±0.07 kPa. This index will be used to consider the excesses of the indoor versus outdoor partial vapour pressure, following Hens' work [38].

$$V\frac{\mathrm{d}c_{\mathrm{in}}}{\mathrm{d}\tau} = G + M + nV(c_{\mathrm{out}} - c_{\mathrm{in}}) \qquad (2.5.4.2.1)$$

where;

τ is the time (s),
V is the volume of the room, m^3
N is the air ventilation rate, s^{-1}
c is the water vapour content, $\mathrm{kg/m}^3$
G is the moisture generation rate, kg/s
M is the sum of moisture quantities contributed by buildings components, kg/s.

$$J_{\mathrm{v}} = -\mu \cdot \mathrm{gradp} \qquad (2.5.4.2.2)$$

where;

μ is the water vapour permeability of the medium (kg/(s m Pa)),
J is the water vapour flux density ($\mathrm{kg/m}^2$ s)
Gradp is the driving potential (Pa/m).

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