Analysis of thermal and olfactory comfort in closed spaces

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Abstract: It is well known that the prediction of comfort in buildings is performed either by direct measurement of each microclimate parametre, or by some comfort indices which synthesize the combined influence of more environmental factors implied. This paper approaches the numerical prediction of thermal comfort in closed spaces on the basis of PMV – PPD model, and its testing to asymmetric or nonuniform thermal radiation, as well as the indoor air quality control. It is developed a computation and testing model of thermal comfort in buildings, as well as a methodology to determine the outside airflow rate and to verify the indoor air quality in rooms, according to the European Standard CEN 1752. Also, it is presented the thermal comfort criteria for design of heating systems. On the bases of these mathematical models there were elaborated the COMFORT 1.0 and COMFORT 2.0 computer programs, implemented on compatible microsystems IBM-PC. The COMFORT 1.0 computer program allows both for the direct computation of the PMV (predicted mean vote) and PPD (predicted percent dissatisfied) indices in different points of a room, and their comparative analysis, and the determination of the mean radiant temperature in isolated points or in a series of points situated on a straight line. The COMFORT 2.0 program computes the outside airflow rate for a room ventilation, the number of air exchanges per hour, and the variation in time of contaminants concentration of room air according to European and national norms and analyses influence of different parametres on these sizes. The performance of the developed computational models and the advantages of the proposed computer programs is illustrated by using some numerical comparative applications for two constructive types building.

Key-Words: Closed spaces, Thermal comfort prediction, Olfactory comfort analysis, Outside airflow rate, Indoor air quality control, Mathematical models, Computer programs, Numerical comparative analysis.

1 Introduction

The greatest majority of people carry on 80…90% of their lives inside buildings, which must satisfy the objective and subjective requests linked to vital functions of the occupants. That is why the closed spaces must insure the possibility for both physical and intellectual work, as well as for some recreation activities, for rest and sleep under most favourable conditions. The achievement of these conditions depends on very many factors that decisively influence the sensation of comfort perceived, the work capacity and man's regeneration capacity. The design of the rooms must take into consideration these conditions and present ten-dencies to reduce the energetic consumption, that are decisively influencing the optimal or admissible values of comfort parameters. Thus the inside microclimate of a building must be the result of a computation of multicriterial optimization, taking into account technical and psychological comfort and the energy saving.

The concept of technical comfort comprises all parameters achieved and controlled with HVAC systems that act upon building occupants senses. This includes thermal, acoustic, olfactory and visual comfort. In accordance with the dissatisfied person percent of the ensured comfort: 10%, 20%, 30%, rooms are classified into three categories: A, B, C.

Subjective comfort of persons in a room depends on many factors: temperature, humidity and air cir-culation; smell and respiration; touch and touching; acoustic factors; sight and colours effect; building vibrations; special factors (solar-gain, ionization); safety factors; economic factors; unpredictable risks. Because of some technical conditions the common influence of these factors can not be analysed, and the adaptation of the human body to a certain environment is a complex process, this one reacting to the common action of more parameters.

In general nationally specified criteria for design of heating systems must be used, but in case of no national regulations international standards [10], [14], [21] gives values for thermal comfort in infor-mative annexes. The recommended criteria are given for general thermal comfort based on PMV-PPD model or operative temperature and for local thermal comfort parameters like vertical temperature differences, radiant temperature asymmetry, draft and surface temperatures.

2 Prediction of thermal comfort

The subjective sensation of thermal comfort is decisively determined by the following parameters [26]: indoor air temperature (t_i); mean radiant temperature (t_{mr}) of bordering surfaces; relative humidity of air (φ_i); partial water vapours pressure (p_a); air velocity (v_i); thermal resistance of clothing (R_{cl} or R_h), and their influence on the vaporization; heat production of human body and human thermoregulation. The first four are physical parameters, and the other two expresses the capacity of the human body to adapt itself in order to maintain the thermal equilibrium. The main factors that influence the thermal equilibrium of the human body are:

– heat production of human body, which depends on the activity level, age, sex, etc.

– body heat loss, which depends on clothing and on the other parameters mentioned previously.

In general, comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized. Surprisingly, although climates, living conditions, and cultures differ widely throughout the world, the temperature that people choose for comfort under similar conditions of clothing, activity, humidity, and air movement has been found to be very similar [8], [11], [16].

In order to evaluate the sensation of thermal comfort we use the thermal sensation scale with seven levels [2]: +3 (hot); +2 (warm); +1 (slightly warm); 0 (neutral); –1 (slightly cool); –2 (cool); –3 (cold).

Numerical prediction of thermal comfort in a room is performed by using the PMV – PPD model, and testing is achieved at asymmetric thermal radiation, caused by building elements with a temperature t_{el} lot different from the mean radiant temperature t_{mr}. Radiant asymmetry is the difference in radiant temperatures seen by a small flat element looking in opposite directions. Four calculation methods of radiant temperature asymmetry are available in the technical literature [19].

2.1 Mathematical model

Taking into account the thermal interaction of the human body with its environment it is possible to write the well known energy balance equation:

$$M - W = Q_i = (C + R + E_{sk}) + (C_{res} + E_{res}) \qquad (1)$$

where: M is the rate of metabolic heat production; W – rate of mechanical work accomplished (zero for most of activities); Q_i – internal heat production; C, R – convective and radiant heat losses outer surface of a closed body; E_{sk} – rate of evaporative heat loss from skin; C_{res} – rate of evaporative heat loss from respiration; E_{res} – rate of convective heat loss from respiration.

All term in the basic heat balance equation are expressed per unit body nude surface area.

• Convective heat loss C is given by equation (2) and radiant heat loss R is expressed in terms of the Stefan-Boltzmann law (3):

$$C = f_{cl}\alpha_c(t_h - t_i) \qquad (2)$$

$$R = 3.96 \cdot 10^{-8} f_{cl}\left[(t_h + 273)^4 - (t_{mr} + 273)^4\right] \qquad (3)$$

where:

$$f_{cl} = \frac{A_h}{A_D} \qquad (4)$$

$$A_D = 0.202\, m^{0.425} h^{0.725} \qquad (5)$$

in which: t_i is the indoor air temperature; t_h – mean temperature of the outer surface of the closed body; t_{mr} – mean radiant temperature; α_c – convective heat transfer coefficient; f_{cl} – clothing area factor dimensionless; A_h – actual surface area of the clothed body; A_D – nude body surface area [13]; m – mass of human body; h – height of human body.

The sensible heat (C+R) is transferred through conduction from the skin surface to the outer clothing surface:

$$C + R = Q_s = \frac{t_p - t_h}{R_h} \qquad (6)$$

where:

$$R_h = 0.155 R_{cl} \qquad (7)$$

in which: Q_s is the sensible heat loss from skin to outer clothing surface; R_h – thermal resistance of clothing, in (m^2·K)/W; R_{cl} – thermal resistance of clothing, in clo; t_p – skin temperature expressed by:

$$t_p = 35.7 - 0.032(M - W) \qquad (8)$$

• Evaporative heat loss from skin E_{sk} may be computed with the equation:

$$E_{sk} = E_d + E_{rsw} \qquad (9)$$

where:

$$E_d = 0.35(1.92 t_p - 25.3 - p_a) \qquad (10)$$

$$E_{rsw} = 0.42(M - W - 58) \qquad (11)$$

in which: E_d is the evaporative heat loss by diffusion of water through the skin; E_{rsw} – evaporative heat loss by regulatory sweating; p_a – water vapour partial pressure in indoor air.

• Respiratory heat loss is often expressed in terms of sensible C_{res} and latent E_{res} heat losses. The C_{res} and E_{res} can be computed as follows:

$$C_{res} + E_{res} = 0.0014M(34 - t_i) + \\ + 1.73 \cdot 10^{-5} M(5870 - p_s) \quad (12)$$

where p_s is the saturated water vapour pressure at the humid operative temperature.

The body's mechanical efficiency is defined by:

$$\eta = \frac{W}{M} \quad (13)$$

The energy balance equation (1) is under the form:

$$Q_{ac} = M - W - E - (C_{res} + E_{res}) - (C + R) \quad (14)$$

where Q_{ac} is the accumulated heat in body.

If Q_{ac} = 0, thermal comfort felt by occupants is adequate. When Q_{ac} > 0, body temperature rises and the person will have a sense of warmth, and if Q_{ac} < 0, body temperature decreases and the person will have a sense of coldness.

Writing the energy balance equation (14), for Q_{ac} = 0, in the form:

$$M - W - E - (C_{res} + E_{res}) = Q_s = C + R \quad (15)$$

and explaining the terms in it with the equations (2), (3), (6), (9), (12), we obtain the thermal comfort equation:

$$M(1-\eta) - 0.35[43 - 0.061M(1-\eta) - p_a] - \\ - 0.42[M(1-\eta) - 58] - 0.0014M(34 - t_i) - \\ - 0.0023M(44 - p_a) = \frac{35.7 - 0.032M(1-\eta) - t_h}{0.155TR_{cl}} = \quad (16) \\ = f_{cl}\alpha_c(t_h - t_i) - 3.96 \cdot 10^{-8} f_{cl}[(t_h + 273)^4 - \\ - (t_{mr} + 273)^4]$$

Considering some parameters as constant in the equation (16), after it is solved and the solutions graphically represented the comfort diagrams are obtained [25]. Substituting the thermal comfort equation (16) into expression of PMV index established by Fanger [15], the predicted mean vote index is obtained, for a certain point of the room:

$$PMV = [0.352\exp(-0.042M) + 0.032] \times \\ \times \{M(1-\eta) - 0.35[43 - 0.061M(1-\eta) - p_a] - \\ - 0.42[M(1-\eta) - 58] - 0.0023M(44 - p_a) - \quad (17) \\ - 0.0014M(34 - t_i) - f_{cl}\alpha_c(t_h - t_i) - \\ - 3.96 \cdot 10^{-8} f_{cl}[(t_h + 273)^4 - (t_{mr} + 273)^4]\}$$

PMV index has the optimum value equal to zero, but according to the prescriptions ISO Standard 7730 it is considered that the domain of thermal comfort corresponds to values between –0,5 and +0,5. The use of PMV index is recommended only for values between +2 and –2. In the Figure 1 are represented the values of operative comfort temperature t_c (corresponding to index PMV = 0), correlated to thermal resistance of clothing (R_{cl} and R_h), metabolic rate i_M and metabolic heat production M.

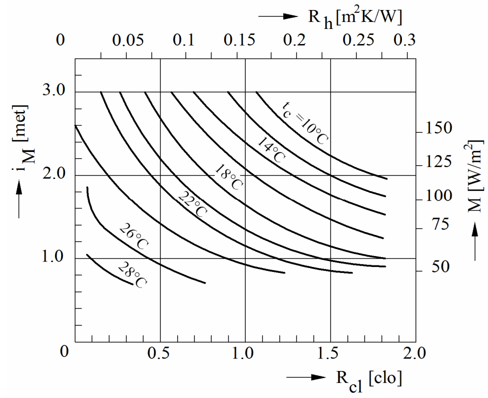

Fig. 1 Operative comfort temperature function of clothing and activity

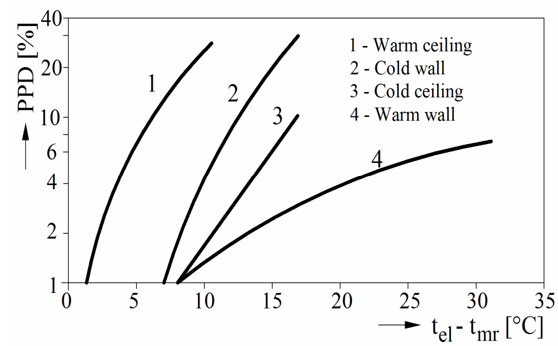

Fig. 2 Variation of percent dissatisfied function of radiant temperature asymmetry

The predicted percent dissatisfied PPD is function of PMV index as follows:

$$PPD = 100 - 95\exp(-0{,}0335PMV^4 - 0{,}2179PMV^2) \quad (18)$$

The optimal value for PPD index is of 5 % and may be obtained only using air conditioning systems, with a high automation degree. A relationship between the radiant temperature asymmetry and the percent dissatisfied was established (Fig. 2). Figure 2 show that people are sensitive, to asymmetry caused by an overhead warm surface than by a vertical cold surface. These data are particularly important when using radiant panels to provide comfort in spaces with large cold surfaces or cold windows.

2.2 Computer program COMFORT 1.0

Computer program COMFORT 1.0 allows both direct computation of PMV and PPD indices in different points of a room, and their comparative analysis. It is also possible to determine the mean radiant temperature in isolated points or in a series of points situated on a straight line.

- The inputs are: geometrical dimensions of a room, in m; thermal transfer coefficients of the bordering building elements, in W/(m^2·K); air temperature out of neighbouring rooms, in ^oC; thermal power of heating source, in W; indoor air temperature, in ^oC; air pressure, in Pa and air relative humidity; air velocity, in m/s; clothing thermal resistance, in clo; metabolic rate, in met; mass in kg and height in m, for occupant.

- The main outputs, in tabular form, are PMV, PPD indices and mean radiant temperature or their variation that may be printed or saved in files.

2.3 Numerical application

The room with geometrical dimensions of 4.4×6×2.7 m from Figure 3 is considered. The following data are known: – heat transfer coefficient of building components: walls [0.7 W/(m^2·K)], ceiling [0.4 W/(m^2·K)], windows and doors [2.9 W/(m^2·K)]; – glass walls surface: 7.5 m^2; – indoor air temperature: 24 ^oC; – thermal power of heater: 1900 W.

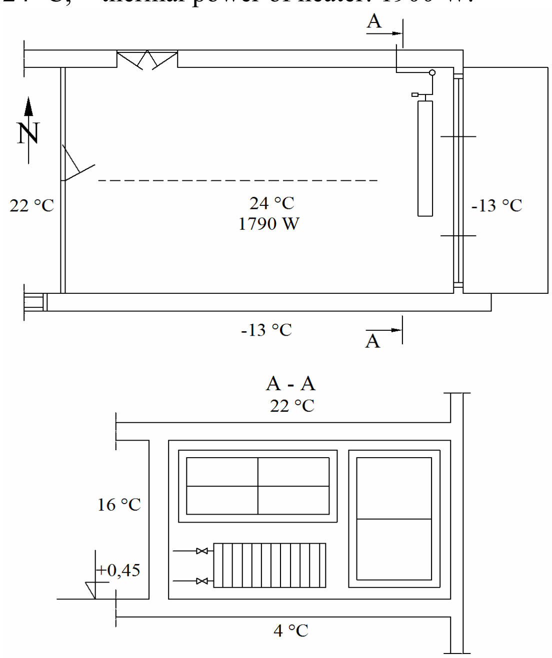

Fig. 3 Room heated.

A comparative study of PMV and PPD indices is performed using the computer program COMFORT 1.0 in several points situated on a straight line (discontinuous), at different distances from the window, function of clothing thermal resistance and metabolic rate.

Results of the numerical solution obtained for the following pair of values: 0.33 clo – 1 met, 1 clo – 1 met, 0.5 clo – 2.8 met, are reported in Table 1. According to the performed study it was established that PMV index has values closed to zero only for the pair of values 1 clo – 1 met. For any other pair of values $R_{cl} - i_M$ the percent dissatisfied people of the thermal comfort would be greater, of 5%.

3 Thermal comfort criteria for design of heating systems

3.1 Criteria for general thermal comfort

For the design of buildings and HVAC systems the thermal comfort criteria (minimum room temperature in winter, maximum room temperature in summer) and required ventilation rates for an acceptable indoor air quality shall be used as input for heating load. The basis for establishing the criteria is ISO Standard 7730 and the use of the PMV-PPD indices, as shown in table 2.

Based on specified type of clothing and activity of the occupants it is possible to calculate the corresponding ranges of operative temperature. As an example, thermal design criteria for different types of space with sedentary activity and typical winter clothing are given in table 2.

The heat emission system must fulfil the requirement that the difference between the operative temperatures in the warmest position in the space must be inside the chose temperature range (3…5 K). Especially in a very deep room with a fully glazed façade the difference may be critical.

By a simplified design method [23] showed that the maximum operative temperature difference in a room at an outdoor air temperature of -12 °C can be calculated according to the equation:

$$t_{c2} - t_{c1} \le 0.96\, U_w \qquad (19)$$

where: t_{c1} is the operative temperature at the coldest position, in °C; t_{c2} – operative temperature at the warmest position, in °C; U_w – average U-value of the facade, in W/(m^2·K).

For a typical moderen standard window (U = 1.5 W/(m^2·K)) the difference in operative temperature is lower than the criteria, because additional variation in operative temperature will be caused by the control system. If the difference is too large it will be necessary to position a heat emitter at the façade (radiator, floor heating, convector), change the design of the façade, choose a higher insulation of the façade.

3.2 Criteria for local thermal comfort

Criteria for draft, vertical air temperature difference, radiant asymmetry and surface temperatures will also impact the design of buildings and HVAC systems [24]. These criteria are included in ISO 7730.

- *Vertical air temperature differences*. One of the main features of radiant floor heating and cooling is the uniform temperature conditions from floor to ceiling.

Table 1. Numerical results of COMFORT 1.0 computer program.

Distance from the window [m]	0.33 clo – 1 met			1 clo – 1 met			0.5 clo – 2.8 met		
	t_{mr} [°C]	*PMV*	*PPD* [%]	t_{mr} [°C]	*PMV*	*PPD* [%]	t_{mr} [°C]	*PMV*	*PPD* [%]
0	1	2	3	4	5	6	7	8	9
1.0	24.43	–1.22	36.28	24.43	–0.18	5.64	24.43	–0.70	15.39
1.5	24.86	–1.18	34.37	24.86	–0.13	5.38	24.86	–0.63	13.42
2.0	25.34	–1.09	30.18	25.34	–0.07	5.09	25.34	–0.56	11.46
2.5	25.67	–1.03	27.45	25.67	–0.02	5.01	25.67	–0.50	10.26
3.0	25.78	–1.01	26.56	25.78	–0.01	5.00	25.78	–0.48	9.89
3.5	25.70	–1.03	27.20	25.70	–0.02	5.01	25.70	–0.50	10.16
4.0	25.63	–1.04	27.77	25.63	–0.03	5.01	25.63	–0.51	10.41
4.5	25.55	–1.05	28.43	25.55	–0.04	5.03	25.55	–0.52	10.68
5.0	25.37	–1.09	29.92	25.37	–0.06	5.08	25.37	–0.55	11.35

Table 2. Examples of recommended categories for design of mechanical heated and cooled buildings

Type of building or space	Category	PPD [%]	PMV	Temp. range for heating, °C (R_{cl}=1 clo)	Temp. range for cooling, °C (R_{cl}=1 clo)
0	1	2	3	4	5
Offices and spaces with similar activity (single offices, open-plan offices, conference rooms, auditoriun, cafeteria, restaurants, classrooms); i_M = 1.2 met	I	<6	-0.2<PMV<+0.2	21.0 – 23.0	23.5 – 25.5
	II	<10	-0.5<PMV<+0.5	20.0 – 24.0	23.0 – 26.0
	III	<15	-0.7<PMV<+0.7	19.0 – 25.0	22.0 – 27.0

Measurements show that floor heating large panel radiators under the window have a very uniform temperature profile. The more convective systems Baseboard under window, Warm air systems, Back wall) or high temperature water systems have larger differences in air temperature. The standard [21] requires the vertical temperature difference lower than 3 K between feet and head (category II) and below 2 K (category I).

• *Radiant temperature asymmetry*. People is most sensitive to radiant asymmetry caused by warm ceiling or cool walls – windows. Difficulties may occur by a temperature asymmetry of 5 K for warm ceiling and 10 K for cold walls (category II [21]). Usually, the radiators/convectors are placed under the window in order to avoid the problems. Due to the development of high isolative windows this is however, normally not a problem anymore. Also, the maximum radiant temperature asymmetry can be calculated at the design stage [23] by the equation:

$$\Delta t_{mr} < 3.96\,U_w \qquad (20)$$

This means that if the façade is a standard double glassing, U = 2.9 W/(m^2·K), the asymmetry will be 11.54 K, which is higher that the category II criteria of 10 K. For a typical modern standard window, U = 1.5 W/(m^2·K), the asymmetry will be less than 6 K.

• *Draft*. Downdraft from cold surfaces is another factor that may cauise discomfort and require a heated surface under the cold wall/window. On the basis of a calculation method, the relation between the window hight, U-value for the wall/window, outdoor air temperature and the maximum acceptable air velocity [21] can be determined [4].

An example is shown in figure 4, where the maximum air velocity in the occupied zone 1 m from a cold vertical surface (wall, window) is shown as a function of window-wall height and U-value by an outdoor air temperature of -12 °C.

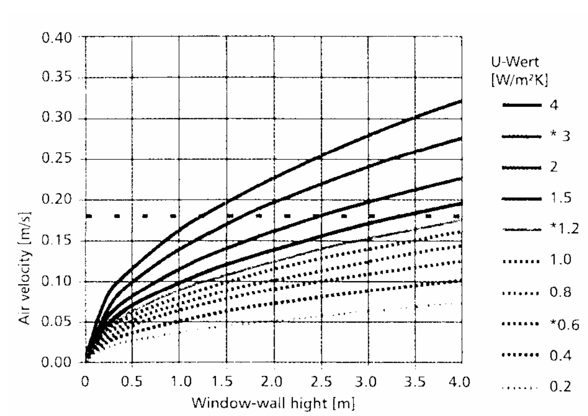

Fig. 4 Variation of maximum air velocity in function of window-wall height and U-value

For category II the criteria for the air velocity, taken into account the calculated lower air tempe-rature and an assumed turbulence intensity of 20%, is 0.18 m/s. This means that if the window is a standard double glassing, U = 2.9 W/(m^2·K), the window should not be higher than 1.5 m. For a typical modern standard window, U = 1.5 W/(m^2·K), the height could be full room height, 3.5 m.

- *Floor surface temperature*. In international standards, a floor temperature range of 19 °C to 29 °C is recommended in the occupied zone for rooms with sedentary and/or standing occupants wearing normal shoes. This is a limiting factor for the ca-pacity of floor systems.

For heating, the maximum temperature is 29 °C, which at a design indoor air temperature of 20 °C will provide a heat output of 100 W/m. Seated persons would prefer 1 K higher floor temperatures and standing persons 1 K lower surface temperatures. In the standard EN 15377-1, it is acceptable to use 35 °C as the design floor temperature outside the occupied zone i.e. perimeter until 1 m from façade.

- *Wall temperature*. For wall heating, the maximum lies in the range 35…50 °C. The maximum may depend on the application of the wall heating system. The risk for burns and pain is a skin temperature of 42…45 °C and depends on the conductivity of the surface layer. Surface temperature of a radiator may also be limited. In some countries it is limited to 55 °C. the difference to a wall heating is that the radiator is invisible and people can be warned.

- *Ceiling temperature*. Requirements to the ceiling surface temperature should not cause a too high radiant temperature asymmetry. The radiant asymmetry depends on the angle factor from a small plane element to the ceiling and the ceiling temperature.

For a room with the dimensions 2.4 m × 4.8 m and 2.7 hight, the angle factor is 0.42 between the ceiling and a sitting person at the centre of the room. For heating, it is assumed that all other surfaces than the heated ceiling has a temperature equal to a design indoor air temperature of 20 °C. The radiant asymmetry from a heated ceiling must then be less than 5 K:

$$0.42 \times t_{ceiling} + (1 - 0.42) \times 20\,°\mathrm{C} - 20\,°\mathrm{C} < 5\,\mathrm{K} \quad (21)$$

This means, that the limitation is a maximum average ceiling temperature of 32 °C.

4 Olfactory comfort

Comfort and indoor air quality (IAQ) depend on many factors, including thermal regulation, control of internal and external sources of pollutants, supply of acceptable air, occupant activities and preferences, and proper operation and maintenance of building systems. Ventilation and infiltration are only part of the acceptable indoor air quality and thermal comfort problem.

Air composition in living spaces differs from that of the outside air. Carbon dioxide (CO_2) concentration in outside air is between 300 and 400 ppm, and in living spaces is of about 900 ppm. The maximum admitted limit of CO_2 concentration in the inhaled air is of 1000 ppm (Pettenkofer's number). Table 3 presents the effect of different CO_2 concentrations on human body.

Table 3. The effect of CO_2 concentration on human body

CO_2 concentration		Effect
[%]	[ppm]	
3	30000	Deep breathing, strong
4	40000	Head aches, pulse, dizziness, psychic emotions
5	50000	After 0.5…1 hours may cause death
8...10	80000...100000	Sudden death

Air quality is prevailingly determined by people's sensations to different odorants.

Odorant perception depends, on one side, on objective factors: concentration and toxicity of air pollutants (bio-effluents), activity level, outside airflow rate, and on the other hand, on psychological factors with subjective character.

The relation between perceived odorant intensity and its concentration conforms to a power function [29]:

$$S = k\,C^{\beta} \quad (22)$$

where: S is odorant intensity (magnitude); C – odorant concentration, in ppm; β – exponent (0.2…0.7) of psychophysical function; k – constant characteristic of material.

The percentage of persons dissatisfied PPD with air polluted by human bio-effluent (1 olf) can be calculated from the equations [18]:

$$PPD = 395\ \exp\left(-3.66 L_p^{0.36}\right) \quad \text{for } L_p \ge 0.32\ \mathrm{l/s} \quad (23)$$

$$PPD = 100 \quad \text{for } L_p < 0.32\ \mathrm{l/s} \quad (24)$$

where: L_p is the outside airflow rate, in l/s.

Outdoor air requirement for acceptable indoor air quality have long been debated. Historically, the major considerations have included the amount of outdoor air required to control moisture, carbon dioxide and tobacco smoke generated by occupants. These considerations have led to prescriptions of a minimum rate of outdoor air supply per occupant.

Tables 4 and 5 present the minimum rate of outdoor airflow per occupant for different activities, according to European Standard CEN 1752, that also takes into consideration the smokers in ventilated rooms.

Table 4. Minimum rate of outdoor airflow

No.	Activity	Outdoor airflow rate [m^3/(h·pers)]
0	1	2
1	Intellectual	30
2	Physical very easy	30
3	Physical easy	40
4	Physical hard	50

Table 5. Minimum rate of outdoor airflow for rooms with smokers

Room category	Outdoor airflow rate [m^3/(h·pers)]			
	Without smokers	20% smokers	40% smokers	100% smokers
0	1	2	3	4
A	36.0	72.0	108.0	108.0
B	25.2	50.4	75.6	75.6
C	14.4	28.8	43.2	43.2

The perceived olfactory sensation depends not only on the pollutant source but also to a great extend, on the dilution degree with outside air.

The olfactory pollution degree of a room is given by:

$$C_i = C_p + \frac{10\,G}{L_p} \qquad (25)$$

where: C_i is the indoor air quality (in decipol); C_p – outdoor air quality (in decipol); G – contaminants concentration of room air.

5 Computation of outside airflow rate and indoor air quality control

5.1 Mathematical model

Computation of outside airflow rate in a room can be performed function of:

- number of occupants, keeping CO_2 concentration under the maximum admitted level (according to Romanian Norm I 5);
- number of occupants and room surface (according to German Norm, DIN 1946);
- indoor air quality (according to European Standard CEN 1752, described below).

Ventilation efficiency ε_v is a criterion for energy and fan performances. This is used to evaluate a ventilation system and is defined by following expression:

$$\varepsilon_v = \frac{c_i - c_p}{c_{zl} - c_p} \qquad (26)$$

where: c_i is the contaminants concentration in the exhausted air; c_p – contaminants concentration the supply outside air; c_{zl} – contaminants concentration in the working area.

The value of ε_v depends on the entrance place and the exhaust way of outside air, and on the difference between the outside air temperature t_e and indoor air temperature t_i (Table 6).

The outside airflow rate L_p, in l/s, can be computed function of IAQ from equation:

$$L_p = 10\frac{G}{\left(C_i - C_p\right)\varepsilon_v} \qquad (27)$$

where:

$$G = G_{oc} + G_{ob} \qquad (28)$$

in which: G is the contaminants concentration of room air, in olf; C_i – indoor air quality, in dp (Table 7); C_p – outside air quality, in dp (Table 8); G_{oc} – contaminants quantity from the occupants (Table 9); G_{ob} – contaminants quantity from room objects (building elements, furniture, carpets, etc.) having the values in Table 10.

Table 6. Ventilation efficiency, ε_v

System type	t_e–t_i [°C]	ε_v
0	1	2
up–up	<0	0.9…1.0
	0…2	0.9
	2…5	0.8
	>5	0.4…0.7
up–down	<–5	0.9
	–5…0	0.9…1.0
	>0	1.0
down–up	<0	1.2…1.4
	0…2	0.7…0.9
	>2	0.2…0.7

Table 7. Indoor air quality, C_i

Room category	C_i [dp]	Percent dissatisfied [%]
0	1	2
A	1.0	15
B	1.4	20
C	2.5	30

Table 8. Outside air quality, C_p

No..	Air source	C_i [dp]
0	1	2
1	Mountain, sea	0.05
2	Locality, fresh air	0.1
3	Locality, mean air	0.2
4	Locality, polluted air	0.5

Table 9. Contaminants quantity from the occupants, G_{oc}

No.	Contaminants source	G_{oc} [olf/pers]
0	1	2
1	Adults resting, if the percentage of smokers is: 0 % 20 % 40 % 100 %	 1 2 3 6
2	Adults, if metabolic rate is: reduced (3 met) medium (6 met) high (10 met)	 4 10 20
3	Children: children under school age (2.7 met) pupils (1…1.2 met)	 1.2 1.3

Table 10. Contaminants quantity from room objects, G_{ob}

No.	Building destination	G_{ob} [olf/m^2]
0	1	2
1	Offices	0.02…0.95
2	Schools	0.12…0.54
3	Kindergarten	0.20…0.74
4	Meeting rooms	0.13…1.32
5	Houses	0.05…0.10

The outside airflow rate L_p, in m^3/h, can be computed and function of hygienic sanitary conditions follow as:

$$L_p = \frac{P}{\left(c_{i\max} - c_{p\max}\right)\varepsilon_v} \tag{29}$$

where: P is the power of the indoor pollutant source, in mg/h; $c_{i\max}$ – maximum admitted concentration of the most critical contaminant of room air, in mg/m^3; $c_{p\max}$ – maximum admitted concentration of the most critical contaminant of outside air, in mg/m^3.

To determine the variation in time of the contaminants concentration c_i in indoor air two hypotheses are assumed:

– constant pollution in time, where the balance equation within an infinitesimal time interval $d\tau$ is:

$$L_p c_p d\tau + P d\tau - L_p c_i d\tau = V dc_i \tag{30}$$

where V is the volume of room.

Integrating the equation (30), with the initial condition $c_i = c_p$, for $\tau = 0$, we obtain:

$$c_i = c_p + \frac{P}{L_p}\left(1 - e^{-n\tau}\right) \tag{31}$$

where n is the air change per hour.

– instantaneous pollution at moment $\tau = 0$; consequently, the contaminant initial concentration is given by the equation:

$$c_o = \frac{P}{V} \tag{32}$$

The balance equation for an infinitesimal time interval $d\tau$ is:

$$L_p c_p d\tau - L_p c_i d\tau = V dc_i \tag{33}$$

Integrating equation (33) with the initial condition $c_i = c_o$, for $\tau = 0$, is obtained following expression:

$$c_i = c_p - c_o e^{-n\tau} \tag{34}$$

5.2 Computer program COMFORT 2.0

The computer program COMFORT 2.0 allows to determine the outside airflow rate and air change per hour for the ventilation of a room and the variation in time of contaminants concentration of room air both on the basis of the mathematical model described above and on some national norms (I 5, DIN 1946), as well as to analyse the influence of different parameters on these characteristics.

- The inputs data are: geometrical characteristics of the room, in m; number of occupants; activity type; room category; outside air quality, in dp; ventilation system type; ventilation efficiency; smokers existence in room.
- The results of program are the following: outside airflow rate; air change per hour; polluting substances of indoor air; variation in time of CO_2 concentration of room air.

5.3 Numerical application

It is considered an A category room with geometrical dimensions 10×10×2,7 m, where there are 11 persons having an intellectual activity, and smoking is forbidden. Production rate of CO_2 for the occupants is of 15 l/(h·pers) and CO_2 concentration in outdoor air has the value of 350 ppm. The floor finishing is made of parquet floor or PVC.

A comparative study for computation of outside airflow rate and indoor air quality control according to the European Standard CEN and national norm I and DIN is performed using the computer program COMFORT 2.0.

The values obtained for the outside airflow rate and the air change per hour are reported in Table 11. In Figure 5 is represented the variation of outside airflow rate function of the indoor and outdoor air quality. Figure 6 shows the variation in time of CO_2 concentration in room air.

Table 11. Outside airflow rate L_p and air change per hour, n

Computation norm	Method	L_p [m^3/h]	n [h^{-1}]
0	1	2	3
CEN 1752	Air quality - parquet floor	2306	8.54
	Air quality - PVC floor	8494	31.46
I5	Number of occupants	330	1.22
DIN 1946	Surface of the room	600	2.22
	Number of occupants	660	2.44

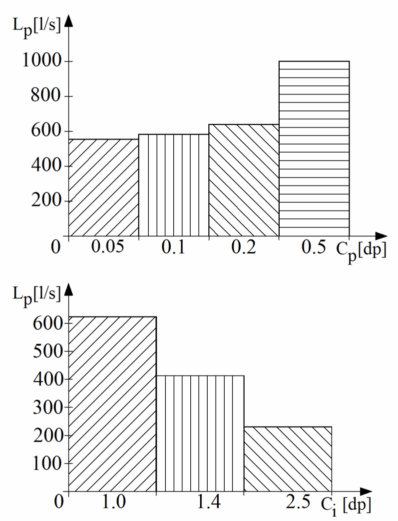

Fig. 5 Variation of outside airflow rate function of outdoor and indoor air quality

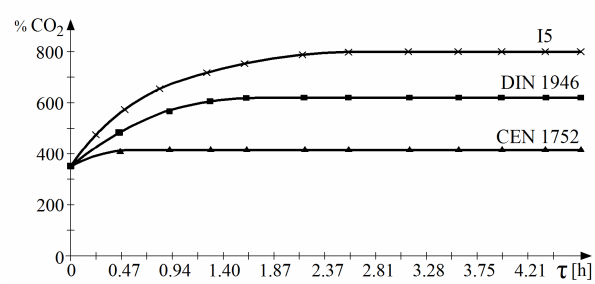

Fig. 6 Variation in time of CO_2-concentration

From Table 11 it is to be seen that the outside airflow rate computed according to norm I5 has the smallest value, leading to the highest values of CO_2 concentration of room air. This CO_2 concentration determines a state of strong tiredness and head aches for the occupants.

6 Conclusions

Computer model developed offer the possibility of detailed analyses and predictions on thermal comfort in closed spaces, being of a real use for the design and research activity and in the environmental studies.

Results show that the microclimate in rooms influence not only the comfort but also the health of the occupants, and preserving the comfort parameters at the optimal values is the mission of the engineer for designing and operating of HVAC systems.

The indoor environment is influenced by the way the equipments are designed, produced and operated. That is why it is necessary to decide upon the equipments that are able to give the proper microclimate conditions, permanently observing the elimination of all secondary effects (professional illnesses) that have negative influences upon human health.

It is possible at the design stage of heating systems and buildings to take into account most of the comfort criteria.

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