Fundamental Aspects of Thermal Comfort

TU Delft

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1. What is comfort?

Knowledge about the thermal sensation of people indoors is needed before being able to state the requirements for the indoor climate.

From ages, people have striven to create a thermally comfortable indoor environment. However, thermal comfort has been defined only from the sixties (1960) on. ISO 7730 standard [1] defines comfort as "That condition of mind which expresses satisfaction with the thermal environment". This definition is not easily converted into physical parameters that can be measured. But we need these physical parameters in order to control the building and the Heating Ventilation and Air Conditioning system (HVAC). Thermal environment should be considered together with other factors such as air quality, light and noise levels. If we do not feel the everyday working and living environment is satisfactory, our working performance and life will inevitably suffer. This reader is about the physical parameters responsible for comfort and about the two most important comfort models.

2. Body temperature control

Before talking about thermal comfort we should first try to understand how our body is controlling its temperature. Humans have a very effective temperature regulatory system, which assures that the body's core temperature is kept at approximately 37 °C, figure 1. When the body becomes too warm, two processes are initiated: first the blood vessels vasodilate, increasing the blood flow through the skin, increasing this way the heat transfer to outside. Second one begins to sweat.

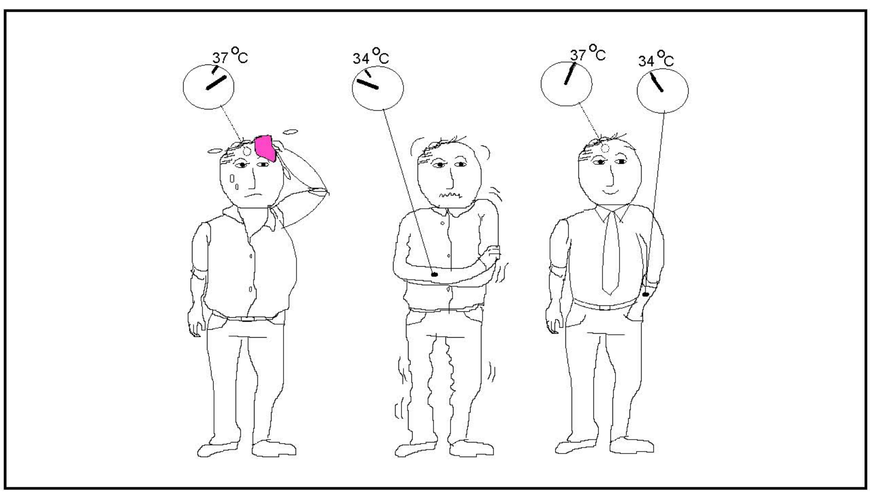


Figure 1, Control values of human temperature regulatory systems, by Jaap Keuvelaar, TU Delft

Sweating is an effective cooling tool, because the energy required for evaporating the sweat is taken from the skin. Already a few tenths of decrees departure from the core temperature can stimulate sweating.

If the body is too cold, the first reaction is for the blood vessels to vasoconstrict, reducing the blood flow through the skin, reducing this way the heat transfer with outside. The second reaction is to increase the internal heat production by stimulating the muscles, which is causing shivering.

Two types of sensors are located in the skin. There are cold sensors, which come in action when the skin temperature falls below 34 °C, and heat sensors coming in action when the skin temperature rises above 34 °C. If the hot and cold sensors give signals at the same time, our brain will block one or both of the body reactions. People consider the environment comfortable if no body reaction is present. In that case a person feels neither too warm nor too cold. This situation with comfort neutrality is the optimal condition. When the skin temperature falls below 34 °C, our cold sensors begin to send pulses to the brain and as the temperature continues to fall, more pulses are sent. Similarly, the heat sensors send pulses when the skin temperature exceeds 34 °C.
At the same time, blood temperature is registered by the hypothalamus and controlled around 37 °C. The brain interprets the signals, and next to triggering conscious reactions like eating, drinking or changing activity and clothing, it puts one of the three following possibilities in operation: muscles movement, sweating or change of blood flow in the skin. It takes some time to change the body's core temperature, therefore the signals from the blood temperature change very slowly compared to the skin sensors. In terms of control the feedback control of the core temperature is tuned to respond more slowly than that of the skin. In figure 2 a block diagram of the control system is shown.

In addition to this, it has been shown in the past years that Brown Adipose Tissues (BAT), also called brown fat, play a role in thermoregulation. Newborns and children have much more brown fat than adults and although the amount of these adipose tissues decreases with age, it remains playing a role in the body's thermoregulation. Basically, brown fat converts an excess of energy in the body (e.g. from food) into heat in a combustion reaction. In a warm environment this combustion is deactivated, while a cold environment or an infection will activate this combustion.

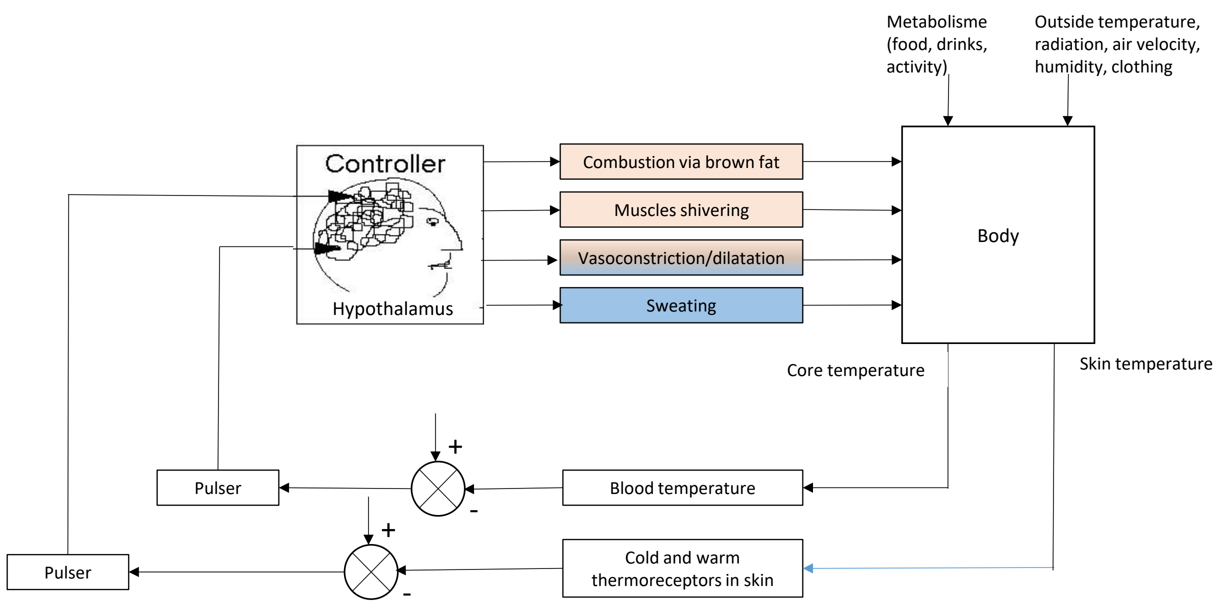


Figure 2, Block diagram of the body temperature control (thermoregulation), adapted from J. Keuvelaar / D. van Paassen, TU Delft

3. Parameters defining comfort

People do not feel the temperature of the surroundings itself, but feel the energy transferred from and to the body. Consequently the parameters that must be measured in order to know if an environment is thermally comfortable are those that affect energy transfer.

These are:

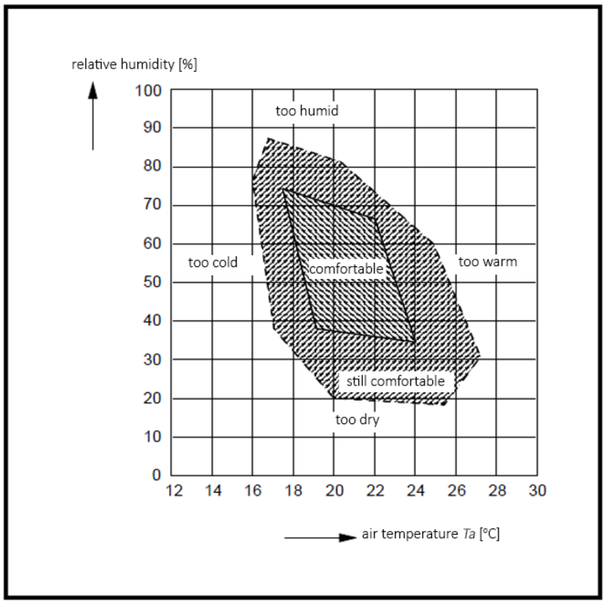
1. Activity level (metabolism)
2. Resistance of clothing
3. Air temperature T_a
4. Relative humidity RH (obtained from Ta and the wet bulb temperature T_n)
5. Air speed v and turbulence
6. Mean radiant temperature (average walls temperature (including walls, windows, ceiling and floor) T_w)

To keep the core temperature of the body at 37 °C, the heat generated inside the body should be increased or removed. This heat flow can be realized by convection, radiation or evaporation. Which of those possibilities dominates depends on the comfort variables. For example: at a high relative humidity and no air movements the evaporative heat flow will be much lower than those by convection and radiation.

In cases where the heat flowing out of the body is not sufficient, there is a risk for overheating and the body itself takes action. By increasing the blood stream and sweating, heat flows will be increased and will help to cool down the body.
When there is too much heat leaving the body, there is a risk for hypothermia. Involuntary actions of the muscles (shivering) help to increase the internal heat generation. Of course then we are far from comfort.

These 6 comfort parameters relate to each other and diagrams have been constructed based on a large number of experiments carried out with people in climate chambers. Because people can have different sensations of the indoor climate, there is not one point that is comfortable, but whole areas indicating the relation between comfort variables.

Following diagrams shows examples for people doing office work and wearing office clothing. Mode, gender, age are not taken into account [2]. Figure 3.1 shows the comfort relation between T_1 and the relative humidity of the air. The same is done in figure 3.2 for the average wall temperature T_{wg} and the air temperature T_1. In both these figures the air speed is 0.15 [m/s] on average.



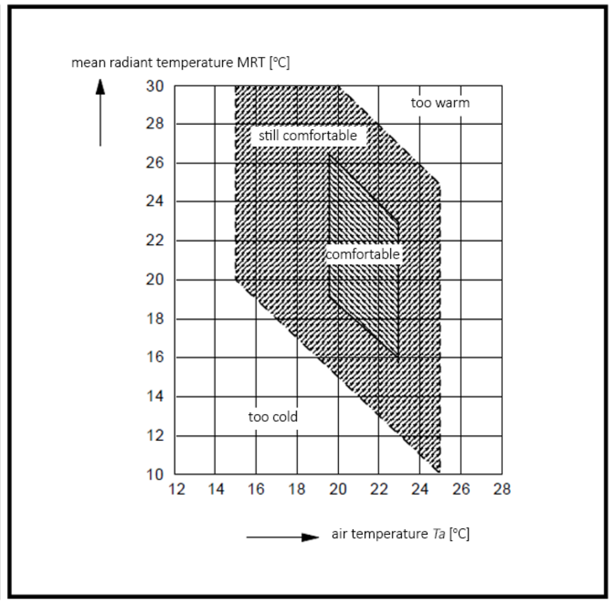


Figure 3.1, Comfort diagram: Humidity - Air Temperature *Figure 3.2, Comfort diagram: Mean Radiant Temperature - Air Temperature*

The indicated comfort area's in figure 3.2 are only valid when the walls temperatures differ no more than a few degrees. If there are large differences between walls, it will feel uncomfortable even though the average wall temperature is right.

From experience we know that at higher temperatures a higher air speed can increase comfort. This is shown by figure 3.3. It is very useful to explain the success of table fans or a cooling breeze through an open window when it becomes too hot inside.

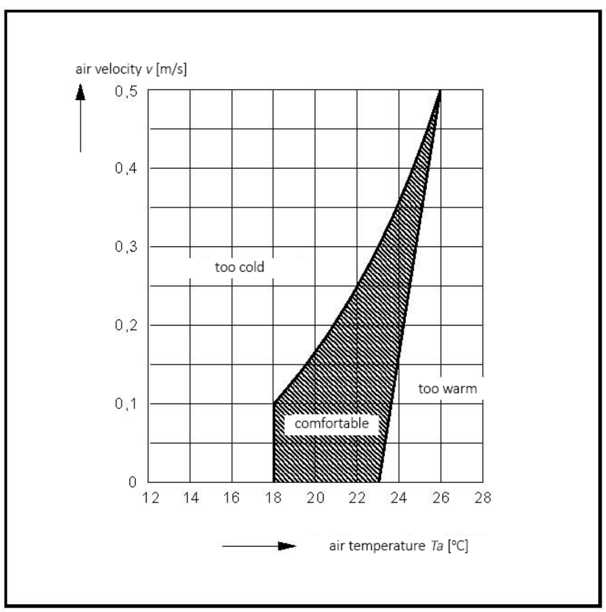


Figure 3.3, Comfort diagram: Air Velocity - Air Temperature

3.1 Activity levels

The activity level is expressed in terms of metabolic rate. It is the amount of energy per unit of time (W or J/s) that a person needs to keep the body functioning and carrying out specific tasks. This metabolic rate may differ per person, but the values will always be around the values given in Table 3.1. The metabolic rate is very often expressed in a specific unit, the so-called MET:

1 Met = 58.1 W/m^2 ,where the square meters relate to the body surface area.

Table 3.1 Metabolism for various activities per m^2 body surface (W/m^2 and MET).

Activity	[W/m^2]	MET
Sleeping	50	0.9
Sitting	60	1.0
Typing	70	1.2
Standing up	80	1.4
Normal standing work in a shop, laboratorial, kitchen	85 - 120	1.5-2.1
Walk slowly (3 km/h)	120	2.1
Walk normal (5 km/h)	150	2.6
Walk fast (7 km/h)	240	4.1
Normal runner - en construction work	350	6.0
Running (10 km/h)	500	8.6

The body surface area of an adult A_{du} is on average 1.8 m^2 and can be calculated using the Dubois formula (1916), relating surface area with body mass and height.

$A_{du} = 0.202.m^{0.425}.h^{0.725}$

With
m = body mass (kg)
h= height (m)

Further Reading:
Measurement of Metabolism: MET can be measured by measuring O_2 consumption and CO_2 production in air inhaled and exhaled. On the assumption that all the oxygen is used to oxidize degradable fuels and all the CO is recovered, it is possible to calculate the total amount of energy produced (conversion of the chemical free-energy of nutrients into the chemical energy of ATP plus energy losses during the oxidation process).

3.2 Resistance of clothing

The resistance of clothing I_{cl} is given in "clo"-units. By definition

1 clo = 0.155 m^2KW^{-1}

The resistance of clothing can be expressed as R= $0.155xI_d$ [m^2kW^{-1}]

1 clo corresponds approximately to male office clothing in temperate/cold countries: trousers, T-shirt, long-sleeved shirt and jacket.

In table 3.2 values for various clothing are given. When calculating heat flow rates from the body (see chapter 4), the body surface area A_{du} must be corrected by the factor f_{cl} representing the increase of heat exchange area (A_{he} arising from wearing clothes ($A_{he}=A_{du}xf_d$, where A_{du} is the body area, see section 3.1)

Table 3.2 Resistance of clothing.

Clothes	I_{cl} [clo]	f_{cl}
Nude	0	1.0
Bikini	0.01	1.0
Bermuda shorts	0.1	1.0
Tropics clothing, shorts, shirt short sleeved	0.3	1.05
Light clothing, pants, shirt long sleeved	0.5	1.1
Light tropic suite	0.8	1.1
Normal office clothing	1.0	1.15
Traditional heavy clothing	1.5	1.15 – 1.2
Danish winter clothing	2.2	1.2 – 1.3
Pole clothing	3 - 4	1.3 – 1.5

3.3 Air temperature (dry-bulb temperature)

When studying comfort, air temperature, also known as dry-bulb temperature, may be the most known and evident parameter. However, keep reminding it is only one of the six relevant parameters and the parameters are interrelated in the comfort diagrams, see for instance figures 3.1 to 3.3.

In general, a comfortable indoor air temperature is in the range 18-28 °C. However this range must be seen with respect to the other comfort parameters and with regional and seasonal preferences:

- In a fitness room or a sports hall temperature preferences are much lower, around 16°C. This is because of the high metabolism during sport activities (see

Table 3.1): more heat must be lost by the body and this is more easy at a lower air temperature.

- In Northern European countries an indoor air temperature of 20°C in winter is considered comfortable, while it will be considered as much too cold in summer, when temperature of 24°C will be considered much more comfortable. In hot countries the indoor temperature preferences may be higher, but also sometimes lower, when people are used to low temperature air conditioning.
- Because people wear different clothing in winter and summer, seasonal variations of the indoor temperature are considered more comfortable than keeping a constant temperature all over the year. An additional advantage of varying the temperature are energy savings: a lower indoor temperature in the cold season means less heating energy use; a higher indoor temperature in the hot season means less cooling energy.

Further reading:
When measuring the air temperature it is essential to take care that radiating surfaces around the temperature sensor are excluded, otherwise the sensor absorbs this radiation, by which its temperature increases and a too high temperature is measured. Figure 3.4 shows an example of air temperature sensor (a platinum thermocouple), with a reflecting shield around to exclude radiation heat flows from surrounding surfaces.

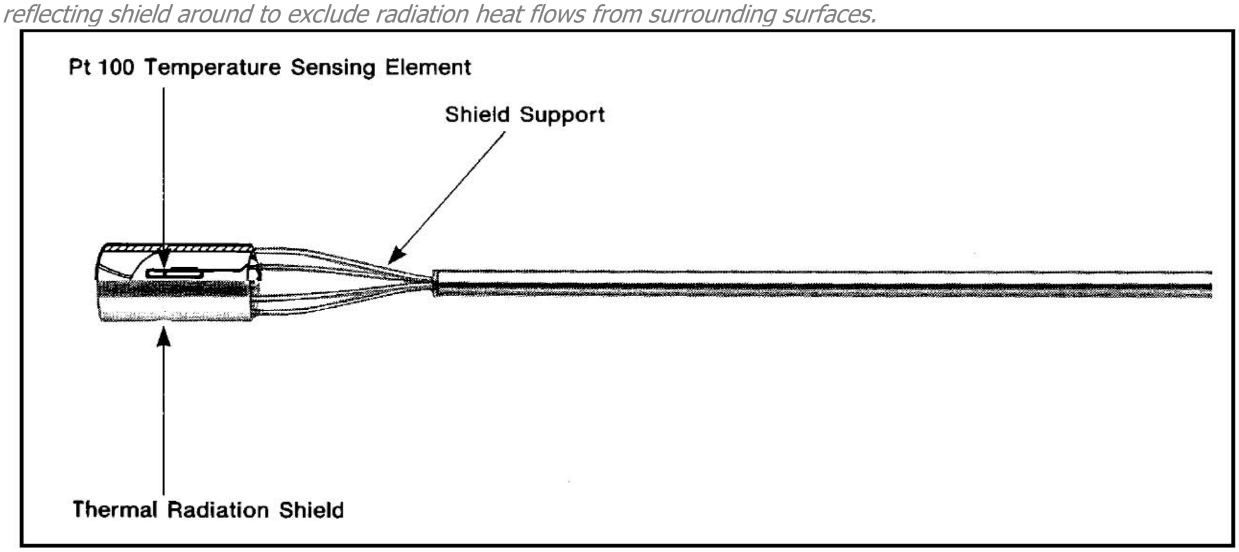


Figure 3.4, Air temperature sensor. Pt100 sensor, shielded from other surfaces by a shining surface

3.4 Relative humidity

Dry air is a mixture of different gases, in mass percentages, 78% Nitrogen, a bit less than 21% Oxygen, 0.9% Argon; 0.03% carbon dioxide. But air in general also contains water vapour up to around 8% in mass.

The maximum quantity of water vapour that can be absorbed by air depends on its temperature and pressure. At atmospheric pressure it therefore depends only on the temperature. For instance air at 0°C can contain a maximum of 3.8 g water/kg dry air (the air is said to be saturated at this water content, meaning that droplets of water start to appear). At 20°C the maximum is 15 g/kg and at 40°C it is 49.1 g/kg. The **humidity expressed in g/kg** dry air is said to be the **absolute humidity**. Note that the mass percentages of water vapour in air are very small.

The sensation of humidity does not relate to these absolute values but to how far the water pressure in the air is from the water pressure at saturation. A relative humidity of 100% corresponds to saturated air. A relative humidity of 0% corresponds to dry air. A relative humidity of 50% corresponds to half the vapour pressure in saturated air at a certain temperature. At a certain temperature the relationship between the relative humidity RH, the absolute humidity X and the absolute humidity at saturation X_{max} can be expressed as follows:

$$RH = 100x(29+18/X_{max})/(29+18/X)$$

Or, conversely by:

$$X = 18RH/[100x(29(1-RH)+18/X_{max})]$$

Where RH is expressed as a percentage and X and X_{max} are expressed in kg/kg dry air (*not in g/kg*). Note also that by approximation $RH=X/X_{max}$ (although this relation has no physical meaning).

In table 3.3. some examples of calculation are given.

Table 3.3: Absolute humidity of air (water content X) at different temperatures and relative humidity's

Water content X (g/kg dry air)	Relative Humidity (%)						
Air Temperature (dry-bulb) (°C)	0%	10%	30%	50%	70%	90%	100% (X_{max})
0°C	*0*	*0.4*	*1.1*	*1.9*	*2.7*	*3.4*	*3.8*
20°C	*0*	*1.5*	*4.4*	*7.4*	*10.4*	*13.5*	*15*
40°C	*0*	*4.6*	*14.0*	*23.6*	*33.6*	*43.8*	*49.1*

For a comfortable and healthy environment, it is recommended to keep the relative humidity between 30 and 70%. Above 70% the risk of mold growth increases a lot and, if the indoor temperature is high, the body will release heat by sweating, which becomes very difficult at a high relative humidity (If the RH is 100%, the air cannot absorb any water vapor, the skin becomes wet and less heat is lost by transpiration).

The lower limit of 30% is set to be prevent electrostatic shocks. In general when people complain about too dry air the complaints relate to particulate matter in the air or a to a combination of particulate matter, draught and poor lighting causing irritation of eyes or throat. It seldom relates to a too low relative humidity.

Figure 3.1 shows the relationship between comfortable (and a bit less but still comfortable) air temperatures and relative humidity. For instance at an air temperature of 24°C and a relative humidity of 35%, people would feel just comfortable. If the relative humidity increases above 65% most people will start feeling very uncomfortable. If the air temperature is 20°C, a relative humidity up to 70% will be comfortable and discomfort will arise only from 80% RH.

Further reading:
Absolute humidity can be measured on different ways. One of them is by putting two thermocouples in the air flow: one measuring the dry bulb temperature (see section 3.2), and one measuring the temperature given by a thermocouple wrapped in a wetted cloth, this is the so called wet-bulb temperature. There is a known relationship between both temperatures, from which the relative humidity can be determined. This relationship is described by the thermodynamic properties of humid air and in the related psychrometric charts or Mollier diagrams.

Another way is based on dew point temperature measurement (see figure 3.5), by which the air is cooled down by a cooling element until the temperature at which condensation occurs (the so-called dew point temperature). The cooling element is placed right under a small mirror. The temperature of the mirror when water starts to condensate is the dew point and is measured by a thermocouple. The exact moment of condensation is determined by using a light emitting diode. The light of such a diode is perfectly reflected by the mirror, but as soon as condensation occurs the light becomes diffuse and the output of the diode decreases, which is the signal that the cooling can be stopped and the temperature of the mirror must be measured. Here too there is a known relationship between air temperature, cooling energy and relative humidity. This relationship is described by the thermodynamic properties of humid air and in the related psychrometric charts or Mollier diagrams.

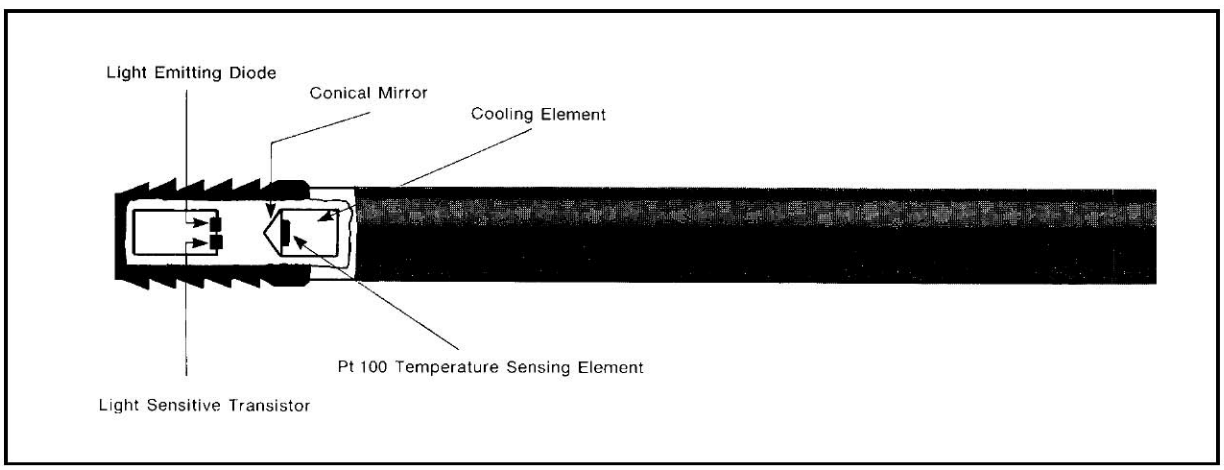


Figure 3.5, Measurement of relative humidity

3.5 Air velocity

Air velocity and turbulence are responsible for draught. Draught may be nice at hot indoor temperatures and helps feeling comfortable even when it is too hot. However, at cold temperatures it increases the heat transfer rate from body to air and can be experienced as very unpleasant.

In general air velocities indoors should be kept below 0.15 m/s during the cold season, but may be increased up to 0.5 during the hot season. For example a desk fan increases the air velocity. It does not cool the air, but because of the increased air velocity, the heat transfer by convection between body and air increases, keeping the body cool. Such a fan also increases the turbulence of the air, which will be treated shortly in section 4.5.

Figure 3.3 shows the relationship between air velocity and air temperature when it comes to a comfortable area.

Further reading
There are many types of air velocity meters on the market, using diverse principles. An accurate one is the one of figure 3.6, based on a little ball that is kept at a constant temperature by an electric heating coil. The temperature is measured and controlled at a constant value. The amount of heat needed to keep it constant is a measure of the velocity of the passing air. The higher the velocity the more heat is required. A similar unheated ball with a nickel wire is used as a temperature reference. Its response is fast so that it can be used to measure the turbulence as well (T_u = standard deviation of the speed / average speed).

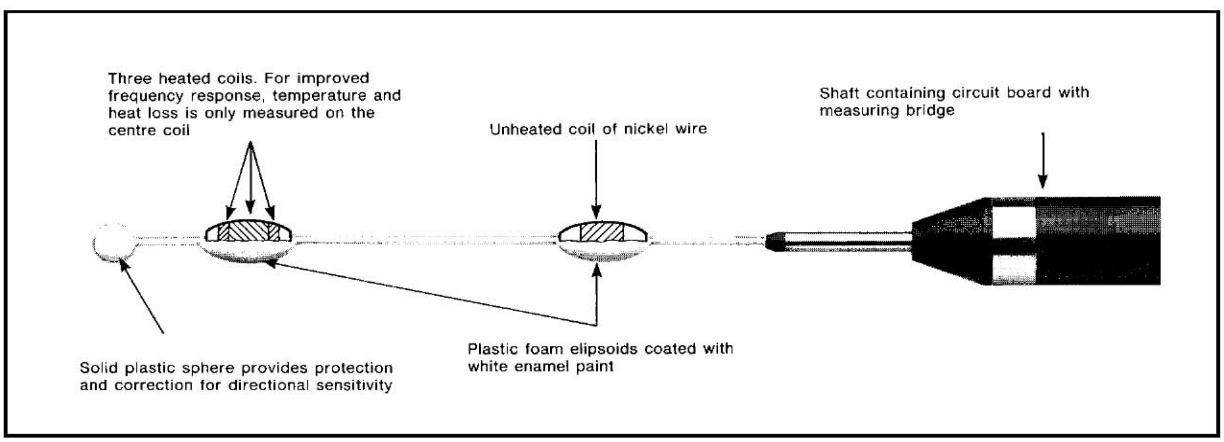


Figure 3.6, Air velocity meter (anemometer)

3.6 Mean Radiant Temperature (MRT)

The skin temperature and the quantity of heat absorbed by the human body do not only depend on the air temperature, but also on the temperature of surrounding surfaces. For instance the human body perceives very well the warmth from the sun, which is radiated from the sun to the earth. This heat cannot be absorbed by air (air is said to be transparent to solar radiation), but is absorbed by the human body (and the ground, trees, buildings etc.). It is why it is possible to feel warm in cold but sunny weather. On the same way the human body can 'feel' (infrared) radiation from every surface having a temperature different from the skin. It is not only a 'feeling', there is real heat exchange and it is called radiative heat transfer. A person in a room will exchange heat with all walls, windows, ceiling and floor and this heat exchange depends on their temperatures to the power of 4. Every surface Ai with a temperature above the absolute zero (0K or -273.15°C) emits radiation and the <u>'radiant' temperature is just another word for 'surface' temperature.</u>

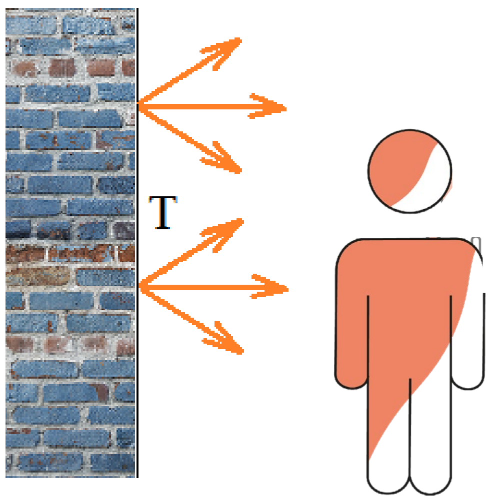


Figure 3.7, Left: Radiation from wall to a person due to wall's temperature and Right: Visualization of wall's & window radiation with an infrared camera. (Pictures by A. Rasooli, R. Lindeman (TUD) and adapted from Sam McAfee, shutterstock Sequence Number: US-0E5D7C35D-1)

A person in a room exchanges therefore radiation heat with all surfaces of the room (this is something completely different than heat exchange by convection with the room air). In order to simplify comfort calculations and comfort diagrams (see figure 3,2 and chapter 4), it is most convenient to work with an average surface temperature of the enclosure (the room) instead of with the temperatures of all separate surfaces of the enclosure. The average surface temperature of the enclosure is called the Mean Radiant Temperature (MRT).

This MRT is not the arithmetic average of the surface temperatures, but is defined as the equivalent uniform temperature of a black enclosure in which the occupant would exchange the same amount of radiant heat as in the actual non-uniform enclosure.

- Non-uniform means that different parts of the enclosure like walls, windows, ceiling and floor have different temperatures
- 'Black enclosure' means that the enclosure has an emissivity of 1; If a surface has an emissivity of 1 it is said to be a black body: it absorbs al radiation it receives. If such a black body is in thermal equilibrium (therefore has a constant temperature) it is also a perfect emitter and will emit more radiation than a non-black body. When the emissivity is lower than 1, the surface is said to be a grey-body and it can only absorb a part of the radiation it receives;

- Calculations of radiation heat transfer are more easy for a black body than for a grey body, that is why the MRT is defined for a black enclosure.

The MRT is not straightforward to calculate. For an enclosure consisting of several (in the equation below there are 'n' surfaces) surfaces A_i, containing a person of surface A_p , MRT it is defined as:

$MRT = (F_{p-1}T_1^4 + F_{p-2}T_2^4 + \ldots + F_{p-n}T_n^4)^{1/4}$ (Equation 3.1)

Where T_i is the temperature of surface A_i and F_{p-i} is a purely geometrical factor called the view factor (or the angle factor) determining the fraction of the total heat emitted by surface A_p that is received by A_i (F_{p-i}=(radiation emitted by p and received by i)/radiation emitted by p). The sum of all view factors must always be 1 ($F_{p-1}+F_{p-2}+\ldots+F_{p-n}=1$). More explanations on the calculation method are given in Appendix A.

If the temperatures of the surfaces of the enclosure are close to each other (not more than 5 to 10 K difference), the equation may be linearized:

$MRT = F_{p-1}T_1 + F_{p-2}T_2 + \ldots + F_{p-n}T_n$ (Equation 3.2)

The MRT calculated with equation 3.2 will be systematically a bit lower than the one calculated with equation 3.1.

If all surfaces of the enclosure have an identical temperature then MRT is equal to this temperature.

The view factors F are difficult to determine, but most energy calculation software packages like Energy Plus, DOE etc. have module making these calculations and calculating automatically the MRT. (see also [3] and
http://www.healthyheating.com/Definitions/Mean%20Radiant_pg5.htm#.YHAIq9yxVhF or
http://www.thermalradiation.net/calc/sectionc/C-10.html)

MRT is difficult to calculate, but more easy to measure, that is why the preference is given to direct measurements, using a globe thermometer, see further reading.

The MRT has an important influence on comfort, see figure 3.2. Lower air temperatures can be compensated by a higher mean radiant temperature. For instance if the MRT is 20°C, an air temperature of 20°C is enough to ensure comfort. If the MRT drops to 17°C, the air temperature must be raised to 22°C. Surface temperatures of walls, windows etc. (and therefore the MRT) depend strongly on the insulation of walls. In a well-insulated building the surface temperatures in winter are higher than in a poorly insulated building. This results in less transmission losses and therefore less heating energy. Furthermore, because the MRT is high, the air temperature can be lowered (e.g. from 22 to 20°C) decreasing even more the transmission losses and the heating energy.

In a summer situation with very hot and sunny outside weather, the surface temperatures may be high. From figure 3.2 we can that read with a MRT of 28°C, an air temperature of 22°C would just ensure a 'still comfortable' environment. While if MRT is decreased to 24°C (by insulating the walls or avoiding solar radiation to penetrate the building), an air temperature of 25°C would be acceptable. Energy savings on cooling energy will be achieved by less transmission losses (because of insulation), less solar gains (if solar shading is applied) and because the allowed indoor temperature is higher.

In cold countries, radiators are generally placed on (outer) walls and under the window. This is because the hot radiator (with temperature between 40-70°C, or even up to 90°C in poorly insulated dwellings) not only brings the needed heat but is also used to compensate for the cold surface temperature of (outer) walls and windows. Very roughly estimated (for a correct estimation MRT should be calculated), if a wall has a temperature of 14°C and a radiator at 80°C is covering 20% of the wall area, this will result in an average wall temperature of 14x0.8+80x0.2=27.2°C, which is much more comfortable than 14°C.

Further reading: Measurement of surface temperature and MRT

MRT can be calculated when knowing all surface temperatures in a room (walls, windows, floor and ceiling). This can be done using surface temperature sensors as shown in figure 3.8.

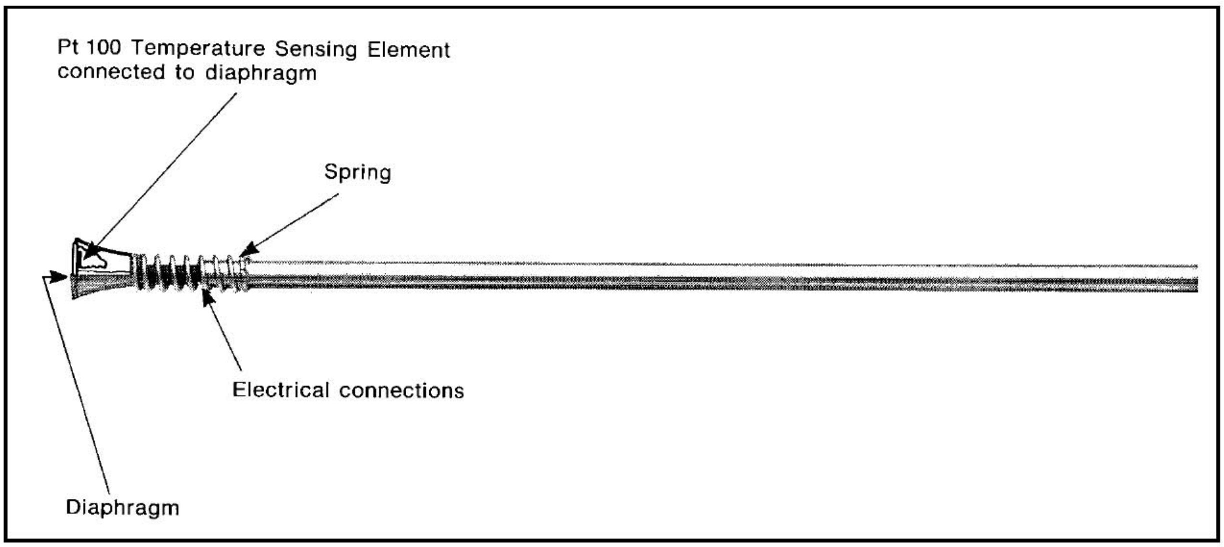
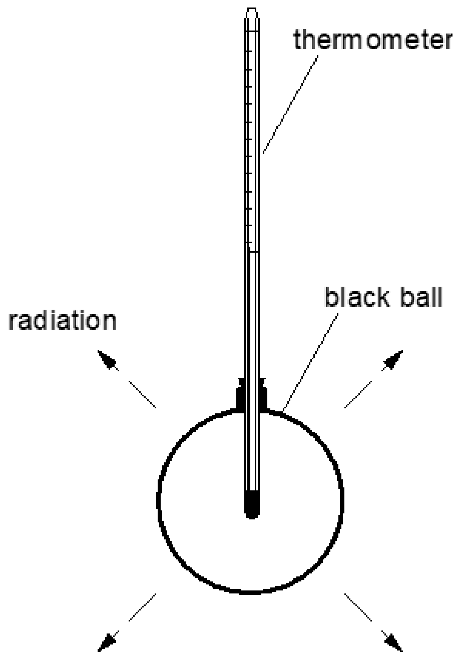


Figure 3.8, Surface temperature sensor. A thin metal disc with a Pt-100 sensor is pushed against the wall surface

However, all view factors must be calculated in order to estimate the MRT from the surface temperatures, and this is very cumbersome.

Less cumbersome is to measure directly the MRT by using a black-ball or globe thermometer see figure 3.9. The thermometer is placed at the location where the MRT must be measured (so generally close to the desk of chair). The temperature sensor itself, is located in the center of the black ball (in the picture left the black ball has a diameter of 15 cm).



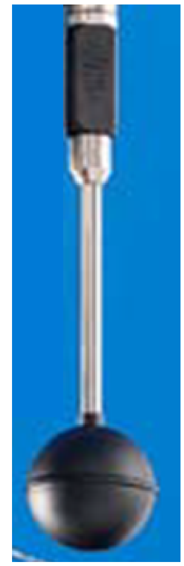


Figure 3.9, Black ball or globe thermometer

The bigger the ball, the less the measurements are influenced by the air temperature. The ball is black, by which it absorbs all radiation and therefore emits radiation following the Stephan Bolzman equation ($Q = \sigma.\varepsilon.A.T^4$ [W], see Appendix A)

By making an energy balance on the globe and by measuring also the air temperature T_a and the heat transfer convection coefficient h, the MRT can be calculated easily using the following balance equation:

$\sigma.\varepsilon.(MRT^4 - T_g^4) + h(T_a - T_g) = 0$

with

- σ = constant of Stefan-Boltzmann, $\sigma = 5.67 \cdot 10^{-8}$ [$W/m^2 \cdot K^4$]
- ε = emission coefficient of black ball, close to 1
- MRT = average radiant temperature [K]
- T_g = temperature of the black ball [K]
- T_a = temperature of the air around the black ball[K]
- h = convective heat transfer coefficient [W/m^2K]
  - Natural convection: $h=1.4((T_a-T_g)/D)^{1/4}$
  - Forced convection : $h=6.3.v^{0.6}/D^{0.4}$
  - With D: diameter of the ball; v: air velocity close to the ball

3.7 Operative temperature and other equivalent temperatures

In various guidelines and publications, see for instance [3], equivalent temperatures combining the effects of two or more comfort parameters are used. The most often used one is the operative temperature.

Operative temperature T_{op}

This is the weighted average of the air temperature and the mean radiant temperature. The weights correspond to the respective heat transfer coefficients by convection and radiation.

$$T_{op} = \frac{h_r.MRT + h_c.Ta}{h_r+h_c} \quad [K]$$

(MRT: Mean Radiant Temperature [K] ; T_a : air temperature [K]; h_r: radiative heat transfer coefficient [W/m^{2}K]; h_c: convective heat transfer coefficient [W/m^{2}K])

As hc, the convective heat transfer coefficient, depends mainly from the air velocity, the operative temperature is in fact a combination of three comfort parameters: radiation and air temperatures and air velocity.

This operative temperature has a physical meaning: if different environments (e.g. rooms) have the same operative temperature, they will be perceived as even comfortable (heat losses and gains from the body will be identical), if their humidity is similar (and people wear the same clothing's and have similar metabolic activity).

Resulting temperature T_{res}

T_{res} is just the arithmetic average between T_a and MRT. If two different environments have the same resulting temperature, they will be perceived as even comfortable, if air velocity and humidity are similar(and people wear the same clothing's and have similar metabolic activity).

T_{res}= (MRT+T_a)/2

Effective temperature T_{eff} This is the same as the operative temperature, but this time the relative humidity is fixed at 50%.

Humid operative temperature T_{oh} It is the same as T_{eff} but now for 100% relative humidity.

Wet-Bulb Globe Temperature WBGT

WBGT is used as index to determine heat exposure as being a combination of wet-bulb temperature T_{wb} (see optional reading in section 3.4) , black globe temperature T_g (see optional reading in section 3.6) and air temperature T_a. So WBGT measures all four environmental parameters determining comfort.

In presence of solar radiation:
WBGT=0.7T_{wb}+0.2T_g+0.1T_a

In absence of solar radiation:
WBGT=0.T_{wb}+0.3T_g, where all temperatures are in degree Celsius.

Heat stress index HIS

The heat stress index is 100 times the ratio of total evaporative body heat losses required for body's thermal equilibrium when the skin temperature is 35°C to maximum possible evaporative heat loss possible in the environment Emax (see sections 3.4 and 4). When HIS is below zero, body cooling takes place. When HIS is above 100, heat stress occurs meaning that the body is not able to lose enough heat to avoid overheating.

Further reading: Simultaneous measurement of comfort parameters

There are instruments on the market allowing for the simultaneous measurement of all comfort parameter. They are sometime called 'thermal mannequins', microclimate or indoor climate analyzers, see figures 3.10 to 3.13

In figure 3.10 the measurement is carried out by a light grey ellipsoid shaped sensor (160mm long and with a diameter of 54 mm). The shape is selected in order to get the same convection and radiant heat transfer as the human body. A standing person is simulated by the sensor in vertical position. On the sensor's surface a temperature sensor is mounted and a heating element is controlled in such a way that the surface temperature is equal to that of a person feeling comfortable. The needed heating power is measured. The sensor measures the combined effect of 3 of the 6 comfort parameters being air speed, air temperature and average radiant temperature. The other parameters clo-value, activity level, and humidity must be adjusted manually. The operative temperature (see section 3.7), resulting temperature, comfort temperature and PMV & PPD-value are calculated automatically (see chapter 4).

Figure 3.10, Thermal comfort meter type 1212 (Bruel & Kjaer)

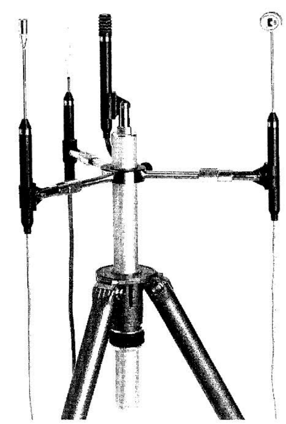


Figure 3.11, Indoor climate Analyzer type 1213(Bruel & Kjaer)

In figure 3.11 the instrument consists of a series of sensors. It measures all parameters separately (air temperature, radiant temperature, humidity, air speed) and calculates automatically the PMV- and PPD-value (see chapter 4).

Figure 3.12 shows a WBGT meter, also measuring MRT and calculating PPD and PMV (see chapter 4). Air velocity, black globe temperature, humidity and air temperatures are measured.


Figure 3.12, Delta Ohm WBGT meter HD32.3

4. The comfort equations of Fanger, PPM and PMV

4.1 Introduction

Based on experiments in climate chambers with large samples of people, Fanger from Denmark was able to develop a set of equations that can be used to assess the thermal environment and to predict the number of people that will not be satisfied with it. First of all he made a physical model based on the heat balance of the human body and the experimental information that, for comfort, the skin temperature should be nearby 34. With this model he was able to define combinations of indoor climate variables delivering a comfortable indoor climate for most people. Later on he could combined this physical model with statistical information obtained by asking people about their thermal sensations under different conditions.
To answer the question how far is a specific indoor climate from optimal comfort and within what limits temperature and humidity should be maintained, he introduced the PMV–index (Predicted Mean Vote). It predicts the mean value of the subjective thermal perception of a group of people in a given environment. To predict how many people are dissatisfied in a specific indoor climate, the PPD-index (Predicted Percentage of people Dissatisfied) has been introduced.

In this chapter the physical model will be discussed and it is explained how PMV and PPD are connected with this model, so that for every possible combination of indoor climate variable it can be predicted what people percentage will complain. This information is extremely important for the designer. For example to know that at least 5% will always be dissatisfied even with the best possible building and HVAC design. The only way to satisfy all people is to give each individual his own climate which can be controlled as desired, and that is in general not possible.

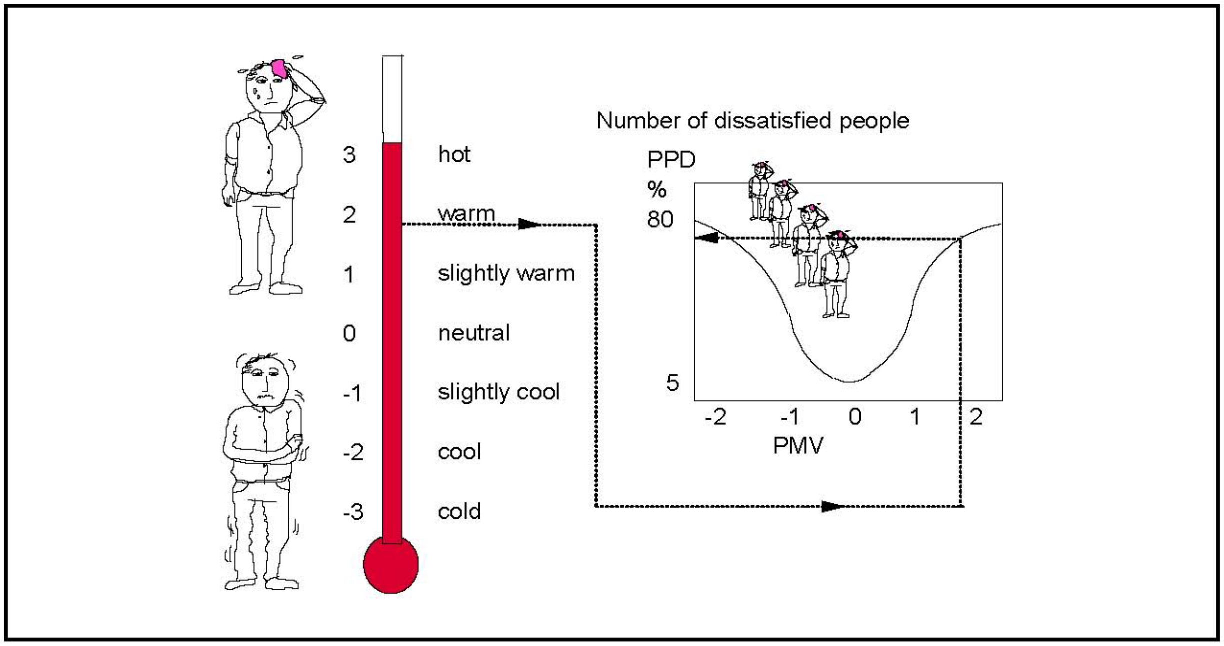


Figure 4.1, Illustration of PMV and PPD (picture by Jaap Keuvelaar/Dolf van Paassen): a rated sensation of feeling warm (PMV 1.8) will lead in this case to around 75% people dissatisfied.

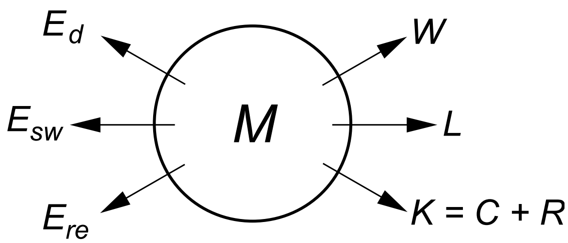
4.2 Heat balance of the human body

The first principle of Thermodynamics is the principle of energy conservation: energy cannot disappear or be created from nothing. It can only be transmitted or transformed from one form of energy to another. This means that if a system is maintained at a certain temperature there is a balance between the heat coming in, the heat stored or discharged internally and the heat going out.

If the system is not in balance, its temperature will decrease if there is a heat deficit, or increase if there is a heat surplus.

In Fangers' model, the system considered is the human body (circle in figure 4.2). Figure 4.2 shows all energy flows to, in and from the human body. There is internal heat production though the Metabolism (M; M is always positive). There is convection and radiation heat transfer from the skin to the clothing and from clothing to environment (K[1]; K is positive if the air temperature is lower than 34°C, negative otherwise). Work is carried out (e.g. any physical activity, W is always positive). Because of air being breathed in and out, there is heat transfer (L; L is positive if the air temperature is lower than 37°C, negative otherwise); The moisture content of the air inhaled and exhaled is different, therefore there is also latent heat transfer (E_{re}, latent heat of vaporization). Finally there is heat produced because of the humidity being released from the body to the outdoor through water diffusion (E_d) and sweating (E_{sw}).

[1] In Fangers' equations heat losses from the body are counted positive. This is the opposite convention as what is usual in heat transfer calculations. Just be aware of it.



M = rate of metabolic heat production
K = Convection & Radiation heat transfer rate
W = rate of mechanical work accomplished
E_{re} = evaporative heat transfer rate from vapor in respiration
L = dry air heat transfer from breathing
E_d = evaporative heat transfer rate by natural water diffusion through skin
E_{sw} = evaporative heat transfer rate by sweating

Figure 4.2: Energy model of human body. All energy rates are in W/m² body area.

If the energy balance is zero (TS=0 in equation 4.1), a constant temperature is maintained inside the body (37ºC) and on the skin (34ºC) and the sensation is comfortable. If TS is negative, there is a deficit of heat, the skin and the core temperatures start decreasing, and people feel too cold (there is discomfort). If TS is positive, there is a surplus of heat, the skin and core temperatures start increasing, and people feel too warm (there is discomfort).

$M-W-E_d-E_{sw}-E_{re}-L-K = TS$ (Equation 4.1)

For complete explanations on the heat balance, see Appendix B.

4.3 Comfort diagrams and practical applications

The comfort equations can be translated to diagrams such as in figures 3.1 to 3.3 and in figures 4.4 to 4.8 in which the (curved) lines indicate optimal comfort.

4.3.1 Effect of humidity and air speed on air temperature:

Figures 4.4 and 4.5 show the effect of humidity for sitting persons wearing clothing with different clo values. In figure 4.4 the clo-value is 0.5, while it is 1 in figure 4.5.

On the x-axis the air temperature is shown and the MRT was equal to the air temperature. The y axis shows the web-bulb temperature, which was only used to construct the diagonal lines of relative humidity (φ=1 means a RH of 100%, φ=0 means dry air).

At he intersection of φ=1 and air speed 0.1, one can read that a comfortable air temperature would be around 24.5ºC. For dry air that would be at 26.5ºC.

So the dryer the air, the warmer the allowed (in terms of comfort) temperature. The picture is the same for all air speeds: very humid air needs a temperature lower by approximately 2 degrees in comparison with very dry air.

The figures also show the importance of the air velocity. For instance at 80% relative humidity (φ=0.8), the comfortable air temperature is 26ºC if the air speed is 0.1 m/s. By increasing the air speed to 1.5 m/s, the comfortable temperature rises to 28.5 ºC.

By making good use of draught it is therefore possible to allow higher temperatures, reducing the demand for cooling.

By comparison, Figure 4.4. shows the very important effect of clothing. With a clo-value of 1 (office clothing), at 80% humidity the comfort temperature is reduced to 22°C if the air speed is 0.1 m/s. By increasing the air speed to 1.5 m/s, the comfortable temperature is around 25 °C. In a cooling situation this means that much more cooling is needed than with a clo of 0.5 (light clothing). In a heating situation on a contrary much less heating will be needed.

4.3.2 Effect of air speed and metabolic activity on comfortable air temperature

Figure 4.5 shows the effect of air speed and activity of the occupants. Here too the x-axis shows the air temperature and the MRT was equal to the air temperature. At low speed (< 0.1 m/s) there is no longer any effect of the speed on the comfort temperature (at constant metabolic activity). The reason is that the convective heat transfer is not affected below that value.

For a certain activity level, the lower the air speed, the higher the air temperature needed to ensure a good comfort. The strong effect of the activity level is shown as well. At an air velocity of 0.1 m/s (which is quite normal in an office room), a sitting person with office clothing (clo=1) would feel comfortable at 23°C. If this person is moving around, doing physical work (metabolic rate 120W/m2), a temperature of 15 °C would be enough.

This explains why in many national building codes, the indoor temperature prescribed depends on the building function (e.g. high in hospitals' bedrooms, low in sport halls).

4.3.3 Effect of metabolic activity and clothing on comfortable MRT and air temperature

In the precedent figures the MRT was equal to the air temperature. But as seen earlier the MRT and the air temperature will mostly not be equal. Figure 4.7 shows how both temperatures relate at different activity levels and clothing (but at a constant air speed).

Line 3 shows how MRT and air temperature should be chosen when the metabolic activity is 70 (sitting person) and the clo value is 1 (office clothing). Choosing 22°C for both MRT and air temperature will lead to a comfortable situation.

If the MRT is only 10°C (which could happen during a cold period if the building is very poorly insulated), then the air temperature should be 30°C, which will cost a lot of heating energy. Conversely, if the MRT is 30°C (which could happen during a warm period if the building is poorly insulated or if there is a lot of incoming solar radiation), the comfortable air temperature will be 17°C, which will cost a lot of cooling energy.

The impact of clothing and activity can be seen by switching to other lines.

This figure shows clearly (like figure 3.2) that high radiation temperatures can be compensated by low air temperatures.

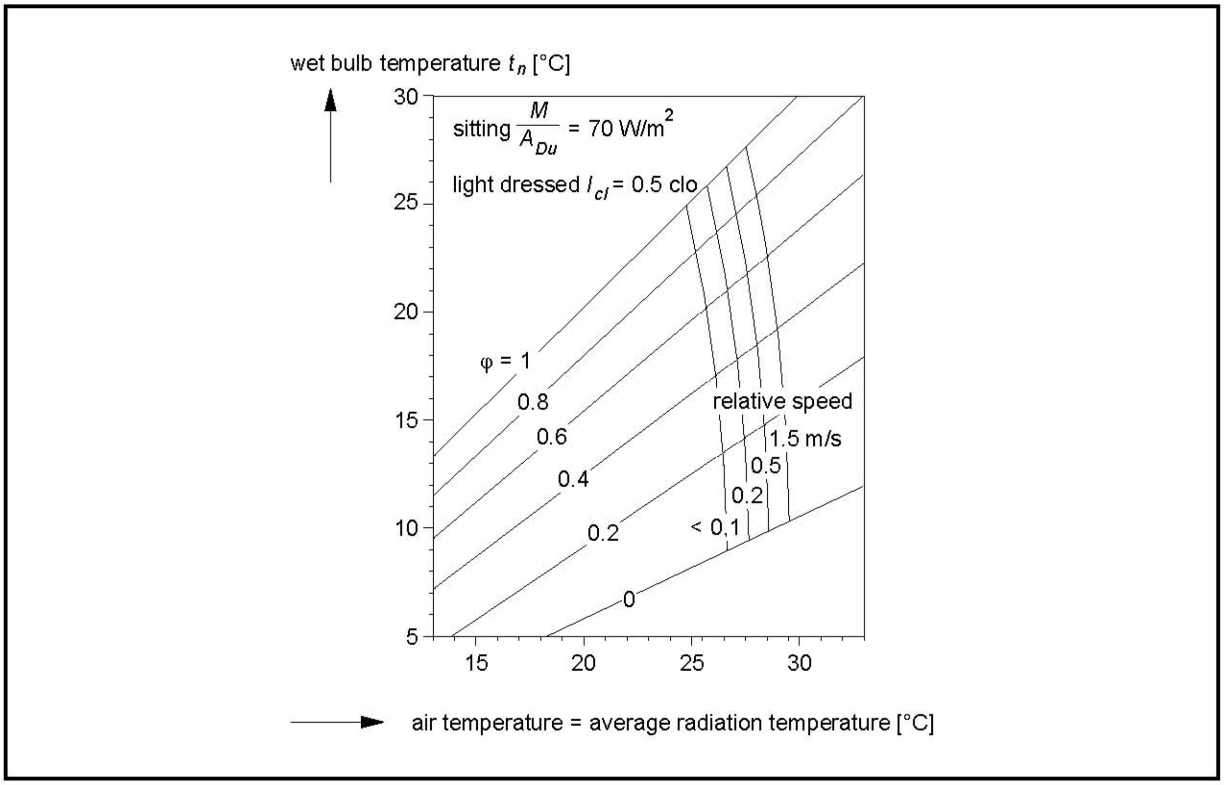


Figure 4.4, Relation between relative humidity, air speed and air temperature for a lightly dressed person

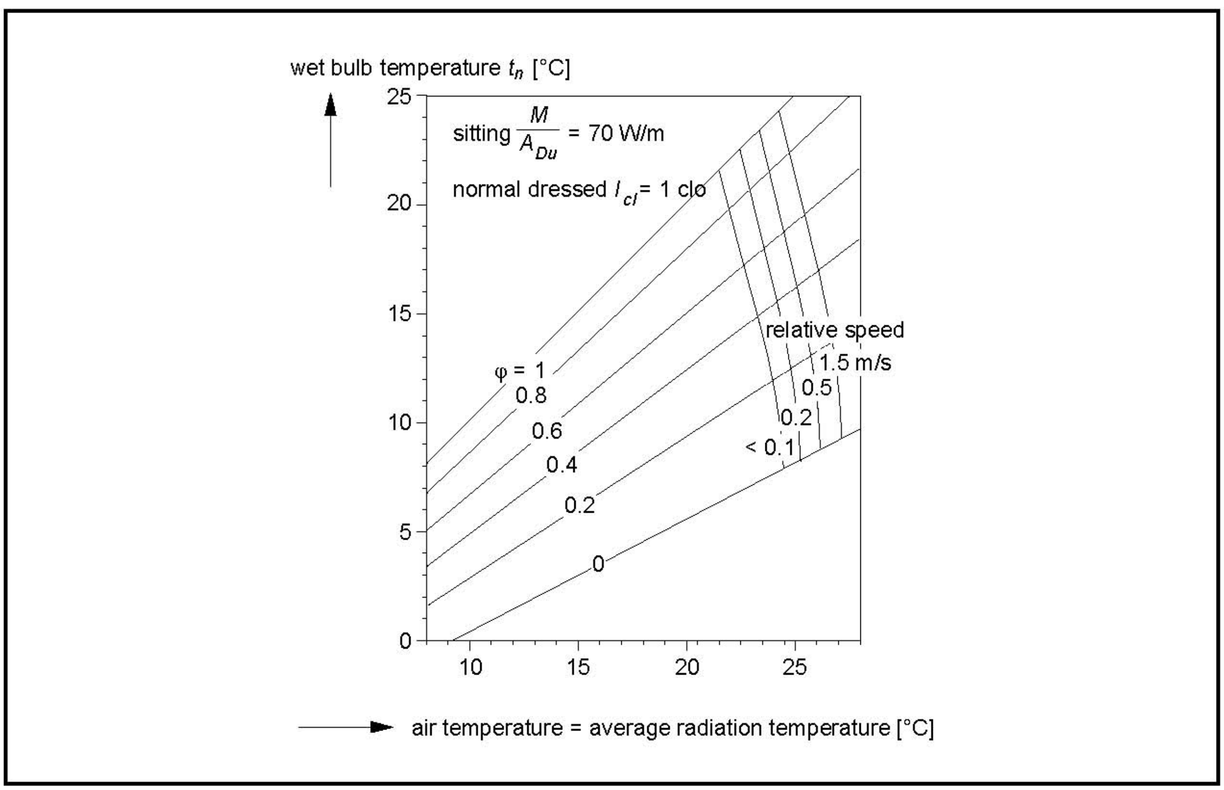


Figure 4.5, Relation between relative humidity, air speed and air temperature for a 'normally' dressed person

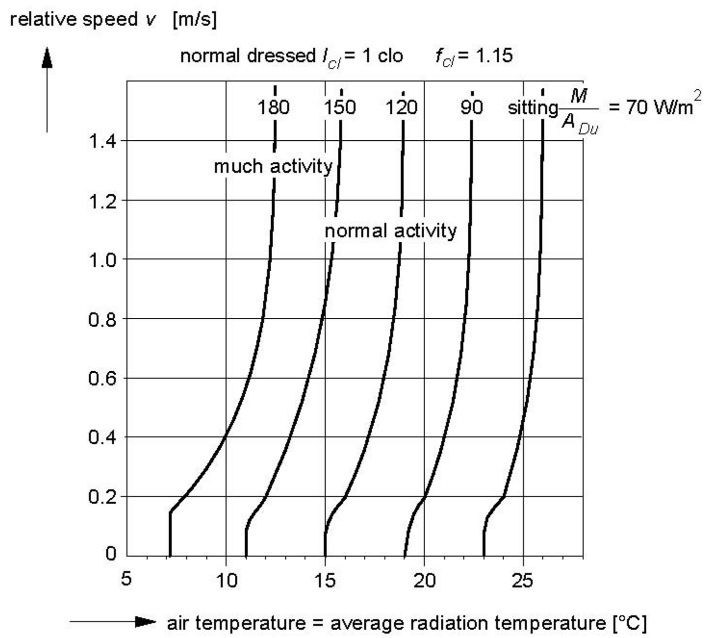
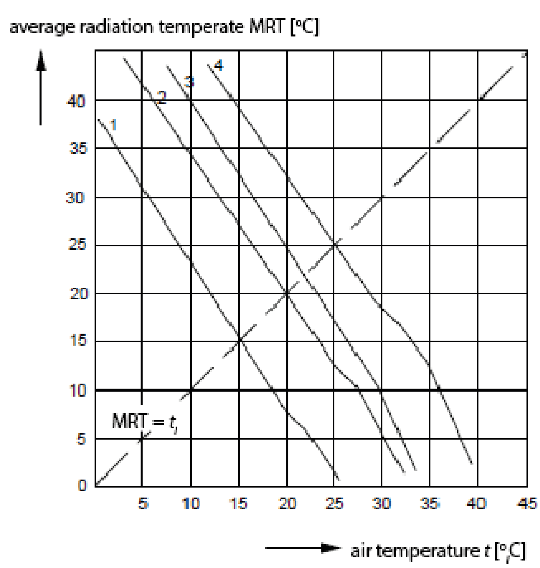


Figure 4.6, Comfort temperature at different air speeds and temperatures for several activity levels



Average airspeed = 0.15 m/s

	$\frac{M}{A_{Du}}$ [W/m^2]	Cloting resistance [clo]
1	110	1.0
2	110	0.5
3	70	1.0
4	70	0.5

Figure 4.7, Relation between MRT and air temperature for different activity levels and clothing

4.3.4 Practical applications

Example 1

Which indoor air temperatures should be chosen in an office building during the heating and cooling seasons? Assume that people are sitting and the clo-value of winter clothing is 1.0 clo and in summer 0.5 clo. The air speed is lower than 0.1 m/s, the relative humidity is 40% in winter and 20% in summer. Assume the MRT is always equal to the air temperature.

Figure 4.5 gives for winter: 23.5 °C and figure 4.4 for summer: 26.5 °C.

Example 2

What is the comfort temperature for shop workers (activity = 90 W/m^2, relative air speed = 0.4 m/s, closing = 1.0 clo).

Figure 4.6 gives: 21.0 °C for air temperature and mean radiant temperature.

Example 3

During winter conditions the average radiation temperature (MRT) inside a car will be 6°C lower than that of the air. What should be the comfortable air temperature for passengers without overcoat (activity M/Adu=70 W/m2 and clo=1.0) assuming an air velocity of 0.15 m/s and relative humidity of 0.5.

In figure 4.7, we have to look at comfort line 3. At 22°C both the MRT and the air temperature are equal. If we move to the lower part of line 3, the air temperature increases and the MRT decreases. At an air temperature of 24 °C the mean radiant temperature should be 18 °C (6 degree lower than the air) to ensure comfort

4.4 Assessing thermal comfort with PMV and PPD

By means of experiments in climate chambers with large samples of people Fanger related the thermal sensation of people to the unbalance in the comfort equation (TS in equation 4.1). He asked people about their thermal sensation in different environments (air velocity, air temperature, MRT, humidity) and for different clothing and activities. People had to report their thermal sensation on a scale from -3 to +3 going from feeling cold to feeling hot, see table 4.1. This actual thermal sensation (TS_{actual}) correlates with the thermal unbalance TS calculated according to equation 4.1. The calculated thermal unbalance TS transformed to the scale from -3 to +3 is called the PMV: Predicted Mean Vote. The PMV therefore predicts the average thermal sensation of large groups (how most people would feel) of people based on comfort equation 4.1.

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

Table 4.1 Thermal sensation scale (Predicted Mean Vote, PMV)

People also had to report if they were satisfied with the thermal environment.

<u>Note that a neutral sensation does not mean thermal satisfaction. It may be that people at that moment would like to feel colder or warmer. They will then report dissatisfaction. It is also possible that people report a cold sensation (-3) and like this sensation at that moment. They</u>

will then report to be satisfied.

In the large scale studies by Fanger, the percentage of people dissatisfied was plotted against the PMV. Fitting the data points resulted in a curve, described in equation 4.2 and in figure 4.8. This equation therefore predicts the percentage of people dissatisfied by the thermal environment (PPD: Predicted Percentage of people Dissatisfied).

$$PPD = 100 - 95e^{-0.03353PMV^4 + 0.2179\,PMV^2} \quad [\%] \quad \text{(Equation 4.2)}$$

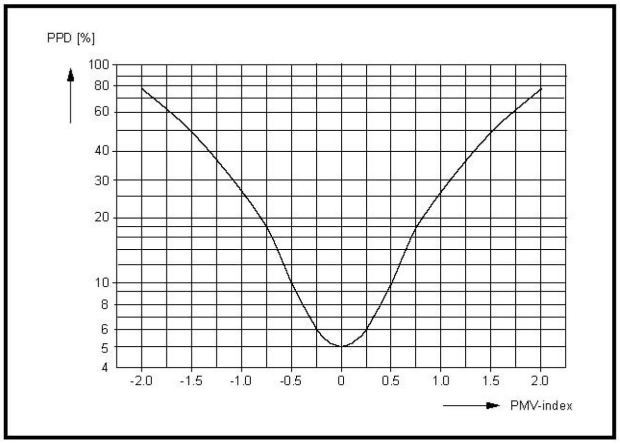


Figure 4.8, Predicted Percentage of people dissatisfied (PPD) against Predicted mean Vote (PMV)

Using the 6 comfort parameters in the comfort equation 4.1 allows to determine the PMV. The PMV in combination with equation 4.2 leads to the PPD. It is this way possible to predict in advance the percentage of (large groups of) people who will be satisfied or dissatisfied with a certain environment.

Figure 4.8 shows that even at neutral conditions 5% of people will be dissatisfied. At warm and cool conditions (PMV +2 and -2 respectively), 80% of people are dissatisfied, meaning that still 20% like it. Note that the curve is symmetrical.

In ISO standard 7730 [1] an thermal indoor environment is classified category A if the PPD is below 6%, category B if it is between 6 and 10% and category C if it is between 10 and 15%.
More information can be found in ISO standard 7730 [1]. There is there an algorithm for calculating PMV and PPD. Most energy simulation software (like Energy Plus, DOE, IES etc..) have a module for the calculation of PMV and PPD too.

There are limitations to the use of PPD and PMV. For valid predictions it should be used only in the following ranges:

- M (Metabolism): 46-232 W/m^2
- Icl (Clothing resistance): 0-2 clo
- Ta (Air temperature indoors): 10-30°C
- MRT (Mean Radiant Temperature): 10-40°C

- v (Air velocity): 0-1 m/s
- Pa (Partial water vapour to measure relative humidity): 0-2.7 kPa, corresponding to a range of absolute humidity of 0-17g/kg.

Besides the six comfort parameters use in Fangers' model, other factors could affect thermal comfort, like age, gender, illness, habits, climate or acclimation. There are not accounted for in Fanger's model.

4.5 Local thermal discomfort

In the previous comfort equation (equation 4.1) and related PPD and PMV, the air temperature was considered homogeneous and the diverse surface temperatures were averaged in the MRT. In reality the air temperature may vary from ground to ceiling, the surface temperatures may differ from each other (think of cold glazing in winter and a warm partitioning wall to another room), and there could be local draught, close to a window for instance.

Additionally the human body was seen as a whole, while it is possible that only a part of the body is heated by radiation or submitted to draught and not all part of a body are equally covered by clothing.

It is therefore possible that although the total body is thermally in balance, the person feels local thermal discomfort. This is especially so at low metabolism (office job). Raising or lowering the average temperature of the enclosure cannot remove this discomfort. It is necessary to remove the cause of the localized thermal discomfort. The causes may be:

- Draught and turbulence
- Large radiation asymmetry
- Large vertical air temperature gradients
- Too low or too high floor temperatures

4.5.1 Draught and turbulence

People at light sedentary activity, with an overall thermal sensation close to or lower than neutral, show to be sensitive for locally too high air speed. Sensitive spots are the neck and the ankles. This discomfort is named "draught". A general rule in air-conditioned buildings is that the air speed should be lower than 0.25 m/s in summer and 0.15 m/s in winter.

Discomfort is also dependent on the turbulence of the air. Turbulence intensity (T_U) is defined as the ratio of the standard deviation of the air velocity to the mean air velocity and is a measure of the changing speed and direction of air flows.

Fanger et all [4] derived a specific equation to predict the percentage of people dissatisfied by draught, the so-called Draught Rating index (DR).

$$DR=(34-T_a)(\tilde{v}-0.05)^{0.62}(3.14+0.37.Tu.\tilde{v}) \quad [\%]$$

With:
T_a: air temperature (^oC) (range is limited to 19-27^oC)

Tu=100.σ/ ṽ (range is limited to below 60%)
σ: standard deviation
ṽ: average air speed (m/s)

Figure 4.9, obtained by Fanger and Christensen [5] who investigated 150 subjects exposed to various average air speeds in the neck, gives the relation between air temperature, turbulence intensity and the average air speed for 15% dissatisfied people (DR=15%). For example at an air temperature of 22oC, air speed of 0.2 m/s, 15% of people will be dissatisfied if the turbulence is 10%. At higher turbulence, more people will be dissatisfied.

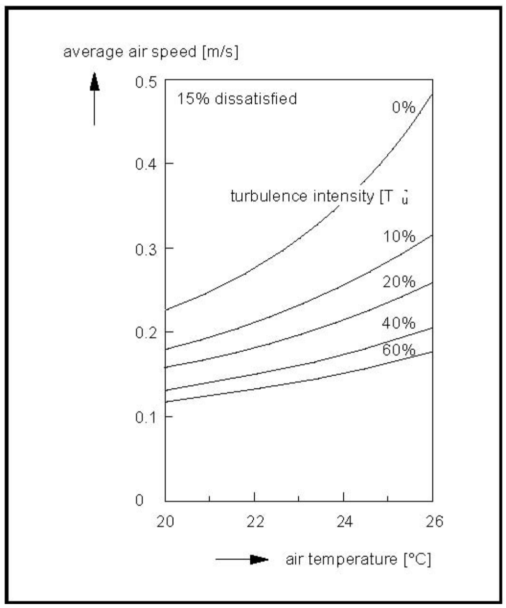


Figure 4.9, Percentage of people dissatisfied (DR) and temperature difference between head and feet

4.5.2 Large radiation asymmetry

Complains about radiant asymmetry arise when there are too large differences between surface temperatures even if the MRT is in the comfort range. This may happen for instance when the ceiling has a too high temperature (e.g. if the ceiling is not insulated and absorbs a lot of solar radiation) or in cases of cold windows [6,7].

In figure 4.10 the relation between the percentage of people dissatisfied (PD) and radiant asymmetry is shown. Radiant asymmetry is defined as the difference in surface temperatures between the surface under consideration and the surface in the opposite direction.

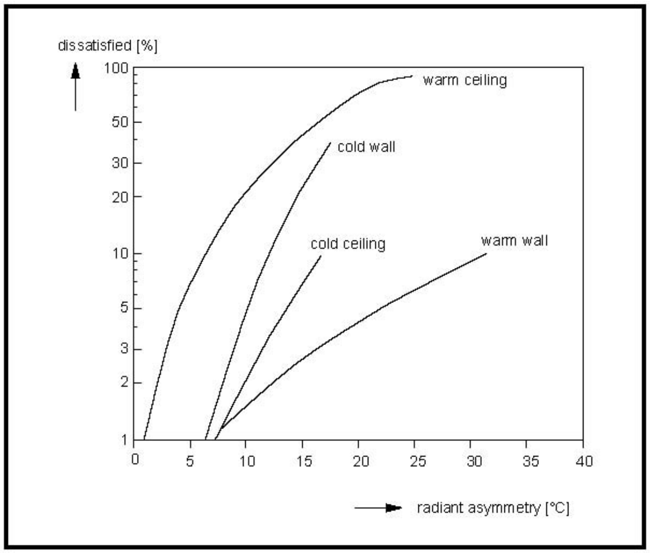


Figure 4.10, Percentage of people Dissatisfied (PD) against radiant asymmetry.

Note, that for the local comfort we use PD and not PPD. PPD is reserved for the comfort equation.

Example 1:
For the warm ceiling, the radiant asymmetry is the temperature difference between ceiling and floor. If the temperature of the (warm) ceiling is lower than the floor temperature plus 5°C, less than 10% of people will be dissatisfied.

Example 2:
For the cold ceiling, the radiant asymmetry is the temperature difference between ceiling and floor. If the temperature of the (cold) ceiling is higher than the floor temperature minus 15°C, less than 9% of people will be dissatisfied. By comparing example 1 and example 2 one can deduce that people are bothered more by warm ceilings than by cold ones.

Example 3:
For a cold wall, the radiant asymmetry is the temperature difference between the wall and the opposite wall. If the temperature of the (cold) wall is higher than the temperature of the opposite wall minus 15°C, less than 30% of people will be dissatisfied. By comparing example 1 and example 2 one can deduce that people are bothered more by warmer ceilings than by colder ones.

Example 4:
For a warm wall, the radiant asymmetry is the temperature difference between the wall and the opposite wall. If the temperature of the (warm) wall is lower than the temperature of the opposite wall plus 15°C, less than 3% of people will be dissatisfied. By comparing example 3 and example 4 one can deduce that people like warmer walls much more than colder ones.

In ISO 7730 [1] following maximum values are advised about the maximum surface temperature asymmetry: for cold walls 10 K, form warm ceiling 5 K. This corresponds with 8%

people dissatisfied.

4.5.3 Large vertical air temperature gradients

Generally it is unpleasant to have a warm head while the feet are cold. Experiments were carried out with people in a state of thermal neutrality, which were exposed to vertical temperature gradients. Olesen et all. [8] found a relation between the percentage of people dissatisfied and the temperature difference between head and feet, see figure 4.11. A temperature difference of 3 K will lead to 5% people dissatisfied, while a temperature difference of 5 K will lead to almost 30% of people being dissatisfied.

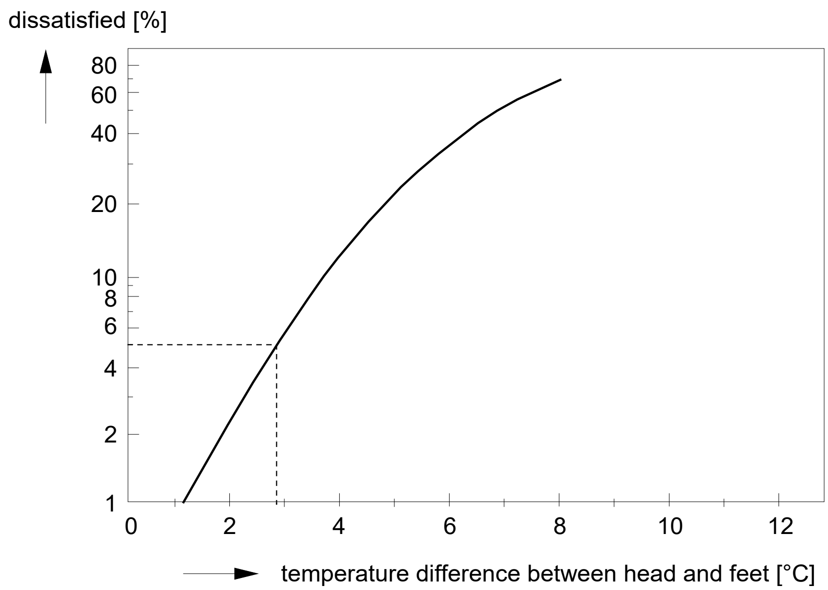


Figure 4.11, Percentage of people Dissatisfied (PD) against air temperature difference between feet and head.

4.5.4 Warm or cold floor

Due to the direct contact between feet and floor, local discomfort of the feet can be caused by too high or too low floor temperature, especially in bathrooms, swimming pools etc… It is caused by conductive heat transfer from the feet to the floor, depending therefore also on the conductivity and the heat capacity of the material the floor is made from. From every day live we know that cork floors are 'warm' and marble floors are 'cold'. In figure 4.12 the relation between PPD and the floor temperature is given. Even at the optimum floor temperature of 24ºC 6% of people are still dissatisfied.

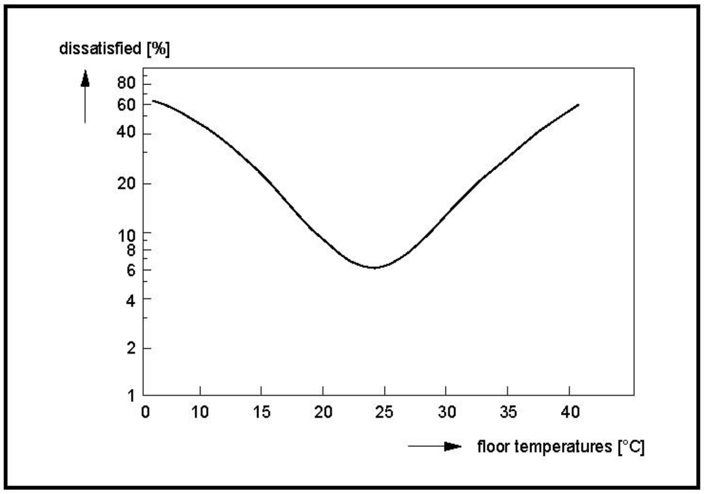


Figure 4.11, Percentage of people Dissatisfied (PD) against floor temperature.

5. Adaptive comfort theory

5.1 Introduction

Fangers' model, with its six comfort parameters does not account for other factors like adaptation. The approach based on thermal balance and thermal neutrality neglects the ability of people to take action to restore their comfort ([9], Humphreys and Nicols 1998). PMV and PPD are calculated in steady-state conditions, meaning that during the measurements that led to Fangers' model people wore specific clothing and carried out specific tasks. However, in real life people can take action when they feel discomfort. For instance they can put on or take off a sweater , open or close a window, a solar blind, or even change activity. If people are able to adapt this way their comfort and environment, they could accept environmental conditions outside of the range given by steady-state models like the Fanger's one and the related PMV and PPD.

Already in 1978 Humphreys [10] noted that people typically use meteorological information about expected daily temperatures, wind and rain to decide what to wear on a specific day. Additionally what they wear is also dictated by the season and the temperatures in the past recent days. In moderate and cold countries people accept lower indoor temperatures in winter, and can accept high indoor temperatures in summer. It was also shown by de Dear and Brager in 2002 [11] that people in naturally ventilated buildings (where they can open the windows) accept a much wider range of conditions than in fully air-conditioned building.

5.2 Operative comfort temperature model and relationship with PMV

A simple mode was proposed relating the operative comfort temperature T_c to the average outdoor temperature in the past days $T_{mean,out}$ in a linear way, see figure 5.1:

$T_c = a.T_{mean,out} + b$ (Equation 5.1)

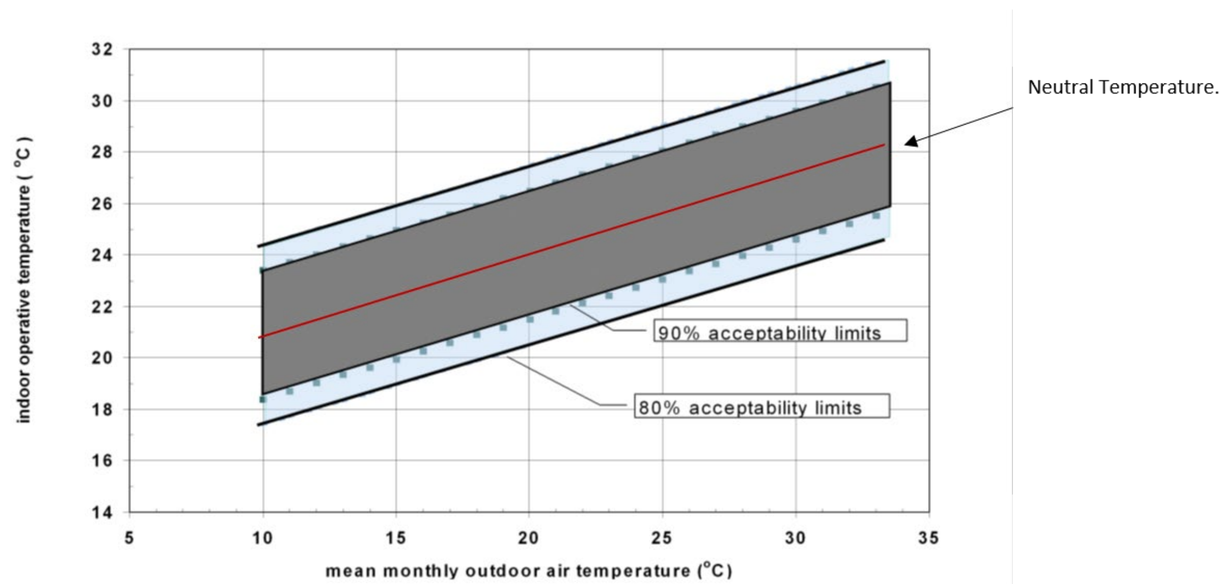
For naturally ventilated buildings the proposed equation in ASHRAE standard 55 [12] is

$T_c = 0.31T_{mean,out} + 17.8.$

This temperature is called the neutral temperature. This equation was obtained by fitting temperature data of hundreds of buildings all over the world. It is therefore an average. Although this equation seems very simple, there are relationships with Fanger's model:

- Tc is not the air temperature, but the operative temperature as defined in section 3.7.
- Using numerous field studies it has been possible to relate the comfort temperature T_c to PMV-PPD. It is usual to show in diagrams the 80% and 90% thermal acceptability, corresponding to 20% and 10% of people dissatisfied. The neutral temperature corresponds to a PMV around zero.

Figure 5.1 shows the comfort operative temperature against the monthly outdoor temperature. The grey range corresponds to 10% dissatisfied people according to PMV-PPD and the blue range corresponds to 20% dissatisfied people.



Based on ASSRAE 55

Figure 5.1, Comfort (operative) temperature against mean monthly outdoor air temperature Tmean,out (source: ASHRAE 55)

5.3 Regional adaptive models

For more accuracy, the constant a and b in equation 5.1 could also be determined for each specific climate zone to account for regional preferences.

For instance in Belgium and the Netherlands, the equations by Peeters et al.[13] can be used [14]

Tc=0.06$*T_{\text{mean,out}}$ +20.4 for $T_{\text{mean,out}}$ < 12.5 oC
Tc=0.36$*$ $_{\text{mean,out}}$ + 16.63 for $T_{\text{mean,out}}$ ≥ 12.5 oC

Where $T_{\text{mean,out}}$ is defined by:

$T_{\text{mean,out}} = (T_{today} + 0.8T_{today-1} + 0.4T_{today-2} + 0.2T_{today-3})/2.4$

With Ttoday being the average of the day's maximum and minimum outside temperatures (°C) and Ttoday-1, Ttoday-2, and Ttoday-3 are the average of maximum and minimum outside temperatures (°C) for yesterday, two and three days before, respectively.

The upper and lower temperature acceptability limits are defined

$T_{upper}=T_{\text{c}}+w.a$
$T_{lower}=$ max $(T_{\text{c}}-w\ (1-a),18)$

The value of w for 90% acceptability is 5°C and for 80% acceptability 7°C. Furthermore, the width of the comfort band is not split symmetrically around the neutral temperature, rather a 70-30% split is recommended, which resulted in an a equal to 0.7.

5.4 Limitations and extensions

In principle the adaptive comfort standard is limited to naturally ventilated buildings (so no air-conditioning), regulated by closing and opening windows, the occupants must have sedentary activities (1-1.3 MET) and must be able to change freely their clothing. The outdoor temperature range is 10-33°C. In figure 5.1 an additional limitation is that no space heating should be used. This limitation does not apply for the equation in section 5.5.

For instance in the Netherlands, an addition has been proposed for summer conditions in air-conditioned buildings, see figure 5.2. In this figure, the remote temperature is $T_{\text{mean,out}}$ defined by RMOT in the figure (it is therefore again a different definition than in section 5.4).

All temperatures between the upper and lower limit are accepted by 90% of the people. The lower limit is 20°C while the upper limits is increasing with the outdoor temperature. It shows that people in buildings accept much higher indoor temperature when they have control possibilities such as opening windows for natural ventilation. At a remote outdoor temperature of 28°C, an indoor temperature of 28°C is accepted, while in buildings with air conditioning and closed facades the limit is 25°C. Therefore two upper limit curves are given.

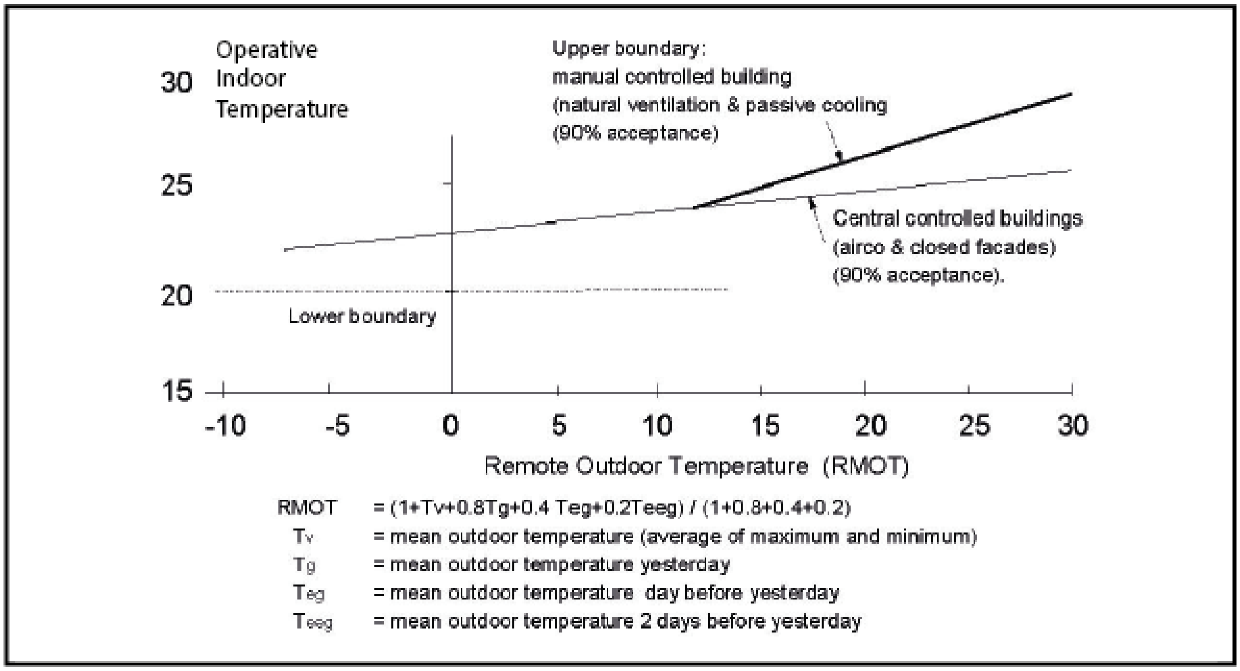


Figure 5.2, Adaptive operative comfort temperatures in the Netherlands

3. Appendix A: Further reading on MRT for the ones with a background in Thermodynamics

A.1 Radiation heat transfer

The equation representing the radiation energy emitted by a surface Ai is (Stefan Boltzmann equation):

$Q_i = \sigma.\varepsilon_i.A_i.T_i^4$ [W]

with:
T_i = surface temperature [K]
σ = constant of Stefan en Boltzmann [$5.67 \cdot 10^{-8}$ W/m^2 K^4]
ε_i = emissivity of the surface. Most materials in buildings can be considered gray and their emissivity is between 0.85 and 1 (a body with emissivity 1 is said to be a black body). Some exception are polished metals like aluminium of copper, with an emissivity of 0.1.

The radiative heat flow Q_{iz} emitted by surface A_i and received by a surface A_z 'seeing' surface Ai is:

$Q_{iz} = F_{i\text{-}z}.\sigma.\varepsilon_i.A_i.T_i^4$ [W]

where $F_{i\text{-}z}$ is a <u>purely geometrical factor</u> called the view factor (or the angle factor) determining the fraction of the total heat emitted by surface A_i i that is received by A_z. ($F_{i\text{-}z}$=(radiation emitted by i and received by z)/radiation emitted by i). The sum of all view factors must always be 1 ($F_{p\text{-}1}+F_{p\text{-}2}+\ldots+F_{p\text{-}n}=1$).

When two surfaces can 'see' each other and have a different temperature, they will exchange heat, by which the hotter surface will give heat to the colder one.

$Q_{iz} = F_{i\text{-}z}.\sigma.\varepsilon_i.A_i.T_i^4$ (emitted by i, received by z)
$Q_{zi} = F_{z\text{-}i}.\sigma.\varepsilon_z.A_z.T_z^4$ (emitted by z, received by i)

The net quantity of heat gained (or lost) by A_z is then

$Q_{zi,net} = Q_{iz}-Q_{zi} = \varepsilon_i.A_i.F_{iz}.\sigma.T_i^4 - \varepsilon_z.A_z.F_{zi}.\sigma.T_z^4$

As already mentioned the emissivity of most surfaces in buildings are quite identical (this is of course a simplification); leading to

$Q_{zi,net} = Q_{iz}-Q_{zi} = \varepsilon.\sigma.(A_i.F_{iz}.T_i^4 - A_z.F_{zi}.T_z^4)$

Looking at a specific case of this general equation will help to determine relationships between geometrical factors and to simplify the equation. When both surfaces have identical temperatures and emissivity (T_i=T_z), there is by definition no net radiative heat exchange, so, in this case

$A_i.F_{iz}.T_i^4$ - $A_z.F_{zi}.T_z^4$ =0; because T_i=T_z this leads to

$A_i.F_{iz}$. -.$A_z.F_{zi}$ =0

This is a general rule about view factors (reciprocity relation):
$A_i.F_{iz}$ = $A_z.F_{zi}$.

By replacing $A_i.F_{iz}$ by $A_z.F_{zi}$ in the equation for $Q_{zi,net}$ we get:

$Q_{zi,net}$ = ε.$A_z.F_{zi}.(T_i^4 - T_z^4)$

A.2 Calculation of the mean radiant temperature MRT

The mean radiant temperature MRT is defined as the equivalent uniform temperature of a black enclosure in which the occupant would exchange the same amount of radiant heat as in the actual non-uniform enclosure.

Let's model the occupant as a surface Z with area A_z . For now it doesn't matter what the shape of this surface is. The occupant receives and emits radiation from all surrounding surfaces (walls, windows, floor, ceiling), so the net radiation flux on Z is the sum of all this net fluxes:

$Q_{z,net} = ε.A_z.Σ.F_{zi}(T_i^4 - T_z^4)$ (a)

By definition MRT represents the equivalent uniform temperature of the enclosure surface A_e (A_e= Σ.A_i), which is considered to be black (ε=1). Therefore:

$Q_{z,net} = A_zF_{ze}(MRT^4-T_z^4)$ (b)

Because the enclosure is completely surrounding A_z (the occupant), all radiation emitted by the occupant is received by the enclosure A_e. In other words the view factor F_{ze} is 1.
Bringing equations (a) and (b) together leads to:

$ε.A_z.Σ.F_{zi}(T_i^4 - T_z^4) = A_z(MRT^4-T_z^4)$

$ε. Σ.F_{zi}(T_i^4 - T_z^4) = MRT^4-T_z^4$

$ε. Σ.F_{zi}.T_i^4 - ε.T_z^4\,Σ.F_{zi} = MRT^4-T_z^4$ (because the terms without index i can be put out of the summation Σ on i).

By definition $Σ.F_{zi} = F_{ze}$ =1; Therefore
$ε. Σ.F_{zi}.T_i^4 - ε.T_z^4 = MRT^4-T_z^4$

$MRT^4 = \varepsilon. \Sigma.F_{zi}.T_i^4 + (1-\varepsilon).T_z^4$

Because the emissivity if generally close to 1, (1-ε) is almost zero. The last term can be neglected. For the same reason, (ε close to 1), we end up with the same formula as in equation 3.1:

$MRT^4 = \Sigma.F_{zi}.T_i^4$

Be aware that this equation contains a lot of simplifications, especially about the emissivity. In specific cases were the emissivity cannot be considered close to 1, or if the emissivity's of the different surfaces differ a lot, the equation should be rewritten on an appropriate way.

A.3 Correction on the MRT for the presence of a high temperature radiant heater
In cold countries, radiators are generally placed on (outer) walls and under the window. This is because the hot radiator with temperature between 40-70°C, or even up to 90°C in poorly insulated dwellings, not only brings the needed heat but is also used to compensate for the cold surface temperature of (outer) walls and windows. In section 3.6 a very rough estimate of the average temperature of the wall with radiator was made and it was mentioned that a MRT calculation should be made.

However, the equation for calculating the mean radiant temperature needs an adaptation when a high temperature radiant source is installed in the space [15]. It is done as follows.

Let $MRT_{unheated}$ be the MRT calculated as in A.2, without taking the radiant surface into account. The MRT with the radiant heater must then be calculated as:

$MRT^4 = MRT^4_{unheated} + 0.182.10^8.f_p.a_{ir}.Q_{ir}$

With:
Qir = radiated heat from the heater (W/m²)
a_{ir} = absorption coefficient of the wall located at the opposite side of the heater
$f_p = A_p/A_{eff}$ where Ap is the surface of the wall opposite to the heater and A_{eff} is

4. Appendix B: Further reading on human body's heat balance equation for the ones with a background in Thermodynamics

See also description in [3].

Metabolism M and work W:
In order to keep the core of the body at 37°C a certain heat flow should be generated in the body by metabolism M (e.g digesting food). Almost 100% of the incoming energy is converted to heat in the body. During hard labor (muscle activity is nothing else than mechanical work) the heat production is reduced to 75%. 25% of the incoming energy is then used for the external mechanical work (W). Only a small part of the metabolism is used for mechanical work (W).

Our metabolism is at lowest while we sleep (0.8 Met) and it is highest during sport activities (10 Met). A rate commonly used is 1.2 corresponding to normal office work, see section 3.1. Domestic work gives Met-values between 2.5 and 2.9. Because of the body heat capacity we should take the average activity during the last hour, so the energy balance will be made on hourly basis.

Convection and radiation K:
K the (dry) heat transferred via the skin and clothing to the surrounding can be calculated by:

$$K = \frac{1}{I_{cl}} \cdot A_{Du} \cdot \left(t_s - t_{cl}\right)$$

with I_{cl} = heat resistance of clothing [$m^2 \cdot K/W$]
A_{Du} = naked body surface area (m^2)
t_s = skin temperature; at comfort it is a function of M/A_{du}[ºC]
t_{cl} = average surface temperature of clothing [ºC]

The heat flow K is transferred to the surrounding by convection C and radiation R (K=C+R)

$$C = A_{Du} \cdot f_{cl} \cdot h_c \cdot \left(t_{cl} - t_a\right)$$

with f_{cl} = ratio between the outside surface of a dressed person and his nude outside surface
h_c = convective heat transfer coefficient

$$\begin{aligned} R &= A_{eff} \cdot \varepsilon \cdot \sigma \cdot \left\{T_{cl}^4 - \left(MRT\right)^4\right\} = \\ &= f_{eff} \cdot f_{cl} \cdot A_{Du} \cdot \varepsilon \cdot \sigma \cdot \left\{T_{cl}^4 - \left(MRT\right)^4\right\} = \\ &= 3.90 \cdot 10^{-8} \cdot A_{Du} \cdot f_{cl} \cdot \left\{T_{cl}^4 - \left(MRT\right)^4\right\} \end{aligned}$$

with: A_{eff} = effective surface that takes part to the exchange by radiation = $f_{eff} \cdot A_{Du}\ f_{cl}$

f_{eff} = effective radiation coefficient of the dressed body =0.71
ε = emission coefficient for radiation of clothing (0.97)
σ = constant of Stefan and Boltzmann = $5.67 \cdot 10^{-8}$ [$W/m^2 \cdot K^4$]
T_{cl} = temperature of the outside surface of clothing
MRT = average radiation temperature in K

Dry air heat transfer rate through respiration L:
The dry heat losses of breathing air are:

$$L = \dot{V} \cdot c_p \cdot (t_{ex} - t_a)$$

with: c_p = specific heat of air at constant pressure [J/kg · K]
t_a = temperature of air breathing in [ºC]
t_{ex}= temperature of air breathing out [ºC]

c_p and t_{ex} are almost constant so that:

$$L = C_5 \cdot M \cdot (C_6 - t_a)$$

with C_5 and C_6 are constants.

Evaporative heat transfer rate from vapor in respiration E_{re}:

$$E_{re} = \dot{V} \cdot r \cdot (x_{ex} - x_a)$$

with: $\dot{V}$ = mass flow of breathing [kg/s]
x_a = water content of air breathing in [kg/kg]
x_{ex} = water content of air breathing out [kg/kg]
r = latent heat of evaporation t of water [J/kg]

Because of the fact that V is proportional with M, x_{ex} is almost constant and x_a is proportional with the partial vapor pressure p_a of the air, the equation can be written as:

$$E_{re} = C_3 \cdot M \cdot (C_4 - p_a)$$

with C_3 and C_4 are constants.

Evaporative heat transfer by water diffusion through skin E_d:

$$E_d = r \cdot \beta \cdot A_{Du} \cdot (p_s - p_a)$$

with: r = latent heat of evaporation of water [J/kg]
β = Vapor transfer coefficient of clothing [$kg/m^2 \cdot Pa \cdot s$]
A_{Du} = Naked body surface
p_s = Water vapor pressure on the skin (saturation pressure at skin temperature) [Pa]
p_a = Water vapor pressure of the air [Pa]
Data about moister transfer of clothing can be found in ASHRAE Handbook Fundamentals [3].

Evaporative heat transfer by sweating E_{sw}:

At comfort conditions it is assumed that all water is evaporated.

$$E_{sw} = C_1 \cdot A_{Du} \cdot \left(\frac{M - W}{A_{du}} - C_2 \right)$$

where C_1 and C_2 are constants and

$$\frac{M - W}{A_{Du}} > C_2$$

By replacing all terms in equation 4.1 by their value, one comes to the comfort equation described in Equation B.

$$\underbrace{\frac{M}{A_{Du}} (1 - \eta)}_{M - W} - \underbrace{3.05 \cdot 10^{-3} \left[5733 - 6.99 \cdot \frac{M}{A_{Du}} (1 - \eta) - p_a\right]}_{E_d} -$$

$$\underbrace{0.42 \cdot \left[\frac{M}{A_{Du}} (1 - \eta) - 58.15 \right]}_{E_{sw}} -$$

$$\underbrace{1.7 \cdot 10^{-5} \frac{M}{A_{Du}} (5867 - p_a)}_{E_{re}} - \underbrace{1.4 \cdot 10^{-3} \cdot \frac{M}{A_{Du}} \cdot (34 - t_a)}_{L} =$$

$$= \underbrace{\frac{1}{I_{cl}} \cdot (t_s - t_{cl})}_{K} =$$

$$\underbrace{3.96 \cdot 10^{-8} \cdot f_{cl} \cdot (T_{cl}^4 - MRT^4)}_{R} + \underbrace{f_{cl} \cdot h_c \cdot (t_{cl} - t_a)}_{C}$$

Equation B, Comfort equation

To come to this equation Fanger determined additional relationships between the skin temperature t_{s},, the saturated vapor pressure p_s at that temperature and the sweat production E_{sw} who appeared to all be functions of the metabolism per m^2 body surface. The function describing the skin temperature at comfort as a function of M was found empirically:

$$t_s = 35.7 - 0,032 \frac{M - W}{A_{Du}}$$

p_s=1.92t_s-25.3

Additionally the mechanical labor efficiency is by definition: η = W/M

h_c is calculated by:

$$h_c = \begin{cases} 2.38 \cdot (t_{cl} - t_a)^{0.25} \text{ als } 2.38 \cdot (t_{cl} - t_a)^{0.25} > 12.1\sqrt{v} \text{ (vrije convectie)} \\ 12.1\sqrt{v} \quad \text{ als } 2.38 \cdot (t_{cl} - t_a)^{0.25} > 12.1\sqrt{v} \text{ (gedwongen convectie)} \end{cases}$$

('vrije convectie'= natural convection'; 'gedwongen convectie'= forced convection)

After elimination of t_{cl} in the previous double equation and substitution of the empirical function for t_s the resulting equation with only 6 variables is found:

M/A_{du}, I_{cl}, ta, MRT, v en Pa

(f_{cl} depends on I_{cl}; the efficiency η= W/M is low and can be ignored)
The resulting equation is rather complex and is not given here.

5. References and further reading

Further reading:

P.M. Bluyssen: The Indoor Environment handbook: How to make building healthy and comfortable; Earthscan and RIBA publishing, 2009

Mean Radiant Temperature calculations:

http://www.healthyheating.com/Definitions/Mean%20Radiant_pg5.htm#.YHAIq9yxVhF or http://www.thermalradiation.net/calc/sectionc/C-10.html)

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