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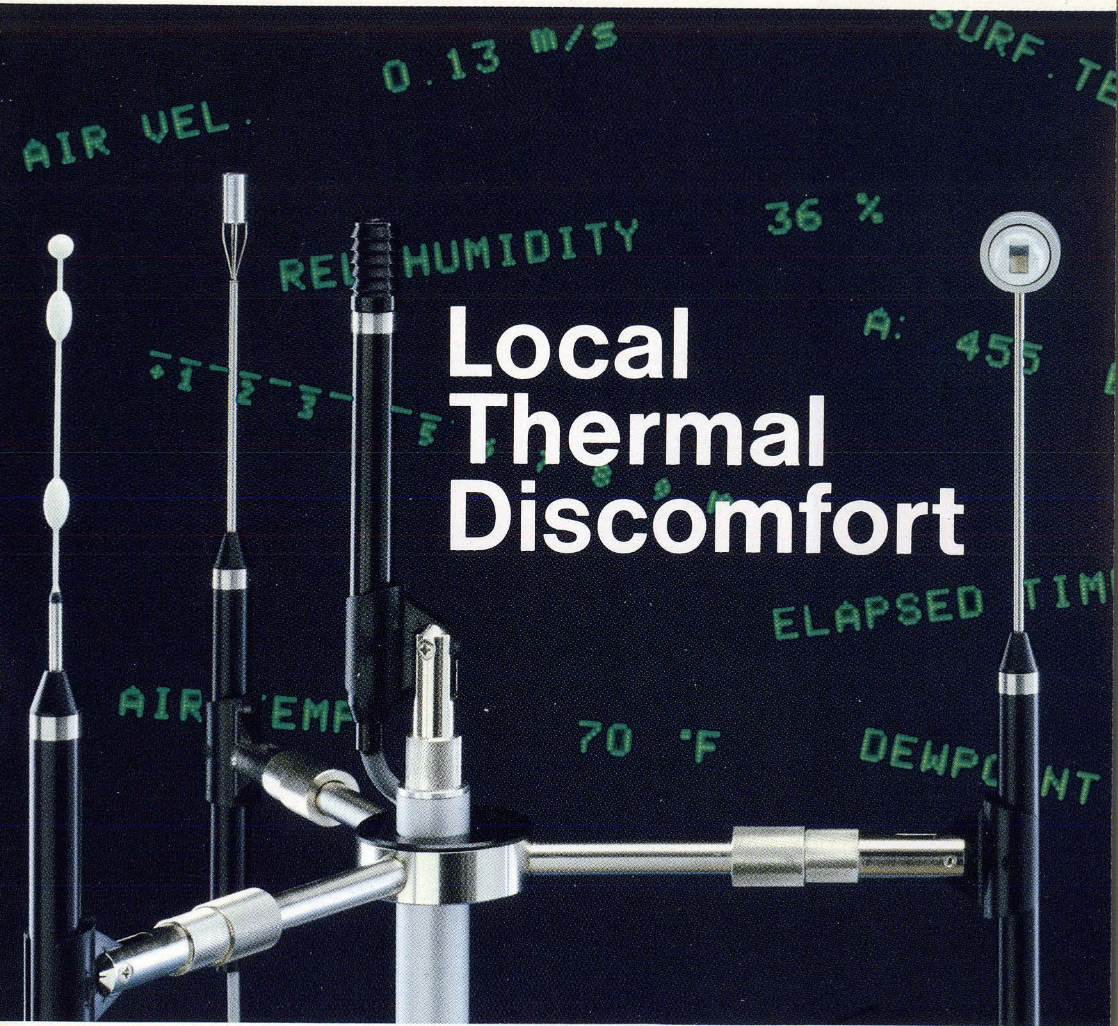

Local Thermal Discomfort

Brüel & Kjær

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(Continued on cover page 3)

TECHNICAL REVIEW

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Contents

LOCAL THERMAL DISCOMFORT

by

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ABSTRACT

The main purpose of most heating and air-conditioning systems is to provide an acceptable thermal environment for human beings. For the design and evaluation of a building/workplace, it is therefore essential to establish international standards which quantitatively specify requirements for acceptable thermal environments.

It is well known that one of the requirements to be fulfilled is that a person be in thermal neutrality as predicted by the comfort equation. This is described and evaluated by the PMV-PPD indices, which take into account the six parameters: activity, clothing, air temperature, mean radiant temperature, air velocity and humidity.

A further requirement is that no local thermal discomfort exists at any part of the human body. Such local discomfort may be caused, for example, by asymmetric thermal radiation, by local convective cooling (draught), by vertical air temperature gradients or by contact with warm or cold floors. Each of these causes of local thermal discomfort will be dealt with in the paper and limits for avoiding such discomfort will be presented.

SOMMAIRE

Le but de la plupart des systèmes de chauffage et de conditionnement d'air est de créer un environnement thermique acceptable pour les hommes. Il est par conséquent essentiel, pour concevoir et évaluer des locaux d'habitation et de travail, d'établir des normes internationales spécifiant quantitativement les conditions nécessaires pour un environnement thermique acceptable.

Il est bien connu que l'une des conditions à remplir, d'après l'équation de confort, est que la personne soit en neutralité thermique. Cette condition est traduite et évaluée par les indices PMV et PPD qui tiennent compte de six paramètres: activité, habillement, température de l'air, température moyenne de rayonnement, vitesse de l'air et humidité.

De plus, il faut qu'aucune partie du corps ne soit en état d'inconfort thermique. Un inconfort local peut par exemple être causé par une asymétrie de rayonnement thermique, un refroidissement local par convection (courant d'air), un gradient vertical de la température de l'air ou par contact avec un plancher chaud ou froid. Cet article traite de chacune de ces causes d'inconfort thermique local et présente des limites pour éviter de tels inconforts.

ZUSAMMENFASSUNG
Die Hauptaufgabe der meisten Heizungen und Klimaanlagen ist es, ein für den Aufenthalt von Menschen behagliches Raumklima zu schaffen. Für die Auslegung und Beurteilung von Räumen/Arbeitsplätzen ist es daher notwendig, internationale Normen und Richtlinien zu schaffen, die quantitativ die Anforderungen an das Raumklima festlegen.
Eine dieser Forderungen ist bekanntermaßen, daß sich Personen in thermischer Neutralität befinden, wie es sich mit der Komfort-Gleichung vorhersagen läßt. Dies läßt sich mit Hilfe der PMV/PPD-Indizes beschreiben und beurteilen, die sechs Parameter berücksichtigen: Aktivität, Kleidung, Lufttemperatur, mittlere Strahlungstemperatur, Luftgeschwindigkeit und Feuchte.
Eine weitere Forderung ist, daß an keiner Stelle des Körpers örtliches Unbehagen auftritt. Örtliches Unbehagen kann verschiedene Ursachen haben, z.B.: Asymmetrische Strahlungstemperaturverteilung, örtliche Kühlung durch Konvektion (Zug), vertikale Lufttemperatur-Gradienten, oder Kontakt mit zu warmen oder zu kalten Fußböden. In dieser Abhandlung werden die genannten Ursachen für thermisches Unbehagen behandelt und Grenzwerte für die verschiedenen Parameter angegeben.

1. Introduction

In a modern industrial society man spends the greater part of his life indoors. A large proportion of the population spends 23 out of 24 hours in an artificial climate – at home, at the workplace, or during transportation.

This has resulted in growing understanding of and interest in studying the influence of indoor climate on man, thus enabling suitable requirements to be established which should be aimed at in practice.

These requirements have now been published in a new standard ISO 7730 "Moderate thermal environments - Determination of the PMV and PPD indices and specification of the conditions for thermal comfort" [1]. Similar requirements have been introduced in ASHRAE - Standard 55–81 [2] and in NKB guidelines [3].

At the same time an increasing number of complaints about unsatisfactory indoor climate suggest, that man has become more critical regarding the environment to which he is subjected. It seems that he is most inclined to complain about the indoor climate of his workplace (offices, industrial premises, shops, schools, etc.) where he is compelled to spend his time in environments which he himself can control only to a very limited degree. Field studies indicate that in practice many of these complaints can be traced to an unsatisfactory thermal environment.

About one third of the world's energy consumption is used to provide thermal comfort for man. It is no wonder, therefore, that efforts towards energy conservation have led to an increased interest in man's comfort conditions in order to assess the human response to different conservation strategies.

A main purpose of most heating and air conditioning systems is to provide thermal comfort for human beings. For the design and operation of such systems and for the thermal design of buildings, it is essential to establish quantitative comfort requirements. Thermal comfort is defined as "that condition of mind which expresses satisfaction with the thermal environment" (ISO 7730 [1], ASHRAE 55–81 [2]). A first requirement for comfort is that a person feels thermally neutral for the body as a whole, i.e. he does not know whether he would prefer a higher or lower ambient temperature level. Man's thermal neutrality depends on the following six factors:

Personal factors:

Activity level, M (met, W/m^2)
Thermal insulation of clothing, I_{cl} (clo, $m^2\,°C/W$)

Environmental parameters:

Air temperature, t_a
Mean radiant temperature, $\bar{t}_r$
Air velocity, v_a
Air humidity, p_a (water vapour pressure)

The thermal effect of clothing, activity and environmental parameters on man has been studied quite intensively (Fanger, [4], Nevins et al. [7], McNall et al. [6], Rohles et al. [8], Gagge et al. [5]). All combinations of the parameters which will provide thermal neutrality may be predicted from the comfort equation and the corresponding comfort diagrams (Fanger, [4]). This method is also used in ISO 7730 together with the PMV-PPD indices, which describe the influence of the thermal environment on the body as a whole. The development and use of the PMV-index has been described in an earlier Technical Review [9]. A PMV value = 0 will be equivalent to thermal neutrality; but in ISO 7730 the recommended limits for an acceptable thermal environment are $-0{,}5 < \text{PMV} < 0{,}5$. With these limits the expected number of dissatisfied persons will be less than 10% (PPD < 10%).

Thermal neutrality as predicted by the comfort equation or described by the PMV-PPD indices is not the only condition for thermal comfort. A person may feel thermally neutral for the body as a whole, but he might not be comfortable if one part of the body is warm and another cold. It is therefore a further requirement for thermal comfort that no local warm or cold discomfort exists at any part of the human body. Such local discomfort may be caused by an asymmetric radiant field, by a local convective cooling (draught), by contact with a warm or a cold floor, or by a vertical air temperature gradient. Each of these cases will be dealt with in the following and limits for avoiding local discomfort will be discussed. Recommended measuring principles and accuracies will also be described.

2. Asymmetric Thermal Radiation

Asymmetric or non-uniform thermal radiation in a space may be caused by cold windows, non-insulated walls, cold products, cold or warm machinery or by warm or cold panels on the wall or on the ceiling.

In residential buildings, offices, restaurants etc. the most common reasons for discomfort due to asymmetric thermal radiation are large windows in winter or heated ceilings.

At industrial workplaces the reasons are numerous, e.g. cold or warm products, cold or warm equipment, high-temperature infrared heaters etc.

First the parameters used for describing asymmetric radiation are introduced and then the latest results and comfort guidelines related to asymmetric thermal radiation are presented.

Radiant temperature asymmetry

The asymmetric radiant field is described by a parameter called "radiant temperature asymmetry", $\triangle t_{pr}$. This parameter is introduced in ISO 7726 "Thermal environments - Instruments and methods for measuring physical quantities", [10]. **Radiant temperature asymmetry,** $\triangle \mathbf{t_{pr}}$ is defined as the difference between the plane radiant temperature of the two opposite sides of a small plane element. **Plane radiant temperature,** $\mathbf{t_{pr}}$ is then defined as the uniform temperature of an enclosure where the irradiance on one side of a small plane element is the same as in the non-uniform actual environment. The plane radiant temperature is a parameter which describes the radiation in one direction. The mean radiant temperature $\bar{t}_r$

describes the radiation from all surrounding surfaces, which influences the radiant heat exchange between the human body and the environment. Mean radiant temperature is therefore defined in relation to the human body (Fanger [4], ISO 7726 [10]) while plane radiant temperature and radiant temperature asymmetry are defined in relation to a small plane element. (ISO 7726 [10], ASHRAE 55–81 [2]).

The plane radiant temperature, which was first introduced by McIntyre [11], depends on the surface temperature of the surrounding surfaces and the angle factor between a small plane element and the surrounding surfaces. As most building materials have a high emittance (ϵ) it is possible to disregard the reflections, i.e. to assure that all the surfaces in the room are radiantly black.

The following equation (1) may then be used to calculate the plane radiant temperature, t_{pr}:

$$T_{pr}^4 = T_1^4 \cdot F_{p-1} + T_2^4 \cdot F_{p-2} + \text{----} T_n^4 F_{p-n} \tag{1}$$

where

T_{pr} = plane radiant temperature in K, t_{pr} + 273°C

T_n = surface temperature of surface n in K, t_n + 273°C

F_{p-n} = angle factor between a small plane element and surface n

$\Sigma\ F_{p-n} = 1$

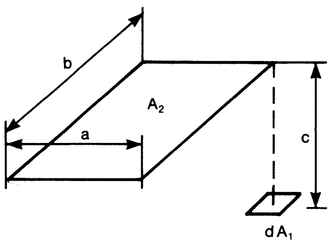
As the sum of the angle factors is unity, the fourth power of the plane radiant temperature can be seen to be equal to the mean value of the surface temperature of the hemisphere to the fourth power, weighted according to the respective angle factors.

If there are only relatively small temperature differences between the surfaces of the enclosure, equation (1) can be simplified by a linear form:

$$t_{pr} = t_1\ F_{p-1} + t_2\ F_{p-2} + \text{------} t_n\ F_{p-n} \quad {}^\circ\mathrm{C} \tag{2}$$

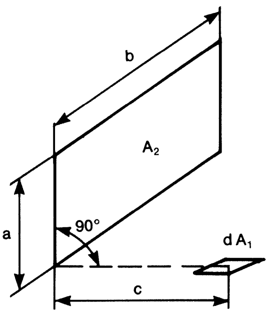
where all the temperatures now are in °C. In other words, the plane radiant temperature is calculated as the mean value of the surface temperatures weighted according to the magnitude of the respective angle factors.

Equation (2) will always give a slightly lower plane radiant temperature than equation (1), but in many cases the difference is small. If, for example, half of the surroundings ($F_{p-N} = 0{,}5$) has a temperature which is 10 K higher than the other half, the difference between the calculated mean radiant temperatures according to equation (1) and equation (2) will thus be only 0,2°C. If, however, there are large differences in temperature between the surfaces, the error by using equation (2) can be considerable. If the temperature difference in the example above is 100 K, the plane radiant temperature will, according to equation (2), be calculated approximately 10 K too low.



$$X = \frac{a}{c} \qquad Y = \frac{b}{c}$$

$$F_{d1-2} = \frac{1}{2\pi}\left(\frac{X}{\sqrt{1+X^2}}\tan^{-1}\frac{Y}{\sqrt{1+X^2}} + \frac{Y}{\sqrt{1+Y^2}}\tan^{-1}\frac{X}{\sqrt{1+Y^2}}\right)$$



$$X = \frac{a}{b} \qquad Y = \frac{c}{b}$$

$$F_{d1-2} = \frac{1}{2\pi}\left[\tan^{-1}\frac{1}{Y} - \frac{Y}{\sqrt{X^2+Y^2}}\tan^{-1}\frac{1}{\sqrt{X^2+X^2}}\right]$$

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Fig. 1. Analytical formulae relating to the calculation of the angle factor for a small plane element

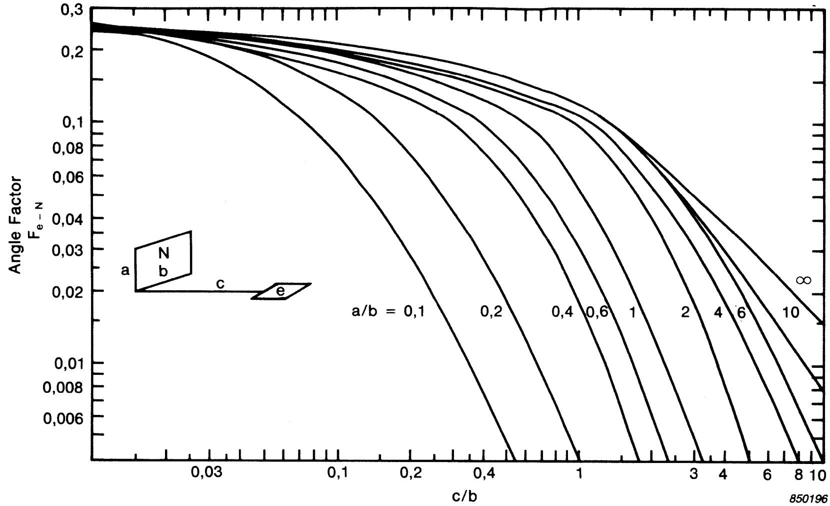

Fig. 2. Diagram for the calculation of the angle factor in the case of a small plane element perpendicular to a rectangular surface

The radiant temperature asymmetry is then calculated as the difference between the plane radiant temperature on the two opposite sides of the small plane element.

The angle factors F_{p-n} can be estimated according to the analytical equations in Fig.1 or the diagrams in Figs.2 and 3, which all are valid for rectangular surfaces. By means of equation (1) or (2) it is possible theoretically to estimate and describe the radiant temperature field in a typical room. The following examples are based on these calculations.

Example 1

In a room (Fig.4) with one double-pane window it is assumed that all surface temperatures are equal to 20°C except for the window which is 8°C (assuming approx. –12° outside temperature). The room is assumed to be heated by warm air heating, so there are no other hot or cold surfaces in the room.

We now want to estimate the radiant temperature asymmetry due to a cold window for a person seated in front of the window (Fig.4). According to ISO 7726 the radiant temperature asymmetry for a seated person is estimated at a level 0,6 m above the floor.

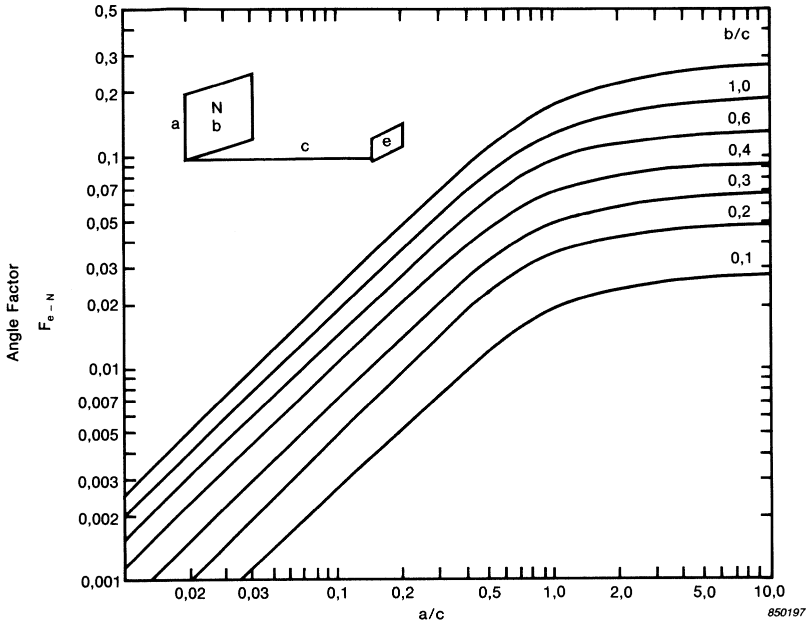

Fig. 3. Diagram for the calculation of the angle factor in the case of a small plane element parallel to a rectangular surface

The angle factor between a small plane element parallel to the window at a distance of 0,6 m and 0,6 m above the floor is estimated by means of Fig.3, $F_{p\text{-}window} = 0{,}428$. As all other surfaces have the same surface temperature, 20°C, and the sum of all angle factors equals one, the plane radiant temperature towards the window is estimated according to equation (2).

$$t_{pr} = 0{,}428 \cdot 8°C + (1 - 0{,}428) \cdot 20°C = 14{,}9°C$$

The plane radiant temperature towards the rear wall is 20°C, since all surfaces in this half room are assumed to have the same temperature, 20°C.

The radiant temperature asymmetry ($\triangle t_{pr}$) is equal to the difference between plane radiant temperatures in the two directions:

$$\triangle t_{pr} = 20°C - 14{,}9°C = 5{,}1°C$$

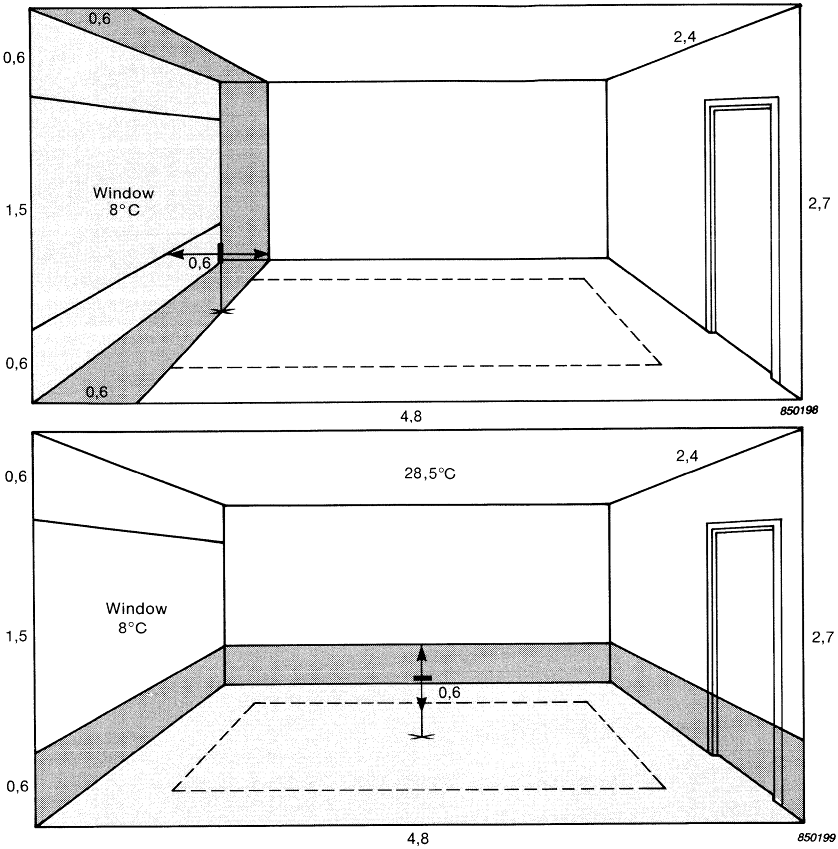

Fig. 4. Room with one outside wall including a window. All dimensions in meters

Example 2

It is now assumed that the same room is provided with a heated ceiling and the ceiling surface temperature is 28,5°C. As in example 1 we want to estimate the radiant temperature asymmetry for a person seated in front of the window. It is now also necessary to calculate the angle factor for the small element towards the ceiling. This is done by Fig.2 because the plane element is perpendicular to the ceiling. The plane radiant temperature towards the window is only influenced by a 0,6 m wide strip of the ceiling along the outside wall. The angle factor towards this part of the ceiling is equal to 0,015. The plane radiant temperature towards the window is then estimated according to equation (2)

$$t_{pr} = 0{,}428 \cdot 8°C + 0{,}015 \cdot 28{,}5°C + (1-0{,}428-0{,}015) \cdot 20°C = 15{,}0°C$$

The plane radiant temperature towards the rear wall is also influenced by the ceiling which now has another surface temperature (28,5°C) than the other surfaces in this direction (20°C). The angle factor towards this part of the ceiling is equal to 0,13.

The plane radiant temperature towards the rear wall is given by:

$$t_{pr} = 0{,}13 \cdot 28{,}5°C + (1-0{,}13) \cdot 20°C = 21{,}1°C$$

The radiant temperature asymmetry is thus:

$$\triangle t_{pr} = 21{,}1°C - 15{,}0°C = 6{,}1°C$$

With a heated ceiling there also exists radiant asymmetry in relation to a horizontal plane. The asymmetry is calculated at the centre of the room (Fig.1) in relation to a small horizontal plane 0,6 m above the floor.

The plane radiant temperature in the direction towards the ceiling is estimated according to equation (2). The angle factor between the ceiling and the small plane element, 0,419, is estimated from Fig.3. The angle factor between the window and the small plane element, 0,039, is estimated from Fig.2.

The plane radiant temperature towards the ceiling is then:

$$t_{pr} = 0{,}039 \cdot 8°C + 0{,}419 \cdot 28{,}5°C + (1-0{,}039-0{,}419) \cdot 20°C = 23{,}1°C$$

In the direction towards the floor all surface temperatures are 20°C, i.e. the plane radiant temperature towards the floor is equal to 20°C.

The radiant temperature asymmetry, $\triangle t_{pr} = 23{,}1-20{,}0 = 3{,}1°C$

These examples show how the radiant temperature asymmetry in a room may be calculated. Later it will also be shown how these parameters can be measured directly.

Local thermal discomfort due to radiant asymmetry

Several studies on the influence of asymmetric thermal radiation on comfort are reported in the literature (McIntyre and Griffiths [12, 13, 14, 17], Chrenko [15], McNall and Biddison [16], Olesen et al. [18]). All studies have been performed with seated subjects. The recommendations in the existing standards (ISO 7730, ASHRAE 55–81) are, however, based on studies at the Technical University of Denmark and reported by Fanger et al. [19, 20, 21, 22]. In the standards there are only guidelines regarding the radiant temperature asymmetry from an overhead warm surface (heated ceiling) and a vertical cold surface (cold window).

To establish a relationship between the radiant temperature asymmetry and the sensation of discomfort, a series of experiments with human subjects has been performed in a climatic chamber. The subjects were seated, dressed in standard clothing ($\sim$ 0,6 clo) and exposed to either an overhead warm surface or a vertical cold surface from one side. During the experiments the radiant temperature asymmetry was increased by increasing the temperature of the overhead warm surface or decreasing the temperature of the vertical cold surface. All the other surfaces in the climatic chamber was kept equal to the air temperature. Changing the temperature on the warm/cold surface also influenced the mean radiant temperature and then the general thermal sensation of the subjects. This was, however, compensated by changing the air temperature according to the subject's wishes. In this way they were always in thermal neutrality and only exposed to the discomfort from an increasing asymmetry.

The subjects gave their subjective reactions on their comfort-sensation and a relationship between the radiant temperature asymmetry and the number of subjects feeling dissatisfied was established. This is shown for an overhead warm surface in Fig.5 and for a vertical cold surface in Fig.6. It must be noticed that these percentage of dissatisfied have nothing to do with the PPD-index, which predicts the percentage of dissatisfied due to a general warm – or cold sensation. In all the experiments the subjects were in thermal neutrality so the percentage of dissatisfied was only related to the radiant temperature asymmetry. It is important to emphasize that the percentage of people feeling local thermal discomfort and the percentage of people feeling generally uncomfortably warm or cold (PPD-value) cannot be added and assumed to be equal to the total number of people feeling dissatisfied.

From Figs.5 and 6 it is seen that people are more sensitive to the asymmetry caused by an overhead warm surface than by a vertical cold surface. The influence of an overhead cold surface and a vertical warm surface was also studied by Fanger et al. [22]. It was found that the subjects were much less sensitive to this form of radiant temperature asymmetry. Therefore it was also decided not to establish any guidelines or limit values for the radiant temperature asymmetry from an overhead cold surface and a vertical warm surface, because in a moderate thermal environment this would in practice never cause any problems.

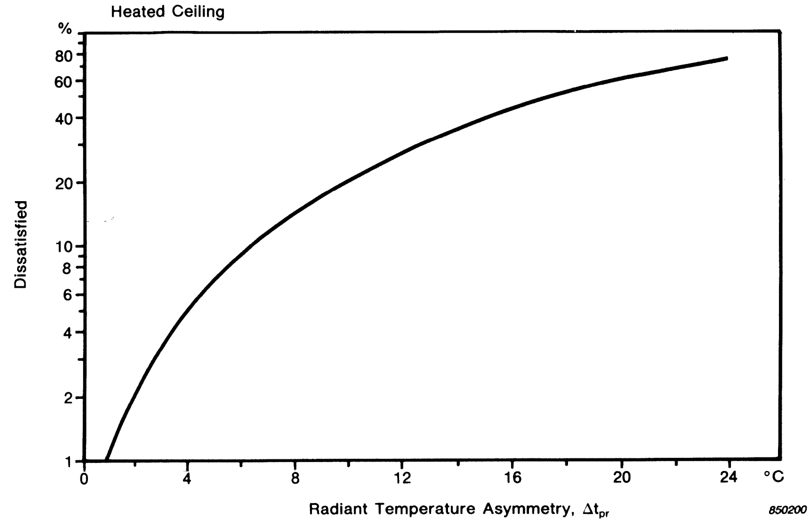

Fig. 5. Percentage of dissatisfied as a function of the radiant temperature asymmetry Δt_{pr} from a heated ceiling

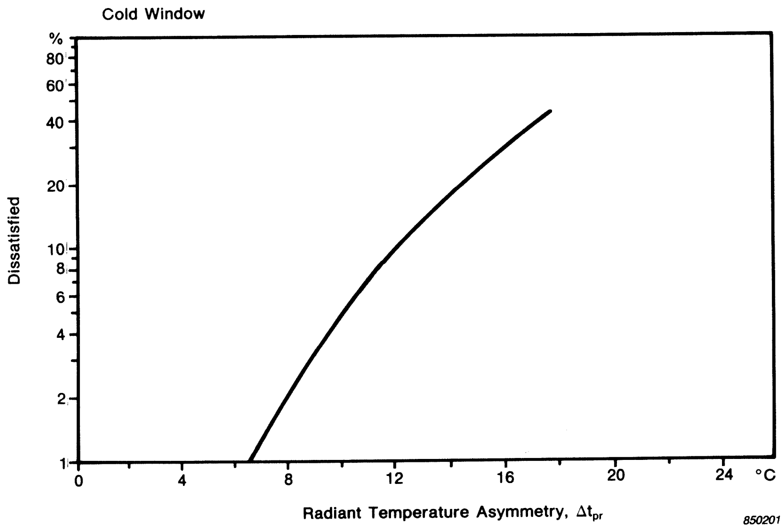

Fig. 6. Percentage of dissatisfied as a function of the radiant temperature asymmetry Δt_{pr} from a cold window/wall

The following recommendations have then been included in ISO 7730, ASHRAE 55–81 and NKB–guidelines:

The radiant temperature asymmetry from windows or other cold vertical surfaces shall be less than 10°C (in relation to a small vertical plane 0,6 m above the floor).

The radiant temperature asymmetry from a warm (heated) ceiling shall be less than 5°C (in relation to a small horizontal plane 0,6 m above the floor).

These recommendations are for people with light, mainly sedentary activity. There are no experimental results yet for people with higher activities. But practice indicates that people with higher activities are less sensitive to radiant asymmetry and thus greater asymmetry is accepted.

As seen from Fig.5 the limit for radiant temperature asymmetry caused by a warm overhead surface will result in approx. 7% dissatisfied while the limit for a cold vertical surface will result in approx. 5% dissatisfied.

Measurement of radiant temperature asymmetry
As described earlier the radiant temperature asymmetry may be estimated by measuring all surface temperatures and calculating the corresponding angle factors. This is, however, often very time consuming and inaccurate. Instead it is recommended to use a direct measurement as performed by the Indoor Climate Analyzer, Type 1213 which is described at the end of this article. This instrument includes a double faced transducer, where the plane radiant temperature is measured in two opposite directions simultaneously. The radiant temperature asymmetry is then given as the difference between the two sides. The sensor is based on a principle using a reflective gold-plate and an absorbing disc (black painted), which will be influenced differently by the radiation from the surrounding surfaces.

In the ISO 7726 Standard measuring accuracies and positions are specified. The recommended measuring range and accuracy are shown in Table 1. When measuring the influence of radiant asymmetry on a person the sensor should be positioned at the abdomen level, i.e. 0,6 above floor level for seated and 1,1 m above floor level for standing persons.

Thermal environ-ment	Measuring range °C	Precision Specified °C	Precision Desirable °C	Comments
Moderate	0 – 20	± 1	± 0,5	Response time as short as possible
Cold and Hot	0 – 20 20 – 200	± 2 $\pm 0{,}1 \cdot \Delta t_{pr}$	± 1 $\pm 0{,}05 \cdot \Delta t_{pr}$	Response time as short as possible

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Table 1. Measuring range and accuracy for radiant temperature asymmetry

3. Draught

Draught is defined as an undesired local cooling of the human body caused by air movement. Draught is a serious problem in practice, not only in many ventilated buildings but also in automobiles, trains and aircraft. Draught has been identified as one of the two must annoying environmental factors in workplaces and the most annoying factor in offices (Bolinder et al. [24], Arbejdsmiljøgruppen [23]). When people are sensing draught it often results in a demand for higher air temperatures in the room, or for stopping ventilation systems.

The following chapter describes the latest research on draught and how it should be measured.

Local thermal discomfort due to draught

Several studies have investigated the convective heat loss for the entire body as a function of the air velocity (Rohles et al. [8], Olesen et al. [25], Burton et al. [27], Østergaard et al [26]). This is important for setting up a heat balance for the body and for predicting the thermal sensation for the body as a whole (Fanger [4]). But such information is of limited value for predicting a local cooling felt as a draught. A person may feel thermally neutral for the body as a whole, yet he may not be comfortable if air movements cause an unwanted cooling of a particular part of the body.

There are few specific draught studies available. Houghton [28] studied ten male subjects exposed to constant local velocities at the back of the neck and at the ankles. McIntyre [29] used a similar method where he exposed the head region of the subjects to constant local velocities. But in ventilated spaces the air velocity fluctuates and is never constant. Fig.7 shows the examples of air velocity measurements. In case of floor-heating systems the air velocity at the floor level is caused by the down draught from a window and air infiltration. The fluctuations are seen to be very small. Another measurement is from a room heated by a radiator below the window. The air velocity is measured at the head level assuming a person seated close to the window. Here the fluctuations are much greater. This is caused by the very turbulent mixing of down draught from the window and upwards warm air from the heater. Fanger and Pedersen [30] exposed subjects to a periodically fluctuating air flow directed towards the back of the neck or the ankles. It was shown that a fluctuating air flow is more uncomfortable than a constant flow. Fig.8 shows the sensation of comfort as a function of the frequency of the fluctuating air velocity.

In real spaces the occupants are not exposed to a well-defined, periodically fluctuating air flow. Fluctuations in practice will be of a stochastical nature. In a field study in Copenhagen by Thorshauge [31] the velocity fluctuation was determined in twelve typically ventilated spaces. The

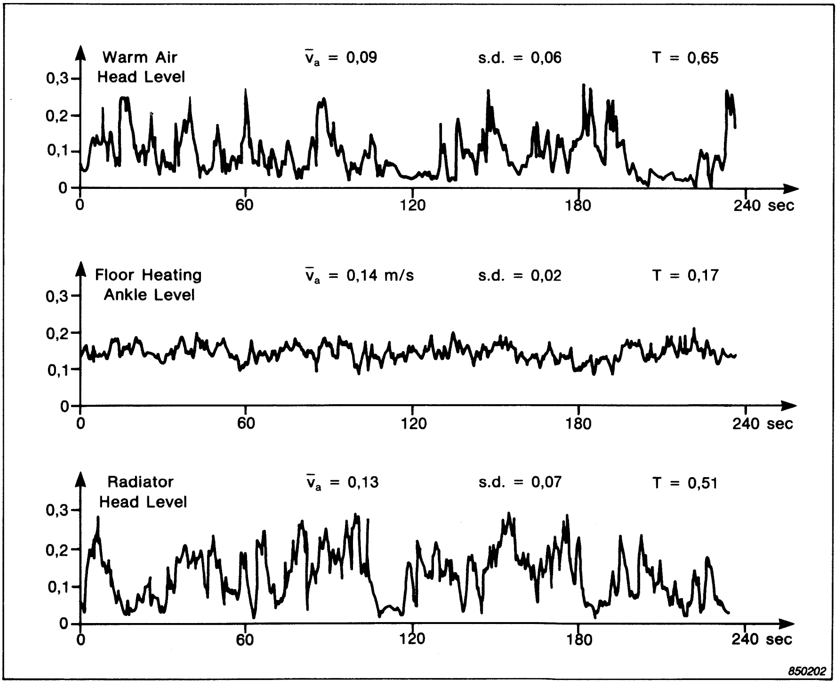

Fig. 7. Air velocities measured in a room where different heating systems were installed

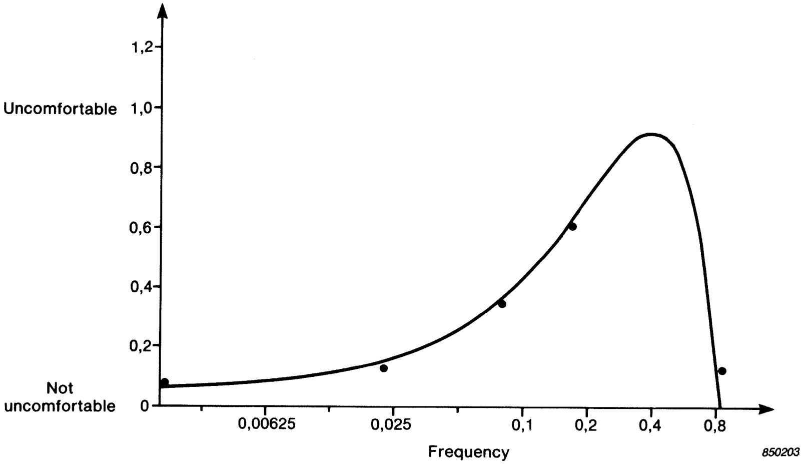

Fig. 8. Mean values of the degree of discomfort expressed by 16 subjects being exposed to a fluctuating air flow, as a function of the frequency. Mean velocity: 0,3 m/s

fluctuations in the occupied zone could be described by the standard deviation of the velocity , Fig.9. From this study it was also found that at least 3 min. of measurements were necessary to estimate a representative mean air velocity and standard deviation.

At the Technical University of Denmark there has recently been performed a study by Christensen et al. [32, 55] with the purpose of establishing the scientific basis required to predict the human response to fluctuating air velocities as they occur in practice. An important aim was to establish the percentage of the population feeling draught when exposed to a given mean velocity.

It was decided to expose 100 subjects to velocity fluctuations. The subjects were exposed to increasing mean velocities up to 0,40 m/s. Each subject was studied during three experiments at an air temperature of 20, 23 and 26°C. The aim was to keep the subject at all three temperatures thermally neutral for the body as a whole by modifying the clothing. Such an adaptation of the clothing will probably also take place in most cases in practice.

It was essential that the subject in the "experimental room" was exposed to velocities fluctuating in the same manner as in typically ventilated spaces in practice, as identified by Thorshauge [31]. Fig.9 shows that the

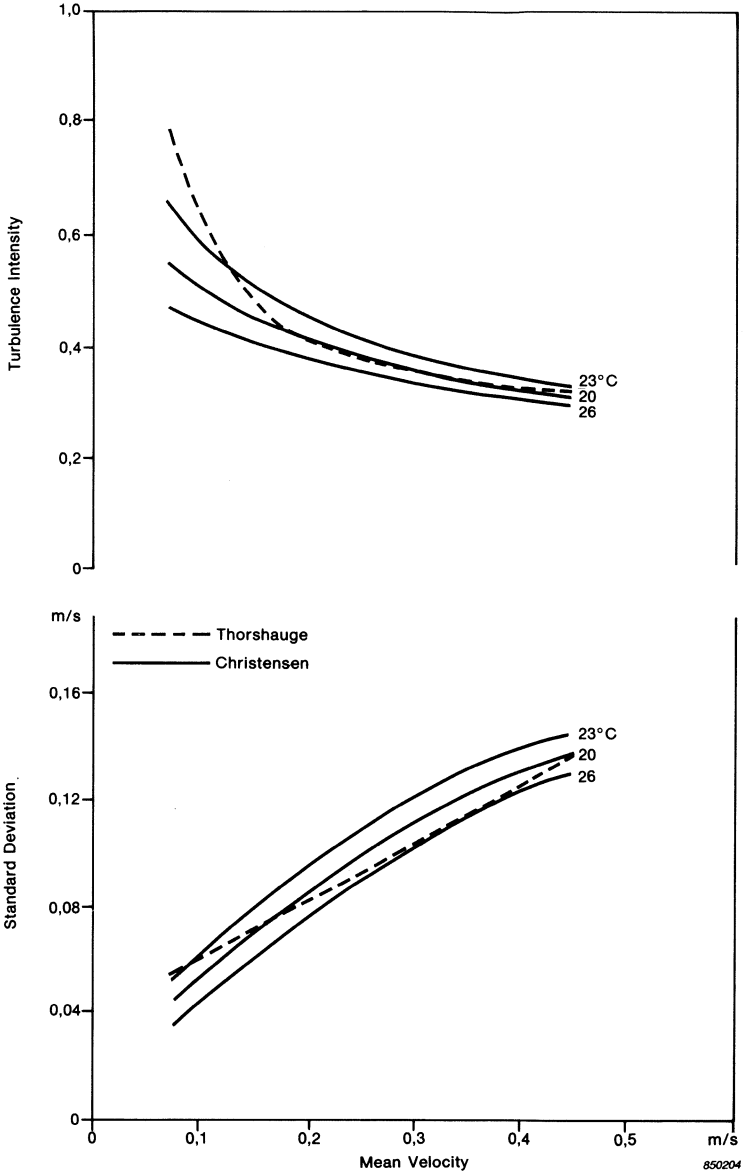

Fig. 9. The relationship between mean air velocity and turbulence intensity for field measured data and for the velocities used in the studies with experimental subjects

velocity fluctuations in the experimental room were very similar to the measurements performed in the field. Since an earlier study [30] had indicated that man is most sensitive to velocities coming from behind, this main flow direction was aimed at where the subject was seated.

Fig.10 shows the percentage of subjects who felt draught on the head region (the dissatisfied) as a function of the mean velocity at the neck. The head region comprises head, neck, shoulders and back. There was a significant influence of the air temperature on the percentage of dissatisfied. There was no significant difference between the draught responses of men and women.

The study was performed with subjects clothed to maintain thermal neutrality at all three air temperatures. The subjects wore normal summer and winter clothing. This means that only the head region, the hands, and in some cases the ankles and lower arms were uncovered. Among these uncovered areas the head region was found to be the most sensitive. The data in Fig.10 applies therefore only to persons wearing normal indoor clothing and performing light, mainly sedentary work. Persons with higher activity levels are not so sensitive to draught. There exists, however, no experimental data on persons with higher activities.

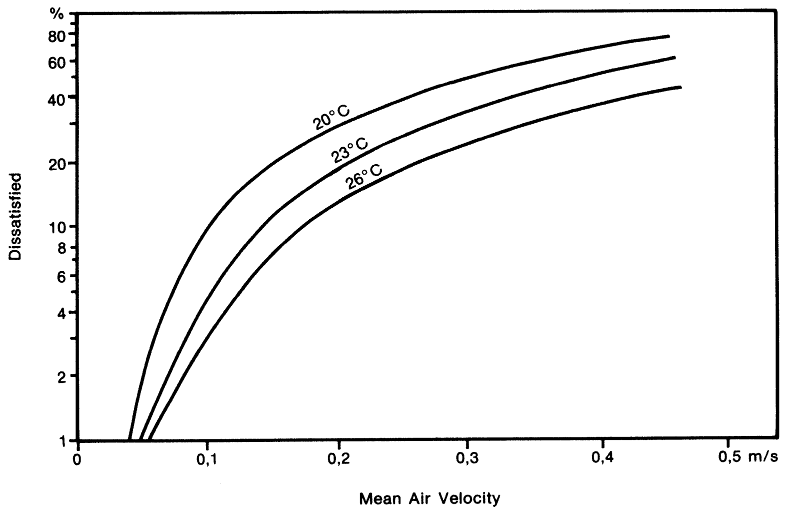

Fig. 10. Percentage of dissatisfied as a function of the mean air velocity

In existing standards there are recommended limit values for the mean air velocity. These limits are illustrated in Fig.11. ISO 7730, ASHRAE 55–81 and NKB-guidelines have agreed on the same limits.

These are however split up in two requirements:

Light, mainly sedentary activity during winter (heating period), i.e. operative temperature between 20 and 24°C

Mean air velocity, v_a less than 0,15 m/s

Light, mainly sedentary activity during summer (cooling period), i.e. operative temperatures between 23 and 26°C

Mean air velocity, v_a, less than 0,25 m/s

At higher air temperatures there may be a benefit from the cooling effect of an increased air velocity. Therefore ASHRAE 55–81 has introduced an extended summer zone, which allows air velocities up to 0,8 m/s (see Fig.11).

The limit values according to the German standard DIN 1946 [33] are also shown in Fig.11. These limits are in the same range as the two limits from ISO 7730. It is, however, a little inconsistent to have two limits in the temperature range 23 to 24°C as recommended by ISO, ASHRAE and NKB.

Also, the results from the previously described experiments are shown in Fig.11 as lines with different levels of percent dissatisfied. The limits in DIN 1946 seem to correspond very well with the line for 20% dissatisfied. The other limits will result in values between 10% and 25% dissatisfied. It is important to stress that the air velocity used is based on a 3 min. mean value, so there may be maximum air velocities much higher than the limits. Even at very low air velocities (0,1 to 0,2 m/s) there will be a surprisingly high number of dissatisfied persons, which may be explained by the effect of fluctuations.

The reason why turbulent air flow is more uncomfortable than laminar air flow could be that the velocity fluctuations cause the skin temperature to fluctuate. According to Hensel [34] the rate of change of the skin temperature initiates signals to the brain. These undesired warning signals may explain the discomfort caused by fluctuating air velocities.

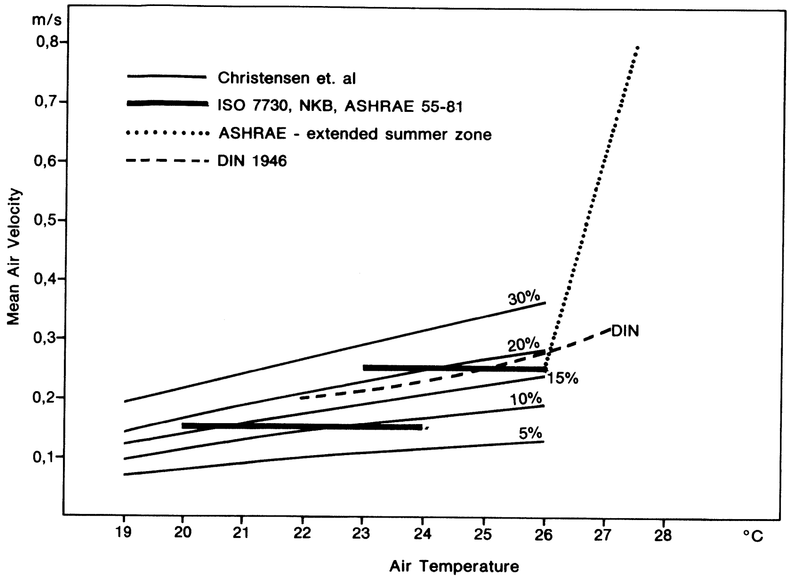

Fig. 11. Comparison of guidelines for air velocities recommended in existing standards (ASHRAE, NKB, ISO, DIN) and the results of the studies by Christensen et al.

This theory has been studied by Madsen [35, 36, 37]. He simulated the human skin including temperature receptors on an electrical analog computer [35]. In the El-model, temperature is simulated by voltage, heat flow by current, thermal capacity corresponds to electrical capacitance and thermal resistance corresponds to electrical resistance. On the El-model the maximum heat flow through the receptors is determined, when the skin is exposed to a number of sine-shaped velocity changes, with constant amplitude, but with different frequencies. In Fig.12 the results from Fanger and Pedersen [30] are seen together with the results from the El-model. The forms of the curves are almost identical, both having a maximum around 0,5 Hz. This result confirms the theory that a local cooling of the body is uncomfortable, when the heat flow through the skin exceeds a certain limit, i.e. the thermoreceptors send so many impulses to the brain that they cause discomfort. The same theory and the El-model were also applied to the fluctuating air velocities used in the experiments described above (Christensen et al. [32]). By use of the El-model the uniform air velocity (constant voltage) which would give the same output from the thermal receptors (heat flow-current) as the actual fluctuating air velocity was estimated. During the tests with subjects

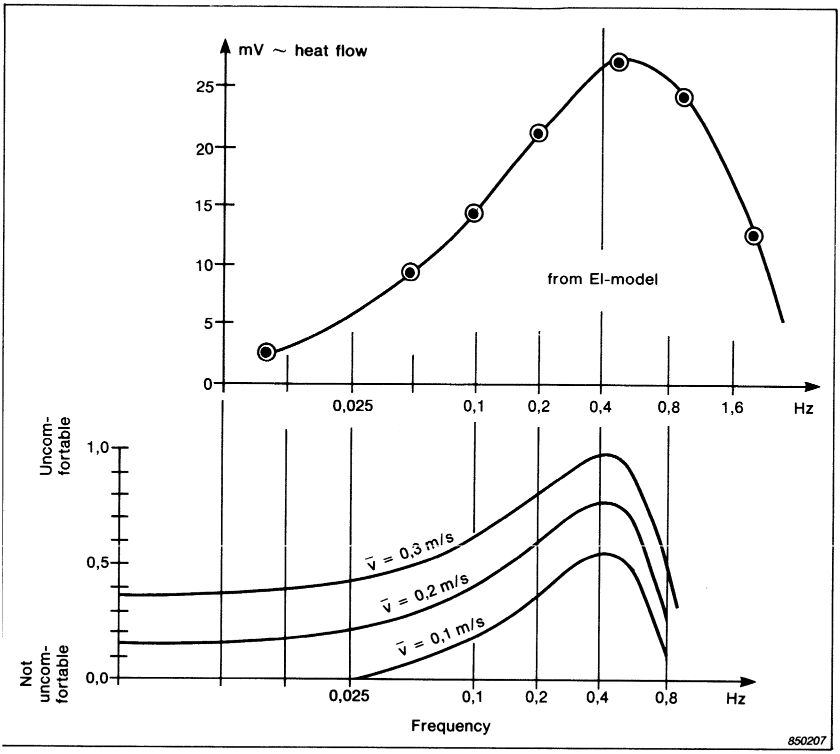

Fig. 12. Comparison between maximum heat flow through thermal receptor and the sensation of draught at different frequencies of fluctuating air movements from Fanger and Pedersen [30]

these velocities had been recorded by a tape recorder as a voltage signal. In Table 2 is shown for five air velocities, the mean air velocity of the fluctuating air velocity used in the experiments with subjects, and the uniform air velocity, which will cause the same output from the thermal receptors as the fluctuating air velocity. This is also illustrated in Fig.13, which shows the correlation between mean air velocity and percent dissatisfied at three different temperature levels. In the same figure is indicated the uniform air velocity which - if the hypothesis is valid - may cause the same percentage of dissatisfied. The results from tests with subjects indicate (Fig.13) that a mean air velocity of 0,15 m/s at 23°C will cause about 10% feeling dissatisfied. The same level of dissatisfied is obtained, when people are exposed instead to a uniform air velocity of 0,30 m/s (Fig.13).

Actual mean air velocity m/s	Equivalent uniform air velocity m/s
0,10	0,24
0,15	0,30
0,22	0,36
0,29	0,45
0,40	0,60

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Table 2. Actual and equivalent air velocity

This theory is very interesting and may be a method for evaluation of the discomfort caused by a fluctuating air velocity. In existing standards the limits are only based on a mean air velocity. Some years ago, however, it was suggested in a proposal for a new German standard DIN 1946, to base the draught limit on the mean air velocity plus standard deviation. In this way the influence of fluctuations, described by the standard deviation, was in a way taken into account. This idea was never accepted. There is a need for further studies of draught and especially how the influence of fluctuations in practice shall be quantified. Until then it has been suggested in ISO 7726 also to measure the standard deviation and use this as a measure of the fluctuations.

Measurement of air velocity

Measurement of air velocity is very difficult and very few instruments which can measure low air velocity correctly are available. As seen from the previous chapter there is a need for measuring air velocity as low as 0,05 m/s, to measure fluctuations as fast as 1 Hz, to give a mean air velocity based on 3 min measurement and to give the equivalent standard deviation. In ISO 7726 these requirements are specified and shown here in Table 3. A further requirement is that the sensor should be omnidirectional, i.e. the measured air velocity shall be measured correctly independent of the direction relative to the sensor (except, of course, for a small angle around the support of the sensor). For example, when using a hot wire, it is very important always to keep the wire perpendicular to the air flow. Especially for low air velocities this may be difficult because the direction may change very often. This also applies to rotating vane anemometers and to some extent also to a heated thermistor.

All the requirements for moderate thermal environments are fulfilled by the air velocity sensor used together with the Indoor Climate Analyzer, Type 1213.

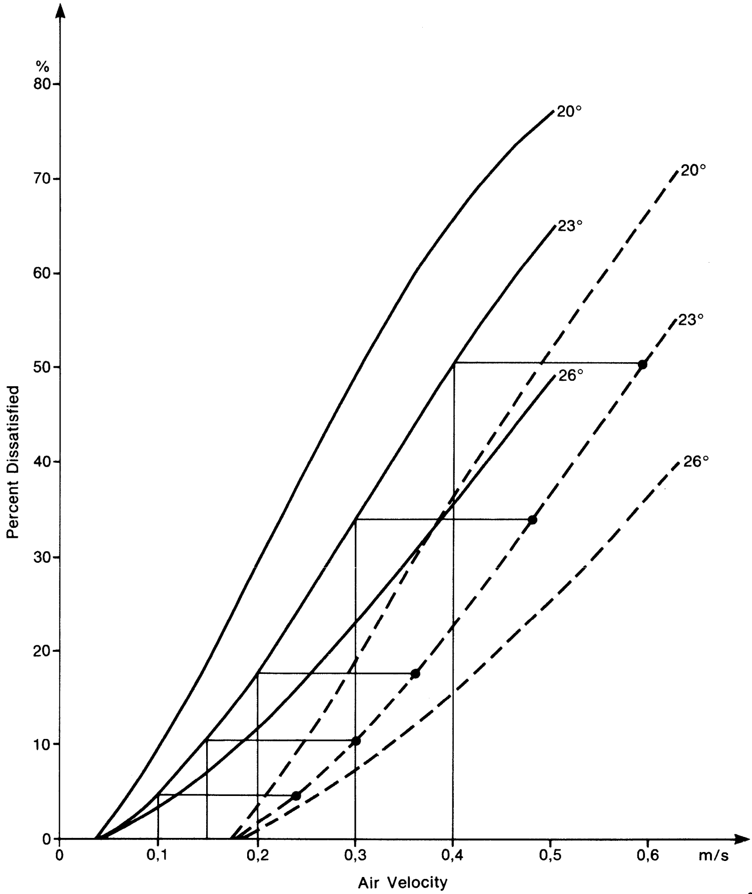

Fig. 13. Correlation between mean air velocity and percent dissatisfied at three temperature levels. The solid line curves from Christensen et al. [32]. The dotted lines are the equivalent uniform air velocity which will cause the same number of dissatisfied. The two lines for 23°C are also listed in table 2.

As the head region and ankle region are the most sensitive places for draught, the air velocity should be measured at these levels, i.e. 0,1 m and 1,1 m above floor level for seated persons and 0,1 m and 1,7 m above floor level for standing persons (ISO 7726 [10], ASHRAE 55–81 [2]). At the same time the air temperature should be measured (see later). For evaluation of the thermal comfort as a whole (PMV–PPD index) it is,

Thermal environment	Measuring range m/s	Precision Specified m/s	Precision Desirable m/s	Comments
Cold and moderate	1 – 10	$\pm \lvert 0{,}1 + 0{,}05 \cdot v_a \rvert$	$\pm \lvert 0{,}05 + 0{,}05\, v_a \rvert$	Precision shall be fullfilled with a solid angle 3π *sr*. Mean value for 3 min. is desirable
	0,05 – 1	$\pm \lvert 0{,}05 + 0{,}05\, v_a \rvert$	$\pm \lvert 0{,}02 + 0{,}07\, v_a \rvert$	
		Response time (90%) 1 *s*	0,5 *s*	Fluctuations are described by the standard deviation
Hot	0,2 – 10	$\pm \lvert 0{,}1 + 0{,}05 \cdot v_a \rvert$	$\pm \lvert 0{,}05 + 0{,}05\, v_a \rvert$	Precision shall be fullfilled with a solid angle 3π *sr*. Mean value for 3 min. is desirable

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Table 3. Measuring range and accuracy for air velocity sensors

however, necessary to measure the air velocity at the abdomen level, i.e. 0,6 m above floor level for seated and 1,1 m above floor level for standing subjects.

4. Vertical air temperature differences

In most spaces in buildings the air temperature is not constant from the floor to the ceiling; it normally increases with the height above the floor. If the gradient is sufficiently large, local warm discomfort can occur at the head, and/or cold discomfort can occur at the feet, although the body as a whole is thermally neutral. In a room a lot of warm air may then be trapped above the occupied zone near the ceiling. This is of no benefit for the people occupying the room and may during the heating season lead to a higher energy consumption. This is indicated in Fig.14. An example of how this vertical temperature difference may be influenced by the heating system is shown in Fig.15. This is from an experiment by Olesen et al. [38, 39, 40], where energy consumption and thermal comfort were studied in a room heated by different heating methods.

Local thermal discomfort due to vertical air temperature differences

Very few studies of vertical air temperature differences and the influence of thermal comfort are reported in the literature (Olesen et al. [41], McNair [42, 43], Eriksson [44]). The results presented here are from the studies by Olesen et al. [41]. In a climatic chamber they exposed seated subjects individually to different air temperature differences between

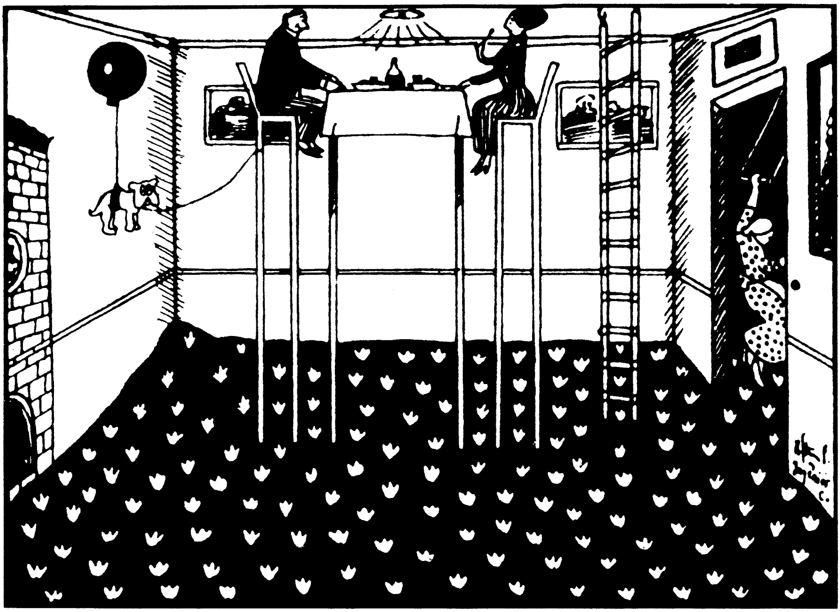

Fig. 14. *The coal shortage 1917*

By occupying the space near the ceiling the heat will be fully utilized - the redundant floor can then be used for growing potatoes. R. Storm.P.

head and ankle. During the tests the subjects were in thermal neutrality, because they were allowed to change the temperature level in the test room whenever they desired; but the vertical temperature difference was kept unchanged. The subjects gave subjective reactions on their thermal sensation and Fig.16 shows the percentage of dissatisfied as a function of the vertical air temperature difference between head (1,1 m above floor) and ankles (0,1 m above floor).

In ISO 7730, ASHRAE 55–81 and NKB the following limit values are recommended:

For light, mainly sedentary activity the vertical air temperature difference between 1,1 m and 0,1 m above floor (head and ankle level) shall be less than 3°C.

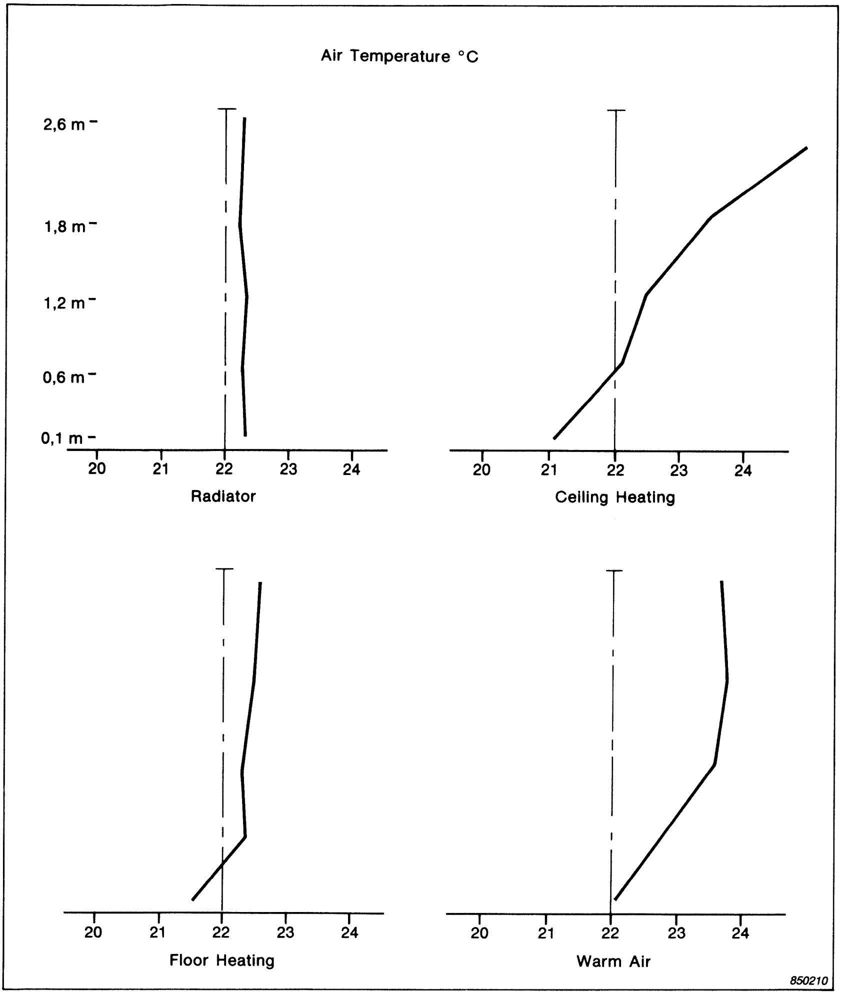

Fig. 15. Vertical air temperatures from tests with different heating systems

This limit of 3°C is mainly for people with sedentary work. According to Fig.16 this will result in less than 5% dissatisfied. Again this value must not be confused with the PPD-index for the thermal comfort for the body as a whole. People with higher activity levels will be less sensitive and can tolerate larger differences. There are, however, no experimental data available for higher activities.

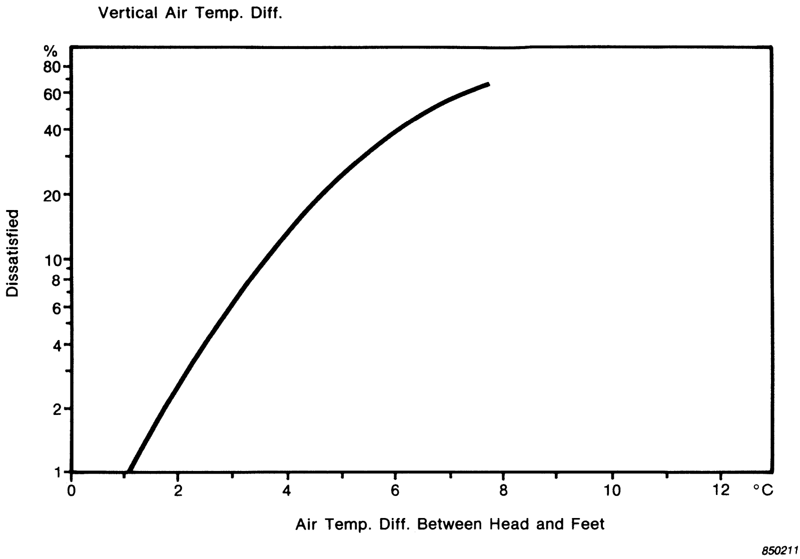

Fig. 16. Percentage of dissatisfied as a function of the vertical air temperature difference between head and ankles

The results from the tests with subjects and the recommended limit is only for an increasing temperature from feet to head. The opposite case where the air temperature at head level is lower than at the ankle level will not be so critical for the occupants. Erikson [44] indicated that his subjects could tolerate much greater differences if the head was cooler. This is also verified in the experiments with asymmetric thermal radiation from a cooled ceiling (Fanger et al. [22]).

Measurement of vertical air temperature differences

Air temperature should be measured at head and ankle level, i.e. 1,1 m and 0,1 m above the floor for seated people and 1,7 m and 0,1 m above the floor for standing people. When measuring air temperature it is very important to minimize the influence of the thermal radiation from the surrounding hot or cold surfaces. The relative influence of the radiation can be decreased by using a small sensor surrounded by a reflective, radiant shield, which allows air to pass across the sensor. In environments with a very strong radiation (steel plants, glass factory) it may also be necessary to increase the air velocity around the sensor and thus increase the heat exchange due to convection.

Thermal environment	Measuring range °C	Precision Specified °C	Desirable °C	Comments
Moderate	10 – 30	± 0,5	± 0,2	Precision shall be valid for $\mid t_a - \bar{t}_r \mid \leqslant 10°C$ Response time shortest possible
Cold and Hot	–40 – 0 0 – 50 50 – 120	$\pm (0{,}5 + 0{,}01 \mid t_a \mid)$ ± 0,5 $\pm (0{,}5 + 0{,}04\ (t_a - 50))$	$\frac{\mathit{Specified}}{2}$	Precision shall be valid for $\mid t_a - \bar{t}_r \mid \leqslant 20°C$ Response time shortest possible

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Table 4. Measuring range and accuracy for air temperature measurement

The measuring range and accuracy for air temperature measurements are specified in ISO 7726 and are shown in Table 4. Here also a limit for the efficiency of a radiant shield is specified . The air temperature sensor used by the Indoor Climate Analyzer, Type 1213 fulfils the requirements to moderate thermal environments. Two air temperature sensors can be connected so that the air temperature can be measured at two different levels.

5. Warm or cold floors

Due to the direct contact between the feet and the floor, local discomfort of the feet can often be caused by a too high or too low floor temperature.

Besides, the floor temperature has a significant influence on the mean radiant temperature in a room. The floor temperature is very much influenced by the way the building is constructed, i.e. insulation of the floor, above a cellar, direct on the ground, above another room, use of floor heating etc. Therefore it is very important to know which floor temperatures are acceptable. If a floor is too cold and the occupants feel cold discomfort in their feet, a common reaction is to increase the temperature level in the room and then in the heating season this will increase the energy consumption.

Local thermal discomfort due to warm or cold floors
Numerous studies on the influence of floor temperature on feet-comfort are reported in the literature (Nevins et al. [45, 46, 47], Olesen [48, 49]. Frank [50], Cammerer [51], Schüle [52, 53], Missenard [54]).

The most extensive studies are performed by Olesen [48, 49] who, based on his own experiments and re-analysis of data from Nevins et al. [45, 46, 47] came to the following results.

For floors occupied by **people with bare feet** (in swimming halls, gymnasiums, dressing rooms, bathrooms, bedrooms, etc.) the flooring material is important. Based on heat transfer theory, Olesen [48] found the optimal temperatures and recommended the temperature intervals given in Table 5 for a number of typical flooring materials. For 10 minutes occupancy about 10% of the persons can be expected to experience discomfort at the optimal floor temperature while less than 15% can be expected to be uncomfortable within the recommended temperature interval. To save energy, flooring materials with a low contact coefficient (cork, wood, carpets) or heated floors should be chosen, to eliminate a desire for higher ambient temperatures caused by cold feet. Also in kindergartens where the children often are playing directly on the floor, these recommendations could be followed.

Flooring Material	Optimal Floor Temperature for 1 min. Occupancy °C	Optimal Floor Temperature for 10 min. Occupancy °C	Recommended Floor Temp. Interval
Textiles (mats)	21	24,5	21 – 28
Cork	24	26	23 – 28
Pinewood Floor	25	26	22,5 – 28
Oakwood Floor	26	26	24,5 – 28
PVC-Sheet with Felt Underlay on Concrete	28	27	25,5 – 28
Hard Linoleum on Wood	28	26	24 – 28
5 mm Tesselated Floor on Gas Concrete	29	27	26 – 28,5
Concrete Floor	28,5	27	26 – 28,5
Marble	30	29	28 – 29,5

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Table 5. Comfortable temperature of floors occupied by people with bare feet

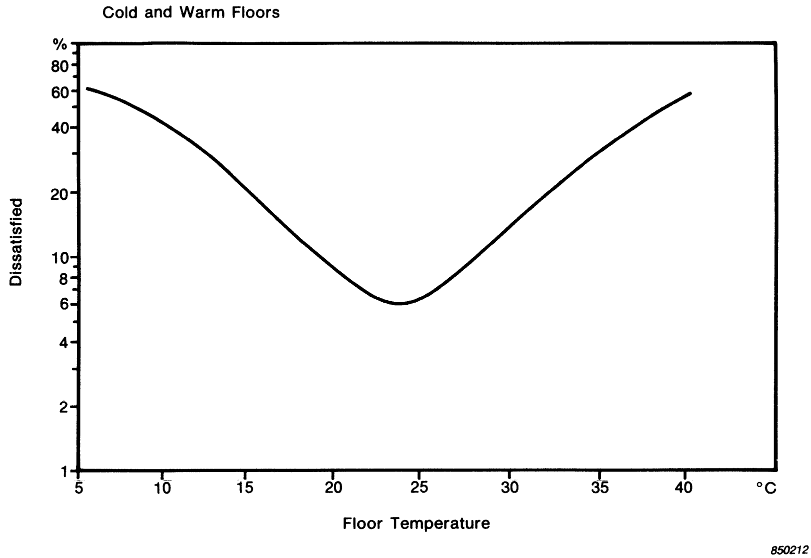

Fig. 17. Percentage of dissatisfied as a function of floor temperature

For floors occupied by **people with footwear** (normal indoor footwear) the flooring material is without significance. Olesen [49] found, based on his own experiments and a re-analysis of the results of Nevins [45, 46, 47], an optimal temperature of 25°C for sedentary and 23°C for standing or walking persons. At the optimal temperature 6% of the occupants felt warm or cold discomfort in the feet. If the data from experiments with seated and standing subjects are pooled together a relationship between floor temperature and percentage of dissatisfied as shown in Fig.17 was found.

In all experiments the subjects were in thermal neutrality so the percentage of dissatisfied is only related to the discomfort due to cold or warm feet.

No significant difference was found in the floor temperature preferred by females and males. The optimal temperature is 24°C and accepting up to 10% dissatisfied will result in a range from 19,5°C to 28°C.

In ISO 7730 and NKB the requirements to the floor temperature are identical:

For light, mainly sedentary activity the surface temperature of the floor shall normally be between 19°C and 26°C, but floor heating systems may be designed for 29°C.

In ASHRAE 55–81 the requirements are: The surface temperature of the floor for people wearing appropriate indoor footwear shall be between 18°C and 29°C.

According to Fig.17 these limits will result in less than approx. 10% dissatisfied. The reason for a higher acceptable floor temperature, when designing floor heating systems, is the fact that this temperature only will occur very few hours a day during the winter at the coldest outside temperatures.

The recommended limits will not at all prohibit the use of floor heating systems. At a floor temperature of 29°C and a room temperature of 20°C the heat output from the floor to the room will be approx. 100 W/m^2, which with current levels of insulation and energy comsumption is sufficient to heat a house.

Measurement of the floor surface temperature

Often the surface temperature of a floor is not uniform, i.e. above the supply end of a floor heating it is warmer than at the return, and just above the heating coils the temperature may also be significantly higher than between two heating coils. In ISO 7730 and ASHRAE 55–21 it is not defined what is meant by the floor surface temperature. In NKB the floor temperature is defined as a mean value for a square with 1 m sides. The intention of the requirements in ISO 7730 and ASHRAE 55–81 is not a point measurement. It would be more realistic to estimate the mean value of an area equivalent to an area of two feet, i.e. an area about 0,25 m × 0,30 m. This will also cover the distance between heating coils, which normally is between 0,15 and 0,3 m. The mean surface temperature is then estimated from measurements of the surface temperature at a couple of representative points. When there is no floor heating installed the floor surface temperature will normally be more uniform.

A sensor for measuring the surface temperature by contact is also provided with the Indoor Climate Analyzer, Type 1213.

6. Definition of occupied zone

All the requirements to the indoor thermal environment are related to the occupied zone, i.e. the positions in a space occupied by people. This

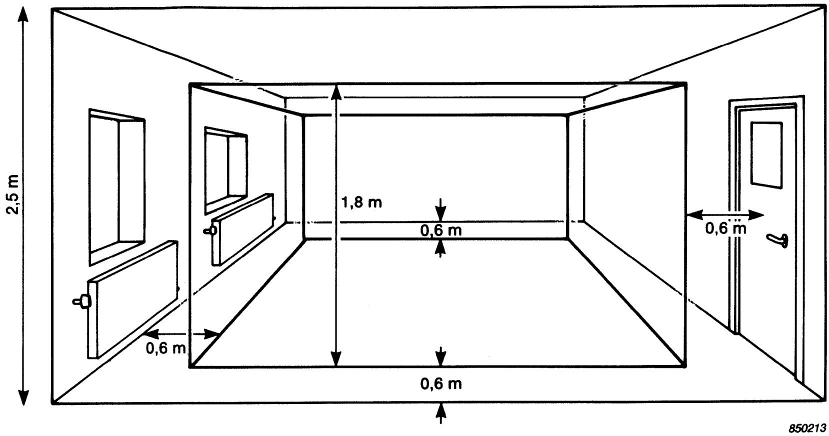

Fig. 18. Occupied zone

may be difficult to define exactly in an industrial situation. Here the general philosophy is that the requirements both regarding the general thermal comfort (PMV-PPD index) and local thermal discomfort should be fulfilled at the place of work, i.e. the measurements shall be performed more or less by substituting the person with the measuring sensors.

For non-industrial premises like residential buildings, offices, schools etc., an occupied zone has been defined by ASHRAE 55–81 and NKB. This zone is shown in Fig.18 and is defined as:

> A region within a space, normally occupied by people, generally considered to be between the floor and 1,8 m above the floor and more than 0,6 m from walls or fixed heating or air-conditioning equipment.

7. General thermal comfort - local thermal comfort

The relationship between the influence of the thermal environment on the body as a whole and how this influences local warm and cold sensations have not really been studied. Also the combined influence of more than one factor causing local thermal discomfort need to be investigated further. Even if it is possible to predict the percentage of dissatisfied due to general thermal comfort (PPD-index) and for each of the local parameters (radiant temperature asymmetry, draught, vertical air temperature differences, warm or cold floor) it is not possible to estimate the total number of dissatisfied. The aim of the requirements in ISO 7730, ASHRAE 55-81 and NKB has been to define an environment which is acceptable for at least 80% of the occupants. That means that even if the requirements are met there may still be persons who are not satisfied with the thermal environment. In every group of people there always exist some persons who are very sensitive regarding the thermal environment.

All the requirements described in the present paper are essentially based on people who are in thermal neutrality, i.e. PMV ~ 0.

And there is no doubt that a deviation from the state of thermal neutrality will in most cases increase the number of people feeling local thermal discomfort. Therefore it is always very important, when evaluating the risk for local thermal discomfort, to measure and evaluate the general thermal comfort (PMV-index).

One example is an occupant sitting close to a large window façade in winter time. He may feel a cold sensation on the side towards the window and feel local thermal discomfort due to the radiant asymmetry. But that may not be the real problem. It will often be so that the mean radiant temperature near a window façade is lower than in the rest of the room. That means the operative temperature and thus the PMV-value is lower than in other positions in the room. If now the heating or air conditioning system is controlled by a thermostat normally sensing air temperature, which may be positioned somewhere else in the room, the person near the window may feel generally cool (PMV < 0). Then if he is generally cool he is much more sensitive for a asymmetric cold radiation from the window. If instead the temperature level in the room was controlled with a sensor taking into account the cold radiation from the window based on the occupant near the window, then this occupant would be close to thermal neutrality and probably not feel any severe local thermal discomfort due to the radiant asymmetry.

The same is true for other factors. A person who feels generally warm will be more sensitive to warm radiation from a heated ceiling or easier get warm feet from a heated floor. A person who is generally cool will be more sensitive to draught than a person in thermal neutrality or a person feeling generally warm.

Local thermal discomfort may also have an influence on the energy consumption. If local cold discomfort in a space occurs during winter, e.g. caused by air velocity, a cold floor, or by asymmetric radiation from a large window, it is likely that the occupant(s) will require a higher ambient temperature level. This will decrease the local discomfort but will at the same time cause a slightly warm thermal sensation for the body as a whole. The higher the temperature level the higher the energy consumption ($\sim$ 5–15% per °C). From the points of view of both comfort and energy it is thus essential that the building as well as the HVAC system be designed so that the occurrence of local cold discomfort on the body is unlikely. For summer conditions the same applies for local warm discomfort.

On the other hand, local heating (cooling) of the body may provide a means of energy conservation during winter (summer). This may be considered primarily under conditions where deviations from optimal comfort would be acceptable.

It is clear that the sensation of local thermal discomfort is influenced by the general thermal comfort for the body as a whole. It is, however, not possible to quantify this influence and further experimental studies with human beings are needed.

The present paper together with the paper on thermal comfort in an earlier Technical Review presents the background and requirements for evaluating a moderate thermal environment. By the use of the Thermal Comfort Meter, Type 1212 and the Indoor Climate Analyzer, Type 1213, it is now also possible in an easy way to perform all the necessary measurements.

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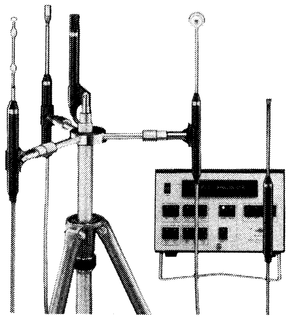
News from the Factory

Indoor Climate Analyzer Type 1213



Brüel & Kjær Indoor Climate Analyzer Type 1213 is a handy, easy-to-operate, portable instrument for evaluating all of the basic parameters which influence the thermal environment and its effect on man. Measurements with Type 1213 are performed in accordance with ISO 7726 criteria and the Analyzer features a clear, easy-to-read, 20-character alphanumeric Display which also provides the user with clearly understood interactive prompts.

Using five transducers Type 1213 can measure air temperature, surface temperature, radiant temperature asymmetry, humidity and air velocity. It can be used either to obtain real-time measurements or data can be recorded and stored in its integral memory for subsequent output to a Level or X–Y recorder. Up to sixty measurements of each preselected parameter can be recorded and 4 different recording periods (1, 6, 24 and 120 hours) are available. Measurements are automatically spaced evenly throughout the recording period so that the Analyzer can be left completely unattended.

In addition to measurement of the basic parameters the Type 1213 provides measurement of:

- — 3 minute mean and standard deviation of air velocity.
- — Radiant temperature either as:
  1. Plane radiant temperature simultaneously in two directions
  2. Radiant temperature asymmetry
  3. Incident power.
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  1. Dew point
  2. Vapour pressure
  3. Relative humidity.

Type 1213 can be ordered as a separate unit together with basic accessories but excluding Transducers. This enables the complete system to be built-up to meet the user's requirements over a period of time by selecting the desired combination of transducers from:

- — Air Temperature Transducer MM 0034
- — Surface Temperature Transducer MM 0035
- — Radiant Temperature Asymmetry Transducer MM 0036
- — Air Humidity Transducer MM 0037
- — Air Velocity Transducer MM 0038

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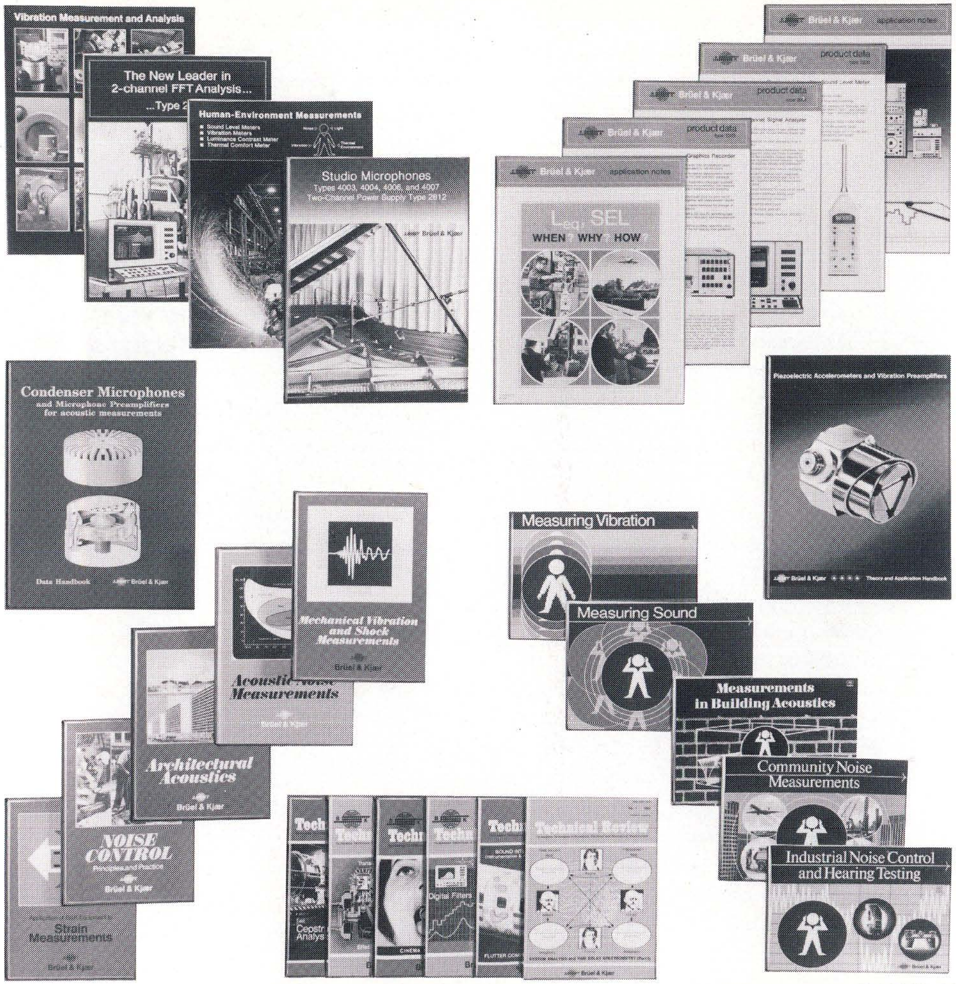

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