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EVALUATING THERMAL ENVIRONMENTS BY USING A THERMAL MANIKIN WITH CONTROLLED SKIN SURFACE TEMPERATURE

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ABSTRACT

This paper describes a method for measuring non-uniform thermal environments using a new thermal manikin with controlled skin surface temperature. The manikin and its control logic are described, and an equivalent temperature based on the thermal manikin (teq) *is proposed and discussed. A method to calculate the PMV index from manikin heat loss is also given.*

To calibrate these methods, fundamental data were collected. The manikin-based equivalent temperature (teq) *is shown to be effective at accounting for the effects of local heat sources and asymmetrical environmental conditions.*

INTRODUCTION

Purpose

Indoor spaces often exhibit vertical temperature differences, radiant asymmetry, local airflows, and local body cooling. There are few places as uniform as the climate chambers used in laboratory studies of comfort. Evaluations of thermal environment are typically conducted with the measurements of several physical parameters such as air temperatures, air velocities, and so on. These physical measurements provide useful information, but sometimes it is difficult to integrate them in order to estimate human sensation under non-uniform conditions. Thermal sensation is closely related to the heat exchange between humans and their environment. Thermal comfort indices such as predicted mean vote (PMV) (Fanger 1970) and SET* (Gagge et al. 1973) are based on the heat balance of the whole body. These indices are inaccurate or inapplicable for the evaluation of non-uniform conditions. A realistically shaped thermal manikin is one of the useful tools with which to directly measure heat exchange between humans and their environment. The purpose of this study is to develop a method for measuring non-uniform thermal environments with a new skin-surface-controlled manikin. To obtain fundamental data, the thermal manikin was first exposed under uniform thermal environments. Then measurements were made in a thermally non-uniform workstation with local air supply from an underfloor air distribution system and with a typical arrangement of local heat sources such as a personal computer and task light.

Thermal Manikin Studies

Thermal manikins, originally developed to measure the thermal insulation of clothing, are heated dummies that simulate the heat transfer between humans and their thermal environment. Winslow and Herrington (1949) developed a standing manikin for clothing studies that became the model for many manikins, including Toda's (1958) standing copper manikin. Kerslake and Clifford (1965) developed a seated thermal manikin, and Madsen (1976a, 1976b) developed a male thermal manikin that can sit, stand, and even move. It has been used to provide a comprehensive data base for clothing insulation. McCullough et al. (1985) also published a comprehensive data base of clothing insulation obtained from a series of manikin measurements over the years.

Mihira et al. (1977) developed a thermal manikin not only for clothing insulation measurements but for the evaluation of thermal environments. However, the relationship between measured heat loss and thermal sensation was unknown. Olesen et al. (1979) and Fanger et al. (1980, 1986) tried to use a thermal manikin for evaluations of the indoor environment. Tanabe et al. (1989) proposed evaluating thermal environments with an aluminum thermal manikin.

Recently, Wyon et al. (1989) proposed and demonstrated the usefulness of the manikin-derived EHT (equivalent

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homogeneous temperature) in evaluating strongly nonhomogeneous thermal environments in automobiles. Bánhidi et al. (1991) used Wyon's manikin for the evaluation of heated spaces. Wyon's manikin is controlled to keep skin temperature constant. Wyon and Sandberg (1990) also predicted discomfort due to displacement ventilation using a thermal manikin.

NEW THERMAL MANIKIN

Construction of Manikin

Basically there are three types of heating systems for thermal manikins. In these systems, the heating element is placed at (1) the outer surface of the manikin, (2) the inside surface of the manikin, or (3) the inside space of the manikin. When using method 2 or 3, high-conductivity materials such as copper and aluminum are often used for the shell to keep the surface temperature uniform. Each method has advantages and disadvantages. The manikin described in this study uses method 1 because it produces a relatively small time constant (less than five minutes).

The manikin consists of a 4-mm fiberglass-armed polyester shell wound with nickel wire of 0.3-mm diameter at a maximum spacing of 2 mm. The wiring is covered by a 0.1-mm to 1.0-mm protective shield. The heating element is placed close to the surface to give the manikin a very small time constant (less than five minutes) compared to other thermal manikins. The time constant is further reduced by the fact that the same nickel wire is used sequentially both for heating the manikin and for measuring and controlling the skin temperature. For the nude manikin with a heat loss of 100 W/m^2, the difference in surface temperature between the hottest point directly above a wire and the coldest point midway between two wires was measured to be less than 0.5°C using infrared thermovision equipment.

Figure 1 is a picture of the thermal manikin (named Anne). The 16 body parts and their respective surface areas are listed in Table 1. Each part is separately controlled and measured by a laptop computer outside the manikin. Data are output from the control computer for storage and spreadsheet analysis.

Heat Transfer between the Manikin and the Environment

Figure 2 shows the heat transfer between skin and environment through clothing. The following relationships were obtained:

$$It = (ts,cl - to)/0.155\ Qt \quad (1)$$

where 1 clo = 0.155 m^2·°C/W,

$$Ia = (ts,n - to)/0.155\ Qa, \quad (2)$$

$$Ia' = Ia/fcl, \quad (3)$$

Figure 1 Picture of thermal manikin (Anne).

TABLE 1
Name and Surface Area of Each Body Part

	Name of part	Area(m^2)
1	Left Foot	0.0446
2	Right Foot	0.0437
3	Left Leg	0.0892
4	Right Leg	0.0879
5	Left Thigh	0.1630
6	Right Thigh	0.1670
7	Crotch	0.1740
8	Head	0.1100
9	Left Hand	0.0397
10	Right Hand	0.0394
11	Left Arm	0.0490
12	Right Arm	0.0500
13	Left Shoulder	0.0736
14	Right Shoulder	0.0778
15	Chest	0.1380
16	Back	0.1270
Total		1.4739

$$Icl = It - Ia/fcl \text{ or } It - Ia', \quad (4)$$

and

$$fcl = 1 + 0.3\ Icl. \quad (5)$$

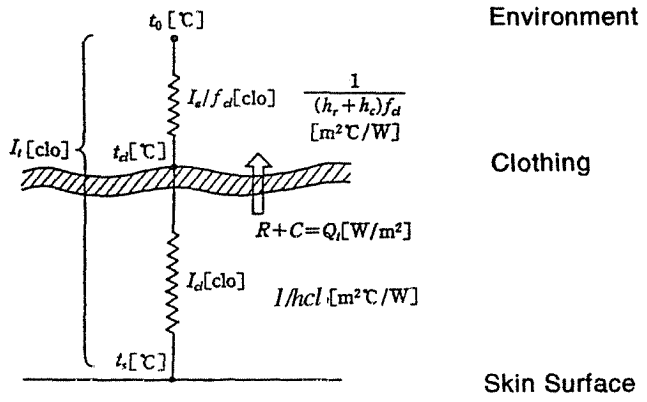


Figure 2 *Sensible heat transfer from skin surface to environment through clothing.*

Under steady-state conditions, heat supplied to the manikin's heating elements equals the heat loss from the skin surface. Thus, heat loss per unit skin surface area can be derived from the electricity consumption of each part. Equation 5 is quoted from McCullough et al. (1985).

Manikin Control Principles

There are two major control methods used for manikins. One is to keep the heater temperature constant and the other is to keep supply power constant. Temperature control may not be required for a manikin with a constant power supply, but the skin temperatures that result when it is used in non-uniform environments can be unrealistically high or low. This new manikin uses a third method.

The control of this manikin is based on human heat exchange with the thermal environment. The manikin differs from real human beings in that it does not evaporate moisture from lungs or skin, and this must be taken into account. Heat loss from the human body is expressed as Equation 6. Under steady-state conditions, metabolic heat production (M) is almost equal to the total heat loss from a human (Qm). For most indoor work, energy flows due to external work may be assumed to be negligible. Total heat loss (Qm) is divided into that from the skin surface (Qs) and that caused by respiration ($Qres$). Heat loss from the skin surface (Qs) consists of sensible heat loss (Qt) and latent heat loss (Es), as shown in Equation 7. Sensible heat loss (Qt) is in turn divided into radiative (R) and convective (C) heat loss, as shown in Equation 8.

$$Qm = M = Qs + Qres \tag{6}$$

$$Qs = Qt + Es \tag{7}$$

$$Qt = R + C \tag{8}$$

Thus, sensible heat loss from the skin surface (Qt) is expressed as Equation 9:

$$Qt = Qm - Qres - Es. \tag{9}$$

According to Fanger (1970) and ASHRAE (1989), heat loss caused by respiration ($Qres$) and evaporative heat loss from the skin surface (Es) can be expressed as shown below. Here, evaporative heat loss from each body part is unknown; it is not necessary for calculating sensible heat loss from each body part.

$$Qres = 1.7 \cdot 10^{-5} M(5867 - \text{Pa}) + 0.0014M(34 - ta) \tag{10}$$

$$Es = 3.05 \cdot 10^{-3}(5733 - 6.99M - \text{Pa}) + 0.42(M - 58.15) \tag{11}$$

Since air temperature (ta) is included in the second term of Equation 10, it is necessary to measure air temperature to estimate respiration heat loss. To avoid this, air temperature is assumed to be 20°C. This assumption affects only heat loss by respiration, causing a maximum 1.6% error of total heat loss within the range of 10°C to 30°C.

According to Fanger's (1970) comfort equation, the mean skin temperature under thermal neutrality may be estimated as Equation 12:

$$ts = 35.77 - 0.028\,Qm. \tag{12}$$

Since this thermal manikin is unable to sweat, Qm cannot be measured directly. For the present purposes, vapor pressure (Pa) is assumed to be 1.5 kPa, which is equivalent to typical indoor conditions at 24°C and 50% relative humidity (RH). Equation 13 may then be derived from Equations 6 through 12:

$$Qm = 1.96\,Qt - 21.56. \tag{13}$$

By inserting Equation 13 into Equation 12, the following equation is obtained:

$$ts = 36.4 - 0.054\,Qt. \tag{14}$$

To simulate Equation 14, the system controls the skin surface to have a thermal resistance offset of 0.054 $m^2 \cdot °C/W$. Figure 3 shows a diagram of skin temperature control. For example, shown as a dotted line, when the heater temperature is set at 36.4°C at the first estimation, the heat loss from the skin surface is measured as the electricity consumption of the heating element. However, this relationship between skin surface temperature and heat loss does not satisfy Equation 14, so the setpoint of the skin surface is iteratively changed until it meets Equation 14.

In this report, the thermal manikin was controlled to satisfy Equation 14. However, this equation may not be applicable under different conditions and different parts of the body. Bischof and Madsen (1991) compared skin temperatures of a thermal manikin like this one with skin temperatures measured on subjects. They showed that the skin temperature of the manikin's feet did not agree with the subjective temperatures, but they found good agreement at other parts of the body. The control equation for individual body parts should probably be adjusted to predict the local skin temperature with more accuracy.

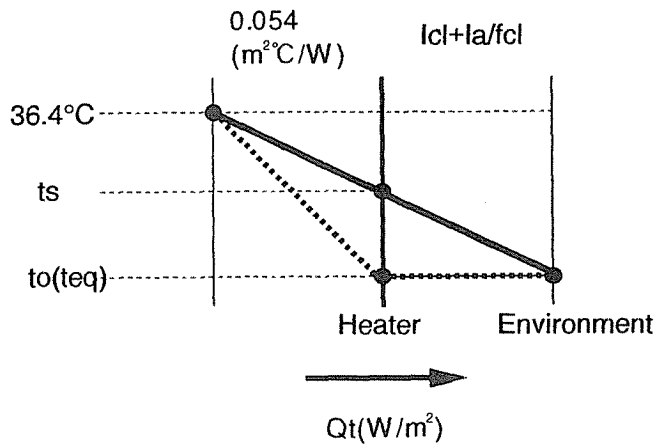


Figure 3 Diagram of skin surface temperature control.

Basic Clothing Insulation

The basic clothing insulation value (*Icl*) is calculated from Equations 1 through 5. In ISO (1991), measurement conditions for measuring the clo value with thermal manikins are described as follows: posture should be standing, air velocity should be less than 0.1 m/s, the difference between air temperature and mean radiant temperature should be less than 5°C, relative humidity should be stable during the experiment and in the range of 10% to 70% RH, heat loss from the manikin should be between 40 and 80 W/m^2, and mean skin temperature should be between 32°C and 34°C.

According to Gagge et al. (1941), the clo value was defined as occurring under thermal neutrality. Namely, they assumed that the sensible heat loss from a sedentary subject would be 44 W/m^2 and the mean skin temperature would be 33°C. To satisfy these conditions, their thermal manikin had to be exposed again and again until they could find a suitable combination of skin temperature setpoint and operative temperature. Since the thermal manikin described in this paper is controlled to satisfy Equation 14, it is rather easy to meet the original definition of Gagge et al.

Since it is hard to measure the combined heat transfer coefficient at the clothing surface, the combined heat transfer coefficient at the nude skin surface is applied instead of that at the clothing surface, as shown in Equations 2 through 4. In ASTM (1991), a method is given by which to measure clothing insulation, but the total insulation value (*It*) is only briefly mentioned. Measurement of the combined heat transfer coefficient at the clothing surface is not addressed in detail. The combined heat transfer coefficient at the nude skin surface is affected by the characteristics of the chamber and the chair when the manikin is in the sitting posture. Further research work may be required to determine combined heat transfer coefficients at the clothing surface and to prescribe measurement conditions.

EVALUATION OF THE THERMAL ENVIRONMENT

Equivalent Temperature Based on Thermal Manikin Measurements

Manikin-based equivalent temperature (*teq*) is defined as the temperature of a uniform enclosure in which a thermal manikin with realistic skin surface temperatures would lose heat at the same rate as it would in the actual environment. The idea of *teq* is closely related to Dufton's historic equivalent temperature (Dufton 1936), which was based on measurements with a prototypical manikin, the Eupatheoscope. This instrument, from which heat loss was measured, was an internally heated cylinder that measured 550 mm high and 190 mm in diameter. Because of resistance between the interior and the surface, its surface temperature varied with ambient temperature.

Equations 15 and 16 are mathematical expressions of *teq*. The values of (*Icli* + *Iai/fcli*) need to be obtained from thermal manikin measurements in order to solve the equations. The values of *fcli* and *Icli* at each body part cannot be obtained from thermal manikin experiments:

$$teq = 36.4 - (0.054 + 0.155(Icl + (Ia/fcl))Qt = ts - 0.155(Icl + Ia/fcl)Qt \quad (15)$$

$$teqi = tsi - 0.155(Icli + Iai/fcli)Qti. \quad (16)$$

The value (*Icli* + *Iai/fcli*) is changed by clothing, body posture, and chamber characteristics. According to the evaluation environment, a reference combination should be chosen. When nude, *Icl* is zero and *fcl* is unity. For the nude condition, the thermal resistance of the skin surface is the inverse of the combined heat transfer coefficient.

It should be noted that there are other "equivalent temperatures" with different physical bases. Bedford (1948) proposed an equation with which to calculate the equivalent temperature from air temperature, mean radiant temperature, and air velocity. This equation was based on the subjective responses of workers in British factories during the winter season and cannot be compared directly with this *teq*. In addition, Madsen (1976a, 1976b, 1979) and Olesen (1988) developed comfort-sensing instrumentation that delivered an "equivalent temperature" output. Madsen et al. (1984) described equations to account for clothing in the calculation of equivalent temperature. As mentioned previously, basic clothing insulation (*Icl*) may be affected by air movement, so a comparison of the equivalent temperatures based on a thermal manikin (*teq*) with those of Bedford, Dufton, and Madsen may be useful.

PMV Based on Equivalent Temperature

A PMV may be easily calculated based on thermal manikin measurements. Since *teq* is defined as being under uniform conditions, PMV is calculated by inserting *teq* into the air temperature and mean radiant temperature of its

program. Air velocity is assumed to be still air (natural convection) and relative humidity is 50%. The actual basic clothing insulation value and activity level may be entered into the calculation. Figure 4 shows the sample relationship between *teq* and PMV for an experimental clothing ensemble in the sitting posture.

CALIBRATION OF THERMAL MANIKIN UNDER UNIFORM CONDITIONS

Effect of Ambient Temperature When Nude

To find the relationship between manikin heat loss and surrounding temperature, the nude thermal manikin was exposed in the climatic chamber at a Japanese university. In the climatic chamber (4.05 m by 4.85 m by 2.5 m [h]), air is supplied upward from the entire surface of a perforated floor and returned to a perforated ceiling. There is a 5-cm-thick air layer inside surrounding walls to keep the mean radiant temperature equal to the air temperature. The nude thermal manikin in a sitting posture was exposed under the operative temperatures of 19.8°C, 24.8°C, and 29.7°C. No vertical temperature difference was observed in the chamber. Mean air velocity was measured as 0.1 m/s using an omnidirectional air velocity sensor. The heat losses and skin temperatures of the 16 parts of the body were measured. Operative temperature was measured with a globe thermometer at 1.6 m from the floor level during the entire experimental period. Relative humidity was kept around 50% RH. An open mesh-surfaced chair was used to expose the whole body surface. The steady-state conditions of chamber and manikin, heat losses, skin temperatures, and globe temperature were recorded at one-minute intervals. The data output from the control computer was already the mean of 60 measurements. An average of five measurements (300 observations) was used for data analysis.

Figure 5 shows heat loss from each part of the nude body; the heat loss from the whole body was 91.2 W/m^2 at 19.8°C, 64.4 W/m^2 at 24.8°C, and 37.7 W/m^2 at 29.7°C. The heat loss from the head was smaller than from other parts of the body because of its hair. Heat losses at the feet and hands were slightly greater than those of other parts, due primarily to their smaller diameter. The relationship between skin temperature and heat loss was controlled to satisfy Equation 14. Mean skin temperature was 31.5°C at an operative temperature of 19.8°C, 32.9°C at 24.8°C, and 34.4°C at 29.7°C.

Figure 6 shows the combined heat transfer coefficient (*hr* + *hc*) for each part of the body. The combined heat transfer coefficient for each part was not affected by the exposure temperatures. The mean of the three conditions for the whole body was 7.9 W/m^2·°C. The combined heat transfer coefficient of feet and hands was greater than for other parts because of the relatively small diameter of these parts. The coefficients at the thighs and crotch were smaller than those at other parts, probably because of greater radiation from adjacent body parts.

In the present manikin study, the combined heat transfer coefficient at the nude skin surface was independent of ambient temperatures within the range of experimental conditions. When the radiative heat transfer coefficient (*hr*) is assumed to be 4.7 W/m^2·°C (ASHRAE 1989), the convective heat transfer coefficient (*hc*) is 3.2 W/m^2·°C. Mitchell (1974) measured the convective heat transfer coefficient for the human body and found it to be 3.1 W/m^2·°C with still air movement and a sitting posture. The value of this thermal manikin measurement is quite close to his result.

Effect of Body Posture When Clothed

Virtually the same experiments were conducted in the controlled environmental chamber at a U.S. university's laboratory. The chamber is 5.5 m by 5.5 m by 2.5 m (h) and is designed to resemble a modern office space while still allowing a high degree of control over the test chamber's thermal environment (Arens et al. 1991). The thermal

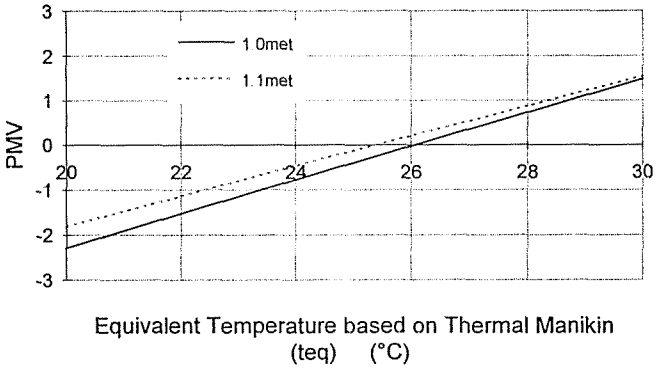


Figure 4 *Relationship between equivalent temperature based on thermal manikin* (teq) *and PMV. The thermal manikin is seated wearing clothing of 0.55 clo and at conditions of 50% relative humidity.*

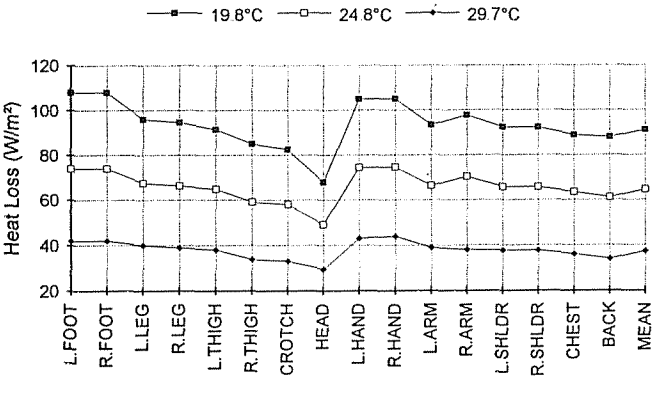


Figure 5 *Heat loss from each part of the nude body in the sitting posture at three different operative temperatures.*

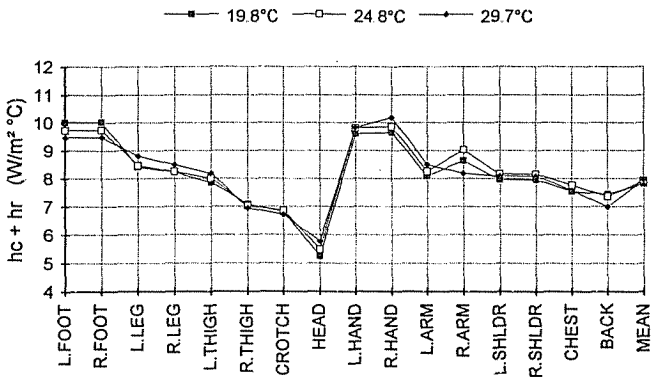


Figure 6 *Combined heat transfer coefficient for each part of the nude body in the sitting posture.*

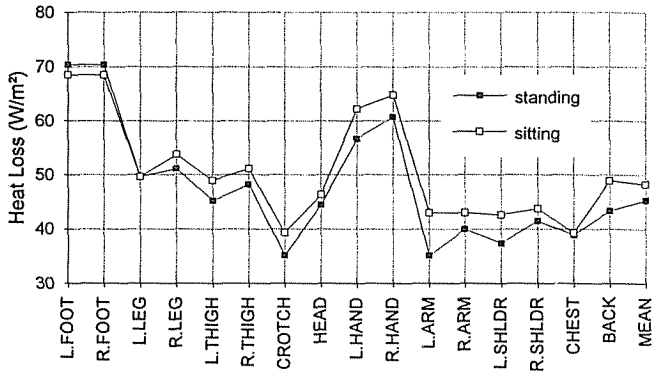


Figure 7 *Heat loss from each part of the clothed body in sitting and standing postures.*

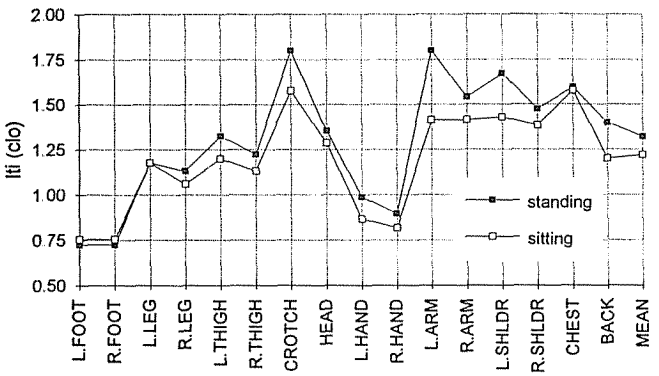


Figure 8 *Total clothing insulation value* (Iti *or* Icli + Iai/fcli) *for each part of the body when sitting and standing.*

manikin with clothing was exposed at an operative temperature of 24.7°C with sitting and standing postures. The clothing ensemble consisted of panties, bra, long-sleeved sweatshirt, sweatpants, and shoes. Air temperatures and velocities were measured after each experiment at 0.1 m, 0.6 m, 1.1 m, and 1.7 m from floor level. Almost no vertical temperature difference was observed. Air movement was almost still during the experiments, and the average of the mean air velocities at the four height levels was 0.05 m/s. Heat losses and skin temperatures were recorded in the same way as in the Japanese tests. A mesh type of chair was used for the sitting experiments.

Figure 7 shows heat loss from each part of the body with clothing. Both sitting and standing conditions are shown. The heat loss from the whole body was 48.2 W/m^2 for the sitting posture and 45.3 W/m^2 for the standing posture. Heat loss when standing was 6% lower than that when sitting. The mean skin temperature was 33.8°C for the sitting posture and 34.0°C for the standing posture.

Figure 8 shows the thermal resistance between the skin surface and the environment (*Iti* or *Icli* + *Iai/fcli*) for each part of the body with clothing. These values were applied to calculate equivalent temperatures. The thermal resistance for the whole body was 0.189 m^2·°C/W (1.22 clo) when sitting and 0.205 m^2·°C/W (1.32 clo) when standing. Since the thermal resistance of the nude body when sitting (*Ia*) was measured as 0.78 clo during another experiment in this chamber, the basic thermal insulation value of the clothing ensemble when sitting (*Icl*) was calculated to be 0.55 clo.

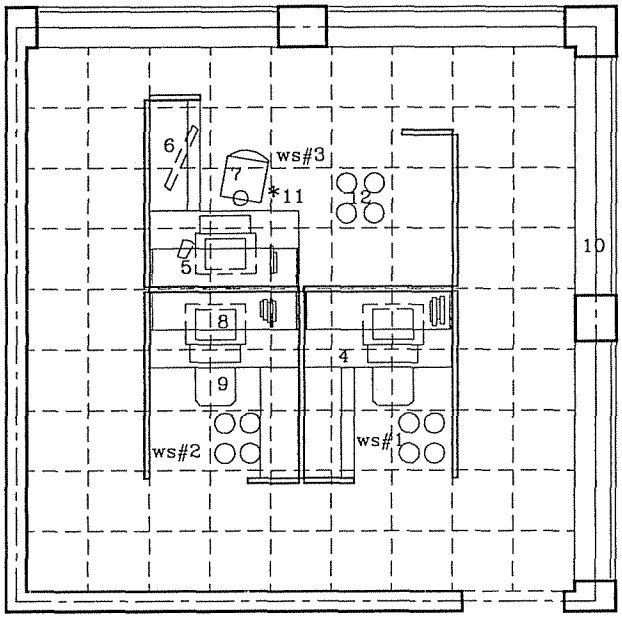
MEASUREMENT OF THERMAL EFFECTS UNDER NON-UNIFORM CONDITIONS

The manikin was then used to evaluate thermally non-uniform environments produced by a floor-based air supply system. More detailed descriptions of the system and experimental results will be published in an upcoming paper (Bauman et al., in press), so not all experimental results are described here.

Experimental Setup and Conditions

The manikin was tested in a mockup of an office with a floor-supply air distribution system. Floor systems are capable of producing a wide range of local environmental conditions by injecting supply air into the occupant's nearby environment. In conjunction with the typical heat-generating equipment found in offices, such systems produce non-uniform local environments around the occupant, with asymmetrical airflows and radiant fields.

The modular workstation configuration shown in Figure 9 was installed in the chamber. The manikin was seated at the desk in a larger workstation (WS #3). Cool supply air was delivered through the floor module on the manikin's left. Heat sources were provided to simulate typical office load distributions and densities. Two different office load densities were studied here. At a medium load, some typical office equipment, including a desktop computer and a task light, was installed. At a high load, a 200-W heater was added under the desk to the right of the manikin to simulate the load of a tower-type computer. The thermal manikin also provided heat. All experiments were carried out under



WS#1	workstation #1	7	manikin
WS#2	workstation #2	8	computer
WS#4	workstation #3	9	chair
4	desk	10	annular space
5	task light	11	measurement location
6	heat source	12	floor supply module

Figure 9 *Chamber plan for underfloor air distribution system.*

steady-state conditions chosen to represent the interior zone of an office building. The test conditions are shown in Table 2. In this table, the test numbers are also shown; these numbers will be referred to in the discussion of the results.

As indicated, the effects of the floor unit's grille position were studied for two orientations: (1) grilles turned inward, toward the center of the module (inward), and (2) all grilles turned toward the desk in the workstation (toward). Measurements using a temperature and velocity sensor array were made at the manikin's position after each experimental session. In Figure 9 the thermal manikin and measurement locations are also shown. Temperatures and air velocities at 0.1 m, 0.6 m, 1.1 m, and 1.7 m were recorded over a one-minute period.

The thermal manikin was sitting or standing during the experiments and was wearing the same clothing ensemble as that used during the uniform tests shown in Figures 7 and 8. After conditions reached steady state, heat losses and skin temperatures were sampled. An average of five measurements (300 observations) was used for data analysis. Table 3 shows the summary of results.

Effect of Supply Temperature

Figure 10a shows the effect of supply temperature and heat load level on the equivalent temperature (*teq*). Both

TABLE 2
Experimental Conditions of Underfloor Air Distribution System

Test Number	Supply Volume (CFM)	Heat Load*	Supply Temp. (°C)	Direction of Outlet**	Posture
Q150/17/T	150	mid	17		
Q210/17/T	210	max	17		
Q210/20/T	210	mid	20	Toward	Sitting
Q270/18/T	270	max	18		
Q270/21/T	270	mid	21		
Q150/17/I	150	mid	17		
Q210/17/I	210	max	17		
Q210/20/I	210	mid	20	Inward	Sitting
Q270/18/I	270	max	18		
Q270/21/I	270	mid	21		

* Heat Load: 'mid' means medium heat load including a computer and task lighting. 'max' means maximum heat load including a computer and a task lighting in addition to a 200W heater.

** Direction of Outlet:' toward' means 4 outlets are aimed toward the thermal manikin and 'inward' means those are aimed inward.

TABLE 3
***Summary of Experimental Results* for Underfloor Air Distribution System**

Test Number	Mean teq (°C)	PMV	Mean Air Temp. (°C)	Mean Air Velocity (m/s)	Δ teq right-left (°C)	Δ teq head-(foot+leg) (°C)
Q150/17/T	22.5	-0.8	23.7	0.07	2.3	3.5
Q210/17/T	23.0	-0.7	24.4	0.05	3.6	-0.3
Q210/20/T	24.9	-0.1	24.9	0.06	0.9	-0.5
Q270/18/T	24.5	-0.2	24.8	0.09	3.1	-1.5
Q270/21/T	24.1	-0.3	24.8	0.17	1.0	-0.1
Q150/17/I	23.2	-0.6	23.2	0.01	1.2	2.6
Q210/17/I	23.2	-0.6	24.3	0.05	2.4	1.0
Q210/20/I	24.4	-0.2	24.6	0.03	1.7	0.0
Q270/18/I	23.8	-0.4	24.3	0.10	2.5	-0.8
Q270/21/I	24.8	-0.1	25.2	0.03	0.1	0.8

Q270/18/T and Q270/21/T are shown. The load was maximum for Q270/18/T (with a 200-W heat source under the desk) and medium for Q270/21/T. The supply volume was 270 cfm for both cases, and the supply air temperature was set at either 18°C or 21°C. Three underfloor air distribution systems were installed, and each system supplied 90 cfm. Grilles were turned toward the desk for both cases. The equivalent temperature of the whole body in the case of Q270/18/T was 24.5°C and that in Q270/21/T was 24.1°C.

A *teq* of 24.5°C means that the thermal manikin exposed in Q270/18/T would lose heat at the same rate as in the uniform environment at an operative temperature of 24.5°C. When PMV is calculated from a whole-body *teq*, using an activity level of 1.1 met to represent typical office work, the PMV for Q270/18/T is −0.2 and that for Q270/21/T is −0.3. Both cases are therefore within the comfort range for whole-body sensation. However, since Q270/18/T included the tower-computer-style heat source under the desk, the individual equivalent temperatures at the right hand, arm, and thigh were much higher than at other parts, and the *teq* at the feet was slightly lower because of the low supply air temperature.

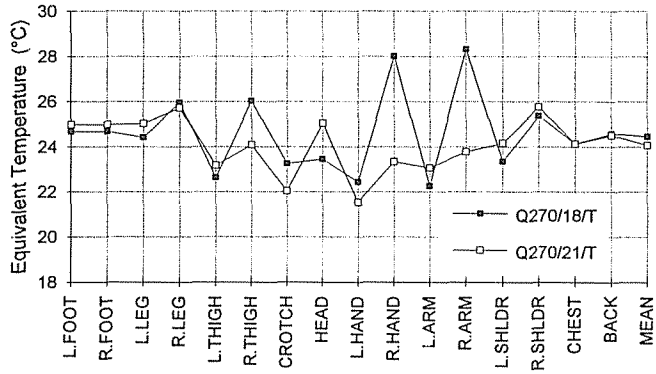


Figure 10a *Effect of supply temperature on thermal-manikin-based equivalent temperature* (teq). *Cases Q270/18/T and Q270/21/T are shown. The maximum load was installed for Q270/18/T and a medium load for Q270/21/T. The supply volume was set constant at 270 cfm and the supply air temperature was set at either 18°C or 21°C. Diffuser grilles were turned at the toward position for both cases.*

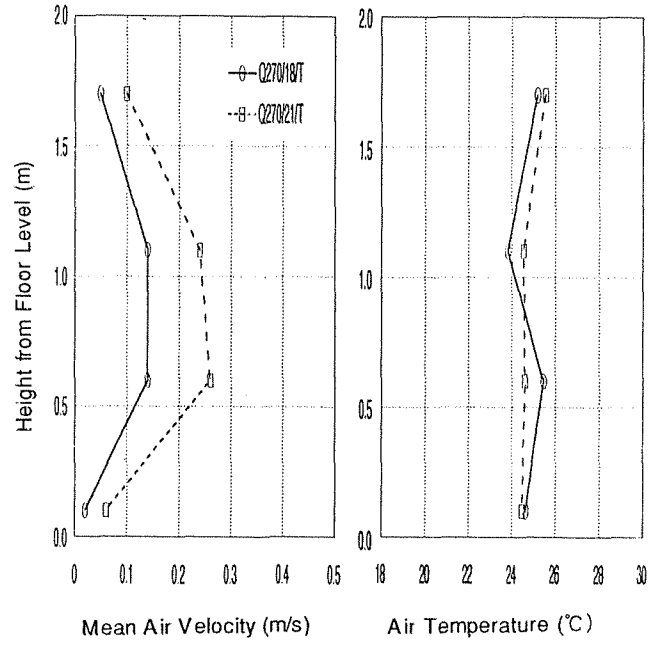


Figure 10b *Air temperatures and velocities at four different heights.*

For comparison, Figure 10b shows air temperatures and velocities at four different heights. The asymmetry caused by the heat sources could not be detected by air temperature measurements, yet had a big impact on heat loss from the manikin.

Effect of Supply Volume

Figure 11a shows the effects of supply volume and heat load level on equivalent temperature (*teq*). The grilles were turned inward for all cases. Three cases—Q150/17/I, Q210/17/I, and Q270/18/I—are shown. The load was maximum for Q270/18/I and Q210/17/I and medium for Q150/17/I. The supply temperature was set at 17°C or 18°C. The *teq* for the whole body in the case of Q150/17/I was 23.2°C, that of Q210/17/I was 23.2°C, and that of Q270/18/I was 23.8°C. PMV calculated from whole-body *teq*, with the activity level set at 1.1 met, was −0.6 for Q150/17/I, −0.6 for Q210/17/I, and −0.4 for Q270/18/I. The equivalent temperatures of the right thigh, hand, and arm were much higher than those of the left parts because of the heat source asymmetry. The equivalent temperatures at the foot and leg were considerably lower in Q150/17/I, and a great vertical temperature difference was observed. The difference between the *teq* at the foot and leg and that at the head was 3.0°C, which could be expected to cause local discomfort (Olesen et al. 1979). In comparing three tests, Figure 11b shows the air temperatures and velocities at four different heights.

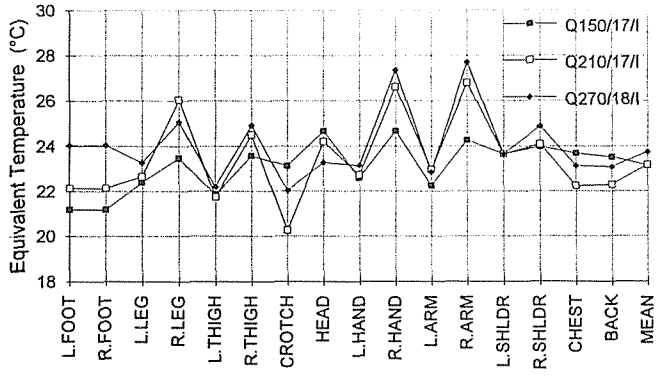


Figure 11a *Effects of supply volume on thermal-manikin-based equivalent temperature* (teq). *The diffuser grilles were turned inward for all cases. Cases Q150/17/I, Q210/17/I, and Q270/18/I are shown. The maximum load was installed for Q270/18/I and Q210/17/I and the medium load for Q150/17/I. Supply temperature was set at 17°C or 18°C.*

Application for Evaluation of Office Environments

As shown in Figures 10 and 11, equivalent temperature (*teq*) was a useful tool with which to determine the effects of local air movement and radiant asymmetry, while measured air temperatures and air velocities provided less detailed explanations. Since the heat loss from the thermal manikin is the final result of realistic heat transfer, it produces an index that can measure the effects of complex thermal environments such as those found in present workstations. A plane radiant temperature sensor might detect radiant asymmetry. For further research, the difference between manikin measurements and plane radiant temperature might be compared.

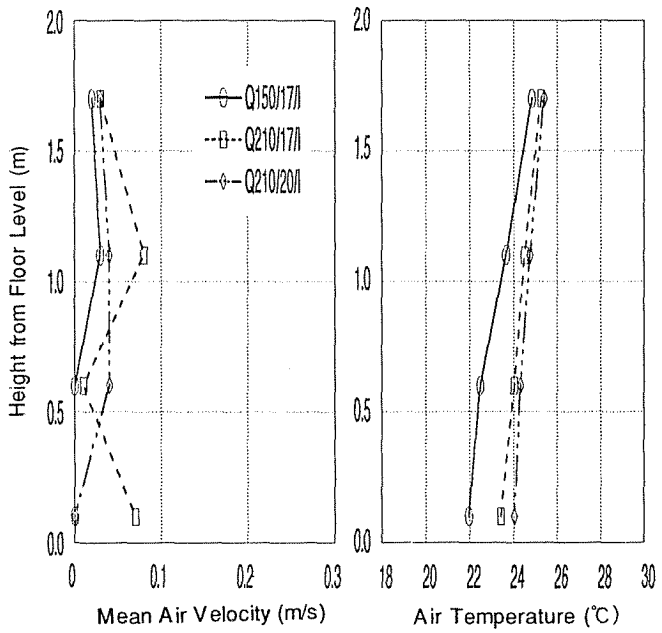


Figure 11b Air temperatures and velocities at four different heights.

Heat sources had a big impact on *teq*. Tower computers and workstations are now often placed under desks. These kinds of office equipment emit heat and need to be considered for their effects on occupants of office spaces. The airflows supplied by underfloor air distribution units affect the occupant directly and control the vertical temperature profile. Such systems need to be evaluated on a more location-specific basis than do conventional ceiling-based air distribution systems.

CONCLUSIONS

1. The control method and structure of a new thermal manikin was described in this paper.
2. Equivalent temperature based on a thermal manikin (*teq*) was proposed to measure and evaluate the thermal environment. A method by which to calculate PMV from manikin heat loss was given.
3. The nude thermal manikin in the sitting posture was exposed in the climatic chamber under three operative temperatures. Heat losses from the whole body were 91.2 W/m^2 at an operative temperature of 19.8°C, 64.4 W/m^2 at 24.8°C, and 37.7 W/m^2 at 29.7°C. Heat losses at the feet and hands were slightly greater than those at other parts. The combined heat transfer coefficient for the nude and sitting manikin was not affected by the exposure temperatures. The mean of the combined heat transfer coefficients for the whole body in the nude was 7.9 W/m^2·°C. When the radiative heat transfer coefficient for the human body (*hr*) was assumed to be 4.7 W/m^2·°C, the convective heat transfer coefficient (*hc*) was estimated to be 3.2 W/m^2·°C.
4. The manikin was also exposed in the climatic chamber in sitting and standing postures with clothing. The heat losses from the whole body were 48.2 W/m^2 for the sitting posture and 45.3 W/m^2 for the standing posture. The total thermal resistance for the whole body was 0.189 m^2·°C/W (1.22 clo) for the sitting posture and 0.205 m^2·°C/W (1.32 clo) for the standing posture. The basic clothing insulation value of the tested clothing ensemble in the sitting (*Icl*) posture was calculated to be 0.55 clo.
5. Applications of equivalent temperature based on the thermal manikin (*teq*) for an underfloor air distribution system were shown. Equivalent temperature based on the thermal manikin (*teq*) was shown to be a useful tool with which to detect the effects of asymmetries in heat sources and airflow.

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NOMENCLATURE

C	=	heat loss by convection (W/m^2)
Es	=	evaporative heat loss from skin surface (W/m^2)
Ia	=	thermal resistance at skin surface per unit total skin surface area (clo)
Ia'	=	thermal resistance at skin surface per unit clothed surface area (clo)
Icl	=	basic clothing insulation (clo)
It	=	total clothing insulation (clo)
M	=	metabolic heat production rate (W/m^2)
Pa	=	water vapor pressure (Pa)
Qa	=	sensible heat loss from skin surface in the nude (W/m^2)
Qm	=	total heat loss from human body (W/m^2)
Qs	=	total heat loss from skin surface (W/m^2)
Qt	=	sensible heat loss from skin surface (W/m^2)
R	=	heat loss by radiation (W/m^2)
Qres	=	heat loss by respiration (W/m^2)
fcl	=	clothing area factor
hc	=	convective heat transfer coefficient (W/m^2·°C)
hr	=	radiative heat transfer coefficient (W/m^2·°C)
tcl	=	clothing outer surface temperature (°C)
teq	=	equivalent temperature based on thermal manikin (°C)
to	=	operative temperature (°C)
ts	=	mean skin surface temperature (°C)

Subscripts

n	=	in the nude
cl	=	with clothing
i	=	individual part of the body

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