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FREE CONVECTIVE HEAT TRANSFER COEFFICIENT OF A HEATED FULL-SCALE MANIKIN

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INTRODUCTION

The significance of the convective heat exchange between the human body and the ambient air is a subject of continuing interest to physiologists, engineers and those concerned with human heat tolerance and thermal comfort during rest and exercise.

Reviewing the heat transfer literature, there are two basic methods of studying this subject. The first is the theoretical method. There are practical difficulties in applying the general theory of heat transfer to the human body because of its complex shape. So, segments of human body are considered as cylinders, spheres and verticle plates, Nishi et al (1970), Rapp (1973). Obviously this method introduces some errors.

The second method of study is experimental. This method is more suitable for determining the constants of proportionality and other specific details, see Mitchell et al (1969), Colin et al (1967), Olesen et al (1986) and Homma et al (1988). In some of these studies a real human body was used. Others use a thermal mankin.

One of the difficulties with using real human bodies for these experiments is that each person has his own physical characteristics - physical condition, health etc. As a result, a large sample has to be taken to make the result significant. To overcome this difficulty, people have tried to use a thermal manikin to measure the convective heat transfer [illegible] n body.

The laws governing radiant and convective heat transfer between the human body and its environments are rigorous physical laws. No physiological laws are involved. This is an additional reason as to why the thermal manikin is a suitable alternative to the human body.

An average heated full-scaled manikin has a general significant.

It is difficult to use the human body to measure convective heat transfer when it is not in a static situation. However, in such circumstances it is quite possible to use the thermal manikin to stimulate this situation.

Referring back to previous investigations, a lot of thermal manikins are composed of several cylinders, spheres, verticle plates and so forth. Usually the manikins are measured with uniform temperature distributions. This has limited the application of the thermal manikin. In reality, temperature distributions are diverse in different physiological situations, Hänel et al (1986), Baborsky et al (1986). See Fig. 1 and Fig. 2.

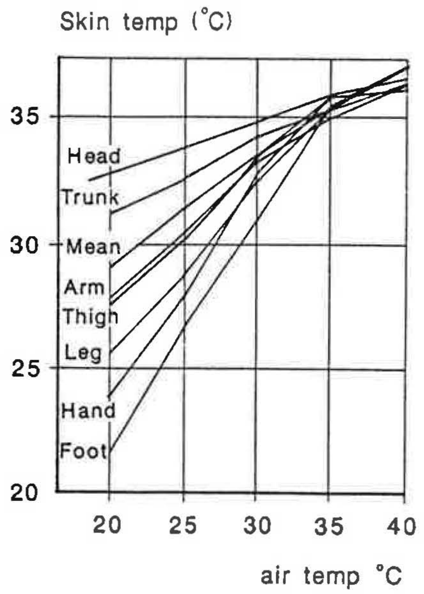

Fig. 1 Temperature distribution when people rest

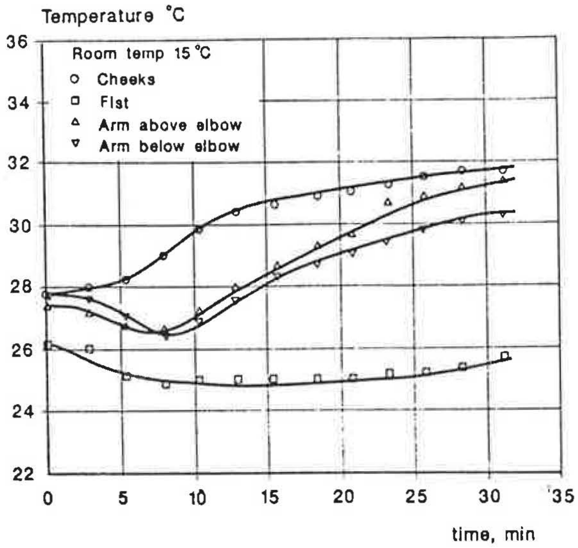

Fig. 2 Skin temperature distribution when people cycle (From Baborsky et al (1986))

This paper uses the everage full-scale manikin to measure convective heat transfer. Every segment has different temperature distribution.

PRINCIPLE AND THEORY

The thermal manikin was separated into several segments in this paper. Every segment was heated by electrical resistance. When the thermal manikin was in thermal equilibrium, there existed the equation as below

$$Q = Q_c + Q_R \qquad (1)$$

where Q is the heat product, it is equal to electrical power W

Q_c is the convective heat exchange, W

Q_R is the radiant heat exchange, W

THE LAW OF RADIATION

The radiation exchanged between the surface of any enclosed convex body and a concave surface surrounding it is related to the temperature of the two surfaces by an expression derived by Christiansen from the Stefan-Boltzmann Law, Jakob et al (1957).

$$Q_R = A_1[1/\varepsilon_1+(1/\varepsilon_2-1)A_1/A_2]^{-1}(T_1^4-T_2^4) \qquad (2)$$

where Q_R is the radiant heat exchange, W

is Stefan-Bultzmann constant 5.67×10^{-8} W/m$^2\cdot$K^4

A_1 is the radiant surface area of enclosed body, m^2

A_2 is the area of the surrounding surfaces, m^2

ε_1 is the emissivity of the surface of the enclosed body

ε_2 is the emissivity of the surrounding surfaces

T_1 is the mean surface temperature of the enclosed body, ^{o}K

T_2 is the mean temperature of the surrounding surfaces, ^{o}K

This expression can be simplified when applied to a manikin in indoor surroundings. First, the area of the surfaces bounding the environment (the wall area of a room, for instance) is generally much larger than the surface area of the segment of the thermal manikin. Hence $A_1/A_2 \leqq 1$. Second, the emissivity of the surfaces of indoor objects at normal room temperatures is close to unity, McAdams (1954), so that $1/\varepsilon_2 - 1 \leqq 1$. The following approximation can therefore be made with very little error:

$$Q_R = \sigma\varepsilon_1 A_1 (T_1^4 - T_2^4) \qquad (3)$$

For the segment of thermal manikin

$$Q_R = \sigma\varepsilon_s\phi A(T_s^4 - T_R^4) \qquad (4)$$

where ε_s is the emissivity of the skin

T_s is the mean skin temperature, ^{o}K

T_R is the mean temperature of the surrounding surfaces, ^{o}K

A is segment surface area, m^2

ϕ is the effective radiation area factor

THE LAW OF CONVECTION

$$Q_c = \alpha_c(\Theta_s - \Theta_a)\cdot A \qquad (5)$$

where α_c is convective heat transfer coefficient, $W/m^2\cdot {}^oC$

Θ_s is mean skin temperature, ^oC

Θ_a is mean air temperature, ^oC

A is the skin area, m^2

For equations (1), (4), (5) we gain:

$$Q = \alpha_c\cdot A\cdot(\Theta_s - \Theta_a) + \sigma\cdot\varepsilon\cdot\phi\cdot A\cdot(T_s^4 - T_R^4)$$

$$\alpha_c = [Q-\sigma\cdot\varepsilon\cdot\phi\cdot A\cdot(T_s^4-T_R^4)]/(A\cdot(\Theta_s-\Theta_a)) \quad (6)$$

So if the values of parameters are known, it is possible to calculate α_c in the right side.

EXPERIMENT

ENVIRONMENT

The thermal manikin was placed in a climatic chamber. The geometrical length was 4400 x 3700 x 2820 mm. The air temperature in this chamber can be regulated from $0^oC \sim 50^oC$. The room temperature distribution was almost uniform. The air velocity was less than 0,05 m/s.

THERMAL MANIKIN

The thermal manikin was established by Johansson et al (1987), see Fig. 3.

Fig. 3 A full-scale manikin (From Johansson et al 1987)

The thermal manikin was divided into seven segments, see Fig. 4.

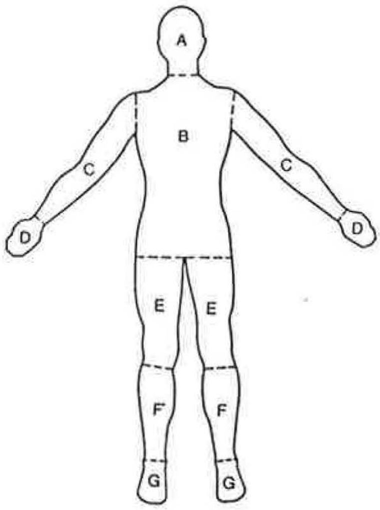

Fig. 4 Segments division of the thermal manikin

Some electrical resistances were arranged in every segment, see Fig. 5. In order to make sure the temperature distribution was uniform, some metal sheets were used. See Fig. 6.

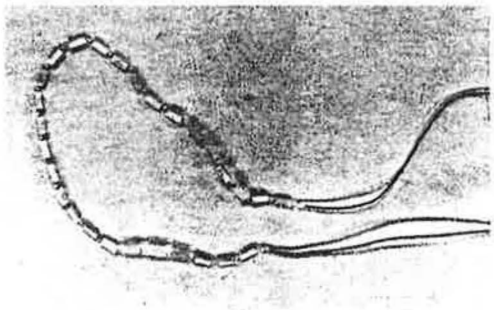

Fig. 5 Electrical resistance (From Johansson 1987)

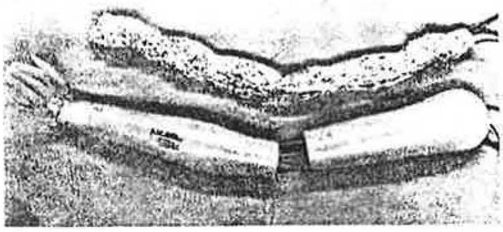

Fig. 6 Assembly of electrical resistence

Nineteen thermocouples were arranged on the surfaces of the thermal manikin to measure the skin temperature, see Fig. 7. The ready manikin is given in Fig. 8.

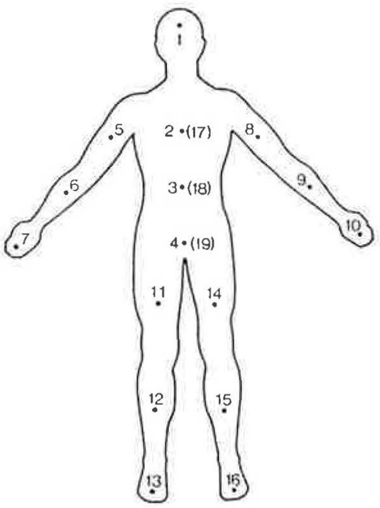

Fig. 7 The positions of thermocouples

Fig. 8 The ready manikin

In order to gain accurate data, the area of every segment surface was remeasured. Some white paper was stuck on the manikin surface. The paper was painted, then the paper was taken away and the painted paper areas measured.

For every segment, values are given in the table 1.

Table 1 Parameters of thermal manikin

Body segment		Area(m^2)	Resistance(Ω)
A	Head	0,133	277,9
B	Trunk	0,521	104,4
C	Arms	0,217	218,9
D	Hands	0,090	693,0
E	Thighs	0,291	161,2
F	Legs	0,224	205,2
G	Feet	0,136	1156

MEASUREMENT

Arrangement of measuring points

Besides the skin temperature, the following data was also required: the surrounding temperature, air temperature and electrical power for every segment.

There are six surfaces surrounding. Six thermocouples were placed on the six surfaces, see Fig. 9.

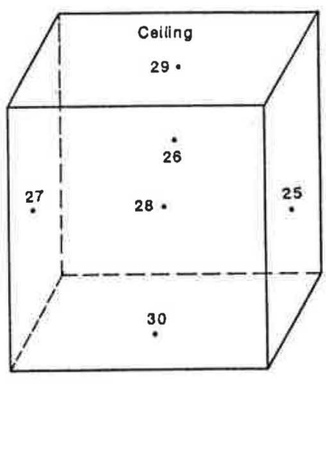

Fig. 9 Measurement points of wall temperature

Five nude thermocouples were hung around the thermal manikin. These five thermocouples were 100 mm far from the manikin, see Fig 10.

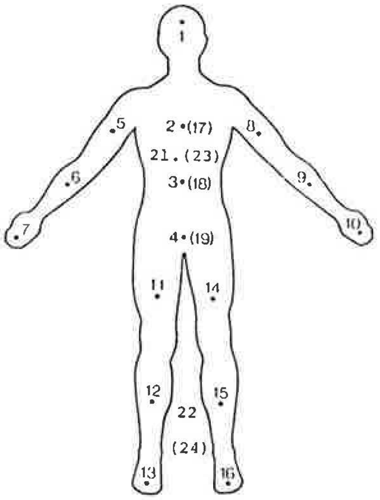

Fig. 10 Air temperature measurement points.

Here, points 20-24 are for air temperature measurement

Temperature measurement

Thermocouples were used to measure to temperature points. The type of thermocouple is Cu-Const. The second instrument is PM 8237 A Multipoint data Recorder. Using internal cold-junction compensation, the error was less than 1°C.

Voltage measurement

Because the pure electrical resistance was used, there was no time lag between the voltage and current. According to the electrical knowledge:

$$P = \frac{U_{eff}^2}{R} \qquad (7)$$

where P is the electric power, W
U_{eff} is the effective voltage, V
R is the electrical resistance, Ω

In fact, the total heat product Q was equal to P. For every segment, the electrical resistance was certain, so we could obtain different values for Q by changing the voltage. Seven transformers were used to seven segments to create different voltages or different heat products.

Air velocity measurement

Air velocity is measured with anemometer. In this experiment, the air velocity was less than 0,05 m/s.

Measuremental conditions

Owing to the fact that the difference between the skin temperature and the ambient air varies from 0°C to 40°C, so the experimental conditions were arranged to cover this range.

Nine different conditions for every segment were measured in this paper, see table 2.

Table 2 Measuremental conditions

Air temp. (°C)	Voltages for every segment(V)
19	40 50 60 70
22	50 60 70 80 90

During the whole measurement, the data for temperature was automatically recorded by PM 8237 A Multipoint data recorder. The interval time was one minute.

CALCULATION

To avoid the random error, the last five minutes data was adapted.

From the measuremental results, the air temperature differences among the five measuremental points were very small, so the mean air temperature was used to calculate the convective heat transfer coefficient instead of local air temperature.

$$\Theta_a = (\sum_{i=1}^{n_1} \sum_{j=1}^{n_2} \Theta_{aij})/(n_1 \cdot n_2) \quad (8)$$

where n_1 is the total repeated times, here $n_1 = 5$

n_2 is the total numbers of measurement points

subscript i is the repeated times

j is the number of thermocouples

For mean skin temperature

$$\Theta_{sA} = (\sum_{i=1}^{n_1} \Theta_{si1})/n_1$$

$$\Theta_{sB} = (\sum_{i=1}^{n_1} (\Theta_{si,2}+\Theta_{si,3}+\Theta_{si,4}+\Theta_{si,17}+ +\Theta_{si,18}+\Theta_{si,19}))/(6n_1)$$

$$\Theta_{sC} = (\sum_{i=1}^{n_1} (\Theta_{si,5}+\Theta_{si,6}+\Theta_{si,8}+\Theta_{si,9}))/(4n_1)$$

$$\Theta_{sD} = (\sum_{i=1}^{n_1} (\Theta_{si,7}+\Theta_{si,10}\ )/(2n_1)$$

$$\Theta_{sE} = (\sum_{i=1}^{n_1} (\Theta_{si,11}+\Theta_{si,14}))/(2n_1)$$

$$\Theta_{sF} = (\sum_{i=1}^{n_1} (\Theta_{si,12}+\Theta_{si,15}))/(2n_1)$$

$$\Theta_{sG} = (\sum_{i=1}^{n_1} (\Theta_{si,13}+\Theta_{si,16}))/(2n_1)$$

The mean wall temperature is used to represent radiant temperature:

$$\Theta_w = (\sum_{i=1}^{n_1} \sum_{j=1}^{6} \Theta_{wi,j+24})/(6n_1) \quad (9)$$

Heat product:

$$Q = U_{eff}^2/R \quad (10)$$

Using equation (4)the convective heat transfer coefficient for every segment can be calculated.

For ϕ and ε in equation (4), because the thermal manikin is painted, we adapt 0,92 for every segment ε, Yang (1979). For an effective radiation area factor the value 0,8 is adapted for every segment.

The manikin is standing on the ground, so heat-insulating material was beneath the feet to avoid heat conduction. The convective area is smaller than the skin-surface. The area beneath the feet was measured using the same method as above. Then we calculated the rate of convective area/the total area. The value is 0,801.

Performing lengthy calculations is tedious and may introduce errors, so a computer program was written for calculations.

RESULTS AND ANALYSIS

RESULTS

From the above calculation, the results are given in Table 3 to Table 11 and Fig. 11 to Fig. 17.

Table 3 Convective heat transfer coefficient, θ_w =19,3°C, θ_a =19,3°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m^2·°C)
A	Head	41,1	26,6	7,2	1,9
B	Trunk	39,6	22,4	3,1	5,0
C	Arms	40,0	23,0	3,6	4,9
E	Thighs	40,2	22,8	3,5	5,5
F	Legs	39,9	23,5	4,1	4,0
G	Feet	40,6	20,8	1,4	4,8

Table 4 Convective heat transfer coefficient, θ_w =19,7°C, θ_a =19,7°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m^2·°C)
A	Head	50,4	30,8	11,1	1,7
B	Trunk	50,0	25,2	5,5	4,0
C	Arms	50,3	25,6	5,9	4,7
E	Thighs	49,9	24,5	4,8	6,7
F	Legs	50,2	27,2	7,5	2,9
G	Feet	50,1	22,2	2,4	3,9

Table 5 Convective heat transfer coefficient, θ_w =21,9°C, θ_a =21,9°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m^2·°C)
A	Head	59,2	35,0	15,1	1,7
B	Trunk	61,2	28,0	8,1	4,1
C	Arms	60,8	28,1	8,2	5,1
E	Thighs	60,6	27,8	7,9	5,5
F	Legs	60,3	30,2	10,3	3,2
G	Feet	60,2	23,6	3,8	3,5

Table 6 Convective heat transfer coefficient, θ_w =21,9°C, θ_a =21,9°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m^2·°C)
A	Head	69,4	39,3	17,4	2,8
B	Trunk	69,3	31,9	9,9	4,3
C	Arms	69,6	31,5	9,6	6,1
E	Thighs	69,8	31,3	9,4	6,5
F	Legs	69,6	32,5	10,6	5,3
G	Feet	69,8	25,4	3,5	6,6

Table 7 Convective heat transfer coefficient, θ_w =15,1°C, θ_a =15,5°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m$^2\cdot$°C)
A	Head	49,6	25,1	9,6	2,5
B	Trunk	49,4	20,4	4,9	4,6
C	Arms	49,1	21,3	5,8	4,3
E	Thighs	49,4	21,2	5,6	4,8
F	Legs	49,8	22,9	7,4	2,8
G	Feet	48,9	17,5	2,0	4,5

Table 8 Convective heat transfer coefficient, θ_w =16,2°C, θ_a =16,2°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m$^2\cdot$°C)
A	Head	61,3	30,7	14,4	2,6
B	Trunk	60,5	24,3	8,0	4,1
C	Arms	60,4	24,9	8,7	4,5
E	Thighs	60,6	24,8	8,6	4,8
F	Legs	61,0	26,9	10,6	3,2
G	Feet	60,6	19,8	3,5	4,0

Table 9 Convective heat transfer coefficient, θ_w =16,7°C, θ_a =16,7°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m$^2\cdot$°C)
A	Head	70,9	34,0	17,3	3,3
B	Trunk	70,9	27,2	10,5	4,4
C	Arms	70,3	28,0	11,3	4,8
E	Thighs	70,5	27,7	10,9	5,3
F	Legs	70,3	31,0	14,3	3,1
G	Feet	70,7	21,5	4,8	4,0

Table 10 Convective heat transfer coefficient, θ_w =17,0°C, θ_a =17,2°C

Body segment		Volt.(V)	θ_s (°C)	$\Delta\theta_{s\text{-}a}$ (°C)	α_c (W/m$^2\cdot$°C)
A	Head	80,3	40,1	22,9	2,9
B	Trunk	80,2	30,0	12,8	4,7
C	Arms	80,7	31,3	14,0	5,2
E	Thighs	80,6	30,7	13,5	5,8
F	Legs	80,4	35,3	18,1	3,2
G	Feet	80,1	23,4	6,2	3,8

Table 11 Convective heat transfer coefficient, θ_w =17,6°C, θ_a =17,9°C

Body	segment	Volt.(V)	θ_s (°C)	$\Delta\theta_{s-a}$ (°C)	α_c (W/m^2·°C)
A	Head	90,3	45,7	27,7	3,1
B	Trunk	90,5	33,3	15,3	5,2
C	Arms	90,5	34,2	16,3	6,0
E	Thighs	90,9	34,3	16,3	6,2
F	Legs	90,4	39,5	21,5	3,5
G	Feet	90,2	25,6	7,6	3,9

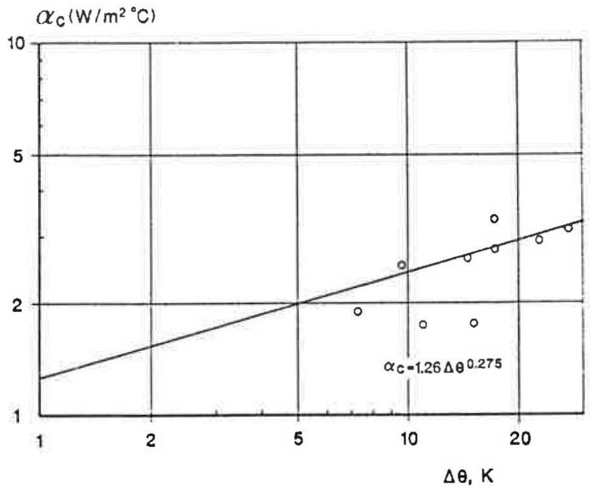

Fig. 11. Convective heat transfer coefficient for head.

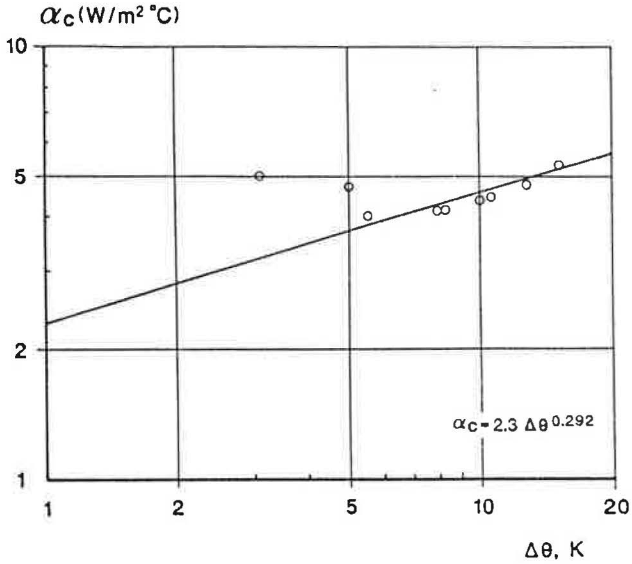

Fig. 12. Convective heat transfer coefficient for trunk.

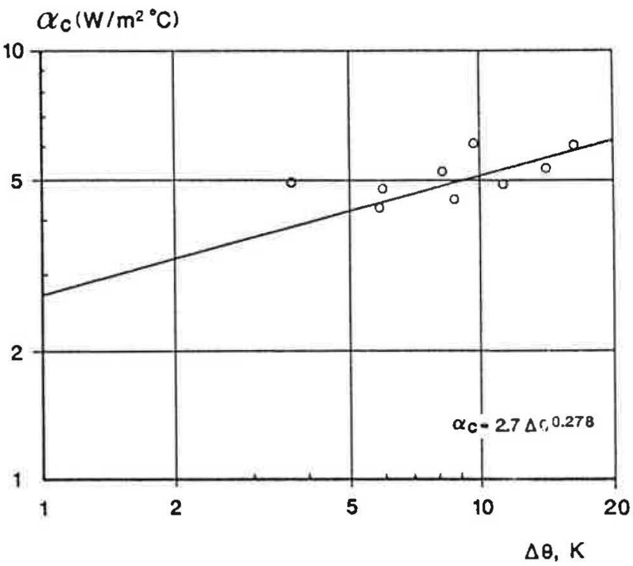

Fig. 13. Convective heat transfer coefficient for arms.

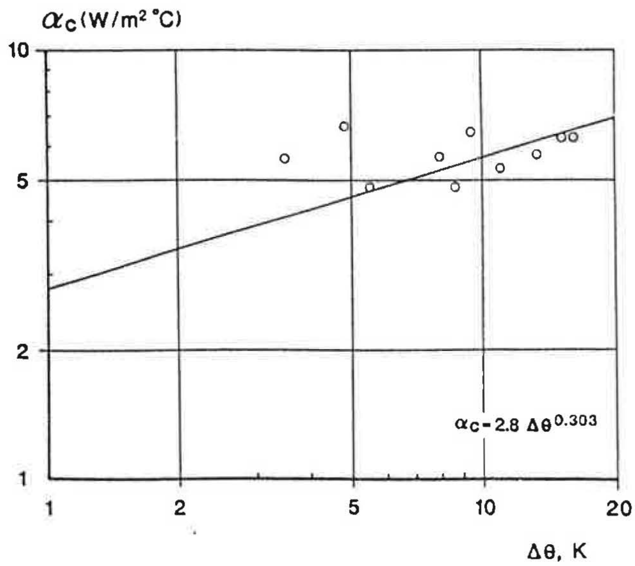

Fig. 14. Convective heat transfer coefficient for thighs.

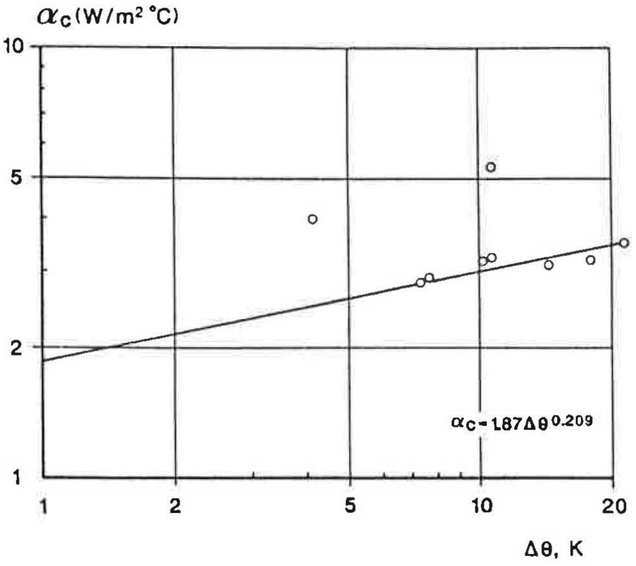

Fig. 16. Convective heat transfer coefficient for feet.

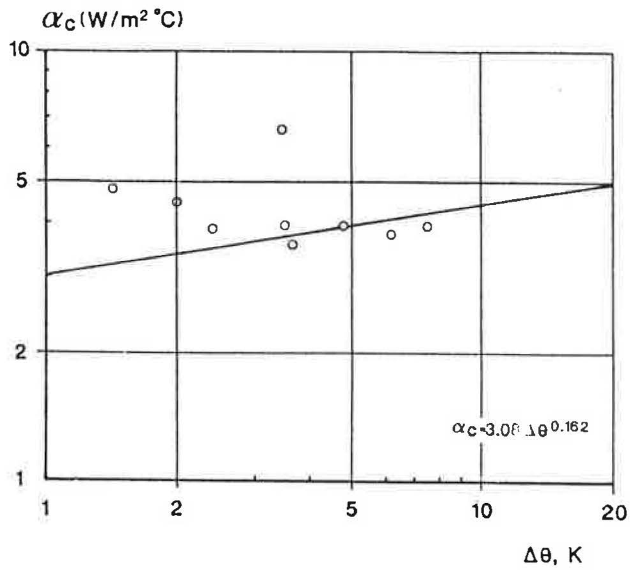

Fig. 15. Convective heat transfer coefficient for legs.

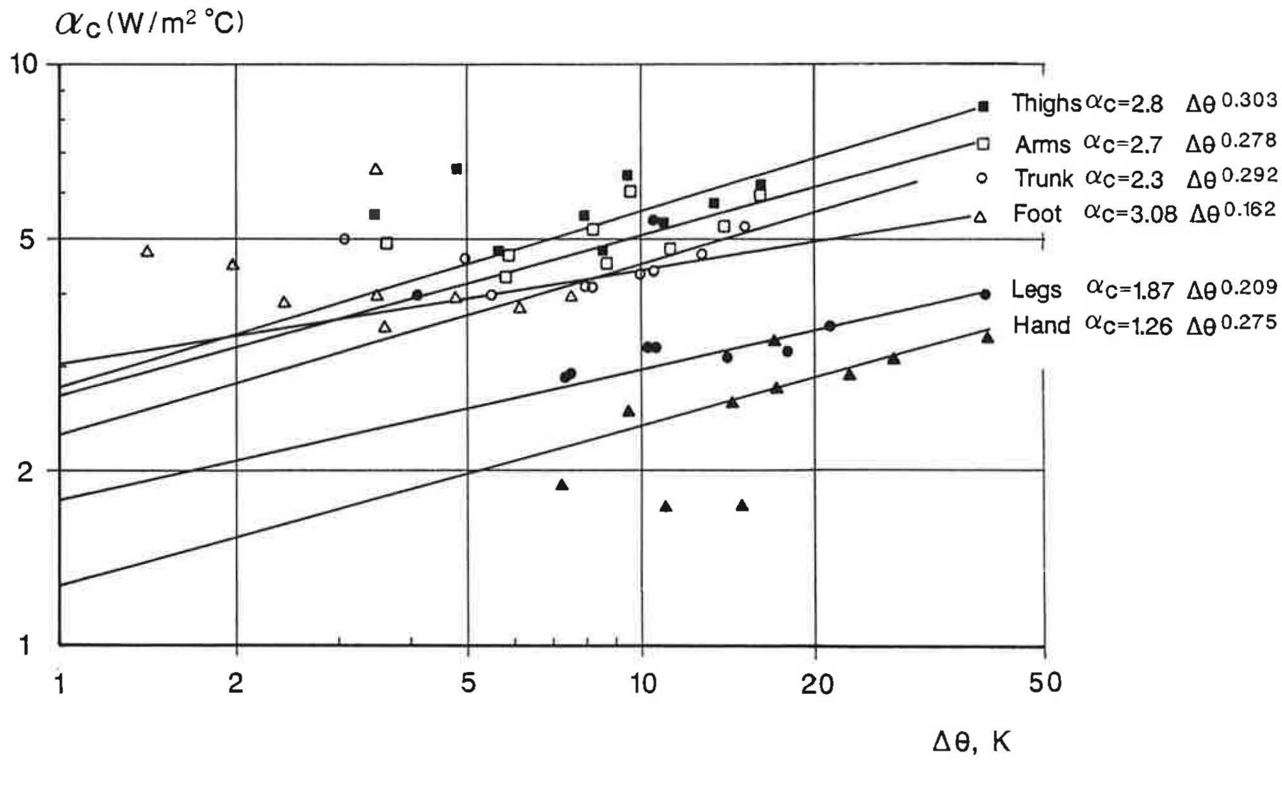

Fig. 17. Comparison of convective heat transfer coefficients for different body segments.

ARROR ANALYSIS

Basic equation (1) is

$$Q = Q_c + Q_R$$

where $Q_c = \alpha_c A \Delta\Theta_{s-a}$

$Q_R = \sigma\varepsilon A_r (T_s^4 - T_w^4)$

From equation (5) we obtain

$$\alpha_c = Q_c / (A \cdot \Delta\Theta_{s-a}) \quad (11)$$

So the relative error is

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = \left|\frac{dQ_c}{Q_c}\right| + \left|\frac{dA}{A}\right| + \left|\frac{d\Delta\Theta_{s-a}}{\Delta\Theta_{s-a}}\right| \quad (12)$$

From equation (2) results

$$dQ_R = \sigma[A_r(T_s^4 - T_w^4)d\varepsilon + \varepsilon(T_s^4 - T_w^4)dA_r + \varepsilon A_r(4T_s^3 dT_s - 4T_w^3 dT_w)]$$

$$\frac{dQ_R}{Q_R} = \frac{d\varepsilon}{\varepsilon} + \frac{dA_r}{A_r} + 4\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4} \quad (13)$$

From equation (1) is derived

$$dQ = dQ_c + dQ_R$$

$$\therefore \frac{dQ_c}{Q_c} = \frac{dQ}{Q_c} - \frac{dQ_R}{Q_c} = \frac{Q}{Q_c}\frac{dQ}{Q} - \frac{Q_R}{Q_c}\frac{dQ_R}{Q_R} \quad (14)$$

$$\left|\frac{dQ_c}{Q_c}\right| = \left|\frac{Q}{Q_c}\right| \cdot \left|\frac{dQ}{Q}\right| + \left|\frac{Q_R}{Q_c}\right| \cdot \left|\frac{dQ_R}{Q_R}\right| \quad (15)$$

From equations (12),(13),(14) we gain

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = \left|\frac{Q}{Q_c}\right| \cdot \left|\frac{dQ}{Q}\right| + \left|\frac{dA}{A}\right| + \left|\frac{d\Delta\Theta_{s-a}}{\Delta\Theta_{s-a}}\right| + \left|\frac{Q_R}{Q_c}\right| \cdot \left[\left|\frac{d\varepsilon}{\varepsilon}\right| + \left|\frac{dA_r}{A_r}\right| + 4 \cdot \left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right|\right] \quad (16)$$

Because

$$Q = \frac{U_{eff}^2}{R}$$

The equation (7) becomes:

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = 2 \cdot \left|\frac{Q}{Q_c}\right| \cdot \left|\frac{dU_{eff}}{U_{eff}}\right| + 2\left|\frac{Q}{Q_c}\right| \cdot \left|\frac{dR}{R}\right| + \left|\frac{d\Delta\Theta_{s-a}}{\Delta\Theta_{s-a}}\right| + \left|\frac{Q_R}{Q_c}\right| \cdot \left[\left|\frac{d\varepsilon}{\varepsilon}\right| + \left|\frac{dA_r}{A_r}\right| + 4\left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right|\right] + \left|\frac{dA}{A}\right| \quad (17)$$

EXPERIMENTAL EVALUATION

Assuming $Q \doteq 2Q_c \doteq 2Q_R$ doesn´t create larger error for error analysis. The equation (17) becomes.

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = 4\left|\frac{dU_{eff}}{U_{eff}}\right| + 4\left|\frac{dR}{R}\right| + \left|\frac{dA}{A}\right| + \left|\frac{d\Delta\phi_{s-a}}{\Delta\phi_{s-a}}\right| + \left|\frac{d\varepsilon}{\varepsilon}\right| + \left|\frac{dA_r}{A_r}\right| + 4\left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right| \quad (18)$$

For items $\left|\frac{dA}{A}\right|$, $\left|\frac{dA_r}{A_r}\right|$, these will create systematic errors. Because areas are remeasured carefully, $\left|\frac{dA}{A}\right|$, $\left|\frac{dA_r}{A_r}\right|$ can be neglected.

For item $\left|\frac{d\varepsilon}{\varepsilon}\right|$, ε is theoretical value, the $\left|\frac{d\varepsilon}{\varepsilon}\right|$ item can be neglected. The equation (18) can be simplified below:

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = 4\left|\frac{dU_{eff}}{U_{eff}}\right| + 4\left|\frac{dR}{R}\right| + \left|\frac{d\Delta\Theta_{s-a}}{\Delta\Theta_{s-a}}\right| + 4\left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right| \quad (19)$$

For U_{eff} value we use $\left|\frac{dU_{eff}}{U_{eff}}\right| < 0{,}1/40 = 0{,}3\%$

For R value we use $\left|\frac{dR}{R}\right| < 0{,}1/161{,}2 = 0{,}1\%$

So these two items can be neglected, equation (19) becomes:

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = \left|\frac{d\Delta\Theta_{s-a}}{\Delta\Theta_{s-a}}\right| + 4\left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right| \quad (20)$$

The temperature measurement itself can not create large errors. The problem is whether the thermocouple measurement point can reflect the real mean skin temperature. The following efforts are made.

Infrared thermograply

See Fig. 18.

Fig. 18 IR-photo for the thermal manikin when surrounding temperature is equal to 22°C, skin temperature

The skin temperature distribution for every segment can be seen. In each case the temperature distribution is reasonable.

Mean skin temperature

In order to know whether the thermocouple measurement value expresses the real mean skin temperature. A C-700 ThermoFlow Thermometer was used to measure many points for every segmer+ See Table 12.

Table 12. Numbers of measurement points using C-700 ThermoFlow Thermometer

Body segment		Numbers of points
A	Head	15
B	Trunk	36
C	Arms	30
E	Thighs	24
F	Legs	24
G	Feet	12

To determine the error created by different instruments and methods, the temperature at the same position was measured using two kinds of instruments. For the whole measurements, the results are given below, see table 13.

Table 13 Comparison with different methods

Body segment		Thermocouple	C-700 (1)	C-700(2)	errors
A	Head	34,1	31,6	31,0	+0,6
B	Trunk	27,0	26,8	26,4	+0,4
C	Arms	27,5	28,2	27,2	+1,0
E	Thighs	27,2	27,8	28,2	+0,6
F	Legs	29,9	28,7	27,7	-0,02
G	Feet	22,2	21,7	22,6	-0,9

Here
C-700(1) is using C-700 to measure the temperatures at the same positions where the thermocouples are located.
C-700(2) is that using C-700 to measure temperatures on many points to get the mean values
And Error=Results(C-700(1))-Results(C-700(2))

From table 13 it is known that values of thermocouples measuring roughly reflect the real mean skin values. Look back to the equation (20).

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = \left|\frac{d\Delta\Theta_{s-a}}{\Delta\Theta_{s-a}}\right| + 4\left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right|$$

In the range of $15°C < \Theta_s < 37°C$, and $15°C < \Theta_w < 40°C$, the item

$$4\left|\frac{T_s^3 dT_s - T_w^3 dT_w}{T_s^4 - T_w^4}\right|$$

is less than 1%.

Actually equation (20) is

$$\left|\frac{d\alpha_c}{\alpha_c}\right| = \left|\frac{d\Delta\Theta_{s-a}}{d\Theta_{s-a}}\right| \qquad (21)$$

The larger the temperature difference, the smaller is the error. When the temperature difference is 10°C, the maximum possible error is about 10%.

CONCLUSIONS AND DISCUSSION

For every segment, the correlation equation is given below:

Head	$\alpha_c = 1{,}26\ \Delta\Theta_{s-a}^{0{,}275}$
Trunk	$\alpha_c = 2{,}30\ \Delta\Theta_{s-a}^{0{,}292}$
Arms	$\alpha_c = 2{,}70\ \Delta\Theta_{s-a}^{0{,}278}$
Thighs	$\alpha_c = 2{,}80\ \Delta\Theta_{s-a}^{0{,}303}$
Legs	$\alpha_c = 1{,}87\ \Delta\Theta_{s-a}^{0{,}209}$
Feet	$\alpha_c = 3{,}08\ \Delta\Theta_{s-a}^{0{,}162}$

In order to compare and analyse the convective coefficient for every segment, all results were entered on the same diagram Fig. 17.

Because the seal is not good for thighs, the convective coefficient in this part seems larger.

Fig. 17 illustrates that the convective coefficients are almost identical for arms, trunk and feet. The value for legs is a little lower, and for the head is the lowest.

These phenomena can be explained using the heat transfer theory.

Considering the posture of thermal manikin, see Fig. 8, the type of air flow are approximately given below for arm and foot, Fig. 19

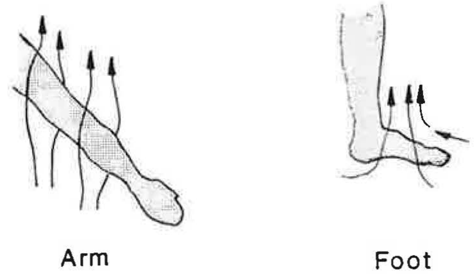

Fig. 19. Air flow pattern for arm and foot

These two parts are at the beginning of boundary layer, so the convective coefficents are bigger than the other parts of the body.

For the trunk the result is bigger than other investigators, such as Nishi et al (1970), Hommer (1988) and so on. If we consider the trunk as a simple geometrical shape, it can be illustrated as follows.

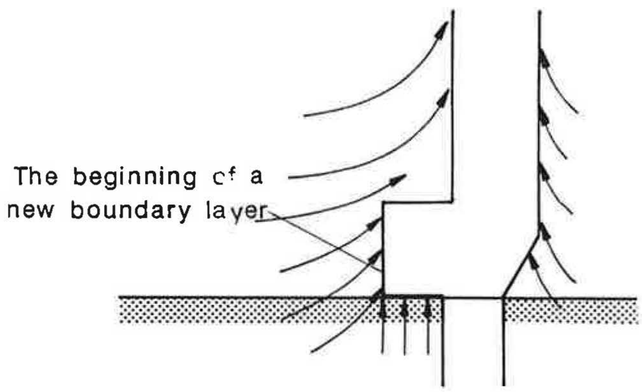

Fig. 20 Air flow type for trunk

There is another new boundary layer to be established for trunk. Although the convective coefficient for upper trunk is lower, the value for lower trunk is higher. Therefore the average convective coefficient is higher.

Usually the convective coefficient for the trunk is for the upper trunk, such as in Nishi et al (1970). In another papers, for instance Homma (1988), a vertical plares or cylinders are used to model the trunk. Because there is no special convex surface, no new boundary layer is established. The value of such a kind of trunk is lower.

Regarding legs, because they are not at the beginning section of a boundary layer, the convective coefficient is lower. However, they are near the feet (the beginning section of boundary layer), and thus have a higher value than the head.

The head is at the top of the body and no new boundary layer is created, so the convective coefficient should be lowest.

For a comparison of results from this study with those of earlier univestigators, see table 14.

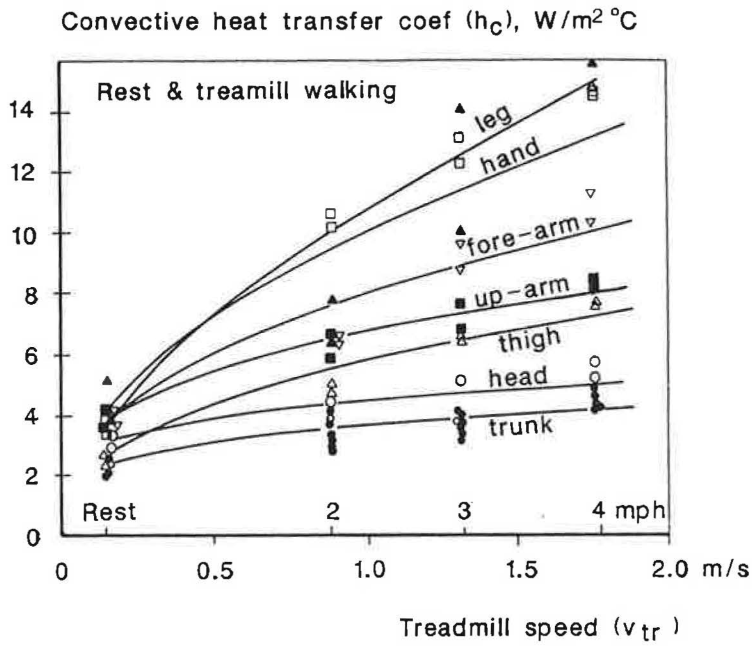

Fig. 21 Regional convective heat transfer coefficients for rest and various treadmill walking speeds. Observed values for head (O), trunk (●), upper arms (■), forearms (∇), hands (▲), thighs (△) and legs (□) are indicated.

Table 14 Comparison of the convective heat transfer coefficients measured by different investigators ($v<0{,}15m/s$)

Investigator	Condition	α_c (W/m$^2\cdot$°C)	Remarks
Nelson et al	Standing	4,8	
Woodcock et al	Standing	4,3	
Neilsen et al	Standing manikin	3,1	$\Delta\theta_{s\text{-}a}$ =5°C
Goldmann	Standing manikin	3,2	
Nishi et al	Standing manikin	3,2	

The results of this paper are in the range of table 14. See Fig.21 for the convective coefficient when people during rest.

In fact, the method used by Nishi et al (1970) can only measure the forced convective coefficient. When the air velocity is very small, the free convective dominates the whole heat transfer coefficient. For the method using plates and cylinders, such as Homma et al, see Fig. 22.

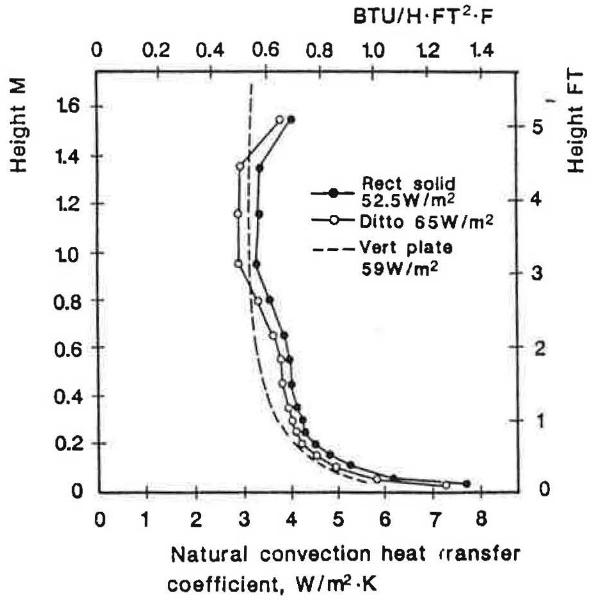

Fig. 22 Local natural convection heat transfer coefficient distributions on heated rectangular manikin placed on floor and on heated vertical plate hung in space.

This kind of manikin will create a different flow field, it is different from the real human body and a full-scale manikin. Consequently, the results generated are not the same.

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