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Thermal Sensation Analysis and its Application to Air-Conditioning

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and
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Abstract

For half a century or more, many attempts have been made to establish thermal conditions based on which optimal promotion of mental and physical state to a task may be produced. However, adequate scales of warmth have not yet been proposed inasmuch as the composition of thermal environment has been too complex to be specified simply with a few thermal factors.

Reviewing the earlier works on the subject, some of them were conducted from a medical point of view while others were carried out from an engineering angle; the former, being empirical scales based on experiments, may not be applicable under unusual conditions, and the latter, theoretical scales based on thermal equilibrium, have its drawback in physical accuracy.

This paper is concerned with the development of a rational comfort index and its application to heating and air-conditioning.

Part I deals with the fundamentals of heat and mass transfer between man and environment, and a rational shape factor for radiation will be induced.

In Part II, the heat balance equations between man and environment are established. In the calculation of the heat exchanges, man must be treated as a whole and mean values of each heat transfer coefficient and of each thermal factor must be used. For such simplification of the situation the physiological properties are consulted. Further, into the experiment each of these approximations is verified.

Part III deals with a rational comfort index and its application. This index evaluates each component of the thermal environment, such as ambient temperature, radiant temperature, humidity, air-movement, heat and vapor resistance and emissivity of clothing, work rate, and physical properties of air.

Content

Part I. Fundamentals of Heat Transfer

Chapter 1. Radiant Heat Transfer 75
1-1. Surrounding Radiant Temperature 75
1-2. Shape Factor between Sphere Element and Rectangular Wall . . 76
1-3. Shape Factor between Cylinder Element and Rectangular Wall . 77

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	1–4.	Comparison of Shape Factors	80
	1–5.	Summary	81
Chapter	2.	Convective Heat Transfer	81
	2–1.	Physical Properties of Air	81
	2–2.	Convective Heat Transfer	82
	2–3.	Convective Heat Transfer Coefficient for Cylinder and Sphere	82
	2–4.	Combined Forced and Natural Convection	83
	2–5.	Effect of Atmospheric Pressure to Convective Heat Transfer Coefficient	83
	2–6.	Summary	83
Chapter	3.	Evaporative Heat Transfer	84
	3–1.	Analogy between Mass and Heat Transfer	84
	3–2.	Mass Transfer Coefficient	85
	3–3.	Effect of Atmospheric Pressure on Evaporative Heat Transfer Coefficient	85
	3–4.	Summary	86
Chapter	4.	Experimental Verification of the Analogy	86
	4–1.	Procedure	86
	4–2.	Results of Measurement	87
	4–3.	Verification of the Analogy	88
	4–4.	Convective Heat and Mass Transfer Coefficient for Short Cylinder	88
	4–5.	Summary	89
		Part II. Heat Equilibrium on Human Body	
Chapter	5.	Radiative and Convective Heat Interchange	89
	5–1.	Radiant Heat Transfer Coefficient	89
	5–2.	Surrounding Radiant Temperature and Mean Radiant Temperature	89
	5–3.	Convective Heat Transfer Coefficient	90
	5–4.	Effect of Clothing	90
	5–5.	Summary	90
Chapter	6.	Evaporative Heat Loss	90
	6–1.	Vapor Pressure over Skin Surface	90
	6–2.	Simplification of Saturated Vapor Pressure	91
	6–3.	Vapor Resistance of Clothing	91
	6–4.	Summary	91
Chapter	7.	Thermoregulatory Mechanism of Man	92
	7–1.	Body Heat Production	92
	7–2.	Body Heat Loss	92
	7–3.	Insensible Perspiration	93
	7–4.	Sweat Secretion	93
	7–5.	Effect of Clothing on Sweat Evaporation	93
	7–6.	Heat Balance Equation of Man	95
	7–7.	Summary	97
Chapter	8.	Experimental Verification of Heat Balance Equation	97
	8–1.	Procedure	97
	8–2.	Experiment-A	98
	8–3.	Experiment-B	99

8-4. Experiment-C 99
8-5. Summary 100

Part III. Development of Warmth Diagram and its Application to Air-Conditioning

Chapter 9. Thermal Sensation and Model Skin Temperature 100
9-1. Definition of Comfort 100
9-2. Expression of Thermal Sensation 101
9-3. Relation between Thermal Vote and Model Skin Temperature . 101
9-4. Temperature-Humidity Chart and Model Skin Temperature . . 101
9-5. Model Skin Temperature as a Scale of Warmth 103
9-6. Summary 104
Chapter 10. Warmth Diagram and its Application 104
10-1. Principle of Warmth Diagram 104
10-2. Graphical Solution of Model Skin Temperature 105
10-3. Warmth Diagram 106
10-4. Evaluation of Thermal Environment with Warmth Diagram . . 114
10-5. Summary 115
Chapter 11. Comfort Detector 116
11-1. Model Man 116
11-2. Thermal Sensation Computer 116
11-3. Summary 117

Appendix

A-1. Discomfort Index 118
A-2. Corrected Effective Temperature 119
A-3. Heat Stress Index 120

Part I. Fundamentals of Heat Transfer

The fundamentals of heat transfer, which form the basis for expressing the heat balance equation between man and environment, are summarized.

Chapter 1. Radiant Heat Transfer

1-1. Surrounding Radiant Temperature

When a body 0 is enclosed with several walls $(1, 2, 3, \cdots, n)$, the radiant heat, emitted from a body 0 and absorbed by wall 1, is given by

$$Q_0 = \varepsilon_w \cdot \varepsilon_0 \cdot \frac{\sigma}{100^4} \cdot S_0 \cdot \varphi_{01} \cdot (T_0 + 273)^4 \qquad (1\text{-}1)$$

In an analogous manner we obtain

$$Q_1 = \varepsilon_0 \cdot \varepsilon_w \cdot \frac{\sigma}{100^4} \cdot S_1 \cdot \varphi_{10} \cdot (T_1 + 273)^4 \qquad (1\text{-}2)$$

where $\varepsilon_0, \varepsilon_w$ = emissivity of a body or wall
S_0, S_1 = surface area of a body or wall
φ_{01} = shape factor for radiation between a body and wall
T_0 = surface temperature of a body

Following Nusselt[1], it is possible in the first approximation to neglect the reflection from either surface. Then, with reciprocity theorem; $\varphi_{01} \cdot S_0 = \varphi_{10} \cdot S_1$.

The quantity of heat exchange per unit time between 0 and 1 is given by

$$Q_{01} = \varepsilon_0 \cdot \varepsilon_w \cdot \sigma \cdot k_{01} \cdot (T_0 - T_1) \cdot S_0 \cdot \varphi_{01} \qquad (1\text{-}3)$$

where $k_{01} = 100^{-4} \cdot \{(T_0+273)^2 + (T_1+273)^2\}\{(T_0+273)+(T_1+273)\}$

Then, we get

$$Q_{01} = \alpha_{r_1} \cdot (\varphi_{01} \cdot T_0 - \varphi_{01} \cdot T_1) \cdot S_0 \qquad (1\text{-}4)$$

where $\alpha_{r_1} = \varepsilon_w \cdot \varepsilon_0 \cdot \sigma \cdot k_{01}$ = radiant heat transfer coefficient

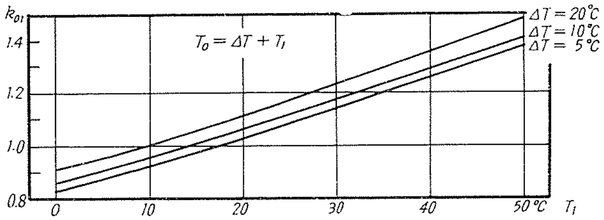

Fig. 1-1. Factor k_{01} and solid body temperature

From Fig. 1-1 we know that factor k_{01} can be treated as a constant value in common use. Thus, the quantity of heat exchange per unit time between body 0 and solid walls may be written by

$$Q_{01} + Q_{02} + \cdots + Q_{0n} = \alpha_r \cdot (T_0 - \sum_1^n \varphi_{0n} \cdot T_n) \cdot S_0 \qquad (1\text{-}5)$$

where $\alpha_{r1} \doteqdot \alpha_{r2} \doteqdot \cdots \doteqdot \alpha_{rn} \equiv \alpha_r$

Here, the surrounding radiant temperature is defined by

$$T_r = \sum_1^n \varphi_{0n} \cdot T_n \qquad (1\text{-}6)$$

And we get simply

$$H_r = \alpha_r \cdot (T_0 - T_r) \qquad (1\text{-}7)$$

where H_r: quantity of radiant heat flow between body 0 and its enclosure, per unit time per unit area

1-2. Shape Factor between Sphere Element and Rectangular Wall

Generally, it is impossible to determine the shape factor between arbitrary surface elements.

The shape factor for the geometrical arrangement, shown in Fig. 1-2, is given by the equation

$$\varphi = \frac{1}{4\pi} \tan^{-1} \frac{(x/d) \cdot (y/d)}{\sqrt{1 + (x/d)^2 + (y/d)^2}} \qquad (1\text{-}8)$$

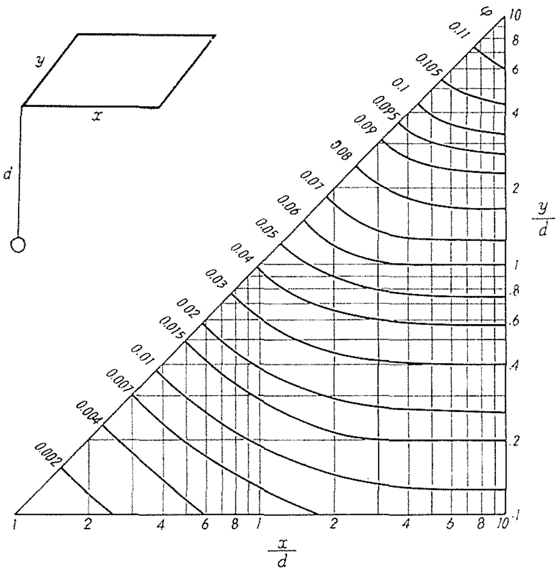

Fig. 1-2. Shape factor between sphere element and rectangular wall

1-3. Shape Factor between Cylinder Element and Rectangle

It is only an expedient to regard man as a sphere element. The authors try to originate a new shape factor, considering man to be a cylinder element.

As shown in Fig. 1-3, turning the small area element 1 around its vertical axis, it may compose the cylinder element. Fig. 1-3 denote the geometrical relation.

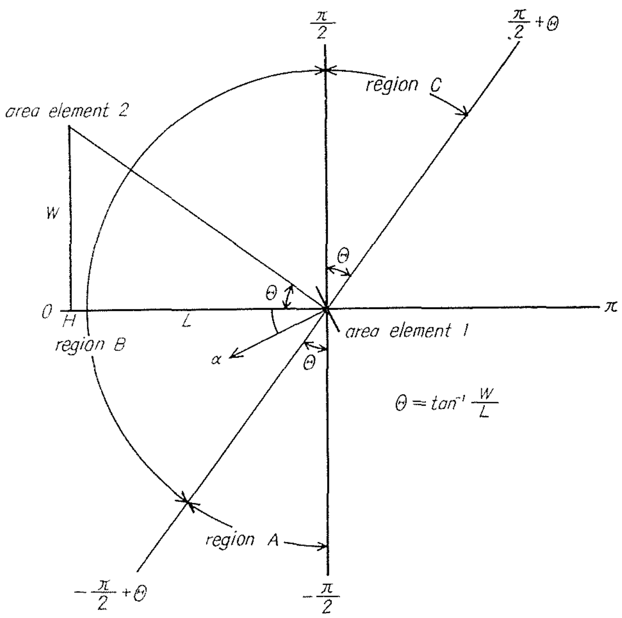

Fig. 1-3. Elevation view of area element 1 and rectangular area element 2

Let element 1 revolve from angular co-ordinate $(-\pi/2)$

(1) **Region A** The shape factor may be written by

$$(\dot{\varphi}_{12})_1 = \frac{1}{2\pi}\left[\cos\alpha\cdot\left\{\frac{H}{\sqrt{H^2+L^2}}\cdot\tan^{-1}\frac{(-L)\cdot\cot\alpha}{\sqrt{H^2+L^2}}\right\} - \tan^{-1}\left(\frac{H}{L}\sin\alpha\right)+\sin\alpha\cdot\tan^{-1}\frac{H}{L}\right] \tag{1-9}$$

(2) **Region B** In the interval $\left(-\frac{\pi}{2}+\Theta\sim\frac{\pi}{2}\right)$ all of the radiant energy leaving element 1 will arrive at the wall 2. The shape factor is given by

$$(\dot{\varphi}_{12})_2 = \frac{1}{2\pi}\left[\cos\alpha\cdot\left\{\frac{H}{\sqrt{H^2+L^2}}\cdot\tan^{-1}\frac{W}{\sqrt{H^2+L^2}}+\frac{W}{\sqrt{L^2+W^2}}\cdot\tan^{-1}\frac{H}{\sqrt{L^2+W^2}}\right\} +\sin\alpha\cdot\left\{\tan^{-1}\frac{H}{L}-\frac{L}{\sqrt{L^2+W^2}}\cdot\tan^{-1}\frac{H}{\sqrt{L^2+W^2}}\right\}\right] \tag{1-10}$$

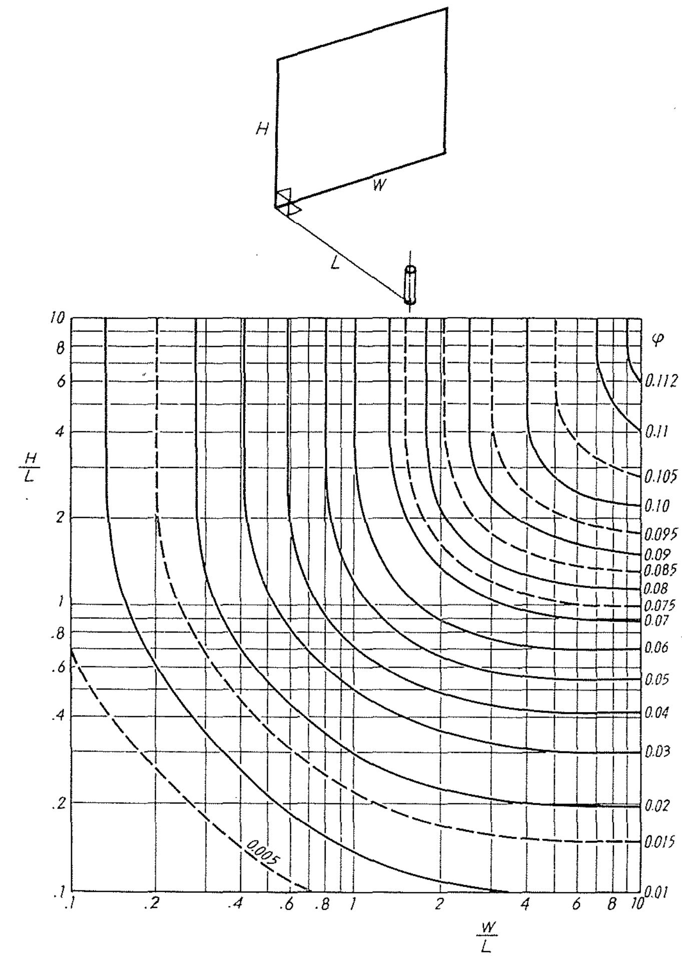

Fig. 1-4. Shape factor between cylinder element and vertical rectangle

(3) **Region C** The shape factor can be written by

$$(\dot{\varphi}_{12})_3=\frac{1}{2\pi}\left[\cos\alpha\cdot\left\{\frac{H}{\sqrt{H^2+L^2}}\cdot\tan^{-1}\frac{W}{\sqrt{H^2+L^2}}\right.\right.$$

$$+\frac{W}{\sqrt{L^2+W^2}}\cdot\tan^{-1}\frac{H}{\sqrt{L^2+W^2}}+\frac{H}{\sqrt{H^2+L^2}}\cdot\tan^{-1}\left.\frac{L\cdot\cot\alpha}{\sqrt{H^2+L^2}}\right\}$$

$$\left.-\sin\alpha\cdot\frac{L}{\sqrt{L^2+W^2}}\cdot\tan^{-1}\frac{H}{\sqrt{L^2+W^2}}+\tan^{-1}\left(\frac{H}{L}\sin\alpha\right)\right] \quad (1\text{-}11)$$

By integrating $(\dot{\varphi}_{12})_1$, $(\dot{\varphi}_{12})_2$ and $(\dot{\varphi}_{13})_3$ over their defined regions and averaging by the angle at the circumference, we get the new shape factor by the equation

$$\varphi_{12}=\frac{1}{2\pi}\left[2\int_{\frac{\pi}{2}}^{\frac{\pi}{2}+\theta}\tan^{-1}(Q\cdot\sin\alpha)d\alpha\right.$$

$$+\frac{1}{2\sqrt{Q^2+1}}\cdot\ln\left[\frac{[P-\{Q\sqrt{Q^2+1}+Q^2+1\}\{\sqrt{P^2+1}+P\}]}{[P+\{Q\sqrt{Q^2+1}-(Q^2+1)\}\{\sqrt{P^2+1}+P\}]}\right.$$

$$\left.\left.\times\frac{[P-\{Q\sqrt{Q^2+1}-(Q^2+1)\}\{\sqrt{P^2+1}-P\}]}{[P+\{Q\sqrt{Q^2+1}+(Q^2+1)\}\{\sqrt{P^2+1}-P\}]}\right]\right] \quad (1\text{-}12)$$

where $P=\dfrac{W}{L}$, $Q=\dfrac{H}{L}$

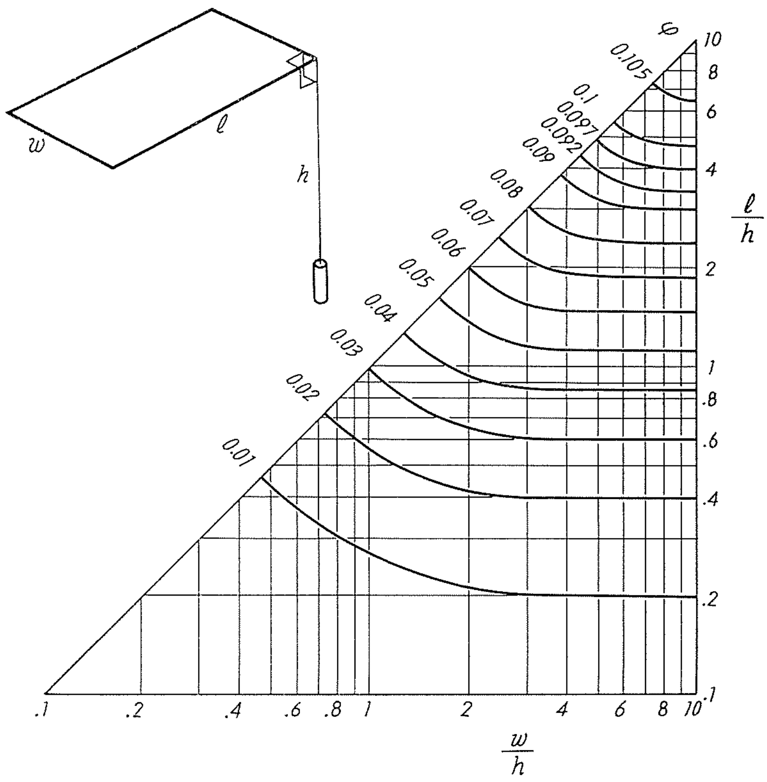

Fig. 1-5. Shape factor between cylinder element and horizontal rectangle

1-4. Comparison of Shape Factors

The characteristics of the shape factor for sphere element and cylinder element may be made clear by the following example.

Example 1-1. Determine the shape factor between sphere or cylinder element and rectangular wall (1, 2, 3, 4), shown in Fig. 1-6.

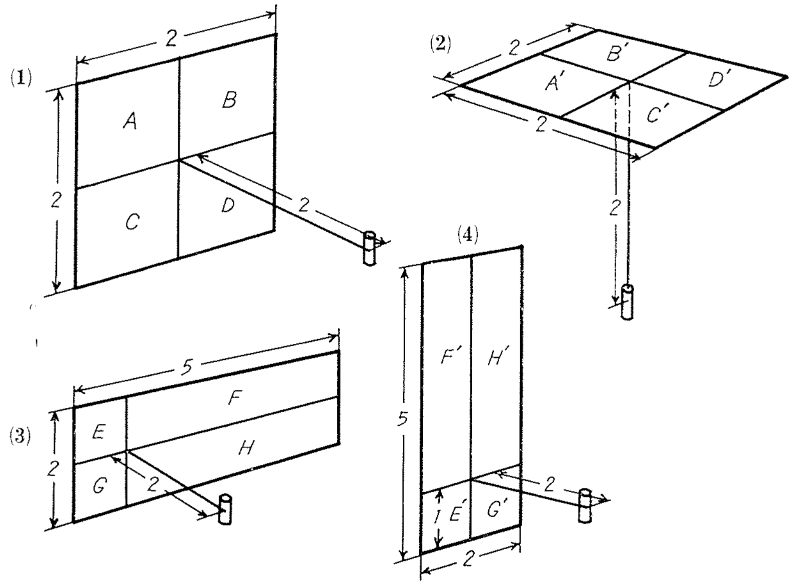

Fig. 1-6. Geometrical arrangement between a body and wall

Solution : Each of the shape factors is shown in Table 1-1.

The example makes clear the followings.

(1) The difference between each of the shape factors is apparent.

(2) If the wall area and the distance are the same, each of the shape factors between the cylinder element and the configurations, shown in (3), (4), is different. These relations give a clear basis for radiant heating in practice.

Table 1-1.

	shape factor for sphere	φ	shape factor for cylinder	φ
(1)	$\varphi_{1A}=0.016$ $\varphi=4\cdot\varphi_{1A}$	0.064	$\varphi_{1A}=0.019$ $\varphi=4\cdot\varphi_{1A}$	0.076
(2)	$\varphi_{1A'}=0.016$ $\varphi=4\cdot\varphi_{1A'}$	0.064	$\varphi_{1A'}=0.012$ $\varphi=4\cdot\varphi_{1A'}$	0.048
(3)	$\varphi_{1E}=0.016$, $\varphi_{1F}=0.033$ $\varphi=2\cdot\varphi_{1E}+2\cdot\varphi_{1F}$	0.098	$\varphi_{1E}=0.019$, $\varphi_{1F}=0.040$ $\varphi=2\cdot\varphi_{1E}+2\cdot\varphi_{1F}$	0.118
(4)	$\varphi_{1E'}=0.016$, $\varphi_{1F'}=0.033$ $\varphi=2\cdot\varphi_{1E'}+2\cdot\varphi_{1F'}$	0.098	$\varphi_{1E'}=0.019$, $\varphi_{1F'}=0.034$ $\varphi=2\cdot\varphi_{1E'}+2\cdot\varphi_{1F'}$	0.106

1-5. Summary

The problems on radiant heat transfer, especially shape factor, have been summarized and the authors have proposed a new shape factor between cylinder element and rectangular wall.

Reference

1) Gröber, H., Erk, S. and Grigull, U.: Fundamentals of Heat Transfer, (1961), p. 454, McGraw-Hill.

Chapter 2. Convective Heat Transfer

2-1. Physical Properties of Air

The influence of temperature, humidity and atmospheric pressure to some physical properties of air are summarized in Table 2-1.

Table 2-1. Influence of temperature, humidity and atmospheric pressure to physical properties of air

	air temp.	atm. pressure	humidity
density	evident	evident	little
viscosity	little	little	little
kinematic viscosity	Fig. 2-1	Fig. 2-1	little
specific heat	little	little	little
heat conductivity	a little	little	little
thermal diffusivity	Fig. 2-2	Fig. 2-2	little
mass diffusivity	Fig. 2-3	Fig. 2-3	negligible
Prandtl number	negligible	negligible	negligible
Schmidt number	negligible	negligible	negligible

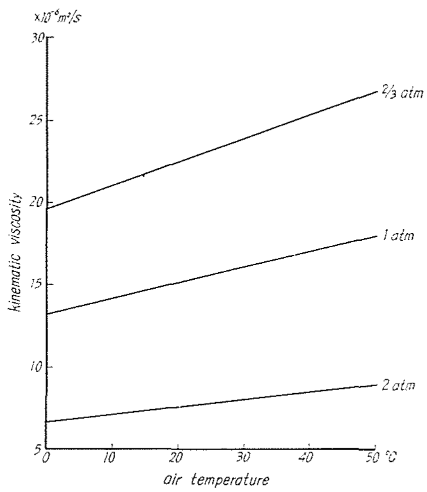

Fig. 2-1. Kinematic viscosity of air

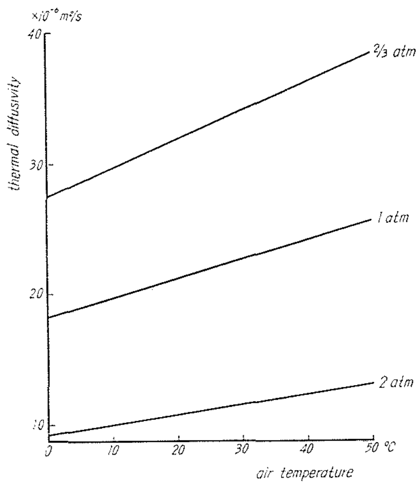

Fig. 2-2. Thermal diffusivity of air

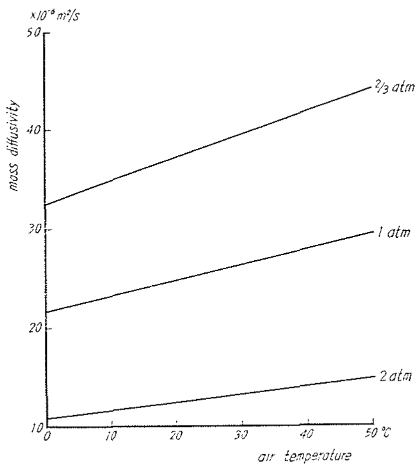

Fig. 2-3. Mass diffusivity of water vapor into air

It is known that density, kinematic viscosity, thermal diffusivity and mass diffusivity are affected by atmospheric pressure, but little by humidity.

2-2. Convective Heat Transfer

Natural convection is caused by the density difference of the air between heated surfaces and fluid. Otherwise forced convection is caused on the air flow. The rate of the heat transfer by convection between a solid boundary and fluid may be evaluated by means of the equation

$$H_c = \alpha_c \cdot (T_c - T_a) \tag{2-1}$$

where H_c = quantity of heat transfered by convection per unit time per unit area
α_c = convective heat transfer coefficient
T_c = temperature of solid boundary
T_a = fluid temperature

The above equation is a definition of the convective heat transfer coefficient α_c.

The convective heat transfer coefficient is a complicated function of the fluid flow, the thermal properties of the fluid medium and the geometry of the system.

2-3. Convective Heat Transfer Coefficient for Cylinder and Sphere

Hilpelt summarized the perimeter-mean heat transfer coefficients for the flow of air normal to a single heated cylinder. These relations are correlated by equations of the form[2)]

$$Nu = C_f \cdot Pr^n \cdot Re^m \tag{2-2}$$

where Nu = Nusselt number
Pr = Prandtl number
Re = Reynolds number

Table 2-2.
Value of "m" and "$C_f \cdot Pr^n$" for Eq. (2-2)

Re	m	$C_f \cdot Pr^n$
1-4	0.330	0.891
4-40	0.385	0.821
40-4000	0.466	0.615
4000-40000	0.618	0.174
40000-250000	0.805	0.0239

$C_f, m, n =$ shown in Table 2–2

Ranz & Marshall correlated the heat transfer coefficients of air flow over a sphere by the equation of the form[3)]

$$Nu = 2.0 + 0.6 \cdot Pr^{\frac{1}{3}} \cdot Re^{\frac{1}{2}} \qquad 1 < Re < 7 \times 10^4 \qquad (2\text{–}3)$$

2–4. Combined Forced and Natural Convection

In any heat transfer process density gradients occur and in the presence of a forced field natural convection currents arise. If the forced convection effect is very large, the influence of natural convection currents may be negligible. When the forced convection effect can be neglected the convective heat transfer coefficients are given by the equation of the form

$$Nu = f(Pr\,Gr) \qquad (2\text{–}4)$$

where $Gr =$ Granshof number

An analysis of the investigation of heat transfer over a small sphere by Yuge states that if the velocity of air flow exceeds 3 cm/s the effect of natural convection current over the sphere of 30 cm in diameter is neglected.[4)]

2–5. Effect of Atmospheric Pressure on Convective Heat Transfer Coefficient

As Eq. (2–2) involves some physical properties of air, at another atmospheric pressure region the convective heat transfer coefficient takes the other value, as shown in Fig. 2–4.

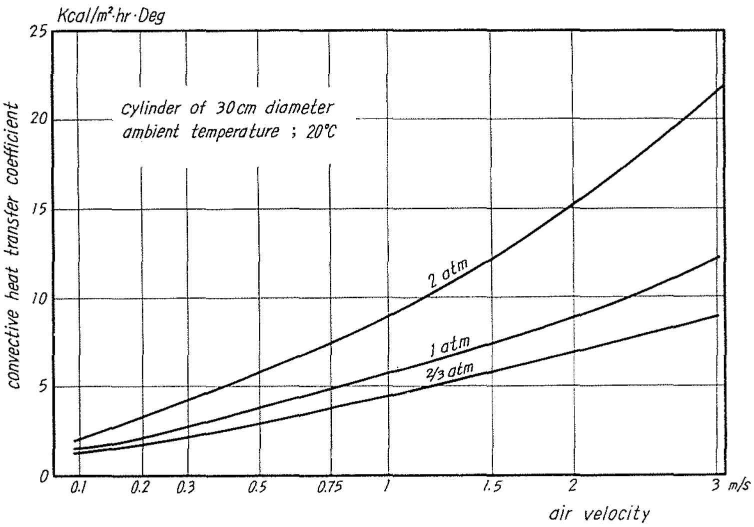

Fig. 2–4. Effect of atmospheric pressure to convective heat transfer coefficient

2–6. Summary

Estimating the rate of convective heat transfer over the human body, the followings must be considered.

The magnitude of the natural convection effects to forced convection may be neglected.

The fact that the atmospheric pressure affects the convective heat transfer coefficient suggests that the design of thermal environment in a pressurized cabin for medical care or in a low pressure space capsule must be differentiated from common design.

References

2) McAdams, W. H.: Heat Transmission, 3rd. ed., p. 260, McGraw-Hill.
3) Gröber, H., Erk, S. and Grigull, U.: Fundamentals of Heat Transfer, p. 412, McGraw-Hill.
4) Yuge, T.: Heat Transfer Experiment over Sphere, Annual Meeting of the Jap. Soc. of Mech. Eng. (1956 April).

Chapter 3. Evaporative Heat Transfer

3-1. Analogy between Mass and Heat Transfer

The equation of convective mass transfer of an incompressible fluid in steady flow, with constant fluid properties and in the absence of a pressure gradient and any external force, can be written in a form identical to the Fourier heat transfer equation

$$\frac{\partial C}{\partial t} = D \cdot \nabla^2 \cdot C \tag{3-1}$$

where C = concentration
t = time
D = mass diffusivity

While, the corresponding equation for convective heat transfer takes the form

$$\frac{\partial \theta}{\partial t} = a \cdot \nabla^2 \cdot \theta \tag{3-2}$$

where a = thermal diffusivity

Further, in steady state mass flux is given by

$$W = -D\frac{dC}{dx} \tag{3-3}$$

The equivalent expression for heat transfer is written by

$$q = -\lambda\frac{d\theta}{dx} = -a\frac{d(\rho \cdot i)}{dx} \tag{3-4}$$

Each of the above equations is similar.

The solutions of Eqs. (3-1) and (3-2) describe the field of concentration and temperature. It is immediately apparent that in the case where the mass diffusivity is equal to the thermal diffusivity, the two fields are identical to one another as long as the boundary conditions for the two equations are the same.

3-2. Mass Transfer Coefficient

The rate of mass transfer, as an equivalent expression of Eq. (2–1), is given by the equation

$$W = k_c \cdot (C_1 - C_2) \tag{3-5}$$

where k_c = mass transfer coefficient based on gas concentration

In a gaseous system the density of the diffusing substance can be replaced by

$$C = \frac{P}{R_v \cdot T} \tag{3-6}$$

Mass transfer coefficients must be evaluated experimentally, but direct experimental data is lacking. Since the mechanism of mass and heat transfer are closely related, one might expect data taken for heat transfer to be useful in predicting the rate of mass transfer.

Colburn j-factor is defined by the following equations

$$j_H = \frac{Nu}{Re \cdot Pr^{\frac{1}{3}}} \tag{3-7}$$

$$j_D = \frac{Sh}{Re \cdot Sc^{\frac{1}{3}}} \tag{3-8}$$

where Sh = Sherwood number
Sc = Schmidt number

The analogy among heat and mass transfer in forced convection systems may be stated by using Colburn j-factors.

$$j_H = j_D \tag{3-9}$$

Further, from Eqs. (2–2) and (3–9), one obtains for the mass transfer

$$Sh = C_f \cdot Sc^n \cdot Re^m \tag{3-10}$$

where the exponent n of the Prandtl or Schmidt number takes the value of 1/3 respectively.

As a result, convective mass transfer coefficient can be written as a function of convective heat transfer coefficient by

$$k_p = \frac{1.033 \times 10^4}{760 \cdot R_v \cdot T \cdot C_p \cdot \rho} \cdot \left(\frac{D}{a}\right)^{\frac{2}{3}} \cdot \alpha_c \quad [\mathrm{kg/m^2 \cdot hr \cdot mmHg}] \tag{3-11}$$

where k_p = convective mass transfer coefficient
C_p = specific heat
T = absolute temperature

3-3. Effect of Atmospheric Pressure on Evaporative Heat Transfer Coefficient

Multiplying the latent heat of evaporation L_h into the mass transfer coefficient, one obtains the evaporative heat transfer coefficient by the equation

$$\beta = L_h \cdot k_p = \kappa \cdot \alpha_c$$
$$[\text{kcal/m}^2\cdot\text{hr}\cdot\text{mmHg}] \quad (3\text{–}12)$$

In the above equation the physical properties, such as density, thermal diffusivity and mass diffusivity are affected by atmospheric pressure.

So, the evaporative heat transfer coefficient decreases at higher atmospheric pressure regions in contrast with the convective heat transfer coefficient.

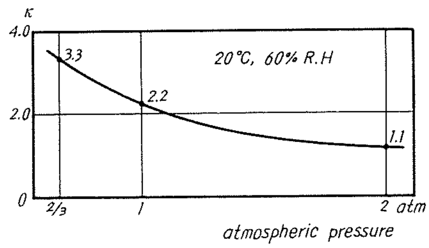

Fig. 3–1. Effect of atmospheric pressure to factor κ

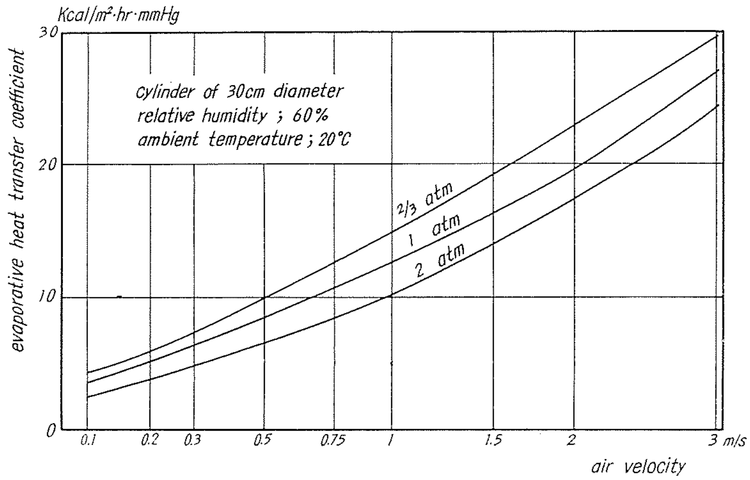

Fig. 3–2. Effect of atmospheric pressure to evaporative heat transfer coeffiicient

3–4. Summary

By drawing an analogy between mass and heat transfer, evaporative heat transfer coefficients can be predicted as the function of convective heat transfer coefficient.

Chapter 4. Experimental Verification of the Analogy

4–1. Procedure

Instead of evaporating liquids, sublimating solids such as naphthalene can be used in mass transfer measurements. The naphthalene was cast in molds that produced different sizes of cylindrical naphthalenes of 2 and 3 cm in diameter and 6 and 9 cm in height respectively. The temperature drop of solid surfaces with the latent heat of the sublimation is calculated to be almost 0.2°C and was neglected.

In the test room, ambient and wall temperatures were maintained at 35±0.5°C for the duration of the experiment.

Both ends of the naphthalene cylinders were either coated with a thin film or exposed.

4–2. Results of Measurement

Measured convective mass transfer coefficients are expressed in Figs. 4–1 and 4–2, and by the following dimensionless equations.

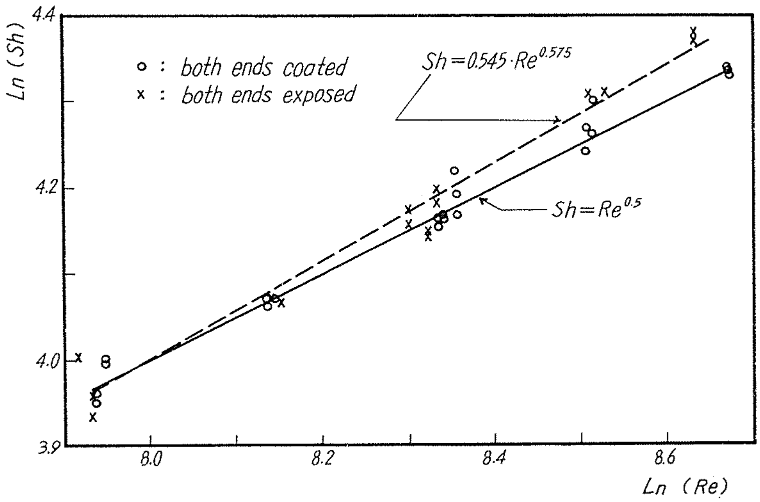

Fig. 4–1. Mass transfer coefficient for naphthalene cylinders in cross flow

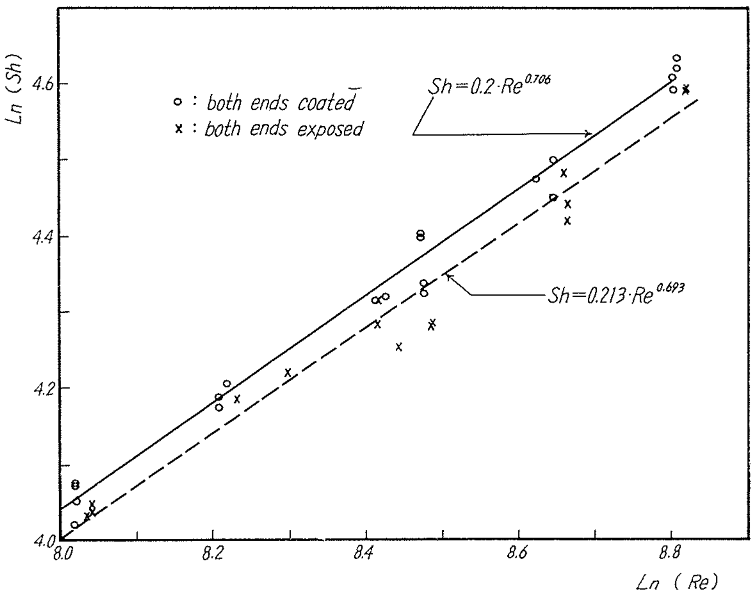

Fig. 4–2. Mass transfer coefficient for naphthalene cylinders in parallel flow

(1) **In cross flow** :

both ends coated

$$Sh = Re^{0.5} \tag{4–1}$$

both ends exposed

$$Sh = 0.545 \cdot Re^{0.575} \quad (4\text{-}2)$$

(2) **In parallel flow** :

both ends coated

$$Sh = 0.20 \cdot Re^{0.706} \quad (4\text{-}3)$$

both ends exposed

$$Sh = 0.213 \cdot Re^{0.693} \quad (4\text{-}4)$$

where $2700 < Re < 7000$

4-3. Verification of the Analogy

If the analogy can be applied, experimental data may correspond to the following equations, derived from Eq. (3-10).

$$Sh = 1.052 \cdot Re^{0.466} \qquad 40 < Re < 4000 \quad (4\text{-}5)$$

$$Sh = 0.298 \cdot Re^{0.618} \qquad 4000 < Re < 40000 \quad (4\text{-}6)$$

The above relation closely parallels the analogy between mass and heat transfer.

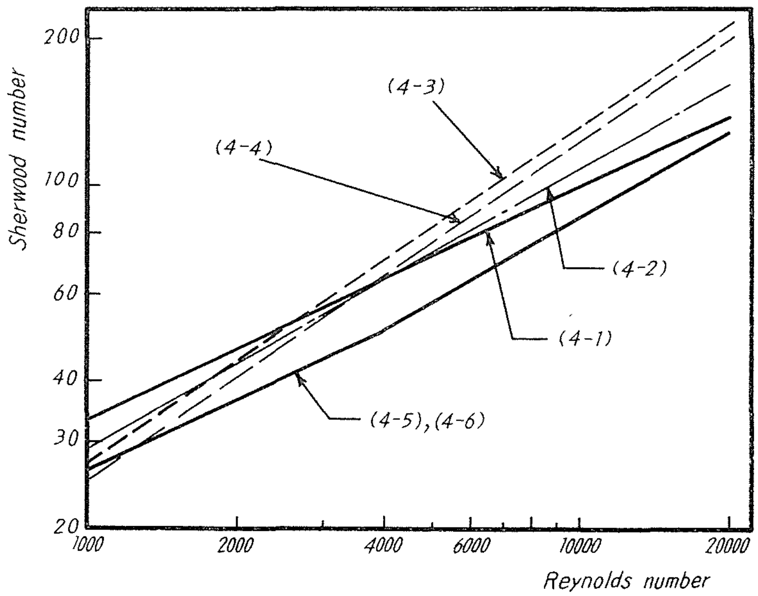

Fig. 4-3. Comparison between experimental data and analogous equations

4-4. Convective Heat and Mass Transfer Coefficient for Short Cylinder

Eq. (2-2) is for long cylinders in cross flow, but the relation for short cylinders in multiple flow is required. As shown in Fig. 4-2, mass transfer coefficients for short cylinders, both ends exposed in parallel flow, closely corresponds to the analogous equations from Eq. (2-2).

As a result, convective heat and mass transfer coefficients for short cylinder can be obtained based on Eq. (2–2).

4–5. Summary

Mass transfer experiments using naphthalene sublimation have verified the analogy among mass and heat transfer, and suggest that convective heat and mass transfer coefficients for short cylinders may be given by Hilpert's equation.

Part II. Heat Equilibrium on Human Body

The characteristics of radiative, convective and evaporative heat interchange over the human body are investigated and steady state heat balance equations are produced.

Sudden changes of environmental condition cause heat shock or heat storage in the body, namely, it is a transient problem, and in such a case the heat balance equations can not be applied.

Chapter 5. Radiative and Convective Heat Interchange

5–1. Radiant Heat Transfer Coefficient

The distribution of black body radiation in a relatively low temperature range is shown in Fig. 5–1 and in an average room situation the radiant intensity shows its maximum value at a wave length of about 10 micron.

It is said that in the long wave length range the absorption of either the nude or clothed body may be taken as complete and the radiation of water vapor or carbon dioxide in the room can be negleced.

Substituting the numerical value at the substantial environment into Eq. (1–4), we get the radiant heat transfer coefficient as

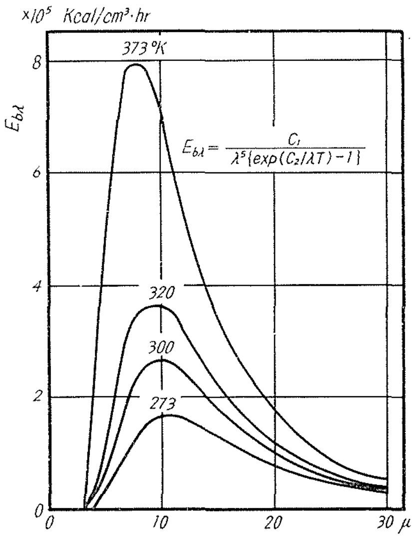

Fig. 5–1. Variation of the intensity of radiation with wave length

$$\alpha_r = 5 \quad [\text{kcal/m}^2\cdot\text{hr}\cdot\text{Deg}] \tag{5–1}$$

where $k = 1.1$, $\varepsilon_0 \cdot \varepsilon_w = 0.93$

On the other hand, if the emissivities of walls are different, "α_r" can not be expressed as simply as the above.

5–2. Surrounding Radiant Temperature and Mean Radiant Temperature

In estimating the surrounding radiant temperature, the problem changes to the question as to what kind of simplified shape factor can be used in place of the complicated shape factor between the human body and its enclosures. In this paper, the authors propose to use the shape factor for the cylinder element as an approximate shape factor for the human body.

While the mean radiant temperature in a wide sense corresponds to the surrounding radiant temperature, its inaccuracy is evident. Moreover the mean radiant temperature derived from the arithmetical average of the boundary surface temperatures or from weighted averaging omit even the shape factor.

5-3. Convective Heat Transfer Coefficient

The geometrical shape of the human body is too complicated to determine the exact convective heat transfer coefficient, so the human form must be treated as some simplified object ; namely a sphere or cylinder.

The authors propose to treat a cylinder of 30 cm in diameter in lieu of the human form.

5-4. Effect of Clothing

(1) On radiation

By decreasing the emissivity of clothing, the rate of radiative heat transfer can be reduced to the ultimate value of zero. This is very effective for protecting against the heat or the cold.

(2) On conduction

"Clo." is one of the practical units of the heat resistance of clothing.

$$1 \text{ Clo.} = 0.18 \quad [m^2 \cdot hr \cdot Deg/kcal] \qquad (5\text{-}2)$$

Table 5-1. Clo. value of clothings

Clothing	Clo. value
nudity	0
semi-nudity	0.2
underwear	0.4
summer clothing	0.6
autumn clothing	1
winter clothing	2

5-5. Summary

In an average indoor situation radiative heat transfer coefficient can be estimated as constant value.

Determining the convective and evaporative heat transfer coefficients for men, the authors have proposed to regard man as a cylinder of 30 cm in diameter.

Chapter 6. Evaporative Heat Loss

6-1. Vapor Pressure over Skin Surfaces

In an ordinary situation skin surfaces are nearly dry, and with the secretion of the perspiration wetted parts gradually appear.

In estimating the vapor pressure over such an indefinite surface the following methods may be applied.

(1) Average vapor pressure

The skin surface is maintained at average vapor pressure and sweat evaporation depends on the gradient between average vapor pressure and ambient vapor pressure, as expressed by the equation

$$W = k_p(\varphi_s \cdot P_s - P_a) \qquad (6\text{-}1)$$

where φ_s = factor for average vapor pressure
P_s = saturated vapor pressure at mean skin temperature

P_a = ambient vapor pressure

The actual value of φ_s can not be determined, but in the case of the skin surface being completely wet it takes the ultimate value of 1.0 and in the case of zero moisture, $\varphi_s = P_a/P_s$.

(2) **Wetted area ratio**

Sweat evaporation may occurs over specified wet skin surfaces, which have been saturated at mean skin temperature.

Here,

$$W = k_p(P_s - P_a) \cdot \varepsilon \tag{6-2}$$

where $\varepsilon = \dfrac{\text{wetted skin surface area}}{\text{total skin surface area}}$ = wetted area ratio

The proportion of wetted area to the total surface area are to be called "wetted area ratio". The actual value of "ε" is also undetermined, but when fully wetted it takes the ultimate value of 1.0 and in the case of zero moisture, $\varepsilon = 0$.

The idea of wetted area ratio becomes more effective for determining the comfort index.

6-2. Simplification of Saturated Vapor Perssure

Over a reasonable range of skin and ambient temperature, the vapor pressure may be approximated by the equations

$$P_s = 2.15 \cdot T_s - 31.91 \quad \text{[mmHg]} \tag{6-3}$$

$$P_{a \cdot s} = 0.031 \cdot T_a^2 - 0.031 \cdot T_a + 5.69 \quad \text{[mmHg]} \tag{6-4}$$

6-3. Vapor Resistance of Clothing

Heat resistance of clothing can be evaluated quantitatively as shown in Table 5-1, but the vapor resistance of clothing, being distinguished from that of fabric, has not been determined.

The authors try to simplify this complicated relation by the following form

$$\beta' = \zeta \cdot \beta \tag{6-5}$$

where ζ = permeance ratio

The actual value of permeance ratio may vary according to clothing type, but in the case of nudity it takes the ultimate value of 1.0, as in the case of perfect damp-proof clothing, $\zeta = 0$. Further, in Chapter 7 the substantial value of permeance ratio may be approximately obtained by the data of some experiments.

6-4. Summary

Estimating the rate of sweat evaporation, the authors have proposed a simplified equation ; the wetted area ratio, the approximation of saturated vapor pressure and the permeance ratio of clothing.

Chapter 7. Thermoregulatory Mechanism of Man

7-1. Body Heat Production

Under the basal metabolic condition it is said that normal man produces at least 40 kilogram calories per unit of skin surface area per hour, and in violent exercise the metabolic rate can be as much as 10 to 16 times that of the basal rate. There is a region of temperature, 25～29°C, over which the metabolic rate is almost constant. When the ambient temperature falls, the bodily heat production rises to maintain a constant temperature, and when the ambient temperature rises the bodily heat production rises also, due to the increased rates of chemical reaction.

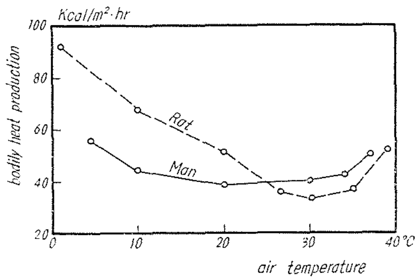

Fig. 7-1. Variation of bodily heat production (Houssay, Human Physiol., p. 518, McGraw-Hill)

Met. and R.M.R. (relative metabolic rate) are the standard expressions of metabolism.

Here,

$$1 \text{ Met.} = 50 \ [\text{kcal/m}^2\cdot\text{hr}] \quad \text{and} \quad \text{R.M.R.} = \frac{\text{energy requirement of work}}{\text{basal metabolism}}$$

Comparing the two indexes, R.M.R. is used as an index of intensity of manual work, but it does not indicate the metabolic rate directly.

The Met. value for typical work categories is shown in Table 7-1.

Table 7-1. Met. value for typical work categories

	Met.	kcal/m²·hr
sleeping	0.8	40
sitting	1.0	50
sedentary work	1.6	80
light work in a standing posture	2.0	100
walking 4 km/hr	3.0	150
5.6 km/hr	4.0	200
5.6 km/hr (with 2 kg weight)	6.0	300

7-2. Body Heat Loss

The heat generated within the body is balanced by losses to the atmosphere as radiation, convection, evaporation and in warming food and inspired air, but the proportion of heat losses by the three principal ways to the total heat loss is large enongh to neglect the others. The rate of heat losses by radiation, convection and evaporation depends on the condition of the environment and the amount of work. Fig. 7-2 is an example, derived from Fahnestock's report.[5]

7-3. Insensible Perspiration

Insensible perspiration is an imperceptible loss of water through the skin and is distinguished from sweat secretion.

At the rate of approximately 23 grams per unit of skin surface area per hour, between 800～1200 grams of water are evaporated by insensible evaporation every 24 hours, and this evaporation accounts for one-quarter of the heat loss in a resting subject. This fact is very important for the analysis.

Insensible perspiration is greatest on the palms of the hands, and the soles of the feet; next on the back of the hand, the neck and the face; and least on the remainder of the body surface. Thus, even with clothing, the rate of insensible perspiration is maintained at a constant value.

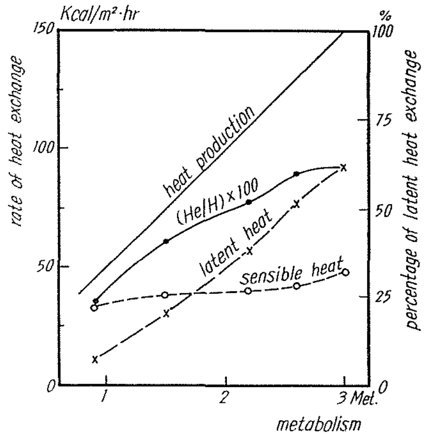

Fig. 7-2.
Variation of the rate of each heat loss

7-4. Sweat Secretion

When the ambient temperature rises the sweat secretion begins over all the skin surface, especially at the exposed parts, with the exception of the palms of the hands and soles of the feet.

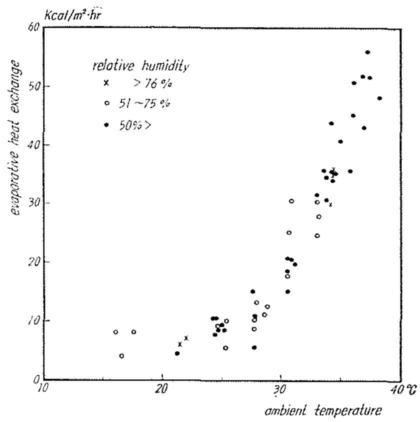

Fig. 7-3. Variation of evaporative heat loss with ambient temperature

Fig. 7-3 shows the relation between evaporative heat loss of man at rest and ambient temperature, derived from Winslow's report[6], and we know that when the ambient temperature rises above 28°C there is an increase in perspiration secretion.

7-5. Effects of Clothing on Sweat Evaporation

Analyzing Gagge and Nagata's experimental data, the magnitude of permeance ratio of clothing, defined in Chapter 6, become evident.

(1) Gagge's experiment[7]

Gagge and his associates presented data on sweat evaportion. The experiment was made with two clothed and four nude subjects in a semi-reclining position. The standard clothing employed consisted of a two piece suit of cotton underwear, a cotton shirt, socks, shoes and a dark gray single-ply suit with three-quarter lined coat and a fully lined vest.

Fig. 7-4 shows that in an average room situation the blocking effect of clothing upon sweat evaporation was almost negligible.

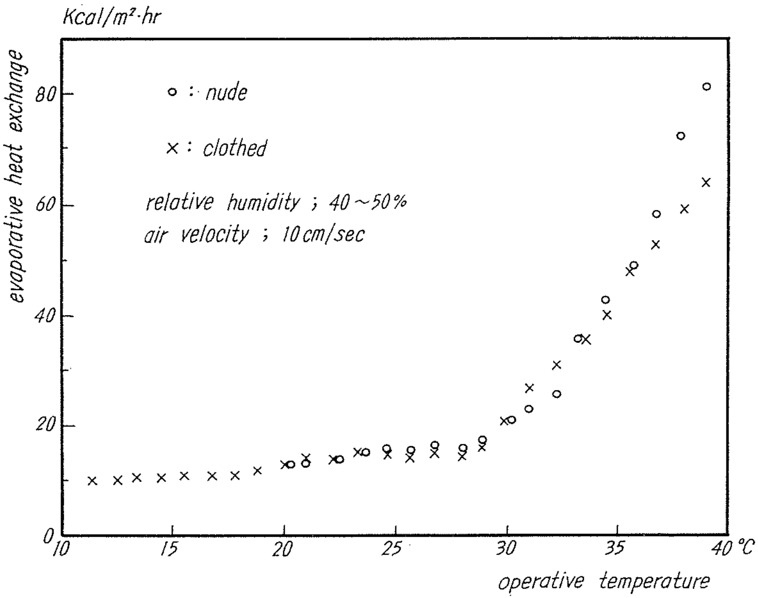

Fig. 7-4. Effect of clothing on sweat evaporation—a

(2) **Nagata's experiment**[8)]

The sweat evaporation of four subjects in the nude and with various types of underwear was observed.

The test room was maintained at approximately 40°C dbt and 50% rh respectively, and both radiant heat and air motion were not sensible.

In that experiment the amount of secreted sweat was 200~250 grams per unit time per person.

Table 7-2 shows that when the rate of sweat secretion reaches several times as much as the normal condition, only 10~15% of secreted sweat remains in the clothing or drips off.

Table 7-2. Effect of Clothing on Sweat Evaporation—b

	nudity	linen	hemp	hemp & rayon	cotton	nylon
$\frac{\text{evaporated sweat}}{\text{secreted sweat}}$ (%)	90.4	81.8	84.2	87.6	86.2	88.2
$\frac{\text{evaporated sweat}}{\text{secreted sw.} - \text{dripped sw.}}$ (%)	100	84	89	88	89	91

Further, Nagata observed the sweat evaporation on the subjects with underwear and outer garment.

The test room was maintained at 30°C dbt and 75~85% rh respectively. Two subjects were clothed in various types of clothing and were made to walk around at a rate of thirty steps per minute for two hours. In this test the amount of perspiration secretion was about 150 grams per unit time per person.

Table 7-3 shows that when the subjects put on damp-proof clothing, they only blcck about 20% of the secreted perspiraticn.

Gagge and Nagata's observations suggest that the vapor resistance of light clothing may be neglected and its permeance ratio seems to take the value of nearly 1.0.

Table 7-3. Effect of Clothing on Sweat Evaporation—c

	C 1	C 2	C 3	C 4	C 5
$\frac{\text{evaporated sweat}}{\text{secreted sweat}}$ (%)	76.4	78.0	78.0	81.4	86.0

The five types of outer garments used were as follows:
- C 1: a knee-length vinyl raincoat
- C 2: a knee-length vinyl raincoat with very fine holes
- C 3: a waist-length vinyl raincoat
- C 4: a waist-length vinyl raincoat with a special rift in the back
- C 5: a knee-length cotton coat

7-6. Heat Balance Equation of Man

The expression of the equation is slightly different in each case of nudity and clothing, because of the existence of heat and vapor resistance.

(1) Heat balance equation of the nude man

In a steady state, the heat produced within the body is balanced by cooling powers of the environment as radiation H_r, convection H_c and evaporation H_e

$$H = H_r + H_c + H_e \quad (7\text{-}1)$$

$$= \alpha_r \cdot (T_s - T_r) + \alpha_c \cdot (T_s - T_a) + \beta \cdot (P_s - P_a) \cdot \varepsilon \quad (7\text{-}2)$$

where H = produced heat per unit of skin surface area per unit time
T_s = mean skin temperature
P_s = saturated vapor pressure at T_s
P_a = partial vapor pressure in the air
ε = wetted area ratio

Generally, the term of heat storage is joined to the right-hand side of Eq. (7-2). The authors do not deny the existence of heat storage, but it appears as an unsteady state heat exchange.

As mentioned in Chapter 6, the vapor pressure over skin surface and in the air can be expressed simply by

$$P_s = a \cdot T_s + b \quad (7\text{-}3)$$

and

$$P_a = (A \cdot T_a^2 + B \cdot T_a + C) \cdot \varphi_a \quad (7\text{-}4)$$

where φ_a = relative humidity

Substituting the above relations into Eq. (7-2) we get

$$H = \alpha_r \cdot (T_s - T_r) + \bar{\alpha}_c \cdot (T_s - \bar{T}_a) \quad (7\text{-}5)$$

where

$$\bar{\alpha}_c = (1 + a \cdot \varepsilon \cdot \kappa) \cdot \alpha_c \tag{7-6}$$

$$\bar{T}_a = \frac{T_a + \varepsilon \cdot \kappa \cdot \varphi_a \cdot (A \cdot T_a^2 + B \cdot T_a + C - b/\varphi_a)}{1 + a \cdot \varepsilon \cdot \kappa} \tag{7-7}$$

$$\beta = \kappa \cdot \alpha_c \tag{7-8}$$

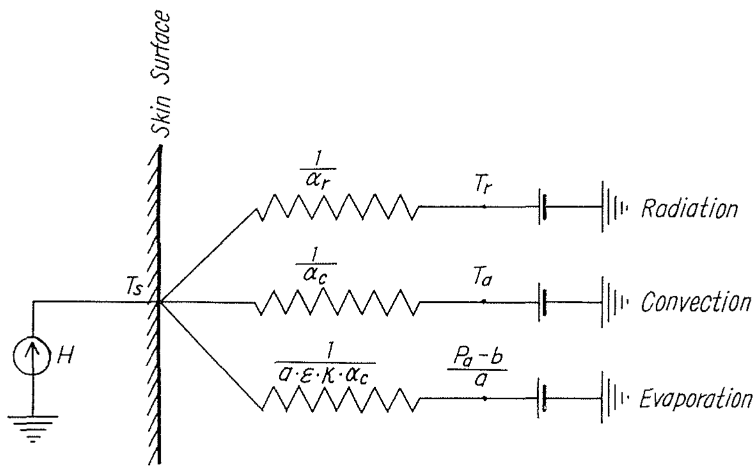

Fig. 7-5. Thermal network for Eq. (7-5)

(2) **Heat balance equation of the clothed man**

When clothed, the following equations may be obtained.

$$H = \alpha_r \cdot (T_c - T_r) + \alpha_c \cdot (T_c - T_a) + \zeta \cdot \beta \cdot (P_s - P_a) \cdot \varepsilon \tag{7-9}$$

and

$$\alpha_r \cdot (T_c - T_r) + \alpha_c \cdot (T_c - T_a) = \frac{1}{R_c} \cdot (T_s - T_c) \tag{7-10}$$

where T_c = surface temperature of clothing
R_c = heat resistance of clothing

and therefore,

$$H = \frac{\alpha_r}{1 + R_c \cdot (\alpha_r + \alpha_c)} (T_s - T_r) + \frac{\bar{\bar{\alpha}}_c}{1 + R_c \cdot (\alpha_r + \alpha_c)} (T_s - \bar{\bar{T}}_a) \tag{7-11}$$

where

$$\bar{\bar{\alpha}}_c = \left[1 + a \cdot \varepsilon \cdot \kappa \cdot \zeta \cdot \left\{1 + R_c(\alpha_r + \alpha_c)\right\}\right] \cdot \alpha_c \tag{7-12}$$

$$\bar{\bar{T}}_a = \frac{T_a + \varepsilon \cdot \kappa \cdot \zeta \cdot \varphi_a \cdot \{1 + R_c(\alpha_r + \alpha_c)\} (A \cdot T_a^2 + B \cdot T_a + C - b/\varphi_a)}{1 + a \cdot \varepsilon \cdot \kappa \cdot \zeta \cdot \{1 + R_c(\alpha_r + \alpha_c)\}} \tag{7-13}$$

The above heat balance equations can quantitatively evaluate each component of the thermal environment, and will be applied in determining the comfort index.

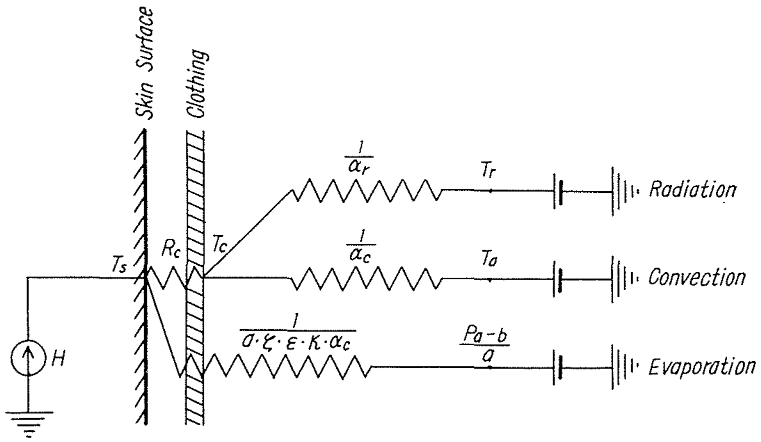

Fig. 7-6. Thermal network for Eqs. (7-9) and (7-10)

7-7. Summary

In expressing the heat equilibrium between man and thermal environment with the engineering method, the physiological characteristics of the human were taken into consideration. Then, on the basis of the above the heat balance equations were proposed.

References

5) Fahnestock, M. K. et al.: Comfort and Physiological Response to Work in an Environment of 75 F and 45% R.H., ASHRAE Journal (1963 March), p. 25.
6) Winslow, C. E. A. et al.: Physiological Reaction of the Human Body to Various Atmospheric Humidities, Am. J. of Physiol. Vol. 120 (1937) p. 290.
7) Gagge, A. P. et al.: The Influence of Clothing on the Physiological Reaction of the Human Body to Varying Environmental Temperature, Am. J. of Physiol. Vol. 124 (1938) p. 30.
8) Nagata, H.: Evaporation of Sweat on Clothed Subject, Jap. J. of Hyg. Vol. 17 (3) (1962) p. 155.

Chapter 8. Experimental Verification of Heat Balance Equation

8-1. Procedure

The skin temperatures and thermal sensations of two male subjects, ages 22, were observed. The detailed physical data on the subjects are summarized in Table 8-1.

The clothing employed was as follows: a) nude with bathing trunks, b)

Table 8-1. Physical data of the subjects

subject	height (m)	weight (kg)	surface area (m^2)	physique	metabolic rate (3 measurements) (kcal/m^2·hr)
D	1.67	54	1.66	slender	50.3
S	1.60	58	1.61	slightly fat	60.7

cotton underwear and underdrawers, c) cotton underwear and underdrawers fully saturated with water.

Skin temperatures were measured with a copper-constantan thermocouple and the mean skin temperatures were calculated by the equations

$$\text{mean skin temp.} = 0.1 \times \text{forehead} + 0.05 \times \text{forearm} + 0.05 \times \text{back of the hand} + 0.4 \times \text{chest} + 0.2 \times \text{thigh} + 0.2 \times \text{leg} \quad (8\text{–}1)$$

and

$$\text{mean skin temp. at wetted part} = \frac{\text{forearm} + \text{chest} + \text{thigh} + \text{leg}}{4} \quad (8\text{–}2)$$

Analyzing the experimental data, some thermal factors were assumed as follows:

1) convective and evaporative heat transfer coefficients are calculated based on Eqs. (2–2) and (3–12) for the cylinder of 30 cm in diameter.

2) radiant heat transfer coefficient is the value of 5 [kcal/m²·hr·Deg].

3) heat resistance and wetted area ratio of the subject with fully wetted clothing are the value of $R_c = 0$ [m²·hr·Deg/kcal] and $\zeta = 1.0$ respectively.

4) metabolic rate of the subject in a seated position is the value of $H = 50$ [kcal/m²·hr] (1 Met.).

5) permeance ratio is the value of $\zeta = 1.0$.

8-2. Experiment A

The heat balance equation suggests that, under very hot circumstances where even radiative and convective heat flow is allowed into the body, man would feel cold so long as capacity of the environment to accept the evaporation is sufficiently large.

The test room was maintained at 47.4°C dbt, 20% rh and 0.7~0.8 m/sec air-movement respectively, and the subjects were in a seated position.

From Table 8–2 we know that in very hot environments, the imposed heat load cannot be compensated for by man's thermoregulatory ability, but the cooling

Table 8–2. Data of Experiment A

condition of clothing	dry, 0.3 Clo.	fully saturated, 0 Clo.
watted area ratio	$\varepsilon = 0.35$	$\varepsilon = 1.0$
sweat secretion	much, but surfaces are drying	none
mean skin temp.	36.5°C (forehead; 35.4°C, back of hand; 37.4°C)	33.0°C (forehead; 39.4°C, back of hand; 40.2°C) 31.3°C (average of wetted parts)
skin temp. from heat balance equation	36.5°C (where $\varepsilon = 0.35$)	31.6°C (where $\varepsilon = 1.0$)
thermal vote	very hot	cold

effect of the evaporation of supplied water is large enough to maintain a heat equilibrium. This fact may be of some value for improving sports conditions or industrial environments.

8-3. Experiment B

The higher the velocity of the air flow, the more the evaporative heat loss increases, but in the case when ambient temperature exceeds the skin temperature, the convective heat gain increases also. When the heat loss and gain offset each other, the mean skin temperature is scarcely concerned with the air movement.

The test room was maintained at 34.4°C dbt and the surrounding radiant temperature, 85% rh and 1.5, 2, 3, 4 m/sec air-movement respectively.

The subjects were in a seated position and were clothed with underwear and underdrawers fully saturated with water.

In spite of changing the air-movement from 1.5 m/sec to 4 m/sec, the drop in mean skin temperature was only 0.4°C. Furthermore, the difference between experimental data and calculated values was sufficiently small. At that time the thermal vote of the subjects gradually changed from cool to slightly cold.

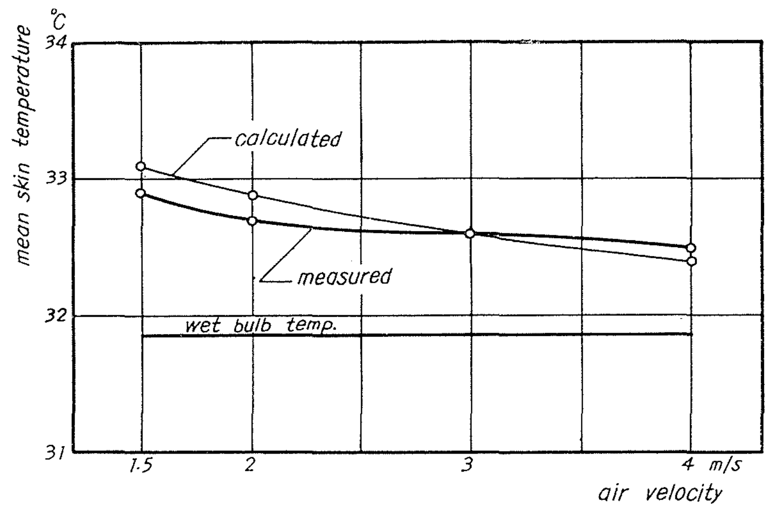

Fig. 8-1. Variation of skin temperature

8-4. Experiment C

The relation between measured skin temperatures and calculated values was examined. 26.4~38.4°C dbt, 17~39 mmHg humidity, and 0.7~4.0 m/sec air-movement, constituted 46 test conditions, and the surrounding radiant temperatures approximately coincided with the ambient temperature.

The subjects were remained seated and put on underwear and underdrawers fully saturated with water.

In Fig. 8-2, if the heat balance equation is idealistically precise, each dotted point must be superposed on the solid line, and should approximate its location.

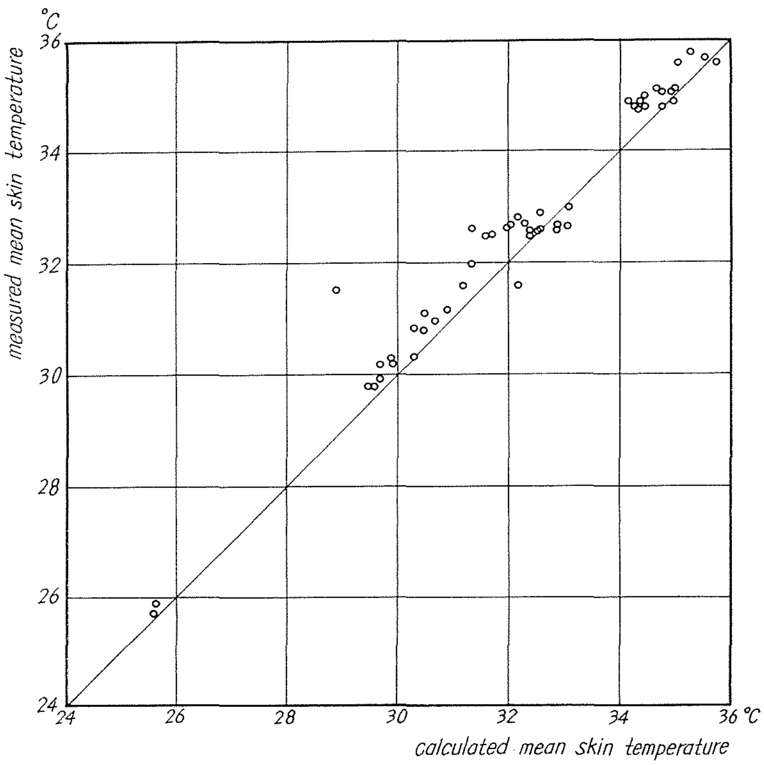

Fig. 8-2. Comparison between experimental data and calculated values

8-5. Summary

The experiments have verified that calculated skin temperatures were approximate to measured values. Thus, it may be said that the heat balance eqnations are rational.

Part III. Development of Warmth Diagram and its Application to Air-Conditioning

On the basis of the heat balance equation "the model skin temperature", that expresses the thermal sensation, may be deduced, and to simplify the calculating process of model skin temperature "the Warmth Diagram" and "Comfort Detector" may be proposed.

Chapter 9. Thermal Sensation and Model Skin Temperature

9-1. Definition of Comfort

From a engineering point of view the authors define "comfort" as follows; "the condition in which man is emitting the entire produced heat steadily at the optimum and constant body temperature without any additional aid of a thermoregulatory mechanism".

The following condition may be assumed to be a comfortable condition.

Environmental side:

ambient temperatare (surrounding radiant temp.)	:	27.5°C
relative humidity	:	50%
air movement	:	10 cm/sec

Physical side:

clothing	:	0 Clo.
work rate	:	1 Met.
mean skin temperature	:	33.0°C
percentage of evaporative heat loss (by insensible perspiration)	:	25%

If the wetted area ratio takes the value of 0.16, the above physical condition is realized at the above thermal environment. This relation suggests that in a comfortable condition without perspiration secretion, the wetted area ratio of the human body may be maintained at the value of 0.16.

In other words, any environment, at which mean skin temperature is 33.0°C with $\varepsilon=0.16$, may be assumed to be a comfortable condition.

9-2. Expression of Thermal Sensation

In hot or cold circumstances men are obliged to set the thermoregulatory mechanism to work in order to maintain a constant body temperature. Then, the quantity of the expended effort is concerned with the thermal sensation.

To extract the quantity of the expended effort we consider "the model man" having the same thermal characteristic as that of the man in the comfortable condition.

In any environment, the difference between the skin temperature of model man and the comfortable skin temperature of 33.0°C seems to be directly related with the quantity of the expended effort.

The authors propose to express the thermal sensation by the following definition; "the difference between the model man's skin temperature, in short, the **model skin temperature** and the comfortable skin temperature of 33.0°C expresses the deviation of the sensation from the comfortable condition."

9-3. Relation between Thermal Vote and Model Skin Temperature

The relation between the thermal sensation and the model skin temperature can be examined by the subjects' thermal votes.

The test conditions were as follows; room temperature: 20～49°C, humidity: 10～18 mmHg, air-movement: 0.1～1 m/sec, work: in a seated position (1 Met.), clothing: in the nude or with underwear and underdrawers (0.3 Clo.), subjects: D and S.

The results were summarized in Fig. 9-1.

9-4. Temperature-Humidity Chart and Model Skin Temperature

On the basis of large scale experiments, Nevins and his associates proposed "the Temperature-Humidity Chart".[9)]

Test conditions were as follows; room temp.: 19～28°C (9 dbt), humidity:

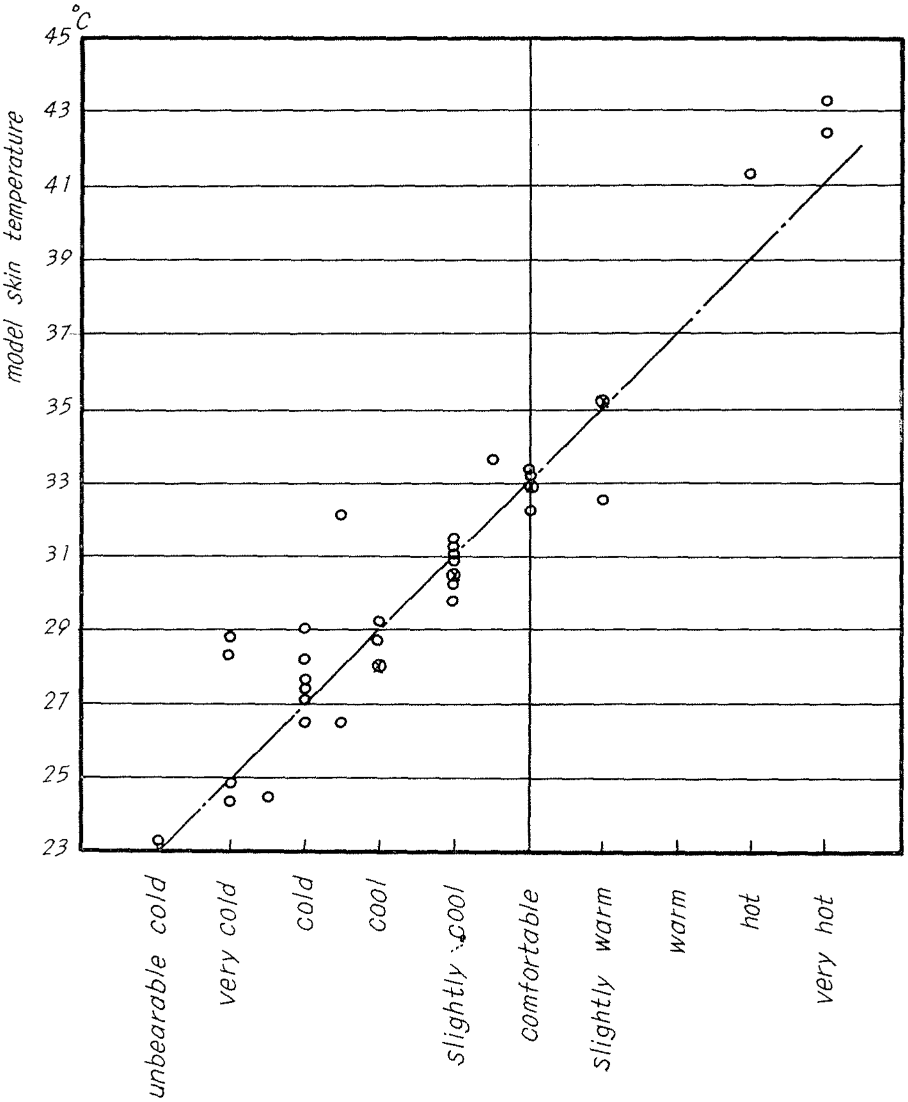

Fig. 9–1. Thermal vote and model skin temperature
(⊗ marks were derived from Nevins' report)

15∼85% (8 rh at each of 9 dbt), air-movement: assumed to be 15 cm/sec, work: in a seated position (1 Met.), clothing: standardized clothing (0.52 Clo.), subjects: 360 male and 360 female students ranging from 18 to 23 in age.

From the thermal votes of the subjects, the following equation has been proposed.

Table 9–1. Comfort vote scale

value of Y_c	thermal sensation
1	cold
2	cool
3	slightly cool
4	comfortable
5	slightly warm
6	warm
7	hot

$$Y_e = -10.749 + 0.183 \cdot T + 0.00017 \cdot T \cdot H \qquad (9\text{-}1)$$

where Y_e = estimated population mean vote for males and females combined in equal numbers. Comfort vote scale is shown in Table 9-1

T = dry bulb temperature °F

H = relative humidity in percentage

Nevins' comfort lines and authors' equal model skin temperature lines are given in Fig. 9-2.

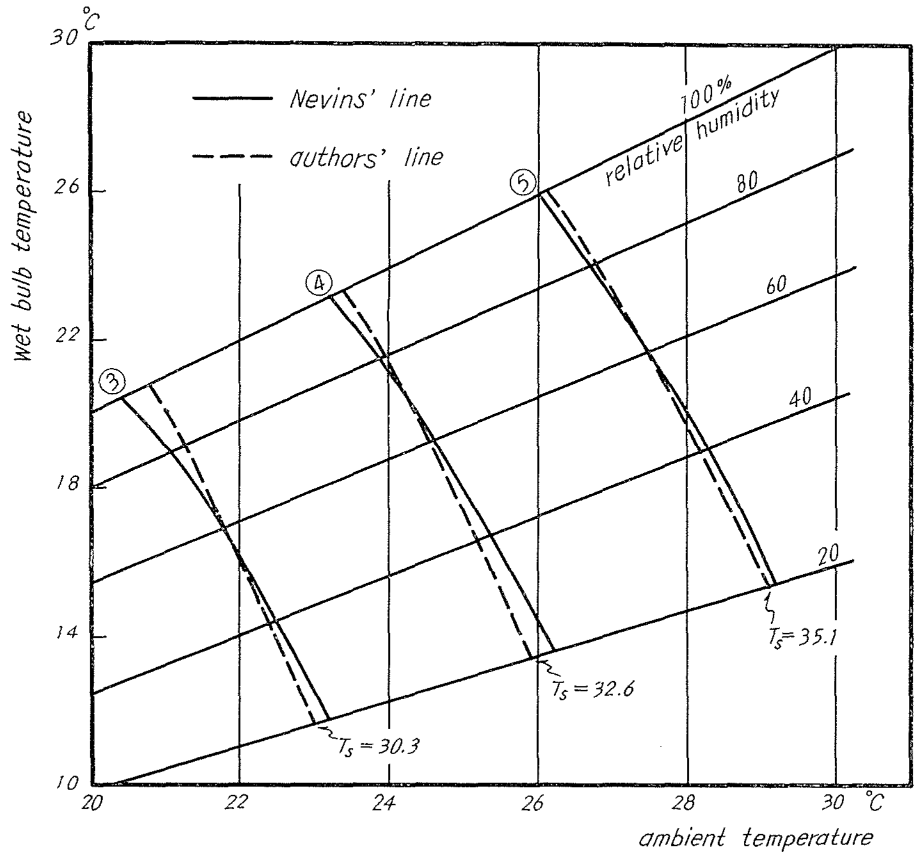

Fig. 9-2. Comfort line and model skin temperature line

9-5. Model Skin Temperature as a Scale of Warmth

From Figs. 9-1 and 9-2, we can recognize that model skin temperature is closely related with thermal sensation.

The model skin temperature does synthetically evaluate many thermal factors, such as ambient temperature, surrounding radiant temperature, humidity, air-movement, heat and vapor resistance and emissivity of clothing, work rate and physical properties of air. So, the model skin temperature may be used as a rational comfort index.

9-6. Summary

By means of a engineering method, the human thermal sensation has been expressed with the model skin temperature that defined by the authors.

Reference

9) Nevins, R. G. et al.: A Temperature Humidity Chart for Thermal Comfort of Seated Person, ASHRAE J. (1966, April) p. 55.

Chapter 10. Warmth Diagram and its Application

10-1. Principle of the Warmth Diagram

K. Ibamota, one of the authors, first proposed the principle of Warmth Diagram.

Radiative and convective heat interchange H_D are expressed by

$$H_D = \alpha_r \cdot (T_c - T_r) + \alpha_c \cdot (T_c - T_a) \qquad (10\text{-}1)$$

$$= (\alpha_r + \alpha_c) \cdot (T_c - T_0) \qquad (10\text{-}2)$$

where $T_0 = \dfrac{\alpha_r \cdot T_r + \alpha_c \cdot T_a}{\alpha_r + \alpha_c}$ = operative temperature

And transformed to

$$T_r = \frac{\alpha_c}{\alpha_r} \cdot (T_c - T_a) + T_c - \frac{H_D}{\alpha_r} \qquad (10\text{-}3)$$

Then, Eq. (10-3) can be expressed in Fig. 10-1.

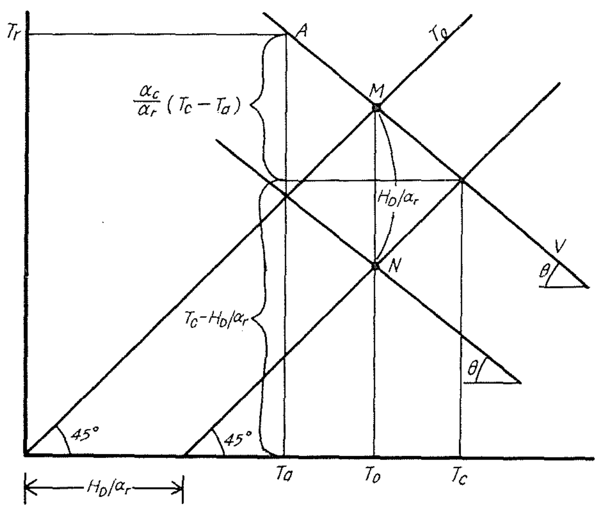

Fig. 10-1. Schematic expression of Eq. (10-3)

In Fig. 10-1, the line V, expressing the air-movement, is characterized by the angle of $\theta = \tan^{-1}\alpha_c/\alpha_r$. The point M expresses the operative temperature, but in the case of the heat production being zero it gives the surface temperature of the body.

From the above relation we get the following concrete scheme.

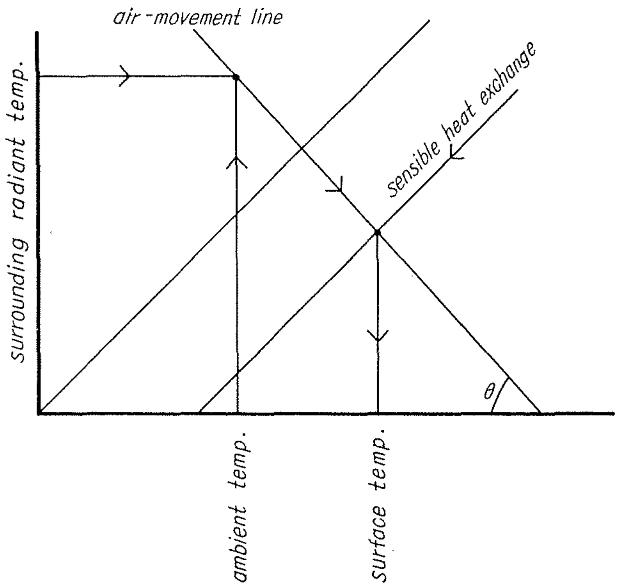

Fig. 10-2. Principle of warmth diagram

10-2. Graphical Solution of Model Skin Temperature

(1) For the nude model

In a similar way as Eq. (10-3), from Eq. (7-5) we get

$$T_r = \frac{\bar{\alpha}_c}{\alpha_r} \cdot (T_s - \bar{T}_a) + T_s - \frac{H}{\alpha_r} \tag{10-4}$$

While, substituting the following Sprung's formula, with respect to wet and dry bulb temperature, into Eq. (7-7) we get Eq. (10-6).

$$P_a = P_{ws} - \frac{0.5}{755} P_{at} \cdot (T_a - T_w) \tag{10-5}$$

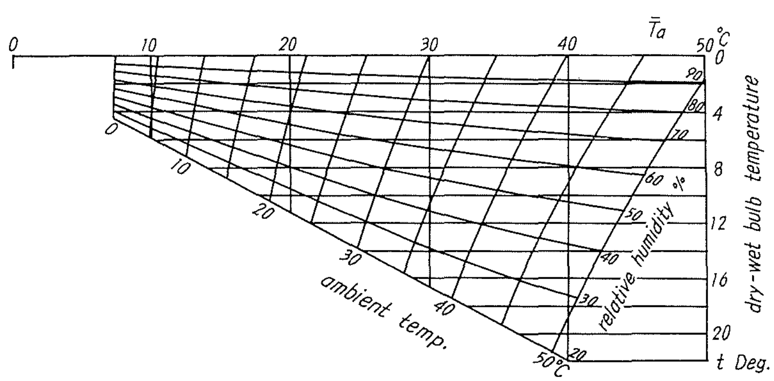

Fig. 10-3. Graphical solution of $\bar{T}_a$

$$\bar{T}_a = \left(\frac{1}{1+a\cdot\varepsilon\cdot\kappa}\right) \times \left\{ \begin{array}{l} A\cdot\varepsilon\cdot\kappa\cdot T_a^2 + (1+2\cdot A\cdot\varepsilon\cdot\kappa\cdot t + B\cdot\varepsilon\cdot\kappa)\cdot T_a \\ \quad + A\cdot\varepsilon\cdot\kappa\cdot t^2 - \varepsilon\cdot\kappa\cdot(B\cdot t + 0.5\cdot P_{at}\cdot t/755) - \varepsilon\cdot\kappa\cdot(C-b) \end{array} \right\} \tag{10-6}$$

Eq. (10–6) involves some variables, such as T_a, $t = T_a - T_w$, P_{at}, κ and ε. If some variables are fixed, as $P_{at} = 1$ atm, $\kappa = 2.2$ and $\varepsilon = 0.16$, the variable $\bar{T}_a$ can be graphically solved with Fig. 10–3.

(2) **For the clothed model**

The Eq. (7–11) can be transformed to

$$T_r = \frac{\bar{\bar{\alpha}}_c}{\alpha_r}(T_s - \bar{\bar{T}}_a) + T_s - \frac{H'}{\alpha_r} \tag{10-7}$$

where

$$H' = \left\{1 + R_c(\alpha_r + \alpha_c)\right\} \cdot H \tag{10-8}$$

The algebraical symbols ; $\bar{\bar{T}}_a$, $\bar{\bar{\alpha}}_c$ and H' involve the heat resistance of clothing R_c respectively. Hence, diagrams should be constructed, treating the variable R_c as a parameter.

Further, $\bar{\bar{T}}_a$ and H' involve the variable α_c. In an average indoor situation, the variation of α_c affects H' in several areas, but has little effect on $\bar{\bar{T}}_a$.

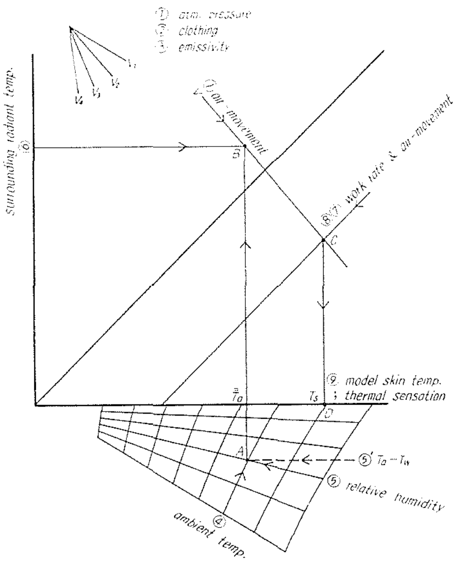

Fig. 10–4. Procedure for determining model skin temperature or thermal sensation

10–3. Warmth Diagram

The completed Warmth Diagrams are shown in Figs. 10–5∼10–11.

(1) Select the proper diagram, according to ① atmospheric pressure, ② Clo. value and ③ emissivity of clothing.

(2) Draw vertical line from intersection A of ④ ambient temperature and ⑤ relative humidity (or ⑤' difference between T_a and T_w).

(3) At ⑥ surrounding radiant temperature draw horizontal line; thus get intersection B.

(4) Passing through intersection B, draw the ⑦ air-movement line V of slope $\theta = \tan^{-1} \alpha_c/\alpha_r$.

(5) From the intersection C of line V and ⑧–⑦ work rate & air-movement line of slope 45°, draw vertical line downward and determine model skin temperature or thermal sensation.

If one adheres to the above regulations the Diagram can be used in many ways to obtain other thermal factors.

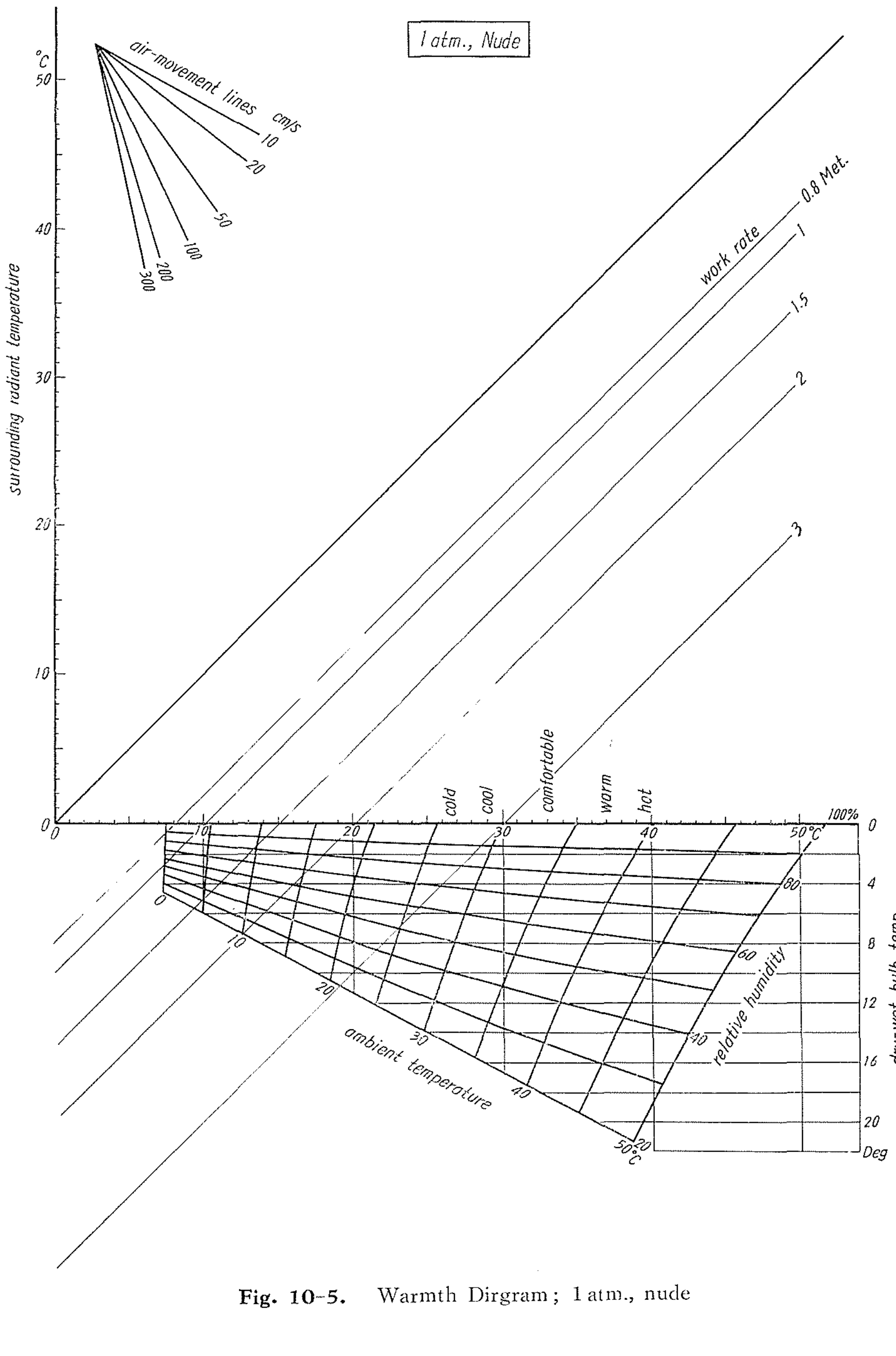

Fig. 10-5. Warmth Dirgram; 1 atm., nude

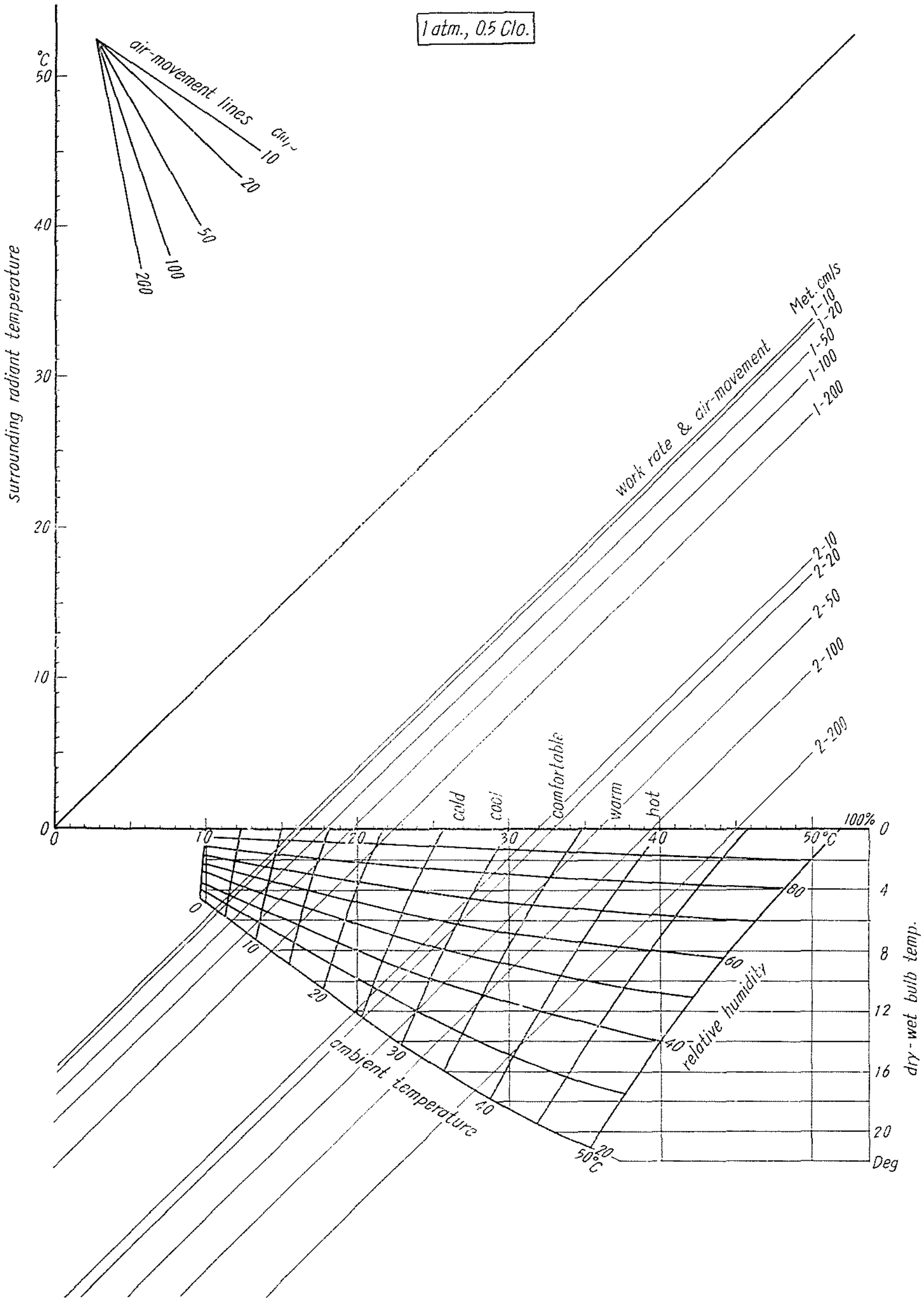

Fig. 10-6. Warmth Diagram ; 1 atm., 0.5 Clo.

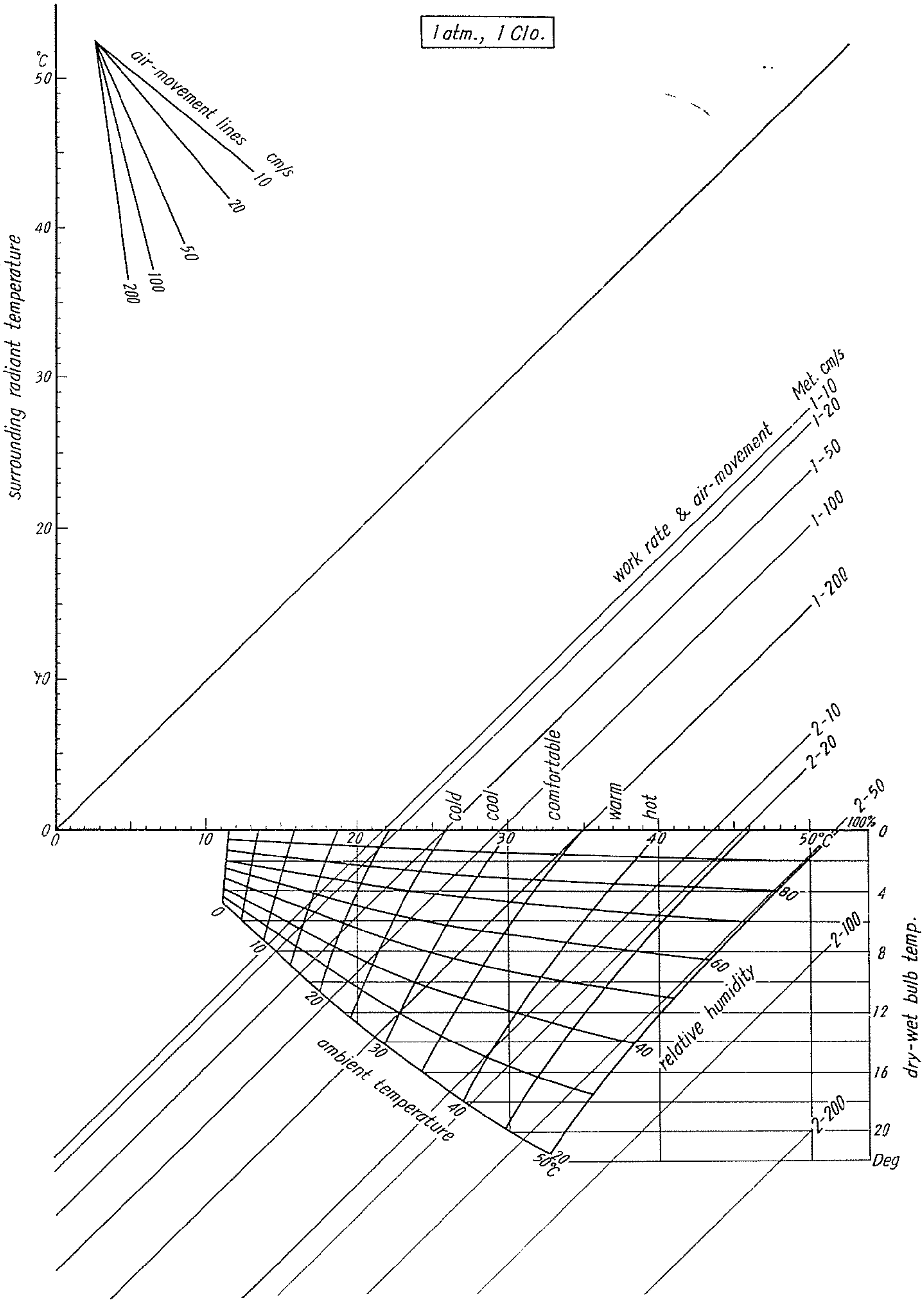

Fig. 10-7. Warmth Diagram ; 1 atm., 1 Clo.

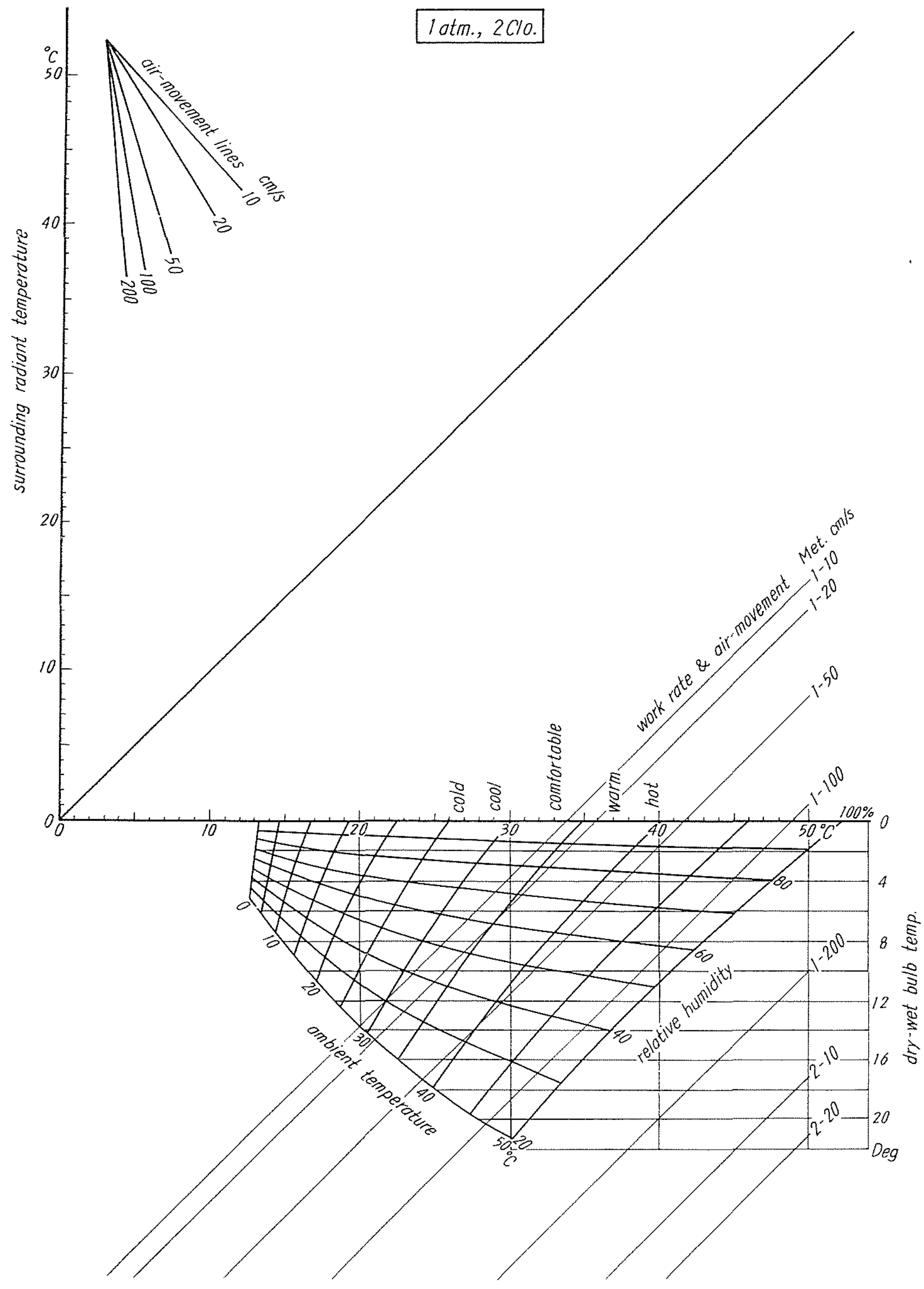

Fig. 10-8. Warmth Diagram; 1 atm., 2 Clo.

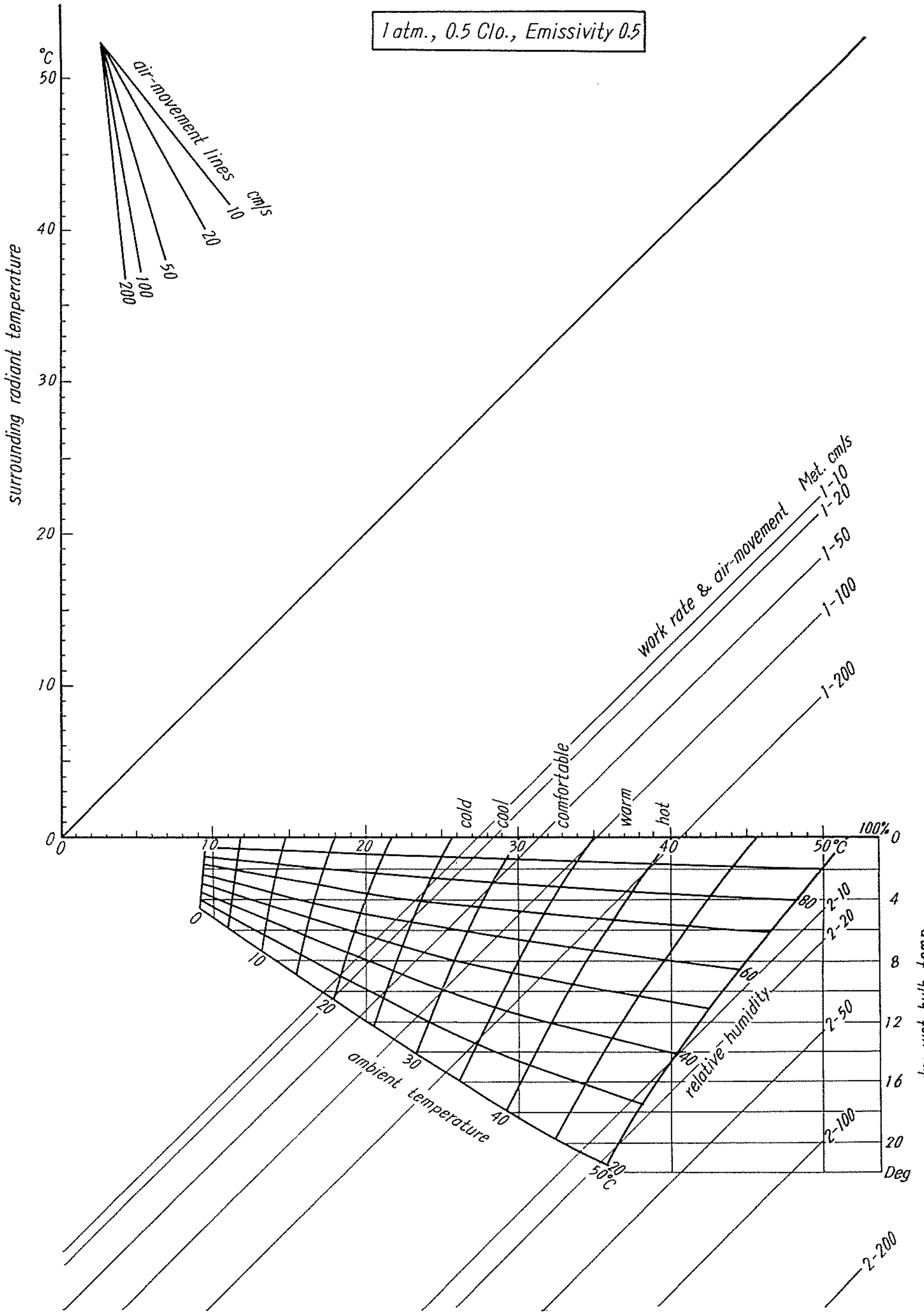

Fig. 10-9. Warmth Diagram; 1 atm., 0.5 Clo., emissivity 0.5

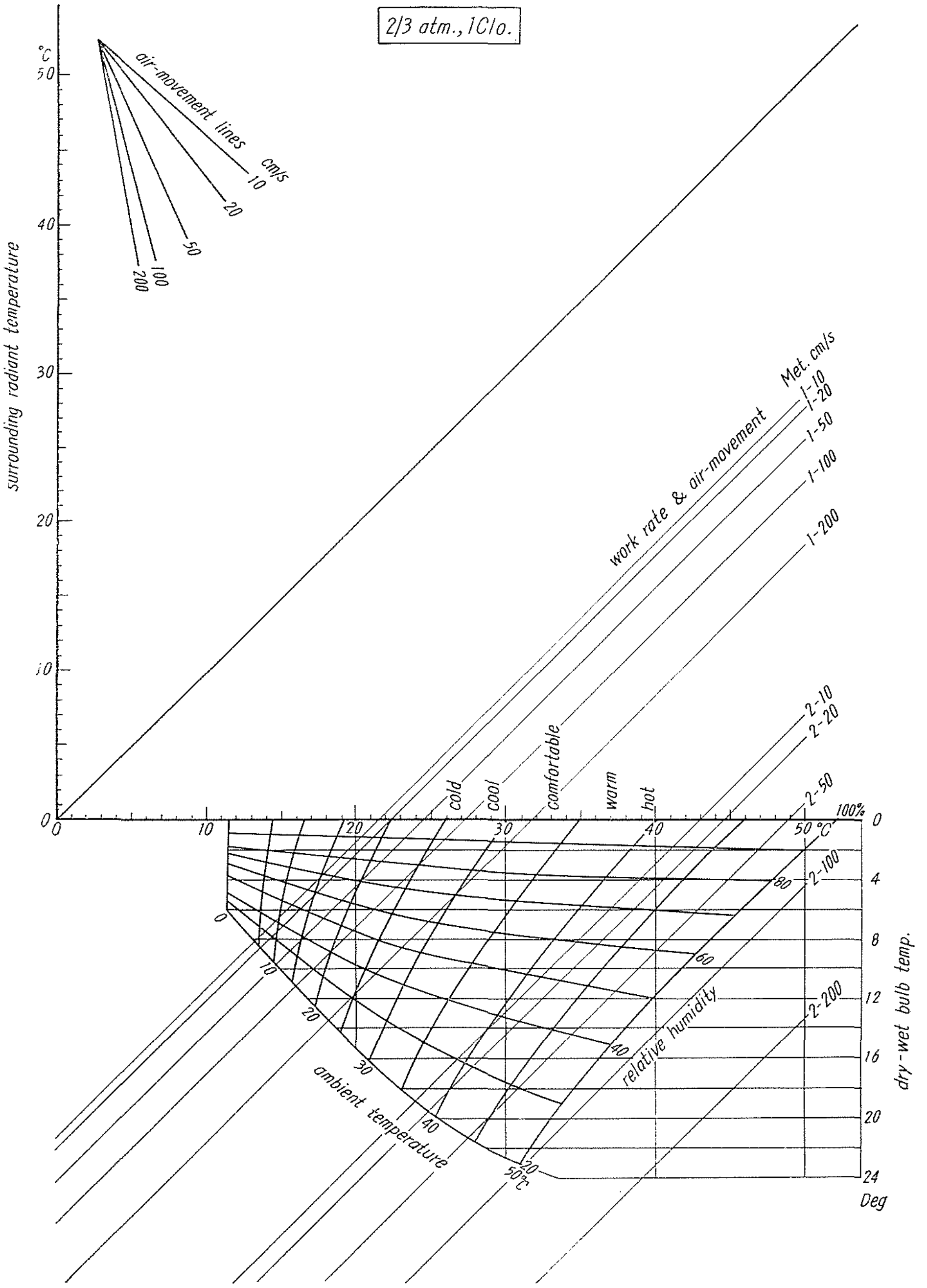

Fig. 10-10. Warmth Diagram ; 2/3 atm., 1 Clo.

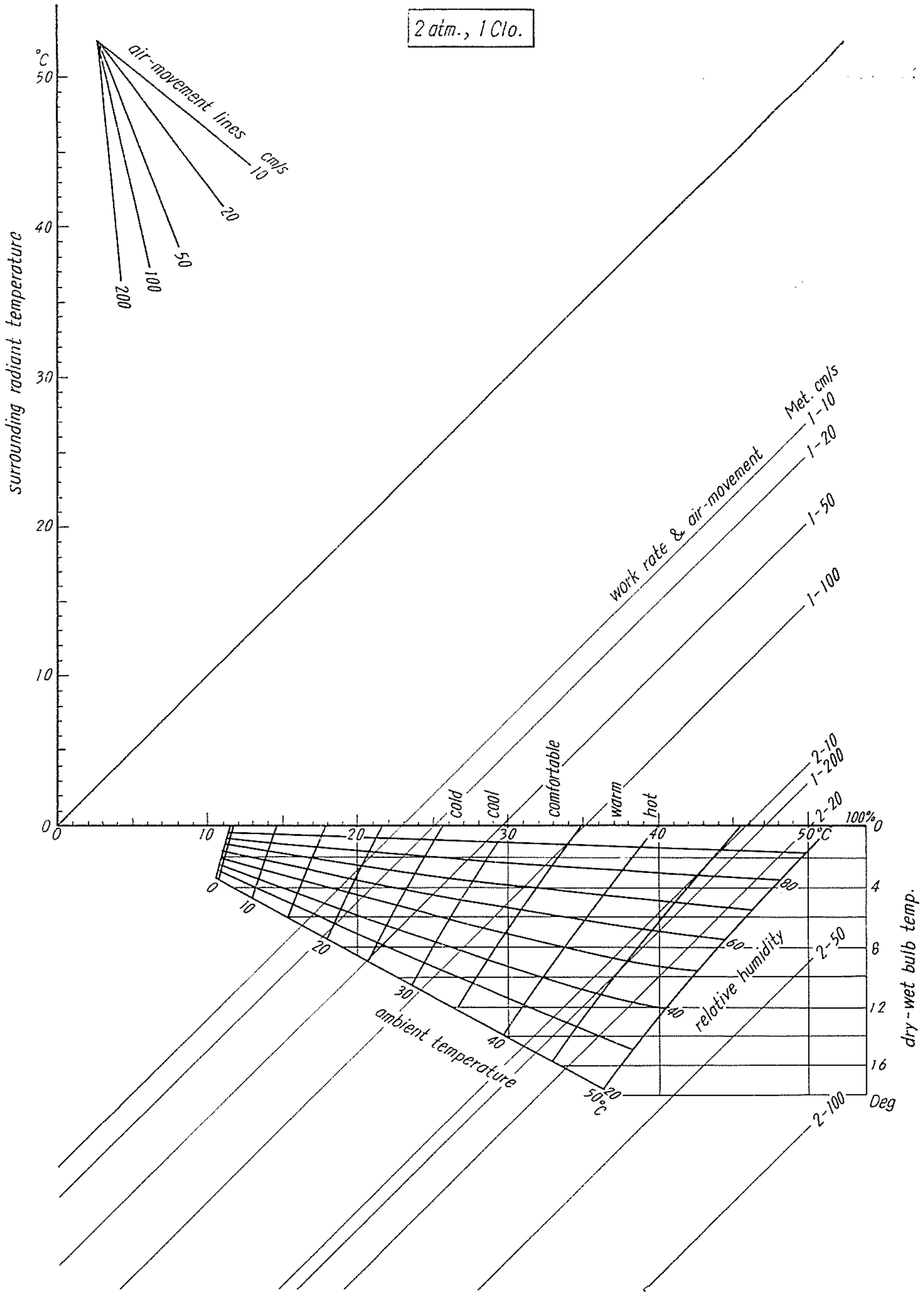

Fig. 10-11. Warmth Diagram; 2 atm., 1 Clo.

10-4. Evaluation of Thermal Environment with Warmth Diagram

By the use of Warmth Diagram one can easily evaluate the condition of the thermal environment.

Example 10-1

In a room at ambient temperature: 25°C, surrounding radiant temperature: 20°C, air movement: 10 cm/sec, a man is in a seated position (1 Met.) with 1 Clo. clothing. If the relative humidity takes the value of 20, 40, 60 and 90% respectively, the comfort sensation in each case may be estimated.

Solution

Using Fig. 10-7, the following relation is derived.

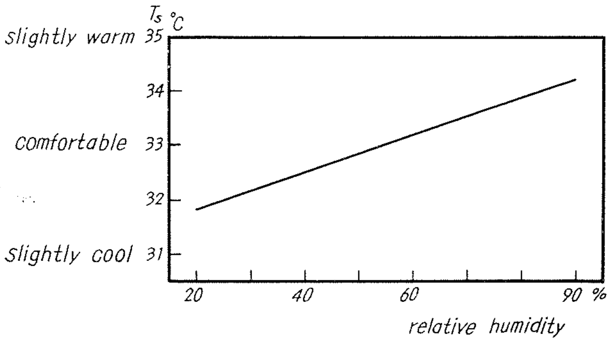

Fig. 10-12. Influence of humidity in the substantial room situation

As shown above, in a substantial room situation the effect of the humidity to comfort sensation is almost negligible.

Example 10-2

In a room at ambient temperature: 35°C, surrounding radiant temperature: 30°C, air movement: 10 cm/sec, a man is in a seated position (1 Met.) with 0.5

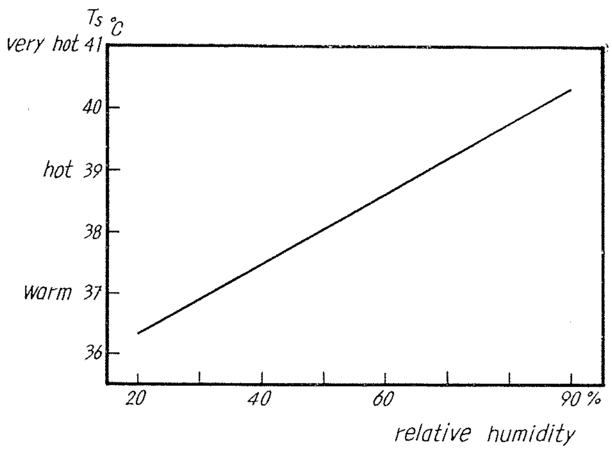

Fig. 10-13. Influence of humidity in the hot environment

Clo. clothing. If the relative humidity takes the value of 20, 40, 60 and 90% respectively, the comfort sensation in each case can be estimated.

Solution

From Fig. 10–6, the result is given in Fig. 10–13.

In the hot environment, the humidity advances to one of the essential thermal factors.

Example 10–3

A man is in a seated position (1 Met.), with 1 Clo. clothing, under the atmospheric pressure of 2/3, 1 and 2 atm. respectively. The other thermal factors for the comfortable condition may be decided, where air movement is fixed at 20 cm/sec

Solution

From Figs. 10–10, 10–7 and 10–11, we get the following illustration.

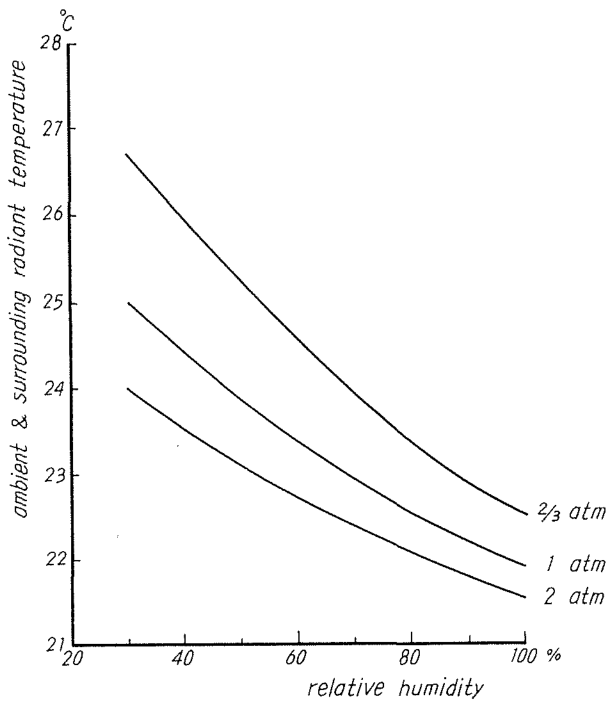

Fig. 10–14. Comfortable conditon and atmospheric pressure

As shown in Fig. 10–14 under the higher atmospheric pressure region, man feels slightly warmer, due to the reduction of the evaporative heat transfer coefficient.

10–5. Summary

The authors have proposed the Warmth Diagram for evaluating the condition of thermal environment or estimating the thermal sensation, and have shown its effectiveness by solving some examples.

Chapter 11. Comfort Detector

11-1. Model Man

In the existing methods of heating and air-conditioning control, the conditions of the thermal environments have been specified in terms of ambient temperature only, or of the temperature and humidity. However, considering the complexities of the thermal sensation the defects of the usual methods are apparent. For detecting the comfort sensation directly the authors propose the rational methods.

The heat balance equation, that induces the model skin temperature, can be realized on the following "model man".

(1) *The model man is the circular cylinder of* 30 cm *in diameter and* 90 cm in height, emitting the same rate of heat as man does, per unit time per unit surface area.

(2) 16 percent of the entire surface is completely saturated with water.

(3) The surface is covered with some material having the same heat and vapor resistance and emissivity as that of clothing.

The inner temperature of the model man indicates the model skin temperature directly.

Further, in an attempt to make a smaller scale model, the sensitivities to each of the thermal factors were found to vary independently, so scaling down may have caused the distortion.

11-2. Thermal Sensation Computer

By detecting each of the thermal factors as an electrical signal, the model skin temperature can be computed automatically.

From Eq. (7-5) the model skin temperature T_s is given by

$$T_s = \frac{\alpha_r \cdot T_r + \bar{\alpha}_c \cdot T_a + H}{\alpha_r + \bar{\alpha}_c} \qquad (11\text{-}1)$$

and from Eq. (2-2) we get

$$\alpha_c = m' \cdot V^m \qquad (11\text{-}2)$$

where V = air-movement

While, if the temperature of the black cylinder thermometer (a sort of globe thermometer) is known, the surrounding radiant temperature may be given by the equation

$$T_r = T_{B\cdot c} + \frac{\alpha_{c\cdot c}}{\alpha_{r\cdot c}}(T_{B\cdot c} - T_a) \qquad (11\text{-}3)$$

where $T_{B\cdot c}$ = temperature of the black cylinder thermometer

The above relations can be expressed by the following computing network.

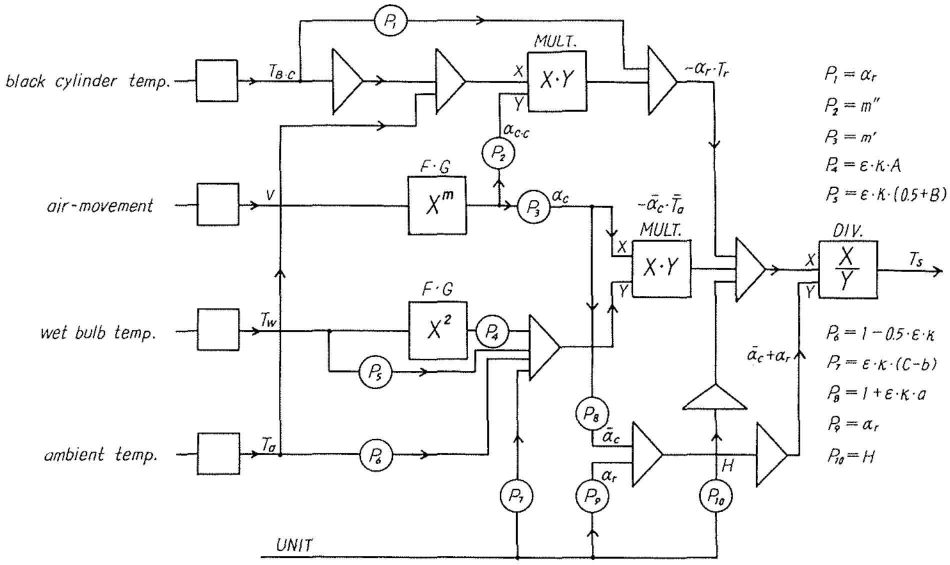

Fig. 11-1. Computing network for thermal sensation

11-3. Summary

Model skin temperature can be obtained by model man or by thermal sensation computer, and these devices may become the fundamentals for rational automatic control of heating and air-conditioning.

Appendix

The authors had presented some reports with respect to comfort indexes. These are summarized below.

A-1. Discomfort Index

The discomfort index (or U. S. Weather Bureau Temperature-Humidity Index) is one of the simple and comprehensible comfort indexes, and is given by the combination of ambient temperature and wet bulb temperature as

$$D.\,I = 0.72(T_a + T_w) + 40.6 \qquad \text{(a-1)}$$

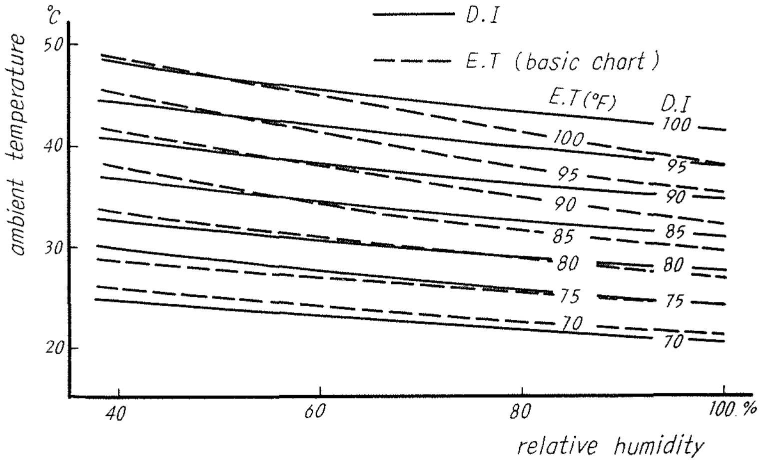

Fig. a-1. Relation between Discomfort Index and Effective Temperature

While, Fig. a-1 shows the relation between Discomfort Index and Effective Temperature, where the solid and broken lines nearly coincide under windless circumstances. From the above, we can perceive that the Discomfort Index is a simplified method to obtain the Effective Temperature under restricted conditions.

Setting aside its non-allowance for radiation, as Yaglou has pointed out[10] the Effective Temperature overestimates the influence of humidity, as shown in Fig. a-2.

As a result it may be concluded that in the presence of controled air motion, cooled or heated panel, significant error will be introduced, if

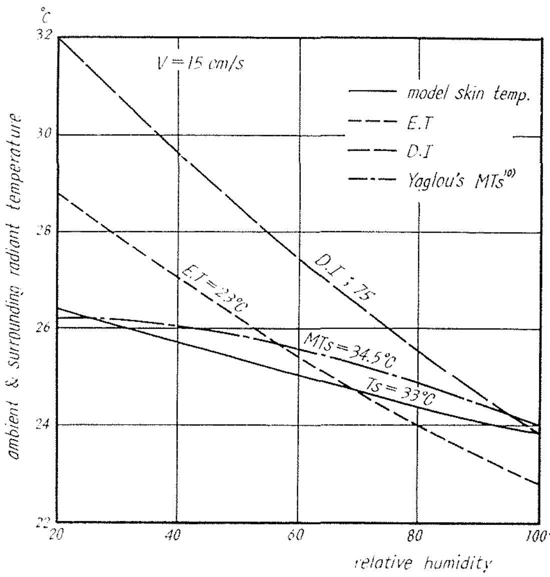

Fig. a-2. Comparison between Discomfort Index and the other Indexes the Discomfort Index is used for evaluating the thermal environment.

A-2. Corrected Effective Temperature Index

Vernon and his associate applied a radiation correction to Effective Temperature by using the globe thermometer in place of the dry-bulb temperature[11]. Further, in the United States the equivalent wet-bulb temperature, as shown in Fig. a-3, was proposed to replace the wet-bulb temperature[12].

The problem in the subject is whether a globe thermometer of 15 cm in diameter can evaluate the same radiation effect as that imposed on man. Fig. a-4 shows that the reading of each globe thermometer depends on its diameter.

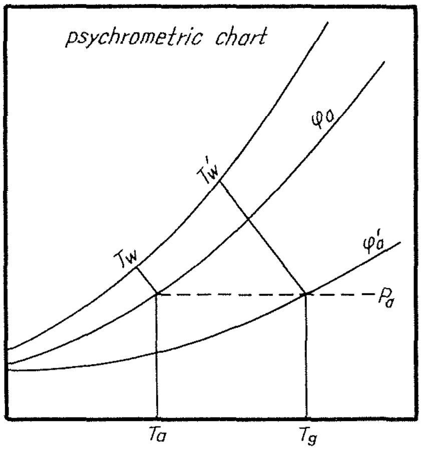

Fig. a-3.
Equivalent wet bulb temperature

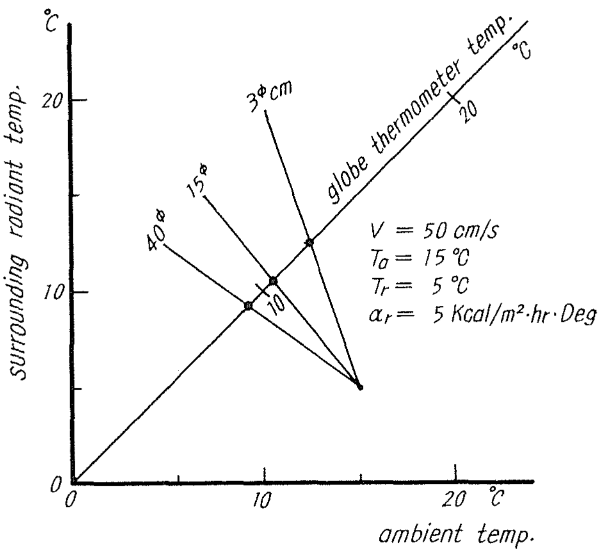

Fig. a-4. Reading of each globe thermometer

As mentioned in Chapter 5, the equivalent body of the human may be assumed to be a circular cylinder of 30 cm in diameter, and in the low air-velocity range, from Eqs. (2-2) and (2-3) we know the corresponding sphere of this cylinder being of 40 cm in diameter.

In other words the reading of a globe thermometer of 40 cm in diameter must be used for the proper radiation correction. The following example shows that the Corrected Effective Temperatures is improved by this modification.

Example A-1 A room is maintained at dbt: 25°C, wbt: 20°C, air-movement: 1 m/sec and the reading of the globe thermometer: 30°C.

Estimate E.T, C.E.T (15^{ϕ}) and C.E.T (40^{ϕ}) respectively.

Solution

We find $T_r = 38.2$°C and $T_g(40^{\phi}) = 31.7$°C by the similar manner on Eq. (11-3). The equivalent wet bulb temperatures are $T'_w(15^{\phi}) = 21.5$°C and $T'_w(40^{\phi}) = 22.0$°C, and the following result is obtained.

In spite of a radiation correction, it seems that the usual Corrected Effective Temperature can not evaluate the radiation effect properly. Besides, we must notice that the modification of this study is useless for correcting the deviation of Effective Temperature itself.

Table a-1. Comparison between E.T, C.E.T(15$^\phi$) and C.E.T(40$^\phi$)

	basic chart (°C)	normal chart (°C)
E.T	18.8	21.2
C.E.T(15$^\phi$)	22.7	24.5
C.E.T(40$^\phi$)	23.8	25.6

A-3. Heat Stress Index

Belding and Hatch proposed the Heat Stress Index which measured the magnitude of heat stress imposed on individuals exposed to different combinations of the four components of the thermal environment, (radiant and air temperature, air-motion and humidity) and operating at different levels of activity.[13)]

Apart from the question of approximations and assumptions in this index, the authors have a few points in question on the physical accuracy of each heat transfer coefficient.

The convective and evaporative heat transfer coefficients for human body are expressed by

$$(\alpha_c)_{H\cdot S\cdot I} = 6.95 \cdot V^{0.5} \qquad [\mathrm{kcal/m^2 \cdot hr \cdot Deg}] \qquad \text{(a–2)}$$

$$(\beta)_{H\cdot S\cdot I} = 11.7 \cdot V^{0.4} \qquad [\mathrm{kcal/m^2 \cdot hr \cdot mmHg}] \qquad \text{(a–3)}$$

where V = air-velocity [m/sec]

Further, in 1963, Hatch corrected these heat transfer coefficients by reexamining the original data.[14,15)]

$$(\alpha_c)_{Ha} = 6.35\,V^{0.6} \qquad \text{(a–4)}$$

$$(\beta)_{Ha} = 13.1\,V^{0.6} \qquad \text{(a–5)}$$

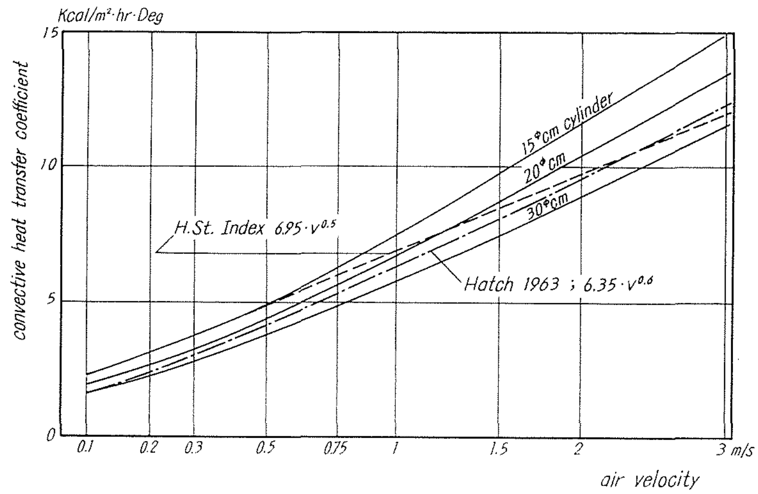

Fig. a-5. Relation between convective heat transfer coefficients

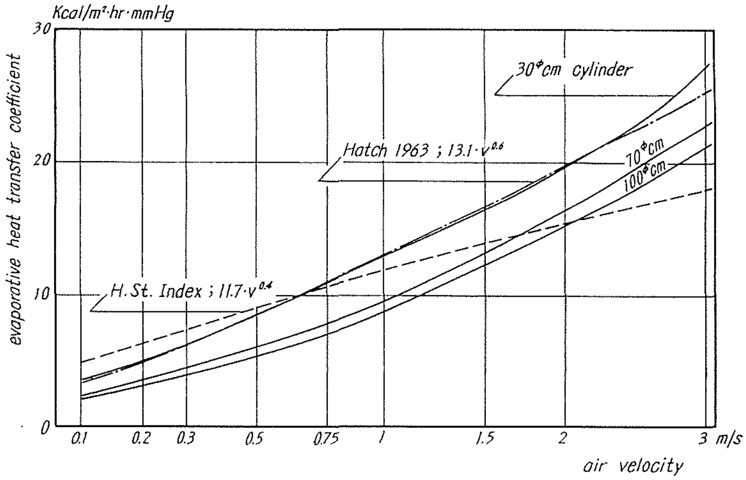

Fig. a-6. Relation between evaporative heat transfer coefficients

Figs. a–5 and a–6 show the relation between the above equations and each heat transfer coefficient of circular cylinders.

From the above we know that on Heat Stress Index the equivalent body to the human with respect to convective heat transfer is a cylinder of 15～25 cm in diameter and that with respect to evaporative heat transfer is a fairly large cylinder in higher air-velocity range. This may be the very cause for overestimating the effect of the air temperature and underestimating the effect of the humidity.

References

10) Yaglou, C. P.: A Method for Improving the Effective Temperature Index, ASHVE Trans. Vol. 53 (1947) p. 307.

11) Vernon, H. M. et al.: The Influence of the Humidity of the Air on Capacity for Work at High Temperature, J. of Hyg. London, Vol. 32 (1932) p. 431.

12) Am. J. of Public Health and Nation's Health Yearbook, Vol. 40 (1950) p. 143.

13) Belding, H. S. and Hatch, T. F.: Index for Evaluating Heat Stress in Terms of Resulting Physiological Strains, Heating, Piping & Air-Conditioning, (1955, August) p. 129.

14) Hatch, T. F.: Assessment of Heat Stress, Temperature, Its Measurement and Control in Science and Industry. Reinhold, New York (1963) p. 307.

15) Nelson, N. et al.: Thermal Exchange of Man at High Temperature, Am. J. of Physiol., Vol. 151, (1947, Dec.) p. 626.