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CONVECTIVE HEAT TRANSFER FROM HUMAN BODY (PART 1. A SIMULATION BY VERTICAL CYLINDER)

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Abstract

Convective heat transfer from human body was studied from a standpoint of heat transfer engineering.

By using a cylindrical heat source set up vertically in a wind tunnel, an experimental formula that predicted convective heat transfer coefficient which was applicable to both forced and natural convections was proposed. When air velocity is high, this formula agrees well with that proposed by Douglas and Churchill in the region of forced convection. By using this formula we derived another useful formula which determined convective heat transfer coefficient for human body as a function of air velocity, diameter of human body and temperature difference between the body and surrounding air. It was found that the coefficients increased rapidly with increasing air velocity. Furthermore, the coefficients decreased with increasing diameter of human body and their rate of decrease became smaller with the increase of diameter.

1. Introduction

Heat transfer from human body to the ambient air is one of the important problems in engineering, especially in a man-thermal environment system, and has been recently studied by many researchers from physiological and/or engineering standpoints. In the physiological studies[1-3], the heat transfer coefficients were directly evaluated by the medical experiments on living persons, the results were therefore considerably influenced by the physiological conditions and air movements caused by body action. In these investigations, the choice of the method to determine the correct heat transfer coefficient from which the above effects are removed is in itself a problem. In the studies based on heat transfer engineering[4-5], the coefficients were determined by the experimental studies using a cylindrical or plane heat source. These studies seem to be exact but have a problem for adapting to the practical human body. Thermal modelling using a heated full-scale manikin[6] is the most realistic method that takes advantage of the above two methods. However, there are some problems in dealing with the physiological phenomena, or the size of the model.

In this paper, from the engineering standpoint, convective heat transfer coefficients are experimentally examined and an experimental formula is obtained as functions of air velocity, surface temperature and size of cylindrical heat sources. Then, the formula is used to determine heat transfer from human body.

2. Experiments

By mounting a cylindrical heat source vertically in a wind tunnel and by measuring the temperature field, convective heat transfer coefficient was determined. Experiments were done under the condition of 20 ± 0.3°C and 65 ± 5% RH.

2.1 Wind tunnel

The schematic diagram of a wind tunnel used is shown in Fig. 1. The tunnel was open-circuit and air flow due to suction by a fan ⑤ was made uniform through a diffuser ①, a flow conditioner ② and a convergent nozzle ③. The suction flow

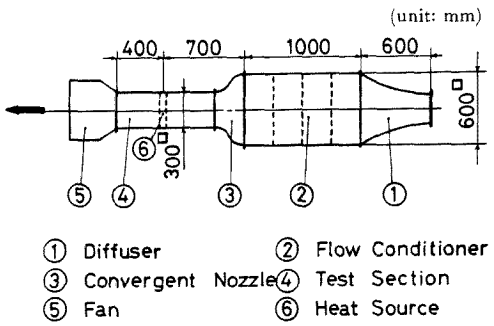


Fig. 1. Schematic diagram of wind tunnel

was used to prevent a temperature rise of air by the fan motor and to make air flow more uniform. Main flow velocity at the test section ④ was controllable by changing the rotational frequency of the fan motor and/or the cross sectional area of the entrance of the tunnel. The velocity was measured by a hot-wire anemometer, and the range was from 0.05 to 4.30 m/s. Inner wall of the tunnel was made of copper plates to prevent heat stagnation, and was paint-sprayed (emissivity: 0.996).

2.2 Heat source

Experiments were carried out by using two cylindrical heat sources, of which the outer diameters were 48 mm and 99 mm, respectively. As shown in Fig. 2, the heat source consists of five parts. The central part is the main heat source, and the upper and lower parts are the sub-heat-sources. Between the main source and the sub-sources, two temperature sensors are inserted for detecting the temperature difference. The main source whose lateral surface is coated with extremely thin paint (emissivity: 0.996) is heated uniformly by a built-in nichrome wire of 0.16 mm in diameter. The surface temperature of the main source is measured by copper-constantan thermocouples of 0.1 mm in diameter which are built in the main source.

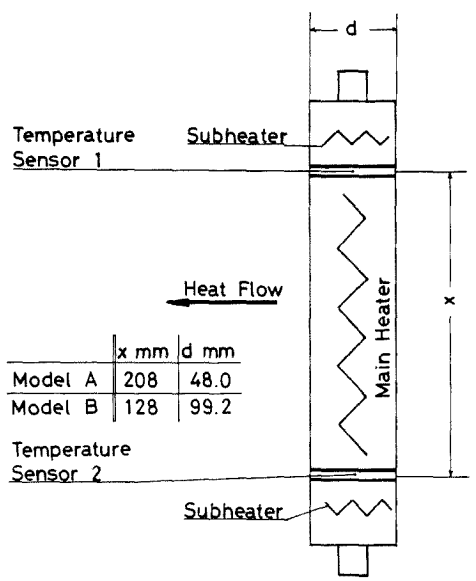


Fig. 2. Outline of cylindrical heat source

As shown in Fig. 3, the temperature sensor is made of an acrylic plate of 3 mm in thickness which was pasted with two copper plate of 1 mm in thickness. The sensors were jointed to electronic control devices. If each sensor detects that the temperature at the surface of the main source is higher than that of the subsource, the lamps mounted on the subsource are lighted up and the subsource is heated until the temperature difference between both the surfaces becomes smaller than a given value. In this way, the temperature difference between two copper plates of the sensor is controlled within 0.05°C. Accordingly, the top and bottom surfaces of the main source is considered to be insulated thermally, so it seems reasonable to consider that the whole amount of heat generated in the main source is liberated from the lateral surface.

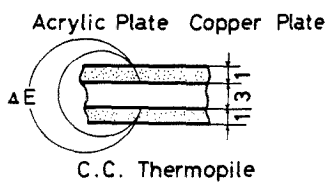


Fig. 3a. Construction of temperature sensor

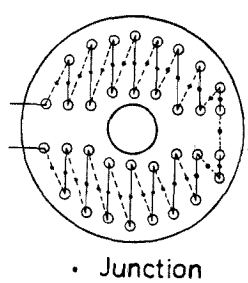


Fig. 3b. Positions of thermocouples in temperature sensor

2.3 Measurements

After heat and fluid flows became enough stable, following quantities were measured.

Amount of heat generated at the main source was determined by measuring voltage and electric current supplied to the main source.

Temperatures at the lateral surface of the main source and at the inner wall of the test section were measured by using a copper-constantan thermocouples of 0.1 mm in diameter. The measured points are shown in Fig. 4. The cold junctions are put in a jar filled with liquid paraffin, so the measured temperatures are the temperature rises from room temperature. Electromotive forces generated by the temperature difference between the hot and cold junctions were recorded on a mv-recorder (YEW TYPE 3046).

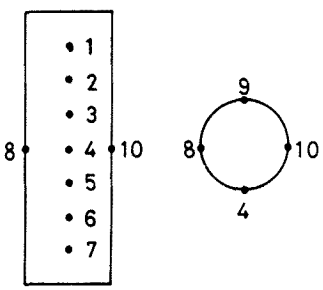


Fig. 4a. Position of thermocouples on main heat source

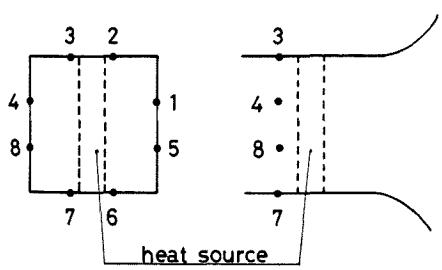


Fig. 4b. Position of thermocouples on inner wall of wind tunnel

The fluid temperature having wide fluctuations with time was measured as a time average value by using a thermocouple mounted in a brass rod of 6 mm in diameter and 100 mm in length.

2.4 Conditions

The ranges of the main flow velocity and the surface temperature of the main source in the experiments were about 0.05 ~ 3.7 m/s and 25 ~ 37°C, respectively. ca. 200 testing conditions were chosen by combining both values.

3. Results and discussion

Let us consider heat transfer problem in a model system shown in Fig. 5. Reynolds number Re, Nusselt number Nu and Grashof number Gr are defined as follows:

$$Re = UD/\nu, \tag{1}$$

$$Nu = hcD/\lambda, \tag{2}$$

$$Gr = D^3 g\beta(T_s - T_a)/\nu^2, \tag{3}$$

here, U = mean velocity of air flow, D = diameter of the main heat source, ν = kinetic viscosity of air, hc = convective heat transfer coefficient, λ = thermal conductivity of air, g = gravitational acceleration, β = temperature coefficient of volume expansion, T_s = surface temperature of the main heat source, and T_a = air temperature.

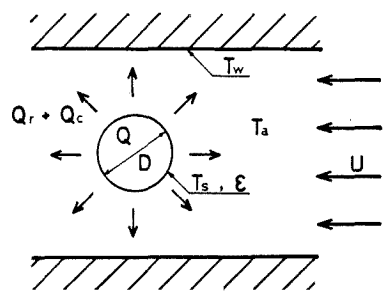


Fig. 5. Model system of heat transfer

In general, total amount of heat transfer Q is given as

$$Q = Q_c + Q_r, \tag{4}$$

here, Q_c and Q_r are total amounts of heat transfers by convection and radiation, respectively, and are given as follows:

$$Q_c = hc(T_s - T_a), \tag{5}$$

$$Q_r = \varepsilon Co\,[(T_s/100)^4 - (T_w/100)^4], \tag{6}$$

here, ε (=0.996) is emissivity of heat source, Co is radiation coefficient of black body and T_w is wall temperature of the wind tunnel. From equations (4), (5) and (6), following equation is obtained,

$$hc = \frac{Q}{T_s - T_a} - \frac{Co\cdot\varepsilon}{T_s - T_a}\left\{\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right\} \tag{7}$$

Then, hc can be determined by measuring Q, T_s, T_a and T_w.

3.1 Effect of the tunnel width on flow around the cylindrical heat source and the correction of main flow velocity

In our experiments, a relation between the main flow velocity and voltage supplied to the fan motor was determined before setting up the heat source in the wind tunnel. Therefore, Reynolds number, Re corresponding to the velocity during tests was obtained by measuring the voltage. When the width of the wind tunnel is not large enough as compared with the source size, the existence of the source and the tunnel wall gives a considerable effect on the fluid flow and the heat flow. This effect is called as the solid blockage effect. It is necessary to correct the velocity by considering this effect because the tunnel width is not so large as compared with the source size in our experiments. There are many experimental or theoretical formulae to determine the main flow velocity by considering the blockage effect. Typical formulae are as follows:

1. $\frac{U_\infty^*}{U_\infty}=(1+D/L)^{5/4}$ (Robinson et al.[7]), (8)

2. $\frac{U_\infty^*}{U_\infty}=1-\frac{\pi D}{4L}$ (Vliet and Leppert[7]), (9)

3. $\frac{U_\infty^*}{U_\infty}=1+\frac{\pi^2 D^2}{12L^2}+\frac{C_D D}{4L}$ (Pope[7]), (10)

4. Experimental (Perkins and Leppert[7]), (11)

5. $\frac{U_\infty^*}{U_\infty}=1+\frac{\pi^2 D^2}{12L^2}+\frac{C_D D}{4L}\left(1+2.83\frac{D^2}{L^2}\right)$ (Hiwada et al.[7]), (12)

here, U_∞^* and U_∞ denote corrected main flow velocity and experimental main flow velocity, respectively and C_D, the drag coefficient is 1.17. These relations are shown in Fig. 6. However, it is not clear which formula is reasonable, so we defined correct coefficients of main flow velocity U_∞^*/U_∞ using the ratio Re/Re', where Re is Reynolds number obtained by substituting experimental Nusselt number into following formula that was proposed by Douglas and Churchill for the case of heat transfer in forced convection from uniformly heated cylinder[8],

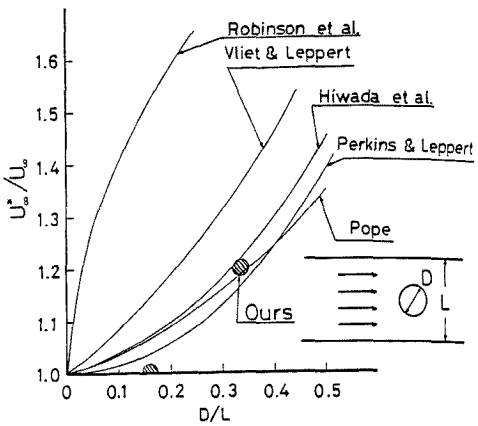


Fig. 6. Correction factor for main flow velocity as a function of D/L

$$Nu = 0.43 + 0.48\sqrt{Re} \quad (2<Re\leqq 500), \quad (13)_1$$

$$Nu = 0.46\sqrt{Re} + 0.00128\,Re \quad (500<Re\leqq 25000), \quad (13)_2$$

and Re' is experimental Reynolds number. These results are also shown in Fig. 6. Hereafter, corrected Reynolds number Re^* based on U_∞^* is used.

3.2 Derivation of experimental formula as a function of forced and natural convections

Relation between Nu and Re obtained is shown in Fig. 7, where, ○ and △ are the corrected experimental results and the solid line denotes the empirical formula (13). The experimental results agree well with the solid line in a large Re^* region, but do not in the region $Re^* \leqq 3000$ because the effect of natural convection becomes large. Fig. 8 which is a magnified figure of Fig. 7 shows that the effect of natural convection increases with increasing Grashof number Gr. This fact indicates that Nu depends not only on Re but also on Gr. Therefore, we will here consider that Nu is a function of Re^* and Gr, that is,

$$Nu = F(Gr, Re^*).$$

Now, let us rearrange the experimental data in Fig. 7 by using a nondimensional parameter Gr/Re^{*2} which means a ratio of buoyancy to inertia force[9]. The rearranged results are shown in Fig. 9, where the ordinate denotes a ratio of Nu to Nu_f, where Nu includes the effect of both

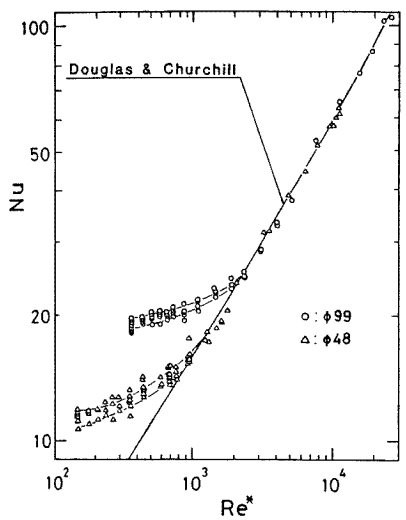


Fig. 7. Relation between Nusselt number and Reynolds number

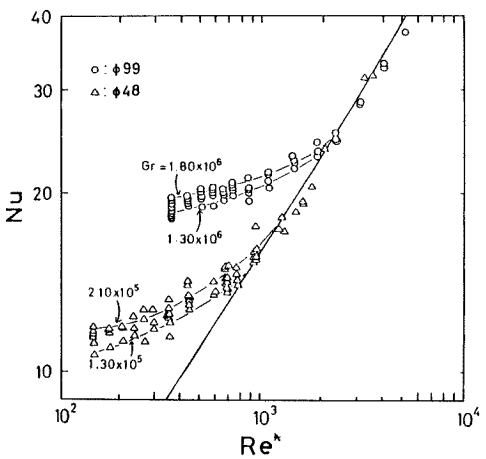


Fig. 8. Magnified figure of figure 7

natural and forced convections and Nu_f the effect of forced convection only. As seen in Fig. 9, we can express the experimental formula by a relation

$$\frac{Nu}{Nu_f} = \frac{1}{3.32/(Gr/Re^{*2})+0.779} + 1, \qquad (14)$$

which was obtained by the least squares method and is shown in Fig. 9 as the solid line.

3.3 Application to heat transfer from human body

Determining formula for convective heat transfer coefficient from the lateral surface of a cylindrical

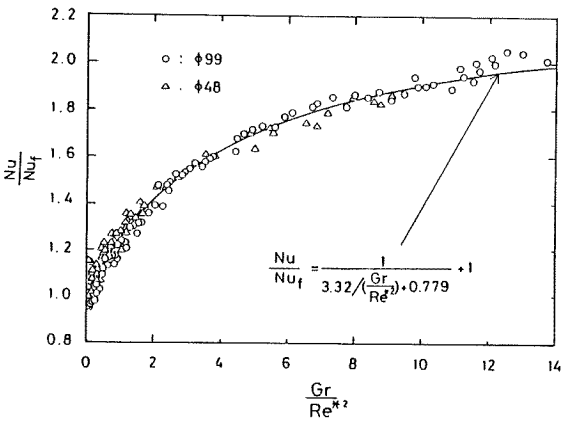


Fig. 9. Nu/Nu_f as a function of Gr/Re^{*2}

heat source is given as follows by transforming equation (14):

$$hc = (0.46\sqrt{U/\nu D} + 0.00128U/\nu)\lambda \cdot \left(\frac{1}{3.78U^2/(D\cdot\beta\cdot g\cdot \Delta T)+0.745}+1\right). \qquad (15)$$

Equation (15) includes the effect of both forced and natural convections.

Now, we will consider heat transfer from human body by utilizing equation (15). Physical properties of air at 15°C (λ=0.0218 kcal/mh°C, ν=0.151 $\times 10^{-4}$ m^2/s, β=0.347$\times 10^{-2}$ 1/K) are used for calculation, because the temperature range in general condition is about 0~30°C and in this range the change of these values are not so large. By substituting these values into equation (15), following equation is obtained.

$$hc = (2.58\sqrt{U/D} + 1.85U) \cdot \left[\frac{1}{97.6U^2/(D\cdot \Delta T)+0.779}+1\right]. \qquad (16)$$

here, $\Delta T (\equiv T_s - T_a)$ and D are temperature difference between human body and surrounding air and diameter of human body, respectively and a notation U is used instead of U_∞^*. Therefore, equation (16) is an experimental formula that denotes the convective heat transfer coefficient from human body. As air velocity U becomes large, the value in the brackets approaches to 1 and equation (16) agrees with equation (13) proposed by Douglas and Churchill.

3.3.1 Effect of temperature difference ΔT between human body and surrounding air

A typical example of effect of ΔT on the relation between hc and U at a fixed D (=0.2 m)

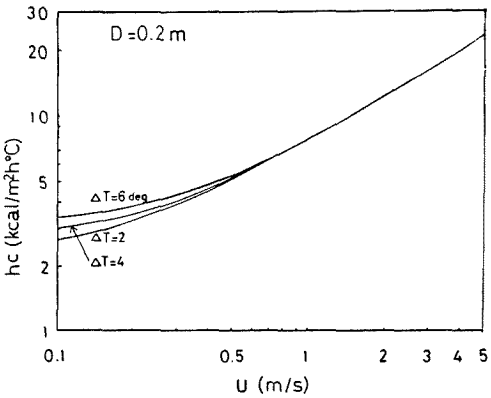


Fig. 10. Effect of ΔT on the relations between hc and U at a fixed D

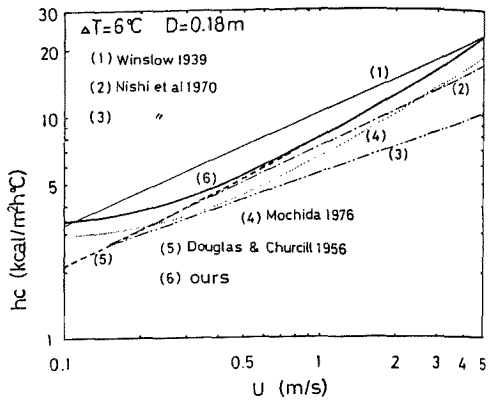


Fig. 12. Typical formulae proposed hitherto and our experimental formula for convective heat transfer

is shown in Fig. 10, where values of ΔT(=6, 4, 2 deg) are chosen so as to be felt pleasant by human body. It is clear that hc increases with increasing U, and that the effect of ΔT on hc becomes small with increasing U and negligibly small for U larger than 0.8 m/s.

3.3.2 Effect of diameter of cylindrical heat source

Relations between hc and U at a fixed ΔT(=4 deg) are shown in Fig. 11. This figure shows that hc decreases with increasing D and the rate of decrease becomes smaller.

In Fig. 12 are shown our experimental formula and typical formulae proposed hitherto for convective heat transfer coefficient. It is clear that our formula agrees almost with the others.

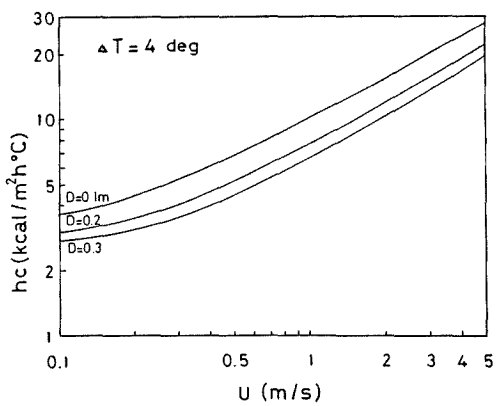


Fig. 11. Effect of D on the relations between hc and U at a fixed ΔT

3.3.3 The ranges of Re number and Gr number

In our experiments, the ranges of Re and Gr numbers are $1.5\times10^2 \leqq Re \leqq 3.2\times10^4$ and $1.3\times10^5 \leqq Gr \leqq 2.5\times10^6$, respectively. Human body may be considered to consist of four important parts, i.e. head, body and upper and lower limbs. Let us consider that the representative diameters of these parts are 0.20, 0.25, 0.10 and 0.05 m, respectively. By using these values as the characteristic lengths L, the air velocity range and the range of temperature difference calculated from the above experimental data are listed in Table 1. The upper limit of the velocity for $L=0.25$ m in Table 1 is 1.9 m/s. However, it is possible to adapt the experimental result of Douglas and Churchill for the upper limit of the range, so the value becomes 15 m/s.

Next, it is a problem that the upper limit of the value of temperature difference of for $L=0.25$ m or 0.20 m is very small compared to a reasonable value (as ca. 6 deg for naked body). This problem may disappear by the following reason. Our data in the range of very low Re number regarded as

Table 1. Ranges of wind velocity and temperature difference obtained from the experiments

Characteristic length L (m)	Range of U (m/s)	Range of ΔT (deg)
0.25	0.09 – 1.9	0.05 – 1.1
0.20	0.11 – 2.4	0.11 – 2.1
0.10	0.23 – 4.8	0.87 – 16.8
0.05	0.45 – 9.7	6.97 – 134

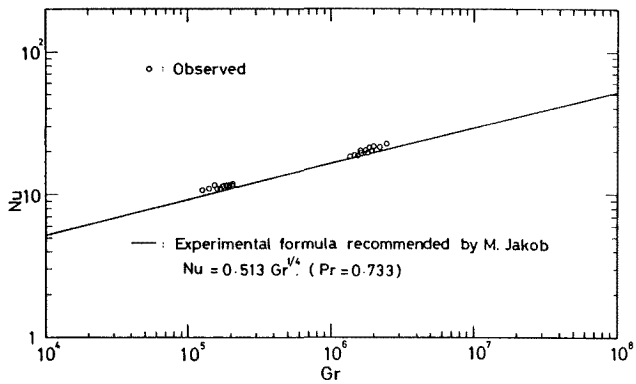


Fig. 13. Experimental relation between Nu number and Gr number in natural convection

natural convection shown in Fig. 13 agree well with other researcher's results[10], which are heat transfer from different surfaces to fluids by laminar natural convection. So, it may be reasonable to take 10^8 as the upper limit of Gr number. Hence the upper limit of the temperature difference for $L=0.25$ m is calculated 43 deg.

As described above, equation 16 can be utilized under ordinary conditions for human body.

4. Conclusions

By setting up a cylindrical heat source in a wind tunnel vertically, an experimental formula was proposed, which determined the heat transfer coefficient from the lateral surface of the heat source including the effect of not only forced convection but also natural convection. When air velocity is high, this formula agrees with that proposed by Douglas and Churchill in the region of forced convection. By using this formula, we derived another useful formula, which determined convective heat transfer coefficient for human body as a function of air velocity U, diameter of human body D and temperature difference ΔT. It was found from this formula that the coefficient increased rapidly with increasing U and that the coefficient decreased with increasing D and the rate of decrease became smaller.

A heat transfer from a clothed cylinder to air will be reported elsewhere.

Reference

1) C.-E. A. Winslow, A. P. Gagge and L. P. Herrington; *Am. J. Physiol.*, **127**, 505 (1939)
2) Y. Nishi and A. P. Gagge; *J. Appl. Physiol.*, **29**, 830 (1970)
3) S. Ohno and J. Kurasaki; *J. Home Econo. Japan*, **27**, 534 (1976)
4) T. Mochida; *Nihon Kenchiku Gakkai*, **258**, 63 (1977)
5) J. L. Spencer-Smith; *Clothing Res. J.*, **5**, 3 (1977)
6) K. Mihira and S. Ohno; *J. Home Econo. Japan*, **28**, 216 (1977)
7) M. Hiwada, K. Niwa, I. Mabuchi and M. Kumada; *JSME*, **42**, 2481 (1976)
8) W. J. M. Douglas and S. W. Churchill; *Chem. Eng. Progr. Symp. Series*, **52**, 23 (1956)
9) P. H. Oosthizen and S. Madan; *J. Heat Trans. ASME*, **92**, 194 (1970)
10) M. Jakob; "Heat Transfer" vol. 1, p. 530, John Wiley, New York (1949)

人体からの対流熱伝達
(第1報 鉛直円柱によるシミュレーション)

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熱工学的見地から，人体からの対流熱伝達率を調べた。風洞内に設置された円筒状熱源を用いて，その対流熱伝達率の実験式を得た。その式は，気流が速い時には，強制対流に対するDouglasとChurchillの式に一致する。その式を用いて，我々は気流速度，人体の太さ及び人体と周囲流体との間の温度差を関数とする人体からの対流熱伝達率を決定するさらに有益な式を導いた。その熱伝達率は気流速度と共に急速に増大する，また，人体の太さが増加するに従い減少するがその減少の割合は次第に小さくなるという知見も得た。