Heat Transfer on Cylinder Covered with Close-fitting Fabrics (Part III, Distribution of Local Heat Transfer Coefficients)*

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The preceding two papers proposed a theoretical formulation of wind penetration and heat transmission through a fabric layer fitting closely to a heated cylinder which is a simple model of clothing system. This paper describes the heat transfer experiments of cylinders covered with 'knicker hose', 'camel underwear', 'body tights' and 'nylon stocking' respectively. The data agree well with the theoretical predictions. Phenomenological features are explained; three regimes of penetration, i.e. front 'flow in' region, middle 'flow out' region, and rear 'stagnate' region. Heat transfer is augmented in the front region, and its behaviour depends primarily on the fabric thickness and the state of contact with the cylinder.

Key Words: Convective Heat Transfer, Wind Penetration in Clothing, Thermal Environment

1. Introduction

Heat transfer on a cylinder covered with a close-fitting fabric has been studied in cross air-flow. Part of the incident air-flow penetrates the fabric, and makes its thermal resistance small. This phenomenon on our clothes is often experienced in windy weather.

On this subject, two papers were published. The first one (1) describes how the wind causes penetration, and proposes a characteristic number for the contact between fabric and cylinder. The second one (2) gives an analysis of heat transfer on the stagnation point. A theory was formulated of the heat transfer on a covered cylinder. The heat transfer coefficient was found to be a function of wind velocity, some properties of air, and such factors as thermal conductivity, permeability and thickness of fabric layer, and characteristic number, α^* , which describes the condition of contact between fabric and cylinder.

α^* has essentially a great effect on heat transfer. Therefore all the published experimental data except ours didn't imply the absolute, physical measure in heat transfer of fabric covering, because the state of contact remained uncertain; they were only used as figures of merit of arbitrary unit.

In the paper, experiments have been conducted with four sorts of real clothings which are closely fitting to a human body, and the data obtained are compared with the theoretical results around the cylinder.

Nomenclature

- C : Fourier coefficient for the pressure distribution around the outer circumference of fabric layer.
- D : diameter of heated cylinder.
- h : local coefficient of overall heat transfer from cylinder to external air, $= q_w / (T_w - T_\infty)$.
- h_a : heat transfer coefficient on a bare cylinder.
- k : permeability of fabric layer, $k^* = k/D^2$.
- N_u : local Nusselt number, $= hD/\lambda_a$.
- $P(r, \theta)$: pressure in the fabric layer, $P^* = 2(P - P_\infty) / (\rho U_\infty^2)$.
- P_∞ : pressure far from the cylinder.
- Pr : Prandtl number of air, $Pr = \mu_a / (\lambda_a / \rho_a)$.
- q_w : heat flux density at heated cylinder surface.
- Re : Reynolds number, $= DU_\infty / \nu$.
- T : temperature in fabric layer, $T^* = T\lambda_a / (q_w D)$.
- T_w : temperature at heated cylinder surface.
- T_∞ : temperature far from the cylinder.
- U_∞ : velocity of incident air flow.
- v_r : radial component of penetration flow rate, $v_r^* = 2v_r \nu D / (k U_\infty^2)$.
- v_θ : azimuthal component of penetration flow rate, $v_\theta^* = 2v_\theta \nu D / (k U_\infty^2)$.
- α^* : characteristic number of contact layer.
- δ : thickness of fabric layer, $\delta^* = \delta/D$.
- ϵ : index number of thermal resistance of fabric layer.
- θ : angle from stagnation point.
- λ_a : thermal conductivity of air.
- λ_m : thermal conductivity of fabric layer.

* Received 22nd December, 1981.

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ρ : density of air.

2. Experiment

2.1 Heated cylinder

Fig.1 shows the structure of a heated cylinder. A thin layer of rubber heater is set around a polyurethane foam insulator, so that the heat flux density at the cylinder surface is uniform. The cylinder is of 82mm diameter, 200mm length of heated part, and 380mm total length. Thermocouple junctions at each 20° point on the surface indicate the temperature, T_w . The cylinder has two pressure taps, one for the pressure at the cylinder surface or at the inner circumference of fabric layer, and the other with an 'L' shaped tube of 1mm diameter for the outer circumference pressure.

The cylinder is put in a test section of wind tunnel. The cylinder is rather large for the test section of 253mm square; the blockage ratio is 0.32. The blockage ratio increases with an increase of fabric layer thickness, and reaches 0.35 for the thickest layer. Therefore, the confinement by the conduit walls cannot be neglected. This is a tedious factor to understand the phenomena analogously, but not a serious disadvantage in applying our formulation.

2.2 Fabric samples

We chose four samples among clothings on the market, which closely fit the human body:

- (1) Knicker hose: knitted stocking hose made of wool yarn,
- (2) Camel: knitted underwear for mountain climbing in winter season, made of wool like camel hair,
- (3) Tights: body tights for skiing, made of stretch yarn,
- (4) Stocking: seamless stoking, made of nylon stretch yarn.

Table 1 shows thickness, permeabilities, thermal conductivities, and characteristic numbers of contact layers in these samples. The significant digits for these values of soft goods are not so many, especially for the thickness. The thickness measured while a sample is fitting to the cylinder is a little larger, at best, than a single significant digit. Permeabilities were measured for samples fitted to a net cylinder which has diameter equal to that of a heated one with the same consideration as in the preceding paper. The permeability of stocking could not be measured, because the pressure difference across the

Fig.1 Heated cylinder

layer was too small to be measured by Betz manometer.

Thermal conductivities were measured by the transient hot wire method⁽³⁾. The application of the method to fabrics has not been authorized yet, though the method is found simpler and more convenient than others we have tried.

Characteristic numbers of contact layer are given in the next section.

2.3 Pressure distribution and characteristic number of contact layer

Pressure distributions were measured at various wind velocities around each clothing sample fitting to the cylinder.

Fig.2 illustrates the distributions around the outer circumference of knicker hose at four levels of Reynolds number. The differences between them are noticed in the region $\theta > 60^\circ$, but the difference is about equal to an error in measuring the dynamic pressure of wind. Therefore, the pressure distribution is rather unchanging in spite of Re , and, from Newton's law of resistance, this assumption is reasonable in the scope of our experiment ($Re \sim 10^4$).

The pressure distributions of Fig.2 and of other materials are the same. Then the mean distribution which is a solid line in Fig.2 is assumed to be representative of all materials. Penetration flows v_{r*} and $v_{\theta*}$, and then the heat transfer coefficients in the next chapter are calculated from this representative distribution.

Around the bare cylinder, the pressure distribution has a minor valley at $\theta = 70^\circ$ of minimum pressure, and the pressure behind the valley is slightly higher than

Table 1 Some pertinent data on clothing samples

clothing	thickness [mm] (δ^*)	permeability [m^2] (k^*)	thermal conductivity [W/mK] (Prm)	characteristic number of contact layer (α^*)
knicker hose	3.5 (0.043)	$3.3/10^{10}$ ($4.9/10^8$)	0.045 (0.41)	(0.022-0.04)
camel	2.0 (0.024)	$4.1/10^{10}$ ($6.1/10^8$)	0.069 (0.27)	(0.009-0.023)
tights	1.0 (0.012)	$9.8/10^{11}$ ($1.5/10^8$)	0.063 (0.29)	(0:estimate)
stocking	0.3 (0.0037)	unmeasurable	0.042 (0.44)	(0:estimate)

Values in parentheses are dimensionless. Prm is a dimensionless reciprocal of λm .

Fig. 2 Pressure distribution around the outer circumference of knicker hose

Fig. 3 Pressure distribution around the outer and the inner circumference of knicker hose

that of Fig. 2.

At the front side $\theta < 60^\circ$, the distributions in bared cylinder and covered one are just the same.

Fig. 3 shows the outer and the inner distributions of pressures around the knicker hose. A difference in the distributions between the outer and inner is noticed at two sites; at the stagnation point and at 60° . However the difference is much less than that of the preceding paper⁽¹⁾, because δ^* of samples is much less than in the preceding paper.

Characteristic number, α^* , of contact layer are determined by the comparison between the outer pressure and the inner one, with the aid of the formula of pressure distribution described in the preceding paper. The formula is the following relation between the pressure difference, δ^* and α^*

$$P^*(0.5 + \delta^*, 0) - P^*(0.5, 0) = \sum_{n=0}^{\infty} c_n \left\{ 1 - \frac{2}{(1+2n\alpha^*)(1+2\delta^*)^n + (1-2n\alpha^*)(1+2\delta^*)^{-n}} \right\} \quad (1)$$

which is illustrated in Fig. 4.

Fig. 4 Differential pressure across fabric layer at the stagnation point

Fig. 5 Scattering of measured characteristic number of contact layer

Values of α^* obtained for each sample are tabulated in Table 1. They scatter as in Fig. 5. Since significant digit of the measured pressure difference is about one, the value of α^* scatters nearly $\pm 50\%$ around its value. However, the reproducibility of α^* between runs was within this $\pm 50\%$, and fortunately a difference of α^* is clearly distinguished between different samples.

The pressure differences across the fabric layer of tights and stockings were unable to be measured, because of their small thickness in addition to their small values of α^* . Then α^* of the tights and the stockings were assumed equal to zero. As the hoop stress of these samples is large, the assumption is natural from the effect of stress⁽²⁾ on α^* as described in the preceding paper.

2.4 Distribution of heat transfer coefficients

Local heat transfer coefficient, h , is calculated from the form $h = q_w / (T_w - T_\infty)$, where q_w is the heat flux density at the cylinder surface, T_w is the temperature at the cylinder surface, and T_∞ is the temperature of incident flow. Heat transfer coefficient is made dimensionless as $Nu = hD / \lambda_a$.

Nu of bare cylinder is shown in Fig. 6,

Fig. 6 Distribution of heat transfer coefficients around the bare cylinder

normalized by the value at stagnation point which is described as $Nu=1.08/\sqrt{Re}$. Normalized distribution is almost constant over the Re range from 9×10^3 to 7×10^4 . Then the local heat transfer coefficient on the bare cylinder can be estimated in the form

$$Nu_s = 1.08 \sqrt{Re} \cdot g(\theta) \quad (2)$$

where $g(\theta)$ is represented in terms of Fourier series of the first 8 partial sums.

Nu on the rear part seems to be a little smaller than that of other reports; e.g. the result of Hiwada et al.⁽⁴⁾. We think that this is due to the position of cylinder, since it is as close to the outlet of wind tunnel as 110mm.

Heat transfer distributions in all cylinders covered with clothings are shown in Figs. 7 to 10. Some features are described below.

Maximum points of heat transfer coefficient appear at the front stagnation point for every fabric, for all Re , except for a single case of stocking at maximum

Re . Minimum coefficients appear at a point on the circumference 80° to 90° . Coefficients in the rear side region are smaller than on the front.

The knicker hose and the camel show small heat transfer coefficients uniformly around the cylinder, when Re is small, i.e. they are cosy clothing. However, when Re is large, coefficients on the front side are remarkably large.

The other group of tights and stocking changes heat transfer coefficient moderately with wind, they show the same shape of distribution for almost all Reynolds numbers experienced.

Heat transfer coefficients increase with an increase of Re . Their behaviours at the front stagnation point and at the rear one are illustrated in Fig. 11. The contrast between the above two groups is remarkable.

These phenomena are mathematically discussed in the next chapter.

Fig. 7 Overall heat transfer coefficient on the cylinder covered with knicker hose

Fig. 9 Overall heat transfer coefficient for 'tights' covering

Fig. 8 Overall heat transfer coefficient for 'camel' covering

Fig. 10 Overall heat transfer coefficient for 'stocking' covering

Fig.11 Overall heat transfer coefficients at the front stagnation point and at the rear stagnation point

3. Theoretical calculation and discussion

3.1 Formulation

The preceding paper (2) introduced a heat transfer equation in the fabric layer and a formula of penetration flow rate. The dimensionless equation for heat transfer becomes

$$\frac{\partial^2 T^*}{\partial r^{*2}} + \frac{1}{r^*} \frac{\partial T^*}{\partial r^*} + \frac{1}{r^{*2}} \frac{\partial^2 T^*}{\partial \theta^2} = \frac{k^* Pr_m R_0^2}{2} \left(v_r^* \frac{\partial T^*}{\partial r^*} + v_\theta^* \frac{\partial T^*}{\partial \theta} \right) \quad (3)$$

where T^* : temperature, r^* and θ : cylindrical coordinates in Fig.12, k^* : permeability, Pr_m : reciprocal of thermal conductivity, Re : wind velocity. v_r^* and v_θ^* are components of penetration flow rate in respective directions induced by the pressure difference around the outer circumference:

$$v_r^* = - \sum_{n=1}^{\infty} \frac{(1+2na^*)(2r^*)^{n-1} - (1-2na^*)(2r^*)^{-n-1}}{(1+2na^*)(1+2\delta^*)^n + (1-2na^*)(1+2\delta^*)^{-n}} \times 2nc_n^* \cos(n\theta) \quad (4)$$

$$v_\theta^* = \sum_{n=1}^{\infty} \frac{(1+2na^*)(2r^*)^{n-1} + (1-2na^*)(2r^*)^{-n-1}}{(1+2na^*)(1+2\delta^*)^n + (1-2na^*)(1+2\delta^*)^{-n}} \times 2nc_n^* \sin(n\theta) \quad (5)$$

where Cn^* are Fourier coefficients for pressure distribution, δ^* is the thickness, and α^* is the characteristic number of contact.

Boundary conditions accompanying the equation are

$$\frac{\partial T^*}{\partial r^*}(0.5, \theta) = -1 \quad (6)$$

: uniform heat flux density on the cylinder surface, and

$$\frac{\partial T^*}{\partial r^*}(0.5 + \delta^*, \theta) = - \frac{Dh_A}{\lambda_m} \cdot T^*(0.5 + \delta^*, \theta) \quad (7)$$

: heat transfer coefficient h_A at the outer circumference.

The formulation has been revised in two points:

- (1) completion of the boundary value problem,
- (2) h_A acts on the conductive heat flux at the circumference, which is not equal to q_w .

The value of h_A is estimated from the formula (2), with substitution of $D+2\delta$ into diameter. In the strict sense, formula (2) does not estimate a correct value on the fabric surface, because the existence of a penetrating flow across the fabric surface does not coincide with the condition of bare cylinder. But the penetrating flow rate is not so large as one hundredth of incident flow rate. Furthermore pressure distributions on the covered cylinder and on the bare one are so similar that flow patterns around both cylinders are to be similar. Then the estimation by the formula (2) may be acceptable.

The blockage ratio is another problem. But it varies from 0.32 of bare to 0.35 of knicker hose at most, then the correction is negligible.

Equation (3) is numerically calculated under the boundary conditions (6) and (7) through iteration.

3.2 Temperature distribution

Fig.12 shows temperature distributions in the layer of knicker hose. The right hand side is for a low wind velocity, and the left side for a high wind velocity.

In the case of low wind velocities (to be exact, in the case of small penetration), isothermal lines are a group of concentric circles slightly distorted. They are swelling out at about 90° , corresponding to the minimum value of h_A around the circumference.

In the case of high velocities (strictly, in the case of large penetration), isotherms make a complicated contour map.

Fig.12 Dimensionless isotherms in the fabric layer of knicker hose; theoretical results for $\alpha^*=0.04$

Fig.13 Distribution of penetration flow rates along $r^*=0.5$

A sharp ridge extends radially in the sector of 70° to 80° , followed by a plateau of rear side. The front side of the ridge is a lowland of nearly zero level. These regions correspond to the penetration regimes.

Fig.13 shows the penetration flow rate distribution in a fabric layer, of which the values of δ^* and α^* are assumed ones and not real ones to give the same order of magnitudes of v_r^* and v_θ^* . However, the feature characterizes all samples.

It is noticed that the nature of v_r^* is divided into three regimes; (A) 'flow in' front region 0° to 40° , (B) 'flow out' middle region 40° to 100° , (C) 'stagnate' rear region 100° to 180° . The air flows into the fabric covering in the front region, streams down through fabric at the rate v_θ^* and through the contact layer, and flows out at the middle region. There is no flow in the rear region. 'Flow in' regime augments heat transfer, while 'flow out' regime keeps good insulation. Then the temperature is low in the front region, high in the middle region, and moderate in the rear region. In the case of large Re , penetration is so much that the temperature corresponds clearly to these regimes as shown in the left hand side of Fig.12.

3.3 Local heat transfer coefficient

Temperature distributions decide heat transfer coefficients around the cylinder, which are the lines in Figs.7 to 10 at three levels of Re .

The heat transfer coefficient increases with an increase of Re , and the fabric samples are classified into two groups according to the behaviour of heat transfer coefficient increase rate:

(1) knicker hose and camel; weak group,
(2) tights and stocking; strong group.
The former is weak and the latter is strong against wind, and the difference between the two groups is apparent from the heat transfer coefficient at the front stagnation point as shown in Fig.11.

The distribution of heat transfer coefficients is just opposite to the temperature distribution; a peak in front part, a valley in the middle, and plateau in the rear. But the valley is not so deep as the theory predicts. There seem to be two reasons; (1) heat generated in the rubber

Fig.14 Thermal resistance of fabric layer in terms of index number ϵ VS the change in α^* and δ^*

heater diffuses to front and to rear, i.e. q_w is less than nominal, so that the temperature is lower there than the predicted, (2) q_w is consumed immediately to raise the enthalpy of air flow in the contact layer, then the heat transfer is made larger than the theory.

It is also predicted by the theory that the latter strong group is not affected much by the wind, because δ^* and α^* are small, not because their permeabilities are small. The 'toughness' of thin layer was described in the preceding paper. δ^* of present samples are so little that the effects are extremely remarkable. Fig.14 shows the thermal resistance of fabric itself; the solid line indicates the resistance of tights relative to the pure conductive resistance. The other lines correspond to an imaginary case of much larger δ^* or of much larger α^* fabrics; their resistances easily decrease with an increase of wind velocity. The region where they hold full resistance can be estimated from the value of modified Reynolds number of abscissa. The tights keeps the full resistance when $Re/\sqrt{k^*Prm} < 5$, that is $Re < 10^5$ from the value of k^* and Prm . The maximum wind of our experiment has no effect yet.

δ^* of the stocking is much smaller, and it is more free from the effect of penetration. Then k^* may be assumed to be zero for stocking if its permeability is

Fig.15 Comparison between samples in terms of heat transfer coefficient at the front stagnation point

Fig.16 Mean overall heat transfer coefficients

not extremely large. This assumption shows that it is not a serious problem whether Darcy's law holds in the layer of very small thickness or does not hold.

The fabric layer resistance plays an important role in the overall resistance. Since the knicker hose and the camel easily lose their resistance at high Reynolds numbers, the tights gets the largest overall thermal resistance there as shown in Fig.15.

The vagueness owing to the scatter of α^* is calculated from its maximum and minimum values, and illustrated by two lines for the knicker hose and the camel in Fig. 11. The deviation owing to the scatter of α^* is not larger than that owing to the measured error of other factors and/or the incompleteness of the theory.

An exceptionally large Nusselt number of stocking at maximum Re in Fig.10 may be caused by the critical phenomena of external boundary layer flow, but it has not yet been detailed.

3.4 Mean of overall heat transfer coefficients

Fig.16 shows the mean heat transfer coefficient defined from the mean temperature over the circumference of cylinder.

The mean coefficient of each sample changes gently with Re, unlike the one at

the stagnation point. The order of samples in insulation effect is generally acceptable. But we cannot depend solely on the order of mean coefficient in choosing a clothing material; we should depend on the results of the above section.

A rather wide discrepancy between the theoretical values and the experimental data is due to the incompleteness of the theory at the minimum heat transfer point described in §3.3.

However, to revise the theory, we must at first know quantitatively better about the exact human metabolism and acclimatization to the cold; the real human body should make the boundary condition (6) into some other form.

4. Conclusions

From the experiments using nylon stocking, tights camel, and knicker hose, a useful formulation of wind penetration and heat transfer has been established. Phenomenological findings of the tests are:

- (1) Wind penetrates in the fabric in the range $\theta < 60^\circ$, where the heat transfer is augmented;
- (2) α^* is an important factor for heat transfer augmentation by the wind;
- (3) δ^* is another important factor for the increase rate of heat transfer coefficient with the wind velocity;
- (4) The tights, which has small δ^* and α^* , is a better insulator than the woollen 'camel' and knicker hose on the windward side at large wind velocity. This agrees with the theoretical prediction about the effect of δ^* and α^* .

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