Pergamon

Building and Environment, Vol. 32, No. 5, |
Printe
0360-132


PII: S0360-1323(97)00012-7

# The Effect of Air Film Thermal Resistance on the Behaviour of Dynamic Insulation

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(*Received* 27 *March* 1996; *revised* 8 *June* 1996; *accepted* 5 *March* 1997)

*The authors' previous analysis of dynamic insulation is extended to include the inner and outer air film resistances with the objective of modelling the variation in surface temperature with air flow. The boundary condition that comes closest to predicting the variation of the surface temperature with air flow is one which assumes that the conduction heat flux at the wall surface, rather than the net heat flux, is equal to the flux incident on the wall from global environmental temperature, $T_{ei}$. Comparison of predictions of the temperature drop across the inner and outer air films with data from a hot box experiment shows that the theory correctly predicts the variation in surface temperature with air flow through a porous wall.* © 1997 *Elsevier Science Ltd.*

## NOMENCLATURE

$A$ relative measure of convective and conductive heat fluxes ($m^{-1}$)
$c_a$ specific heat of air (J/kg · °C)
$H$ height of cavity (m)
$L$ distance between outer and inner surfaces (m) or width of cavity (m)
Nu mean Nusselt number for vertical surface of cavity (m)
Pr Prandtl number
$q$ total heat flux (W/$m^2$)
$q_{bL}$ radiant heat flux between baffle and inner surface of insulation
$q_{co}$ conduction heat flux at outer surface (W/$m^2$)
$q_c$ conduction heat flux (W/$m^2$)
$q_i$ heat flux through inner air film applied to wall (W/$m^2$)
$q_s$ heat flux without air flow (W/$m^2$)
$q_{ui}$ heat flux through inner air film applied to air (W/$m^2$)
$R$ total thermal resistance of building element with air flow ($m^2$ · K/W)
$R_a$ thermal resistance of outer air film ($m^2$ · K/W)
$R_i$ thermal resistance of inner air film ($m^2$ · K/W)
$R_s$ total thermal resistance of building element with no air flow ($m^2$ · K/W)
$Ra_L$ Rayleigh number for cavity of width $L$
$T$ temperature of the air at a point within the envelope (°C)
$T_1$, $T_4$, $T_5$ temperature of probes 1, 4 and 5 (°C)
$T_a$ temperature of the ambient air (°C) or temperature of probe a (°C)
$T_{ai}$ temperature of the indoor air (°C)
$T_b$ temperature of baffle (°C or K)
$T_{bc}$ temperature of hot box chamber (°C)
$T_{ei}$ global environmental temperature (°C)
$T_o$ temperature of the outer surface of the external wall (°C)
$T_L$ temperature of the inner surface of the external wall (°C or K)
$T_m$ mean surface temperature of all internal surfaces (°C)
$T_s$ temperature of the internal surfaces (°C)
$x$ distance through building element from outer surface (m)
$u$ air velocity (m/s)
$\epsilon$ emissivity
$\lambda$ thermal conductivity (W/m · °C)
$\rho_a$ density of air (kg/$m^3$)
$\sigma$ Stefan–Boltzmann constant

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## 1. INTRODUCTION

In a previous paper [1] the authors presented an analytical study of dynamic and diffusive insulation for multi-layer, porous envelopes which clarified the mechanisms of heat and mass transfer through the envelope. Dynamic insulation, as it is referred to in Scandinavia and Canada [2], turns on its head the conventional building practice in the U.K. and other countries of trying to prevent air flowing through walls. By deliberately designing the wall to be porous, and by ensuring that the building is always slightly depressurised, the conductive heat loss through the building fabric can be considerably reduced. Other benefits include the control of water vapour transport through the wall without using a vapour barrier and improved indoor air quality through higher ventilation rates.

The authors' previous paper achieved a major simplification by showing that the net heat and mass fluxes depended only on the total thermal and diffusion resistances respectively of the wall and the air flow. Two porous walls composed of materials differing in their individual thermal resistances but having the same total resistance would show the same variation in heat transfer coefficient with air flow rate. The authors' analysis assumed that the inner and outer surface temperatures of the envelope were constant. This assumption is a good approximation for low air flows or in cases where there are heat sources within the room that will maintain the wall surface temperature constant. However, dynamic insulation is most effective in buildings which require high air change rates such as swimming pools and sports halls, and so it is necessary to be able to predict how the wall internal surface temperature will vary with air flow rate through the wall.

Measurements on test walls [3] and in a hot box [4] indicate that, with the air flowing through a porous wall from outside to in, the inner wall surface temperature, and to a lesser extent the outer wall surface temperature, both decrease with increasing air flow. This phenomenon, whilst previously acknowledged in the literature, has not been included in mathematical models of dynamic insulation. Bartussek [5] did include the resistance of the air film on the inner surface but his resulting equations, as will be shown, do not correctly describe the variation of surface temperature with air flow. Also, in his extensive thesis he did not report any experiments to determine how the wall surface temperature varied with air flow, nor did he explore the consequences for wall surface temperature implied by his theoretical analysis.

The wall surface temperature is an important contributing factor in determining human comfort. Surveys of various building types in different climates indicate that during occupation, the radiant temperature exceeds the air temperature by 0–2°C [6]. However, there are exceptions. In a conventional "impermeable" building envelope the air temperature can greatly exceed the mean radiant temperature during cold weather, as evidenced in poorly insulated rooms employing warm air heating. Dynamically insulated walls will always have internal surface temperatures lower than the room air temperature and this surface temperature will decrease as the air flow increases. As a step towards understanding how a dynamically insulated envelope will integrate with the building heating system and the occupants it is therefore necessary to develop a theoretical understanding of how the internal surface temperature varies with the air flow through it.

In conventional buildings, the pragmatic approach to the design of heating systems is to define a notional internal environmental air temperature, $T_{ei}$, which can be estimated from the following weighted average of the mean wall surface temperature, $T_m$, and the indoor air temperature, $T_{ai}$ [6]:

$$T_{ei} = \frac{2}{3}T_m + \frac{1}{3}T_{ai}. \quad (1)$$

One of the contributors to this weighting is the radiant heat transfer coefficient which, at 5.7 W/m$^2\cdot$K, is almost twice that of the convection heat transfer coefficient (3.0 W/m$^2\cdot$K). The mean surface temperature $T_m$ is approximated by

$$T_m = \frac{5}{6}T_s + \frac{1}{6}T_L, \quad (2)$$

where $T_L$ is the surface temperature of an external wall and the remaining five internal surfaces are assumed to be at temperature $T_s$. Davies [7] gives a good explanation and critique of the concept of environmental temperature. Overall the weighting of the three temperatures that help define a global environmental temperature is given by

$$T_{ei} = \frac{5}{9}T_s + \frac{3}{9}T_{ai} + \frac{1}{9}T_L. \quad (3)$$

The surface temperature of an external wall makes a small but not insignificant impact on the global environmental temperature. These weightings will change for a room with more than one external wall. Close to a wall the surface temperature will have an increasing impact on human comfort and condensation risks. Furthermore, air entering the room through this wall will also be at the surface temperature. Therefore heat will be required to warm this air up to the room air temperature $T_{ai}$. This paper extends our previous analysis with the objective of modelling the variation in surface temperature with air flow since it is clearly an important consideration in the design of buildings using dynamic insulation.

## 2. HEAT TRANSFER THROUGH A DYNAMIC BUILDING ELEMENT WITH INNER AIR FILM

As mentioned, the authors' previous paper showed that a multi-layer envelope could, for the purposes of calculating the heat loss, be treated as being equivalent to a single layer with the same total thermal resistance. Since in the present paper we are interested in the behaviour of the wall surface temperatures we shall present the theory in terms of a single layer of material without loss of generality. The thermal resistances of the air films adjacent to the envelope will be assumed to be constant and independent of air flow through the wall. The speed of the air as it issues from a porous wall in practical dynamic insulation applications is typically in the region of 0.5–5 m/h. This speed should not significantly affect the convection heat transfer coefficient of the wall surface since it is two orders of magnitude smaller than the CIBSE limit of 0.1 m/s for the air speeds at the wall surface beyond which its recommended values of convective heat transfer coefficients would not be valid. Measurements with a hot wire anemometer indicate that air speeds in the centre of a moderately sealed room heated by a radiator under the window are typically 0.05–0.1 m/s. Furthermore, the air speed will have no impact on the radiant heat transfer coefficient which is the more dominant of the two.

With reference to Fig. 1 the governing equation for steady state heat transfer in one dimension is

$$\frac{d^2T(x)}{dx^2} - A\frac{dT(x)}{dx} = 0, \quad (4)$$

where $A$ is defined as

$$A = \frac{u\rho_a c_a}{\lambda}. \quad (5)$$

The air flow $u$ is taken to be positive in the direction of increasing $x$. For the case where the wall surface temperatures are constant the solution to equation (4) is

$$\frac{T(x) - T_0}{T_L - T_0} = \frac{\exp(Ax) - 1}{\exp(AL) - 1}. \quad (6)$$

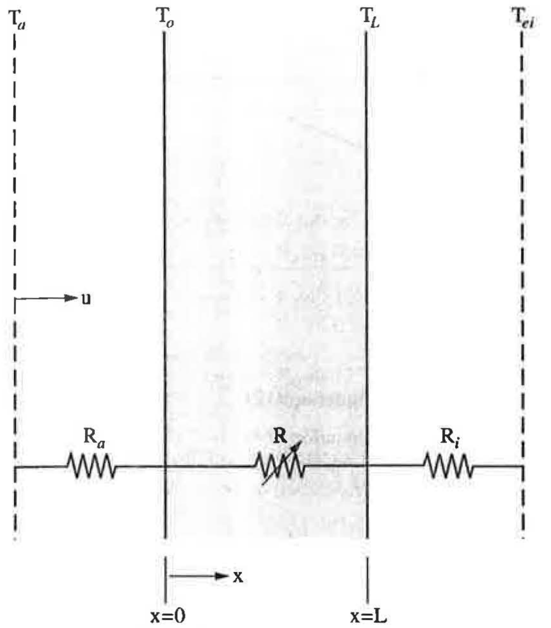

Fig. 1. Dynamic wall element.

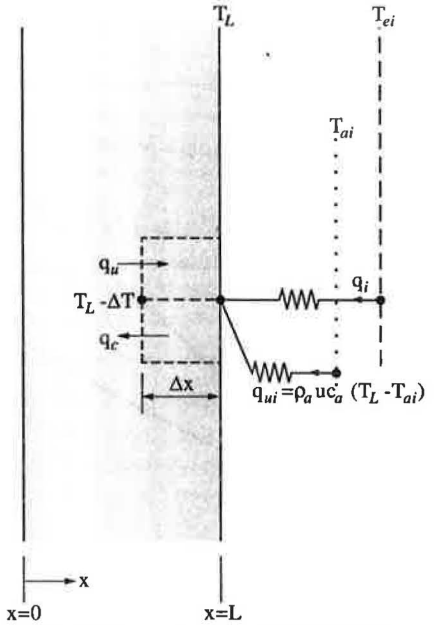

Fig. 2. Energy balance at inner surface.

If the wall surface temperatures are allowed to vary then the boundary conditions to be applied need a little consideration. On the cold side at $x = 0$, the temperature is $T_o$ and the heat flux is given by

$$q_{co} = \frac{T_0 - T_a}{R_a} = -\lambda \left.\frac{dT}{dx}\right|_{x=0}, \tag{7}$$

where $q_{co}$ is the conductive heat flux at $x = 0$. Since the convective flux is zero at this point $q_{co}$ is also the total heat flux. At the inner surface similar conditions will apply, which will result in four equations for the two unknown constants in the general solution of equation (4). There is only one independent boundary condition at each surface and either may be chosen. However, it is not immediately obvious which of the three heat fluxes, conduction, convection or the resultant of the two, is the appropriate one to use.

Consider the volume element enclosing a thin layer in the wall, $\Delta x$, and bounded on one side by the wall surface at temperature $T_L$, as shown in Fig. 2. The heat flux into the wall at temperature $T_L$ comprising both convection and radiation is determined by the film resistance, $R_i$

$$q_i = \frac{T_{ei} - T_L}{R_i}. \tag{8}$$

If it is assumed that the heat flux $q_i$ incident on the wall at temperature $T_L$ is equal to the net flux $q$ which is constant through the wall and decreases with increasing air flow [5], then as the air flow increases $q_i$ will tend to zero and $T_L$ will tend to $T_{ei}$, which is contrary to experimental observation. Whereas if $q_i$ is set equal to the conduction flux $q_c$, then $q_i$ increases with increasing airflow, thus driving down the wall surface temperature. For these reasons it is postulated that the appropriate boundary condition is to set the incident heat flux equal to the conduction flux.

$$q_c = \frac{T_{ei} - T_L}{R_i} = -\lambda \left.\frac{dT}{dx}\right|_{x=L}. \tag{9}$$

As shown in Fig. 2, the air leaves the wall at temperature $T_L$ and additional heat $q_{ui}$ must be supplied to raise the temperature of the air to $T_{ai}$.

Equation (4) can be solved for the boundary conditions $T = T_0$ at $x = 0$ and equation (9) at $x = L$ to give the temperature profile through the wall. Since this result is important for calculating the condensation risk within a wall, it is worthwhile stating it as a matter of record

$$\frac{T(x) - T_0}{T_{ei} - T_0} = \frac{\exp(Ax) - 1}{\lambda A R_i \exp(AL) + \exp(AL) - 1}. \tag{10}$$

However, in the present paper the focus of attention is on the wall surface temperatures since they are the temperatures most readily measured and have a direct effect on the heat gain and loss from the internal and external walls respectively. Consideration of the wall surface temperature also provides a direct comparison between Bartussek's assumed boundary condition and that of the authors.

The relationship between the temperature drop across the inner air film $(T_{ei} - T_L)$ and the parameters describing the envelope and the air flow can be derived as follows. By noting that

$$\begin{aligned} T_{ei} - T_a &= (T_{ei} - T_L) + (T_L - T_0) + (T_0 - T_a) \\ &= q_{c0} R_i \exp(AL) + (T_L - T_0) + q_{c0} R_a \end{aligned} \tag{11}$$

and applying the boundary conditions to eliminate $q_{co}$, we obtain

$$\frac{T_{ei} - T_L}{T_{ei} - T_a} = \frac{R_i \exp(AL)}{R_i \exp(AL) + \dfrac{\exp(AL) - 1}{\lambda A} + R_a}. \tag{12}$$

In the limit of zero air flow equation (12) reduces to the expected

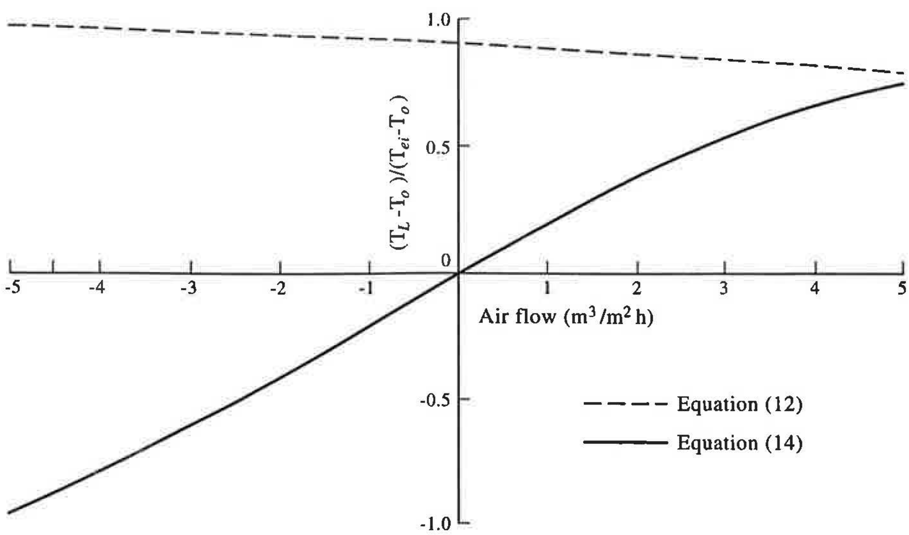

Fig. 3. Wall surface temperature.

$$\frac{T_{ei}-T_L}{T_{ei}-T_a}=\frac{R_i}{R_i+R_s+R_a}, \quad (13)$$

where $R_s$ is the thermal resistance of the wall with no air flow.

The solution based on Bartussek's assumption is

$$\frac{T_{ei}-T_L}{T_{ei}-T_a}=\frac{R_i}{R_i+\dfrac{\exp(AL)-1}{\lambda A}+R_a}. \quad (14)$$

Figure 3 shows how the wall surface temperature varies with air flow for equation (12) and equation (14) for $R_a = 0$. This clearly shows that the assumption that incident heat flux $q_i$ is equal to the net heat flux leads to physically impossible conclusions. It implies that: (i) the inner surface temperature tends to the outer surface temperature at zero air flow; (ii) for air flowing out through the wall the inner surface actually becomes colder than the outer surface; and (iii) at high air flows the inner surface temperature tends to the global environmental temperature, all of which are contrary to experience. The assumption that $q_i$ is proportional to the conductive heat flux at the wall surface leads to a more realistic variation in wall surface temperature and so for the rest of this paper this will be taken as the working hypothesis.

The temperature drop across the outer air film can be derived in a similar fashion to above

$$\frac{T_0-T_a}{T_{ei}-T_a}=\frac{R_a}{R_i\exp(AL)+\dfrac{\exp(AL)-1}{\lambda A}+R_a}. \quad (15)$$

Note that there is no ambiguity here about which heat flux to use since at the outer surface, $x = 0$, the net heat flux and the conductive heat flux are equal [1].

## 3. HEAT TRANSFER THROUGH A DYNAMIC BUILDING ELEMENT WITH INNER AND OUTER AIR FILMS

The heat flux from the wall is found by substituting equation (15) into equation (7)

$$q=\frac{T_{ei}-T_a}{R_i\exp(AL)+\dfrac{\exp(AL)-1}{\lambda A}+R_a}. \quad (16$$

Without air flow the heat loss per unit area is simply

$$q_s=\frac{T_{ei}-T_a}{R_i+R_s+R_a}, \quad (17$$

to which equation (16) simplifies for this special case Eliminating the temperature difference between these equations then gives the very useful relationship betweer the heat loss for dynamic insulation with air flow anc that without air flow.

$$\frac{q}{q_s}=\frac{R_i+R_s+R_a}{R_i\exp(AL)+\dfrac{\exp(AL)-1}{\lambda A}+R_a}. \quad (18$$

To see how the air flow rate through a wall affects the heat flow, consider the effect of the inner air film resistance alone ($R_a = 0$) in equation (18). Figure 4 shows tha the inner film resistance is insignificant for both wel insulated ($R_s = 6\,m^2 \cdot K/W$) and poorly insulatec ($R_s = 1.2\,m^2 \cdot K/W$) walls. This means that when assessing the relative change in the heat loss of dynamic insulation over the static equivalent the effect of the inner ai film can safely be neglected. The outer air film having a smaller resistance than the inner can also be neglected When considering the surface temperatures, however, the air films cannot, in general, be neglected, as shown in the following section.

## 4. VARIATION OF WALL SURFACE TEMPERATURES WITH AIR FLOW AND THERMAL RESISTANCE

It is of interest to explore the implications of equation (12) and (15) for the temperature differences across the inner and outer air films in buildings using dynamic insulation. Taking the thermal resistance of an interna vertical surface as $0.123\,m^2 \cdot K/W$ and that of an outside vertical surface of normal exposure as $0.06\,m^2 \cdot K/W$ [6]

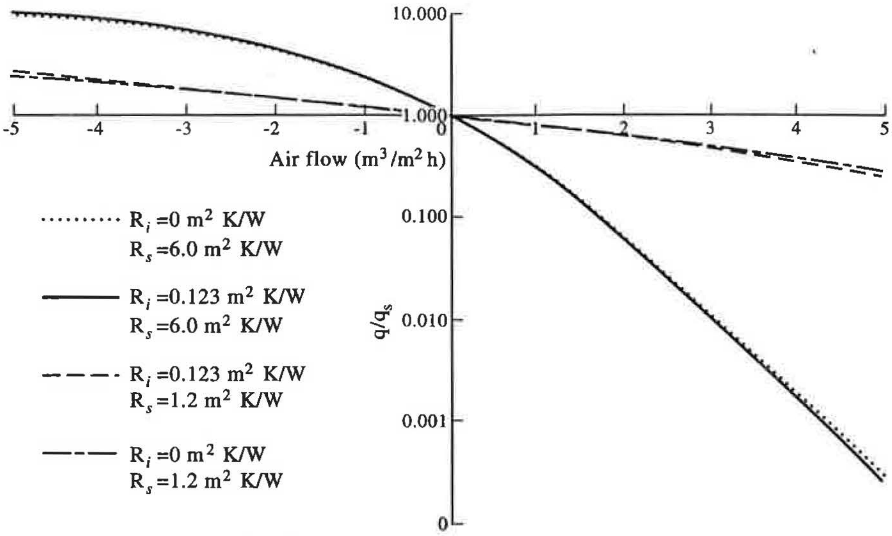


Fig. 4. Ratio of dynamic to diffusive heat flux.

the temperature drops across the inner air film from equation (12) and the outer air film from equation (15) are plotted against air flow rate for a temperature difference $T_{ei}-T_a$ of 10°C in Figs 5 and 6 respectively. The temperature difference across the inner air film at 1 m$^3$/m$^2\cdot$h (typical for dynamic insulation in dwellings) is about 0.4°C for a well insulated wall (thermal resistance of 6 m$^2\cdot$K/W) and over 1°C for a poorly insulated wall (1.2 m$^2\cdot$K/W). As the air flow increases the temperature difference increases and the difference between well insulated and poorly insulated walls decreases. For a swimming pool where the air flow through the wall could be greater than 5 m$^3$/m$^2\cdot$h, the wall surface temperature could be 2°C lower than the environment temperature. When internal air flows to outside through the wall the inner film temperature difference decreases to zero. For the well insulated wall this temperature difference is practically zero for an air outflow of 1 m$^3$/m$^2\cdot$h. A similar pattern is observed for the temperature difference across the outer film except that the temperature difference decreases with increasing air flow. Figure 6 shows that with air flowing into the building the temperature difference across the outer air film could, for practical purposes, be neglected.

## 5. COMPARISON WITH EXPERIMENTAL RESULTS

Crowther [4] has measured the performance of dynamic insulation using a hot box (Fig. 7). His experiments were designed to measure the temperature profile through the wall as function of air flow and to establish whether dynamic insulation operating in the contraflux mode (air flowing in the opposite direction to the heat flux) with heat recovery does in fact achieve a net saving in energy. His results confirmed that the temperature profile is in accordance with equation (6) and that the energy used to maintain the hot box at a constant temperature of 38.9°C with air flowing in through wall A

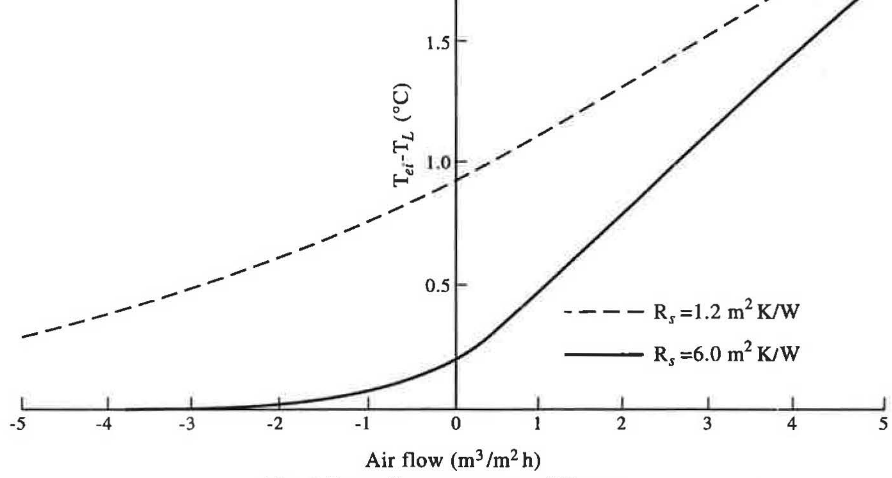


Fig. 5. Inner film temperature difference.

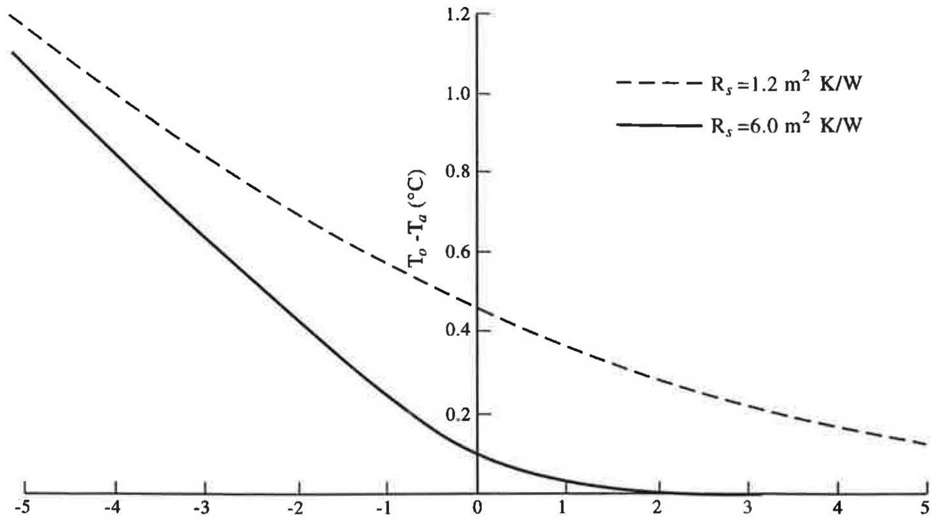

Fig. 6. Outer film temperature difference.

and out through the vent pipe in wall B is up to 7% less than that for a conventionally insulated box.

Crowther's experimental apparatus is a 1.2 m long 0.5 m × 0.5 m square section box made of plywood and covered with 100 mm of mineral wool insulation. The arrangement of the apparatus relevant to this discussion is as shown in Fig. 7. Wall A is a tightly fitting batt of mineral wool and is air permeable. Air is pumped into a plenum on the left of wall A. The air flow rate was measured by variable area flow meter. The pipe connecting the plenum to the heated chamber to the right of wall A is, in this instance, capped. The internal dimensions of the heated chamber are 0.5 m × 0.5 m × 0.5 m. The chamber is heated by an electric cable underneath a thick aluminium plate forming the floor of the chamber. The inside of the chamber is thermostatically controlle at 38.9°C. Air can flow out of the heated chambe through the vent pipe in wall B. The temperature of th air in the inlet plenum varies with the ambient condition in the laboratory.

The mean temperatures measured at one location o the inner, and outer surfaces, in the cavity and th chamber (Fig. 7), are given in Table 1. Figure 8 show how the outer and inner air film temperature difference: $T_1 - T_a$ and $T_5 - T_4$ respectively, vary with the air flo rate through the porous wall (Crowther, pers. comm 1996). The theoretical curves are calculated using equa tions (12) and (15) assuming $R_i = 0.14\,m^2 \cdot K/W$ an $R_a = 0.19\,m^2 \cdot K/W$ for the inner and outer film resist ances respectively. These values not only give good agree

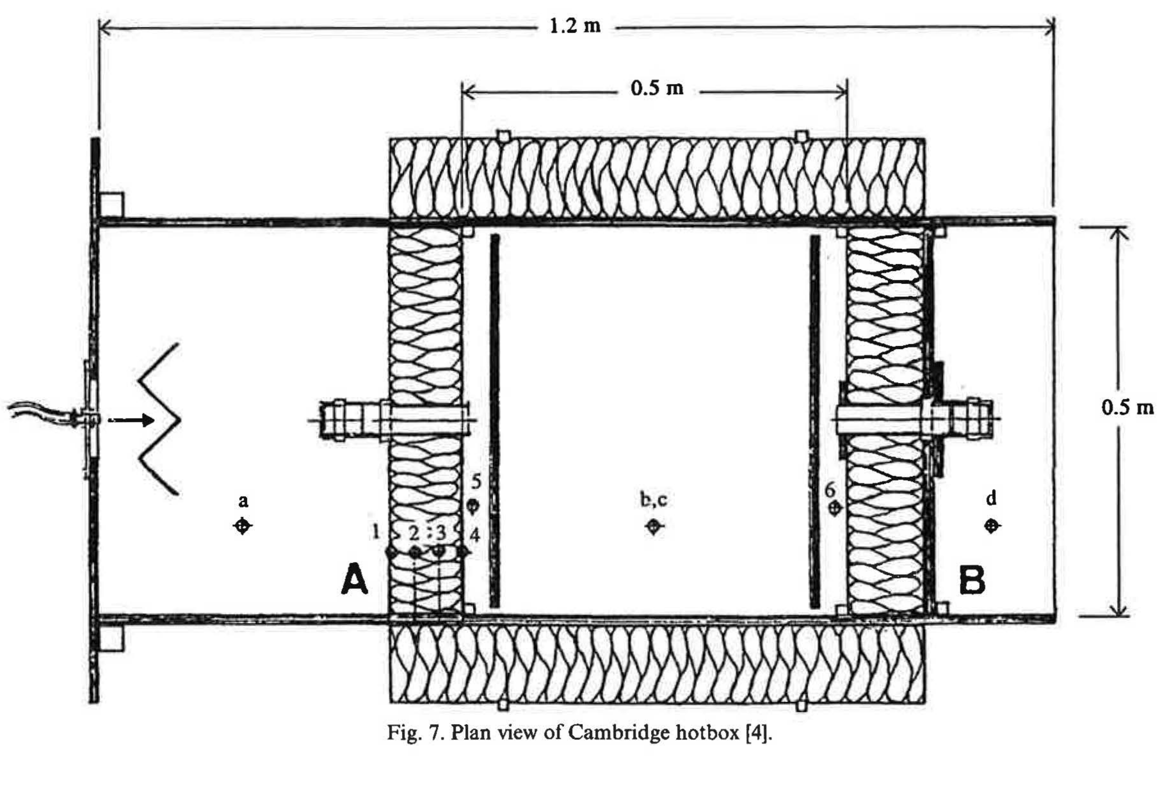

Fig. 7. Plan view of Cambridge hotbox [4].

Table 1. Temperatures of air films close to porous wall

| Air flow (m/h) | Mean temperature (°C) Probe a | Probe 1 | Probe 4 | Probe 5 | Probes b,c |
|---|---|---|---|---|---|
| 0.6 | 20.08 | 20.92 | 36.68 | 37.88 | 39.04 |
| 1.32 | 20.37 | 20.84 | 36.20 | 37.84 | 38.99 |
| 2.76 | 20.88 | 21.22 | 35.28 | 37.38 | 38.92 |
| 4.2 | 20.98 | 21.14 | 34.32 | 37.04 | 38.89 |
| 5.5 | 22.02 | 22.00 | 33.68 | 36.80 | 38.92 |

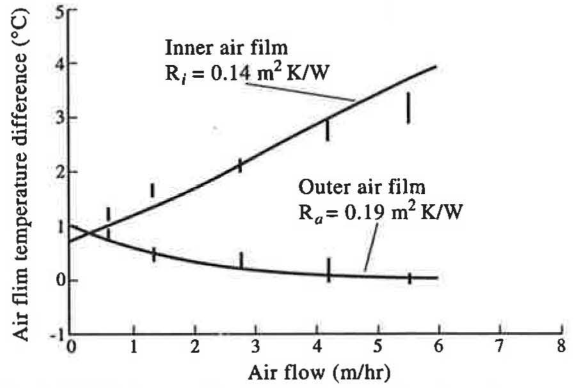


Fig. 8. Air film temperature difference vs air flow through porous wall.

ment between the theory presented in this paper and Crowther's experimental data, but are also quite realistic for the experimental apparatus as will now be demonstrated.

Consider first the internal cavity. The unshielded temperature probe, $T_5$, positioned within the cavity will measure a temperature intermediate between the surface of the wooden baffle (assumed to be the same as the chamber temperature $T_{bc}$) and the inner surface of the insulation $T_4$. The heat transfer mechanisms within the cavity will be natural convection and radiation, both driven by the temperature difference between the plane parallel surfaces. As discussed above, it is unlikely that the air flow issuing from the porous wall will have a significant effect on the natural circulation; the following correlation will be used to estimate the natural convection heat transfer coefficient within the cavity [8]:

$$\mathrm{Nu}_L = 0.042\mathrm{Ra}_L^{1/4}\,\mathrm{Pr}^{0.012}\left(\frac{H}{L}\right)^{-0.3}. \tag{19}$$

For a cavity aspect ratio $H/L = 12.5$ the natural convection heat transfer coefficient varies from 1.34 to $1.64\,\mathrm{W/m^2\cdot K}$ due to the temperature dependence of the Rayleigh number $\mathrm{Ra}_L$. The temperature difference across the cavity increases with air flow from 2.3 to 5.3°C. The convective heat transfer coefficient within the cavity is about half that for a vertical wall in a room.

The radiant heat transfer coefficient can be estimated from the radiant heat exchange between parallel planes [8]

$$q_{\mathrm{bL}} = \frac{\sigma(T_{\mathrm{b}}^4 - T_{\mathrm{L}}^4)}{\dfrac{1}{\varepsilon_{\mathrm{b}}} + \dfrac{1}{\varepsilon_{\mathrm{L}}} + 1}. \tag{20}$$

Assuming that the surface emissivities $\varepsilon_{\mathrm{b}}$, $\varepsilon_{\mathrm{L}}$ are both equal to 0.9, the radiant heat transfer coefficient decreases from 5.59 to $5.51\,\mathrm{W\cdot m^2/K}$ as the insulation surface temperature increases from 33.7 to 36.7°C. Combining the radiant and convective heat transfer coefficients in parallel, the total calculated thermal resistance for the cavity is $0.142 \pm 0.002\,\mathrm{m^2\cdot K/W}$ for the measured range of air flow. This gives excellent agreement with the value derived from the experimental data.

The thermal resistance on the inlet side of the insulation can be estimated with less certainty because the flow regime within the cavity is unknown and there are five radiant surfaces whose temperatures one can only infer from the known experimental conditions. Treating the plenum as a cavity with an aspect ratio of 1.25, the appropriate correlation for the natural convective heat transfer coefficient is [8]

$$\mathrm{Nu}_L = 0.18\left(\frac{\mathrm{Pr}}{0.2+\mathrm{Pr}}\mathrm{Ra}_L\right)^{0.29}. \tag{21}$$

This implies that the natural convective heat transfer coefficient could be $0.97\,\mathrm{W/m^2\cdot K}$ at an air flow of 0.6 m/h, decreasing to $0\,\mathrm{W/m^2\cdot K}$ at 5.5 m/h due to the temperature dependence of the Rayleigh number $\mathrm{Ra}_L$. In addition to natural convection there is also forced convection over the surface of the insulation due to the effect of the air inlet to the plenum. Where the air flow over the surface is in the same direction as the natural convection flow, such as on the lower part of the insulation, then the forced convection will enhance the natural convection. However, in the upper part where the natural and forced flows are likely to be in opposite senses they will tend to cancel one another. The resultant convective heat transfer coefficient could vary over the surface from 0 to, say, $2\,\mathrm{W/m^2\cdot K}$.

The radiant heat losses from the insulation to the five other surfaces of the plenum were calculated and the radiant heat transfer coefficient varies from $5.8\,\mathrm{W/m^2\cdot K}$ at 0.6 m/h to $12.8\,\mathrm{W/m^2\cdot K}$ at 5.5 m/h. Combining the radiant and heat transfer coefficients gives overall film resistances of 0.13 to $0.065\,\mathrm{m^2\cdot K/W}$. This does not represent very good agreement with the experimental value of $0.19\,\mathrm{m^2\cdot K/W}$. However, the calculations indicate the circumstances which might give rise to such a high film resistance: a very low or even zero convective heat transfer coefficient over most of the surface and a radiant heat transfer coefficient of $5.26\,\mathrm{W/m^2\cdot K}$. The latter could come about if the insulation emissivity were about 0.88 instead of the 0.9 assumed or one of the other surfaces of the plenum, say the floor, were at approximately the same temperature as the surface of the insulation. We can conclude then that the experimental value of $0.19\,\mathrm{m^2\cdot K/W}$ for the outer air film thermal resistance is perfectly feasible.

## 6. CONCLUSIONS

Consideration of the air films on either side of a dynamically insulated wall has provided insight into the heat transfer processes at the wall surfaces. The boundary condition that comes closest to predicting the variation of the surface temperature with air flow is to assume that the conduction heat flux at the wall surface, rather than the net heat flux, is equal to the flux incident on the wall from global environmental temperature $T_{ei}$. The theory predicts that a well insulated wall will have a higher inner surface and lower outer surface temperature than a poorly insulated wall for the same air flow. A very practical result from this work is that, when assessing the relative change in the heat loss of dynamic insulation over the static equivalent, then both the outer and inner air films may be neglected. The effect of the air films cannot be neglected when calculating the surface temperatures and how they may affect human comfort and the heat transfer processes with the other surfaces of the room.

Comparison of predictions of the temperature drop across the inner and outer air films with data from a hot box experiment shows that the theory predicts the variation in surface temperature with air flow through a porous wall.

**Acknowledgements**—This work is the result of a collaborative effort between the Scott Sutherland School of Architecture at Robert Gordon University, and the Department of Engineering at the University of Aberdeen. It is funded by the Engineering and Physical Sciences Research Council, U.K. The authors thank Dr D. Crowther for providing his experimental data and the diagram of his hot box.

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