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Validation of heat transfer coefficients on interior building surfaces using a real-sized indoor test cell

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Abstract—Experimental studies are carried out to investigate the heat transfer coefficients on interior building surfaces (such as vertical walls, ceilings and glazing) using a real-sized indoor test cell which measures $2.95 \times 2.35 \times 2.08$ m (length × width × height). A total of 142 tests, each one lasting about 24 h, are conducted under controlled steady-state conditions to cover nine of the most widely used heating configurations in buildings. Ten new heat transfer correlations are developed and these are presented in a way suitable for use by building thermal modellers. The correlations are found to differ by up to a factor of two from those which are currently being used in building thermal models.

1. INTRODUCTION

THE VALUES of the heat transfer coefficient on interior building surfaces used by dynamic simulation models are generally based on those reported in the literature for isolated, and in most cases small, surfaces. The difference between these values and those which occur in real buildings is not known. The problems of two- and three-dimensional natural convection have been addressed in several studies. However, most of these were carried out in small enclosures, and in many cases with water as the working fluid. Comparison between the available correlations [1] indicated that a discrepancy of up to a factor of five can occur in the values of the heat transfer coefficient on vertical surfaces using the different correlations, up to a factor of four on horizontally heated surfaces facing upward, and up to a factor of eight on horizontally heated surfaces facing downward.

The survey of the literature revealed very few experimental studies for heat transfer coefficients on surfaces of real-sized enclosures. Min *et al.* [2] carried out experimental investigations in three different sized rooms, $7.35 \times 3.6 \times 2.7$ m, $7.35 \times 3.6 \times 3.7$ m and $3.6 \times 3.6 \times 2.4$ m (length × width × height). The data reported were obtained with the entire floor area or ceiling area used as a heated panel where all surfaces other than the heated panel were held at a uniform temperature. A number of correlations were reported which cover the heat transfer between the room air and each of the vertical walls, the floor and the ceiling for the two heating configurations mentioned above. The room air was taken at a height of 1.5 m at the centre of the room, which implies an assumption of uniform air temperature, i.e. no temperature stratification. Both assumptions, of uniform wall and uniform air temperatures, are not realistic. For this reason the correlations of Min *et al.* may have some level of uncertainty which cannot be accurately estimated.

The convection heat transfer coefficient from a floor heated by warm water to the room air was investigated by Schlapmann [3]. The measurements were performed in a closed room 4.95 m long, 4.0 m wide and 2.70 m high as well as in a small-scale model. No work was conducted to investigate the heat transfer coefficient on the other elements of the room such as vertical walls and ceiling. Correlations for convection heat transfer from a vertical wall of an office room to the room air were developed by Li *et al.* [4]. A room 3.4 m wide by 4.0 m long by 2.6 m high was used. The room was occupied by three people and the experiment was carried out under normal working conditions with no control on the convection in the room. A wall–air temperature difference of only up to 1.5°C was covered by this study. Furthermore, the heat transfer coefficient was evaluated from readings of just two thermocouples. One was used to measure the wall temperature and the other the air temperature. Because of the known variation of both the air and wall temperature in the vertical direction, the result of this study cannot be taken as an accurate estimation of the average heat transfer coefficient on a wall of a real-sized enclosure. Many other investigators report studies in small-scale three-dimensional enclosures. Bohn *et al.* [5] and Bohn and Anderson [6] investigated the convection heat transfer in a cubical enclosure of interior dimensions of 0.305 m using water as the working fluid. Heidt and Streppel [7] investigated the heat flow from an irradiated interior

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NOMENCLATURE

h	convection heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity of the wall material [W m K^{-1}]
q	heat transfer through the wall [W]
t	thickness of the wall [m]
T_a	temperature of the air in the hot zone [°C]
T_{si}	temperature of the interior surface of the wall [°C]
T_{so}	temperature of the exterior surface of the wall [°C]
ΔT	air to surface temperature difference [°C]
u	wind speed [m s^{-1}].

surface of a 0.3 m cubical enclosure. These small-scale results, however, need to be confirmed for real-size enclosures.

Therefore the main objective of this work is to develop, through careful experiments, correlations for convective heat transfer coefficients which occur on the interior surfaces of the different elements of buildings, such as vertical walls, ceiling and glazing and to study the air temperature stratification and the variation in the local heat transfer coefficient near the vertical wall of a real-sized enclosure. This study differs from previous studies by investigating nine different heating configurations under realistic operating conditions.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The tests were carried out in a real-sized test cell which consists of two separate zones as shown in Fig. 1. The larger zone (the hot zone) was constructed

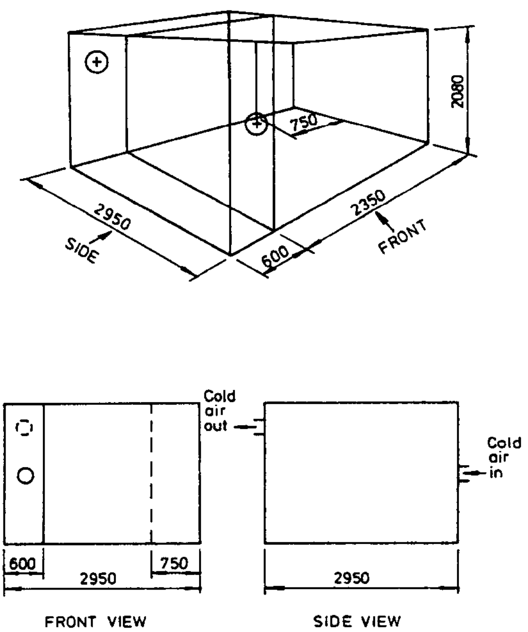


FIG. 1. The dimensions of the test cell.

to represent a large enclosure such as a living area with interior dimensions of 2.95 × 2.35 × 2.08 m (length × width × height) and with a 2.08 m high and 0.75 m wide access door. In order to control the air temperature on the exterior surface of one of the vertical walls, another zone was constructed (the cold zone). The cold zone has approximately the same height and length as does the hot zone but with a width of 0.60 m. The two zones are kept at two different temperatures so as to achieve a controllable and variable temperature difference across the wall separating the two zones (the test wall).

All four vertical walls and the roof of the hot zone are constructed from a 50 mm thick isocyanurate board covered with aluminium foil on both sides. The aluminium foil helped to minimize the effect of the longwave radiation exchange on the temperature and heat flux measurements. The floor of the hot zone is constructed from a 100 mm thick styrofoam board covered with a 19 mm thick chipboard on both sides. The cold zone is constructed from a 3 mm thick hardboard. Both zones are contained in an aluminium frame held on castor wheels. The test cell is located within the University of Wales College of Cardiff Solar Simulator Laboratory which measures 13 m long, 6.5 m wide and 5.8 m high approximately.

The air temperature in the cold zone was kept relatively low by pumping cold air from the ambient by means of an extractor fan, as shown in Fig. 2. The cold air was slightly heated to the maximum temperature expected during the test before it was delivered to the cold zone to compensate for the fluctuation in the outdoor ambient air temperature. The heating effect was achieved by passing the extracted air through an in-line heating element located just before the air entrance to the cold zone. A proportional temperature controller was used to control the input power to the heating element so as to maintain air flow of steady temperature. The air temperature in the hot zone was also kept under control by using another proportional temperature controller to control the input power to the heating device of this zone. The air temperatures in both zones were controlled to better than ± 0.1°C for the 'fan heater' and the 'foil' configurations. However, due to the storage effect of the oil-filled radiator used in some heating configurations, the fluctuation in the temperature of the hot zone for these con-

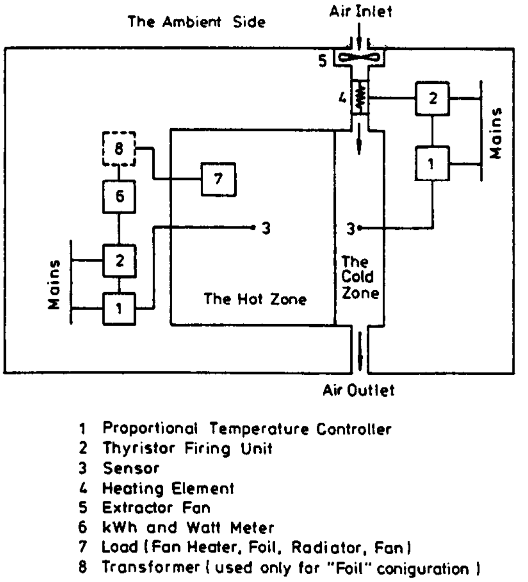


FIG. 2. Details of the experimental apparatus.

figurations was up to a maximum of ± 1.5°C with a typical value of ± 0.3°C.

The wall separating the two zones is instrumented by a total of 21 thermistors (see Fig. 3). A total of 7 thermistors are mounted on the interior surface of the wall (facing the hot zone); a similar number are mounted on the exterior surface of the wall (facing the cold zone), and another 7 are suspended at a distance of 60 mm from the interior surface of the wall to measure the air temperature outside the thermal boundary layer. Another vertical wall and the roof are also instrumented, but with a total of 9 thermistors on each. These are mounted in the same planes explained above, but with only 3 thermistors in each plane (see Fig. 3). The air temperature in the laboratory which contains the test cell is measured by 3 thermistors located around the test cell. At a later stage, the test wall was fitted with a window which is instrumented with a total of 12 thermistors, 4 on each of the three planes mentioned previously (see Fig. 4(h)). All the thermistors were encapsulated in a heat-shrinkable tubing so as to avoid the effect of moisture on the temperature measurements as well as to give some rigidity to the thermistor assembly. The thermistors were also shielded by aluminium foil to minimize the effect of the longwave radiation exchange. Heat sink compound was used with the surface thermistors to ensure a good thermal contact with the surface.

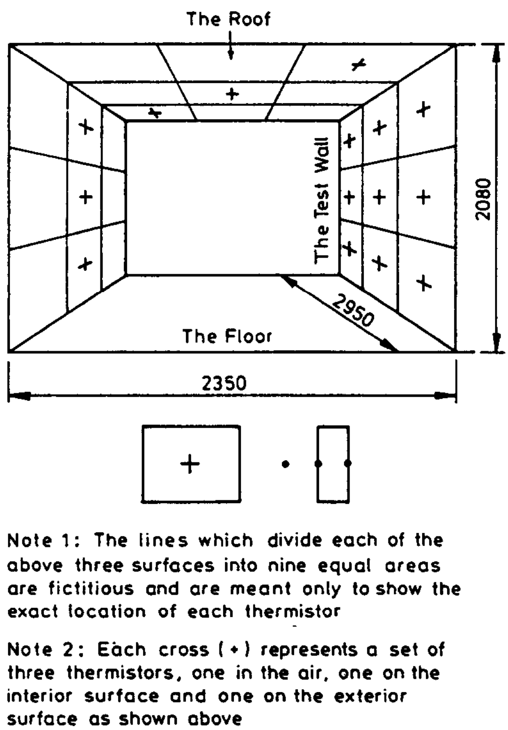


FIG. 3. Thermistor locations in the test cell.

The tests were carried out under steady-state conditions with the test cell empty, unlit and tightly sealed. To achieve the steady-state condition, each experiment was allowed to run for about 24 h between any two different temperature settings, with the air temperature in both zones under control. The steady-state condition is indicated by a steady air temperature in the hot zone, a steady air temperature in the cold zone, and a steady level of the input power to the heating device of the hot zone as indicated by the Wattmeter reading.

The readings are taken by means of 10 K thermistors, every minute, and are averaged and stored in a 60-channel data-logger over 5 min intervals. A microcomputer was used to recover the data which were stored on floppy discs. For each test, readings of about 12 continuous hours were collected (i.e. between 7000 and 8760 readings). The data logging for each test was made overnight between 22.00 and 10.00 the following day. Logging during this period has the advantage of avoiding the effect of lighting, solar gain and movement of people on the conditions inside the laboratory which contains the test cell. Accordingly, the fluctuation in the air temperature inside the laboratory during the logging period was within ± 0.5°C.

The data of each test were analysed by means of a BASIC computer program written for that purpose. The results of each test included the following values for each of the elements under investigation: the heat transfer coefficients on the interior and exterior surface, the heat flux and the U-value of each element, the maximum possible uncertainty associated with the measurement of the heat transfer coefficient on each element, and the local heat transfer coefficients at seven different locations on the test wall.

3. THE DETERMINATION OF THE HEAT TRANSFER COEFFICIENT

The determination of the convection heat transfer coefficient involves the measurement of the heat flux.

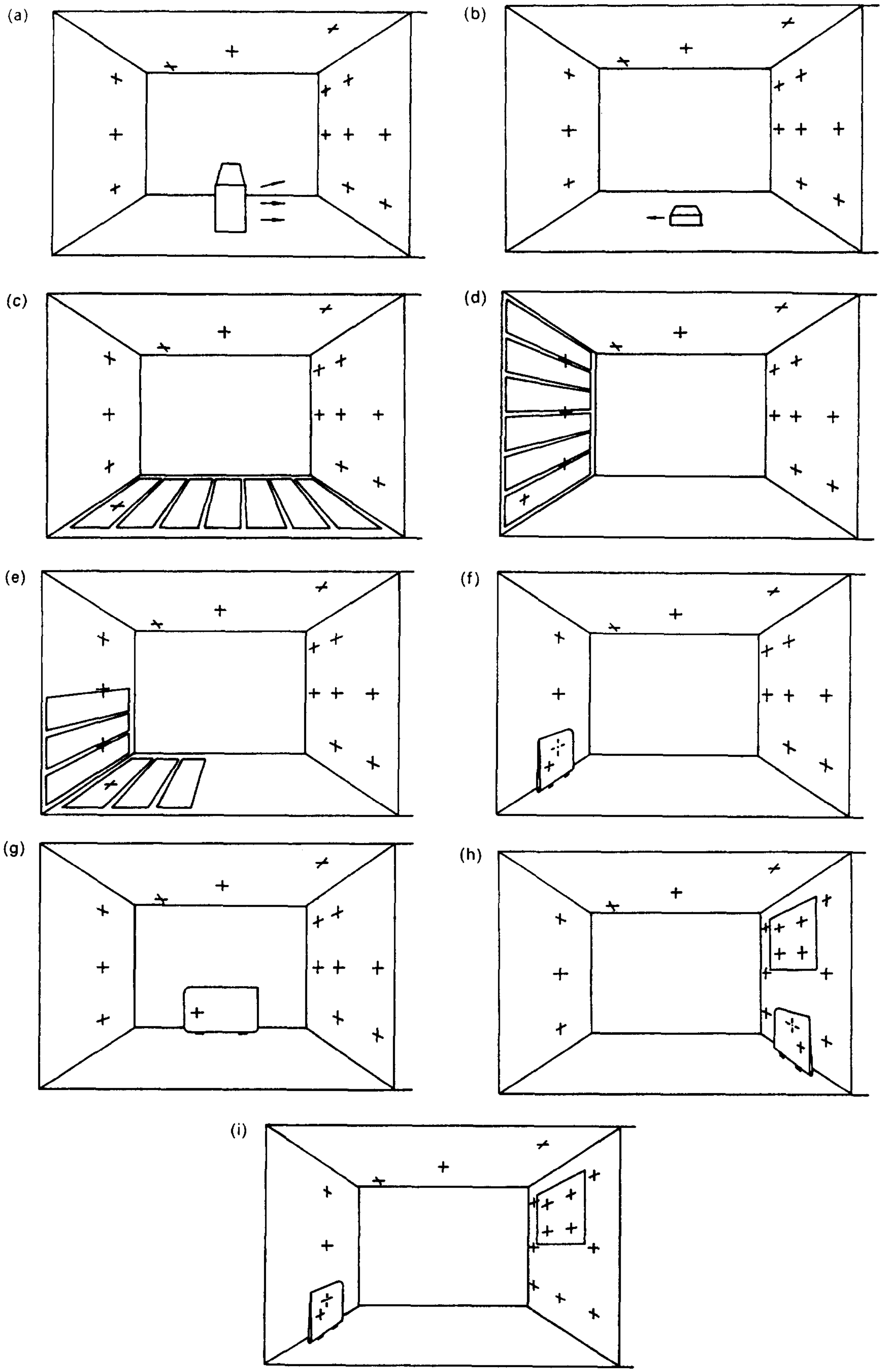


FIG. 4. (a) Simulated forced convection on an interior surface using a large fan. (b) A room heated by a fan heater. (c) A room with a uniformly heated floor using strips of heating foil to simulate the heating effect. (d) A room with a uniformly heated vertical wall using strips of heating foil to simulate the heating effect. (e) A room with a partially heated floor and a partially heated vertical wall (edge) to simulate the heating effect. (f) A room heated by an oil-filled radiator located in front of the test wall. (g) A room heated by an oil-filled radiator located adjacent to the test wall. (h) A room heated by an oil-filled radiator located under a window. (i) A room heated by an oil-filled radiator located in front of a window.

In this study the wall material itself forms the heat flux meter. In principle, most heat flux devices require the measurement of the temperature across a slab of known thermal conductivity and of known thickness. Although the value of the thermal conductivity (k) was specified by the manufacturer, it was checked using a 'Guarded Hot Box' and the value of 0.0215–0.0255 W m^{-1} K^{-1} (an average value of 0.0235 W m^{-1} K^{-1}) obtained was very close to the value of 0.023 W m^{-1} K^{-1} specified by the manufacturer.

In the steady-state condition, the heat flowing from the air to the wall surface by natural convection plus the net heat flow to the surface by longwave radiation exchange should equal the heat loss by conduction through the wall. For the test cell case, the longwave radiative exchange was found to be small enough to be neglected [8], since radiation between enclosed surfaces of very low emissivity (the aluminium foil) is involved. In this case the heat balance on the interior surface of the wall can be given as

$$q = (k/t)(T_{si} - T_{so}) = h(T_a - T_{si}). \qquad (1)$$

The convection heat transfer coefficient can thus be determined from the following equation:

$$h = (k/t)(T_{si} - T_{so})/(T_a - T_{si}). \qquad (2)$$

Equation (2) can be used to determine local heat transfer coefficients by using local values. The integrated average (bulk) heat transfer coefficient can be determined either from the average of the local values or by using integrated average temperatures in equation (2).

The results from both methods were found to be close, with the coefficient obtained from the local values higher by an average value of 2%. The results used in the correlations were those obtained from the integrated average temperatures.

A complete error analysis is done in ref. [8] where it was shown that the total uncertainty in the heat transfer coefficient varies from around 21% at a wall to air temperature difference of 1°C to around 6% at a wall to air temperature difference of 5°C.

4. THE CONFIGURATIONS TESTED

Nine of the most widely used heating configurations in buildings were covered by this study. These included the following configurations:

(1) A simulated forced convection on the interior surface of the test wall using a large fan as shown in Fig. 4(a). Different fan speeds, and hence different temperature levels, were achieved in the hot zone by varying the input power to the fan.

(2) A cell heated by a small low speed 1 kW fan heater as shown in Fig. 4(b). The fan heater was located on the floor at the centre of the room with the air from the fan heater directed towards the opposite direction of the test wall.

(3) Metallized plastic foil was used in three different heating configurations to simulate: (i) a uniformly heated floor as shown in Fig. 4(c), (ii) a uniformly heated vertical wall as shown in Fig. 4(d), (iii) a uniformly heated edge (i.e. a partially heated floor and a partially heated vertical wall) as shown in Fig. 4(e). The foil consists of plastic substrate (polyethylene) coated with a very thin layer of aluminium on both sides, which can be used as a heating element. The foil has to be broken down into individual strips isolated from each other to reduce the current density in each strip to a safe level. Seven strips, 0.30 × 2.80 m each, were needed to cover the floor area, while six were required to cover the vertical wall area. In both cases, the strips were connected in parallel to a 500 VA transformer with a maximum output voltage of 35 V.A.C. For the floor case, the strips were left to lie on the floor area, while for the vertical wall case the strips were mounted as close as possible to the interior surface of the wall, but not in direct contact with the surface.

(4) A cell heated by a radiator was located (i) opposite the test wall as shown in Fig. 4(f), and (ii) adjacent to the test wall as shown in Fig. 4(g). A 1.5 kW oil-filled radiator which measures 1.28 × 0.60 m (length × height) was used as a heating source. The location, shape and size of the radiator matches that which is usually used in a house with central heating radiators, convectors or storage heaters. The temperature of the radiator was controlled by the proportional temperature controller and not by the radiator's own thermostat. Aluminium foil was used to cover the whole surface of the radiator to minimize the longwave radiation exchange.

(5) After the tests in the above heating configurations were completed, the test wall was fitted with a 1.34 × 0.84 m single glazed window. Two configurations were examined in this case, the first with the radiator located beneath the window as shown in Fig. 4(h), and the second with the radiator located opposite the window as shown in Fig. 4(i).

5. RESULTS AND DISCUSSION

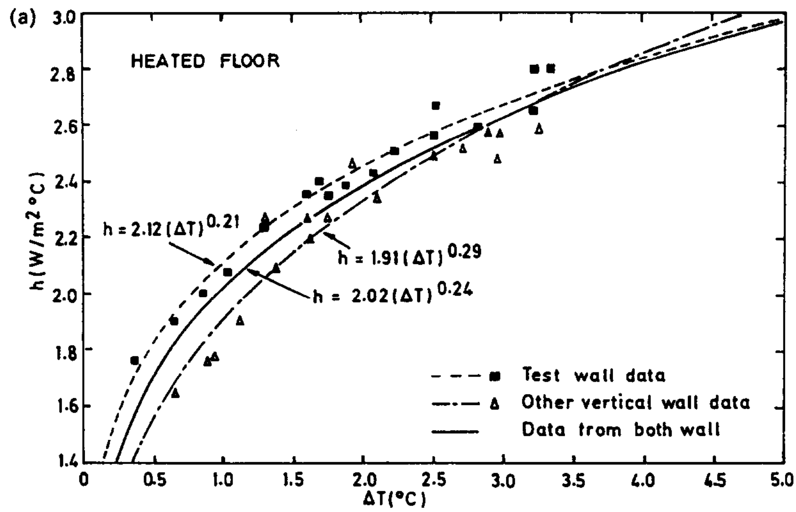
The heat transfer coefficient (h) obtained on the different elements of the test cell were correlated against the temperature drop (ΔT) across the air layer adjacent to each element. A multi-regression computer program was used to correlate the data to yield correlations in the form $h = C(\Delta T)^n$, where ΔT is the air to surface temperature difference. For most of the configurations, four correlations were developed: one for the interior surface of each of the 'test wall', the 'other vertical wall', the 'ceiling', and one from the combined data of the two vertical walls. Correlations were also developed for the interior surface of the glazing for each of the two 'window' configurations.

Since the heat transfer coefficients on the exterior surfaces of the test cell are largely independent of the heating configuration inside the test cell, the convection heat transfer coefficient data for the exterior surface of the 'other vertical wall' for each configuration were combined together and are correlated against the temperature drop across the air layer adjacent to the exterior surface of the wall. Those data obtained on the roof were correlated together in the same way.

A total of 36 correlations were developed. Because of limited space, however, only a number of these will be presented in this paper. The interested reader is referred to ref. [8] for the complete list of the correlations. The data from similar correlations were combined together to obtain a number of general correlations, each one covering more than one configuration, and these will be presented in full in this paper.

The results for the 'heated floor' configuration (Fig. 4(c)) are shown in Fig. 5(a) for both vertical walls. The heat transfer coefficients on both walls are expected to be close, since the same type of air flow pattern is expected on each wall, which is similar to that shown in Fig. 11(a). The difference in the data for the two walls may be attributed, however, to different measurement locations. For that reason the correlation from the combined data of both walls is considered to be more realistic. The data and the correlation obtained for the ceiling are shown in Fig. 5(b).

Figure 6(a) shows the data and correlation of the test wall for the 'heated vertical wall' configuration (Fig. 4(d)). The scatter in the data of the 'other vertical wall' (the heated wall) was found to be large, and no reasonable correlation was developed. The scatter in the data of the ceiling (Fig. 6(b)) was also found to be high and no reasonable correlation was developed. The reason for the scatter in the data is expected to



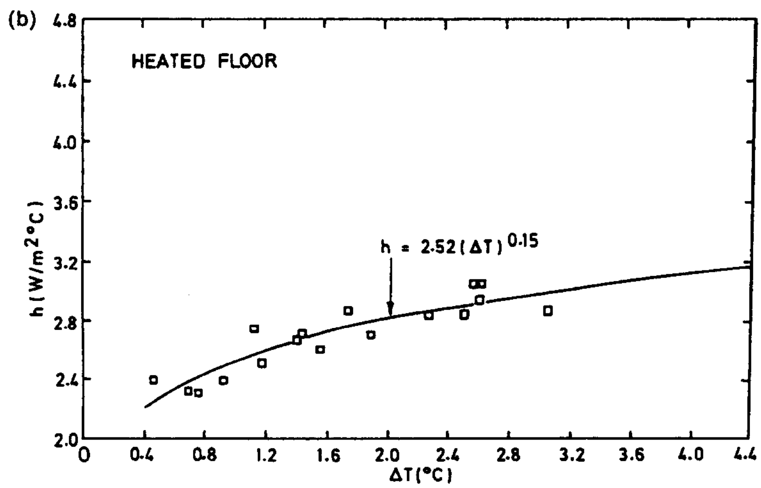
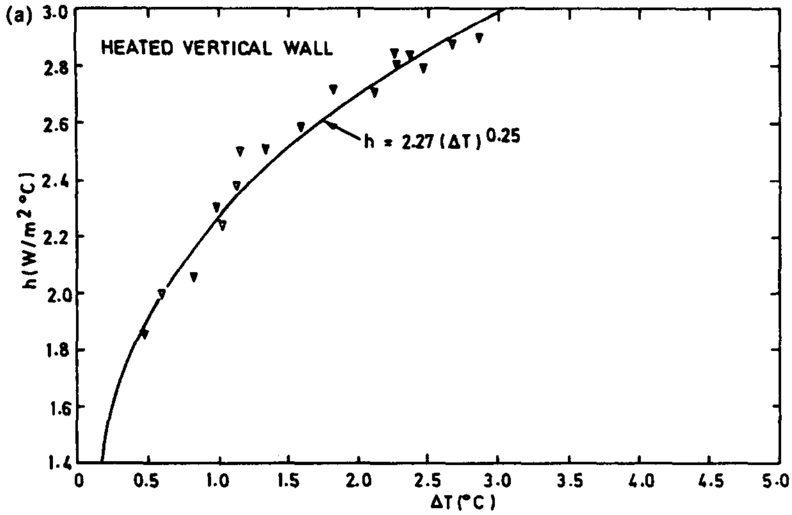


FIG. 5. (a) Heat transfer data and correlations for the two vertical walls—the 'heated floor' configuration. (b) Heat transfer data and correlation for the ceiling—the 'heated floor' configuration.



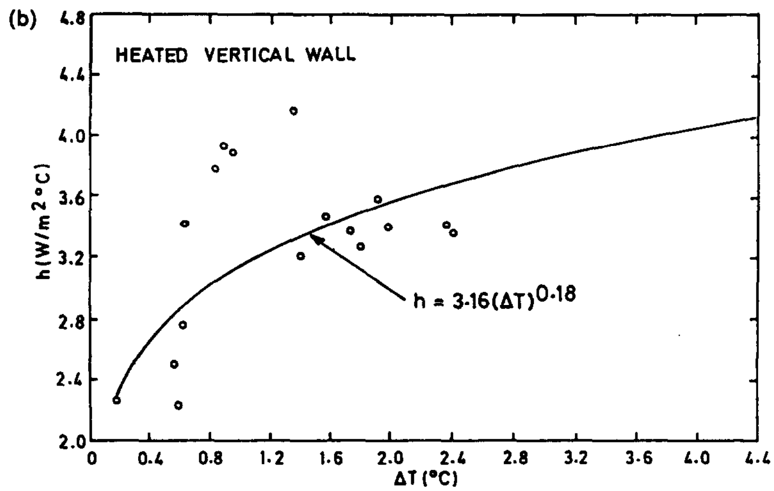


FIG. 6. (a) Heat transfer data and correlation for the test wall—the 'heated vertical wall' configuration. (b) Heat transfer data and correlation for ceiling—the 'heated vertical wall' configuration.

be caused by the type of mounting of the heating foil. To avoid an electric short circuit between the heating foil and the aluminium foil which covers the interior surfaces, the heating foil had to be mounted close and not in direct contact with the surface. This resulted in an unpredictable air flow pattern between the wall and the foil. The air flow pattern in the cell, however, is expected to be similar to that of Fig. 11(b), although no flow visualization study was made.

Figure 7(a) shows the result for the 'radiator adjacent to the test wall' configuration (Fig. 4(g)). The heat transfer coefficients on both the 'test wall' and the 'other vertical wall' are expected to be close. However, it was found (as can be seen in Fig. 7(a)) that the coefficient on the 'test wall' is slightly higher. The heat transfer coefficient on the 'other vertical wall' was measured at the middle of the wall (see Fig. 4(g)). This part of the wall is expected to be at the relatively inactive part of the core for the type of air flow pattern induced by the location of the radiator in this configuration. Lower values are expected in this area compared with those which occur on the rest of the wall, due to the relatively slower air movement. The data and the correlation obtained on the 'ceiling' are shown in Fig. 7(b).

When the test cell was fitted with a window, the radiator was first located under the window (Fig. 4(h)) to simulate one of the most widely used heating configurations in buildings. The results for this configuration are shown in Fig. 8(a). The heat transfer coefficient on the 'test wall' (which was close to the radiator) was found, as would be expected, to be higher than that on the 'other vertical wall'. On the 'ceiling', the data and the correlation shown in Fig. 8(b) were obtained.

The radiator was then located opposite the 'test wall'

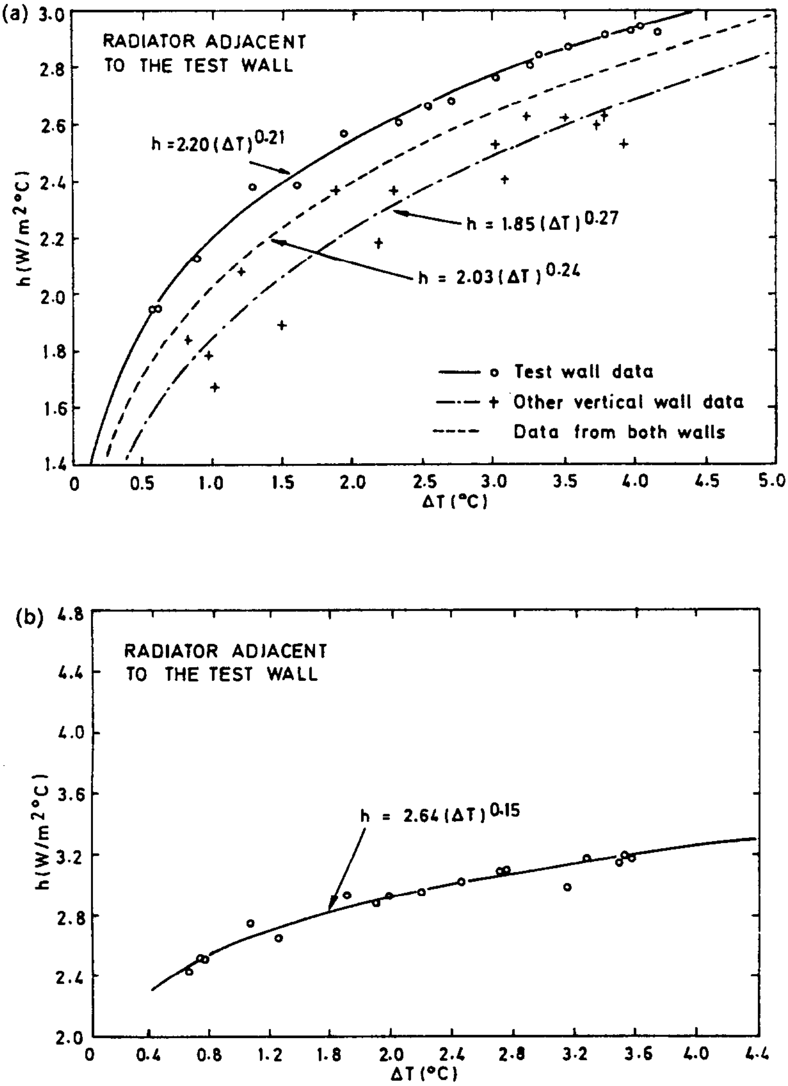


FIG. 7. (a) Heat transfer data and correlation for the two vertical walls—the 'radiator adjacent to the test wall' configuration. (b) Heat transfer data and correlation for ceiling—the 'radiator adjacent to the test wall' configuration.

and the window (Fig. 4(i)). The results from this configuration are shown in Fig. 9(a) for the two vertical walls. The data of the other wall were found to be too scattered to fit in the figure. The data and the correlation obtained on the ceiling are shown in Fig. 9(b). The data and the correlations obtained on the interior surface of the glazing for the two 'window' configurations are shown in Fig. 10.

One correlation was developed for each of the exterior surfaces of the 'other vertical wall' and the 'roof'. Because the test cell is located inside another large enclosure (the laboratory), a condition of still air existed during the vast majority of the tests. Therefore, the heat transfer coefficient on the exterior surface of the vertical wall is expected to be comparable to that of an isolated large vertical heated plate. Likewise, the heat transfer coefficient on the exterior surface of the roof is expected to be similar to that of a large horizontal heated plate facing upward.

Table 1 gives the values of the heat transfer coefficient on the 'test wall' for the forced convection

Table 1. The experimental results for the forced convection tests

Air speed, u (m s^{-1})	Heat transfer coefficient (W m^{-2} K^{-1}) Experimental results	Range in the literature
0.6	7.52	2.3[a] to 10.1[b]
1.1	8.44	4.3[a] to 11.5[b]
1.5	10.52	5.9[a] to 12.4[b]

[a] Using $h = 3.91u$ [9].
[b] Using $h = 8.55+2.65u$ [10].

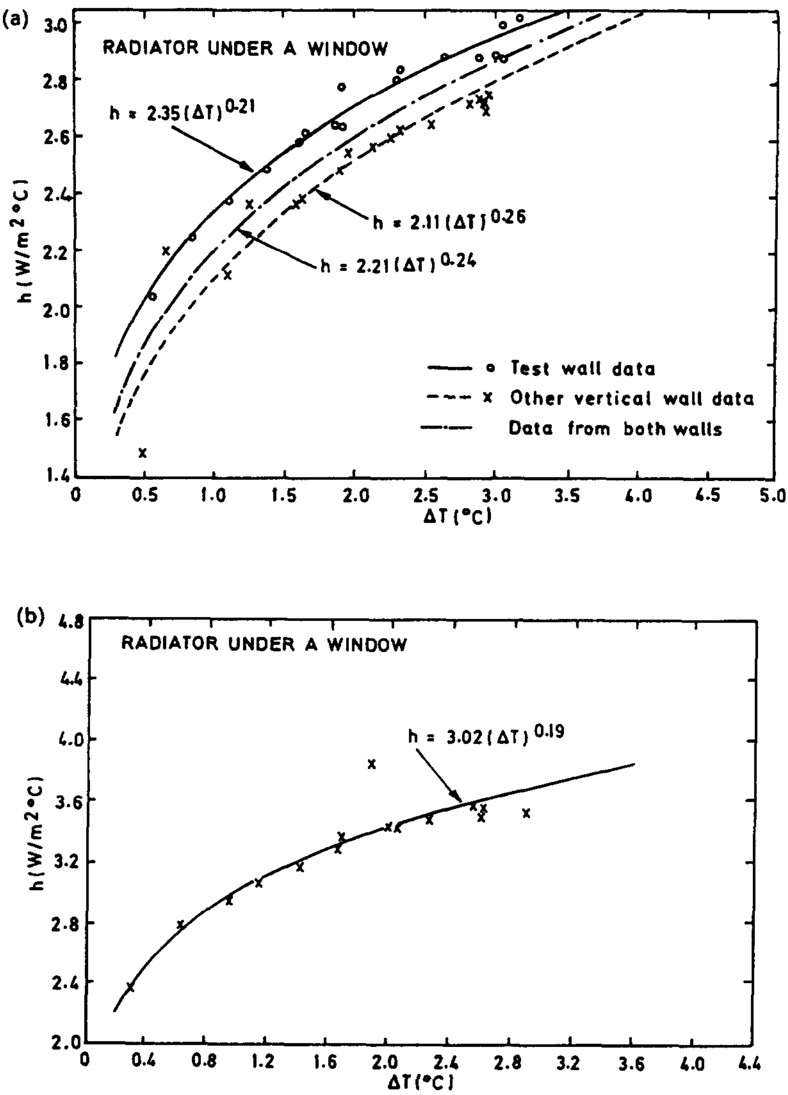


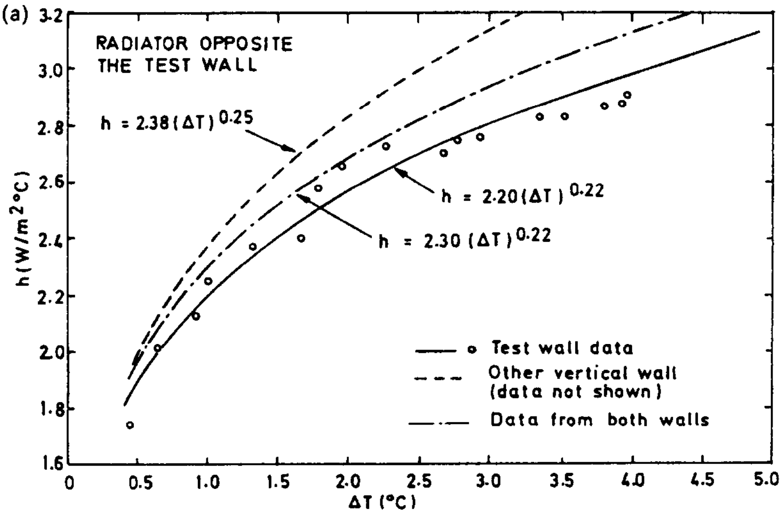
FIG. 8. (a) Heat transfer data and correlation for the two vertical walls—the 'radiator under a window' configuration. (b) Heat transfer data and correlation for ceiling—the 'radiator under a window' configuration.

tests where a large fan was used (see Fig. 4(a)). Since no results were found in the literature for forced convection on interior building surfaces, the results of the present study are compared with some of those reported in the literature (Hand [9] and Soltau and Angermeier [10]) for forced convection on exterior surfaces. However, it should be noted that the two cases are not quite similar. The comparison shows that the results of the present study are well within the range of those reported in the literature for forced convection on exterior surfaces. However, this does not necessarily indicate that the latter are suitable for the estimation of the convective coefficients on interior building surfaces, since the band is too large. The equation [$h = 5.34+3.27u$] was found to best represent the three data points of Table 1.

5.1. *Comparison between all the correlations*

The correlations developed for the 'test wall' from the eight different configurations are shown together in Fig. 12(a), while all those developed for the 'ceiling' are shown in Fig. 12(b). A comparison between the 'foil' configurations shows that the convective heat transfer coefficients on the 'test wall' and on the 'ceiling' are the largest for the 'heated vertical wall' configuration, followed by those of the 'heated edge' and the 'heated floor' configurations. In the light of the expected air flow pattern in the test cell for each configuration and the different air speed which occurs in each one, this order is much like what would be expected.

For the two radiator configurations (without a window), the heat transfer coefficients on the 'test wall'



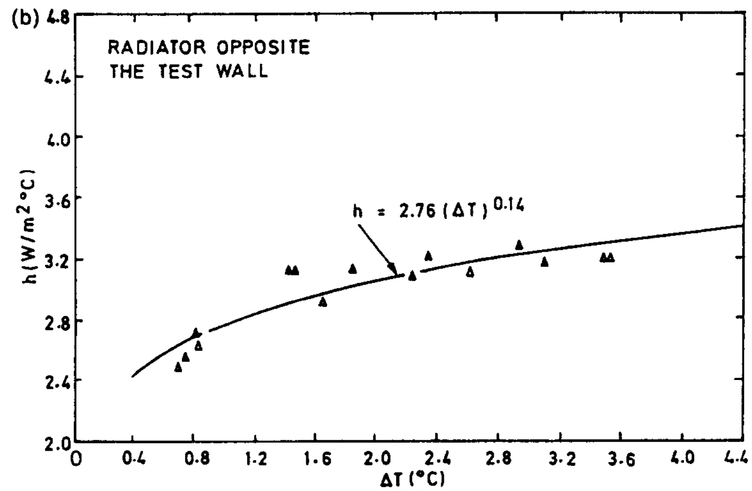


FIG. 9. (a) Heat transfer data and correlations for the two vertical walls—the 'radiator opposite the test wall' configuration. (b) Heat transfer data and correlations for two vertical walls—the 'radiator opposite the test wall' configuration.

and on the 'ceiling' were found to be slightly higher for the configuration where the radiator was located opposite the 'test wall' compared with that where it was located adjacent to the 'test wall'. In both cases, the air flow pattern in the rectangular cell is expected to be close to that of Fig. 11(b). However, the direction of the air movement with respect to the 'test wall', in addition to the shorter distance between the radiator and the opposite wall in the first configuration (2.35 m for the first configuration and 2.95 m for the second) is expected to cause higher air speeds on the 'test wall' and the 'ceiling', and therefore higher heat transfer coefficients are expected.

A comparison between the two 'window' configurations shows that the heat transfer coefficient on the 'test wall' (not including the window) is significantly higher for the case where the radiator was beneath the window (close to the 'test wall') compared with that where it was opposite the window. This is consistent with the result from the 'radiator opposite the test wall' configuration which indicated that the heat transfer coefficient is larger on a wall close to a radiator. The difference between the heat transfer coefficient on the 'ceiling' for the two configurations (see Fig. 12(b)) was found to be unexpectedly large. With the two configurations identical except for the location of the radiator, the reason must be a significantly different air flow pattern in the enclosure in each case. The air flow pattern in the enclosure for the 'radiator opposite a window' configuration is expected to be close to that of Fig. 11(b). That of the 'radiator under a window' configuration is difficult to predict because of the interaction between the upward moving air heated by the radiator and the downward

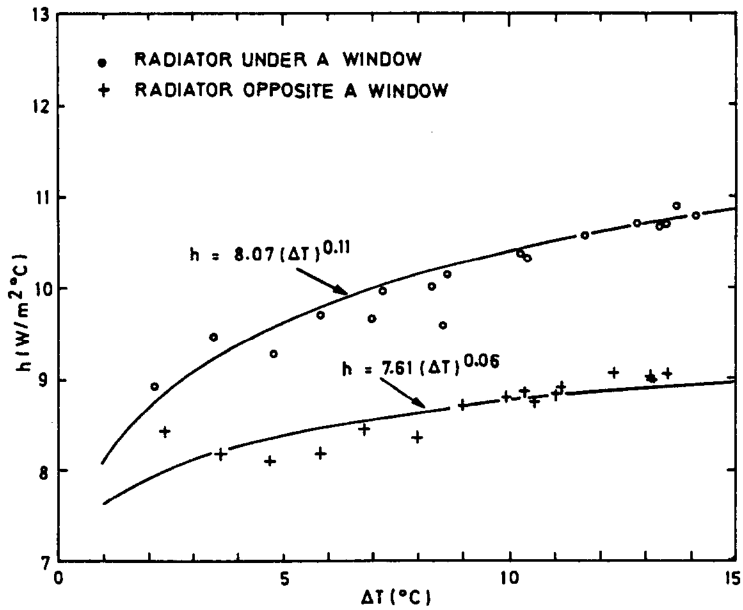


FIG. 10. Heat transfer data and correlations for the glazing—the 'radiator under a window and radiator opposite a window' configuration.

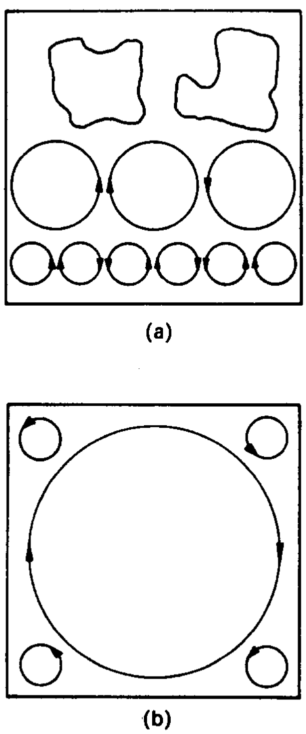


FIG. 11. (a) Air flow pattern in an enclosure with a heated floor. (b) Air flow pattern in an enclosure with a heated vertical wall.

moving air cooled by the window. The difference between the results from the two configurations, however, suggests that the air flow on the 'ceiling', and probably in the enclosure as a whole, has been enhanced by having the radiator under the window.

The effect of the radiator location in the test cell on the heat transfer coefficient on the interior surface of the glazing can be judged from Fig. 10. In this figure, the data and the correlation obtained on the glazing are shown for both window configurations. It can be seen clearly that the heat transfer coefficient is significantly higher for the configuration where the radiator was under the window. Again, the type of air flow pattern induced by the location of the radiator under the window is expected to be the reason. The results from the above comparison suggest that the heat losses from the roof and the glazing of the test cell can be reduced by moving the radiator from under the window. Based on the correlations developed, the reduction was found to be around 15% for the ceiling and around 13% for the glazing at a temperature drop of 5°C across the air layer adjacent to each element. However, it should be noted that the existence of curtains and furniture in the room may change the air flow pattern in the room and hence the magnitude of the losses mentioned above.

For the 'fan heater' configuration, the heat transfer coefficient on the 'test wall' (which was not opposite the fan heater) was found to be the smallest of those obtained from the rest of the configurations, though very close to that of the 'heated floor' configuration, as can be seen in Fig. 12(a). The heat transfer coefficient on the wall which was opposite the fan heater (the 'other vertical wall'), however, was found to be significantly larger (see equation (8) of Fig. 13). The larger difference between the values of the heat transfer coefficient on each wall suggests that no smooth circulation (such as that of Fig. 11(b) for example) has occurred in the enclosure. The effect of the fan heater (with a fan power of only 20 W at full power)

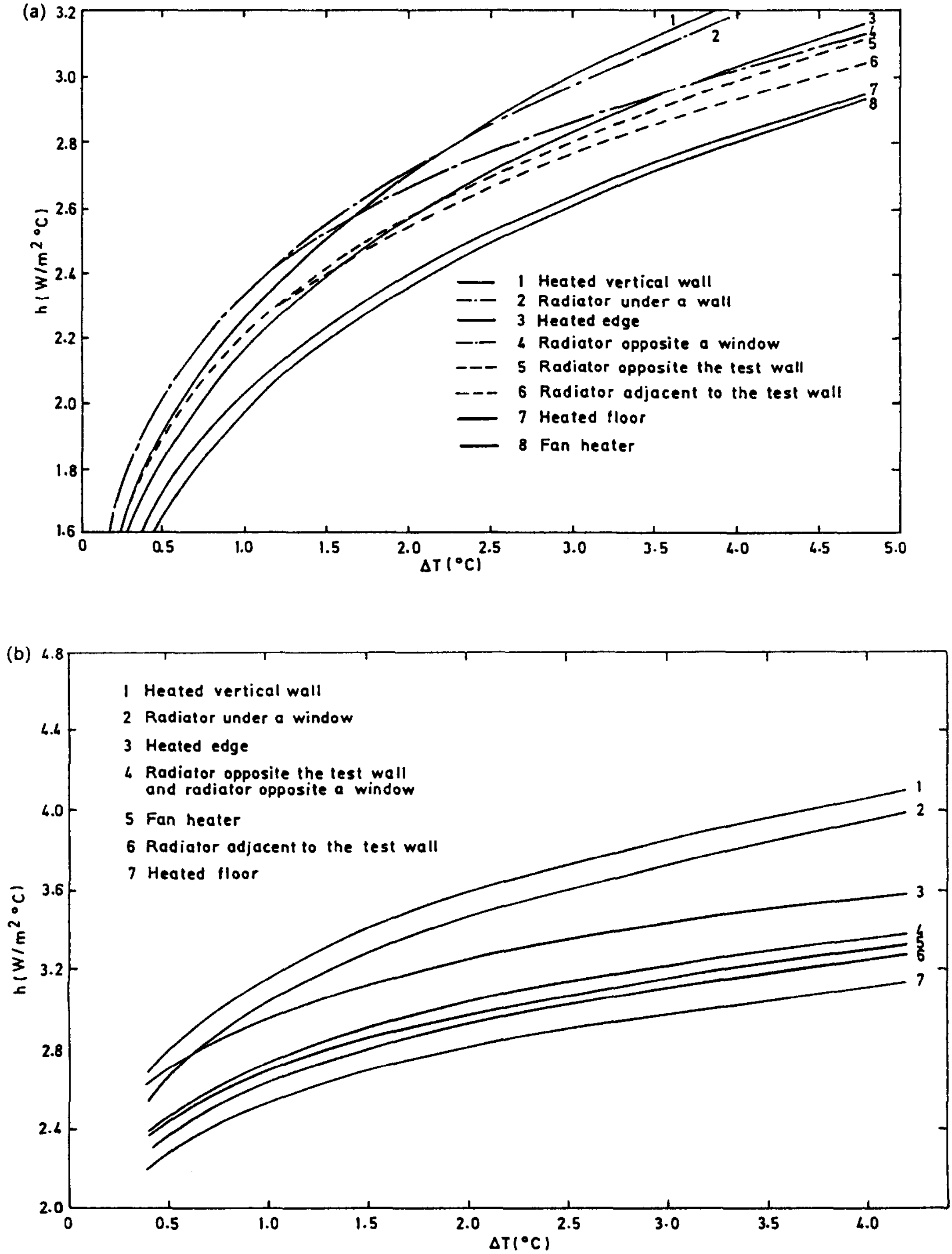


FIG. 12. (a) Heat transfer correlations for the test wall—all configurations. (b) Heat transfer correlations for the ceiling—all configurations.

was probably confined to the 'other vertical wall' (and maybe to part of the ceiling). Elsewhere in the enclosure, and especially on the 'test wall', a mainly buoyancy-driven convection is expected to have occurred.

A comparison between all the correlations of the 'test wall' shown in Fig. 12(a) indicates that the heat transfer coefficient increases with an increase in the area of the heated vertical surface (excluding the configuration where the radiator was located under the window because the large heat transfer coefficients were caused mainly by the radiator being close to the wall as discussed before). The largest values of the convective coefficient were those for the 'heated ver-

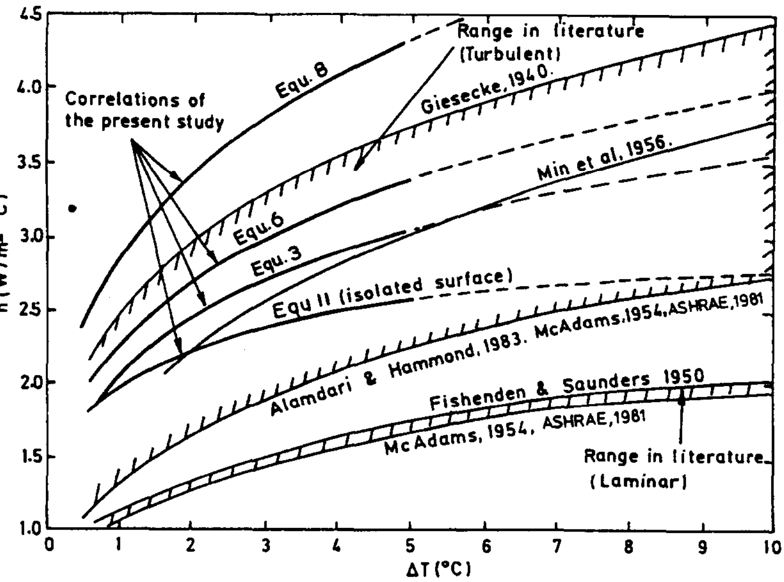


FIG. 13. Comparison between the correlations of the present study and those reported in the literature for vertical surfaces.

tical wall' configuration (heated vertical area of about 6 m^2), followed by those of the 'heated edge' configuration (heated vertical area of about 3 m^2), followed by those of three of the configurations where the radiator was used as a heating device (heated vertical area of about 0.77 m^2 on each side of the radiator), followed by those of the 'heated floor' and the 'fan heater' configurations.

The same conclusions can be drawn from the comparison of all the ceiling correlations shown in Fig. 12(b) where (except for the correlation of the 'fan heater' configuration which has been discussed before) the same trends as in Fig. 12(a) are seen.

The above discussion suggests that one way of reducing the heat losses from buildings is by minimizing the vertical area of the heating device in order to achieve lower heat transfer coefficients in the enclosure. The 'heated floor' configuration (which should be similar to an underfloor heating system) seems to have an advantage in this respect because no heated vertical surface is involved. Comfort requirements, space limitations or cost effectiveness may, however, impose some other constraint.

5.2. *Approximate correlations*

As shown from the above discussion, the correlations obtained from the eight different configurations covered two of the four vertical walls of the enclosure, the ceiling and the glazing. For each one of these elements, a number of correlations were developed: each one is suitable for a certain configuration. It is useful, however, to have some approximate and more general correlations which can be applied to more than one configuration. The procedure used for obtaining such correlations was based on the results from each individual correlation and on how close the results are after accounting for the maximum uncertainty associated with the experimental data of each correlation. The procedure used for estimating the maximum uncertainties in the experimental data of each configuration is given in ref. [8]. When the results from two or more correlations overlap, the experimental data of the correlations are considered close enough to be combined for the purpose of obtaining a new and more general correlation. When this procedure was applied to the correlations for the 'test wall' (shown in Fig. 12(a)) and those of the 'ceiling' (shown in Fig. 12(b)), the correlations of both elements were found to separate into two different groups. The first included the correlations from two of the configurations, namely the 'heated vertical wall' configuration and the 'radiator under a window' configuration, while the second group included the correlations of the remaining six configurations. Two correlations were developed for each group. one for the vertical wall and one for the ceiling as shown in Table 2. This table also includes the correlations for the cases which are not covered by the correlations mentioned above.

5.3. *Comparison between some of the approximate correlations and those reported in the literature*

Table 2 shows a total of ten correlations which cover some of the most widely used heating configurations in buildings. Two of the most general correlations of this table (equations (3) and (6) which are applicable to a vertical wall of an enclosure) are compared in Fig. 13 with some of those reported in the literature. It can be seen clearly that both correlations yield higher heat transfer coefficients (up to a factor of about 1.70) from those which are currently being used in building thermal models such as those recommended by McAdams [11], ASHRAE [12], and

Table 2. Heat transfer correlations recommended for building thermal modellers

Configuration	Element	No. of tests	No. of points	Correlation $h = C(\Delta T)^n$	SE in C (%)	SE in n (%)	r.m.s. dev.	Max. dev.	Equation No.
Enclosure heated by a fan heater [not valid for a wall opposite the fan heater (see below)], enclosure with a heated floor or a heated edge [not on the heated area], enclosure with and without a window heated by a radiator [not valid if the radiator is under the window (see below) and not valid for the wall close to the radiator (see below)]	Vertical wall	105	137	$h = 2.07(\Delta T)^{0.23}$	1.06	3.83	0.13	0.41	3
	Ceiling	105	102	$h = 2.72(\Delta T)^{0.13}$	0.94	8.84	0.19	0.64	4
Wall close to a radiator [radiator not under a window (see below)]	Vertical wall	17	17	$h = 1.98(\Delta T)^{0.32}$	4.22	10.53	0.13	0.25	5
Enclosure with a heated vertical wall [not for the heated wall] enclosure heated by a radiator under a window	Vertical wall	34	34	$h = 2.30(\Delta T)^{0.24}$	0.68	3.31	0.06	0.14	6
	Ceiling	34	34	$h = 3.10(\Delta T)^{0.17}$	1.61	15.29	0.32	0.91	7
Wall opposite a fan heater	Vertical wall	27	27	$h = 2.92(\Delta T)^{0.25}$	9.80	54.82	0.99	2.74	8
Radiator under the window	Glazing	17	17	$h = 8.07(\Delta T)^{0.11}$	1.04	9.48	0.21	0.63	9
Radiator away from the window	Glazing	17	17	$h = 7.61(\Delta T)^{0.06}$	1.15	17.66	0.17	0.43	10
Large isolated vertical surface	Vertical wall	139	123	$h = 2.03(\Delta T)^{0.14}$	2.57	13.19	0.24	0.83	11
Large heated surface facing upward	Horizontal surface	139	134	$h = 2.27(\Delta T)^{0.24}$	2.99	12.71	0.36	1.57	12

Alamdari and Hammond [13]. Nevertheless, a reasonable agreement can be noticed between the Min *et al.* [2] correlation (which is the result of a study in real-sized enclosures) and equations (3) and (6). Figure 13 shows also the correlation developed for the wall which was opposite the 'fan heater' (equation (8)). This correlation was found to differ by up to a factor of two from those reported in the literature for isolated surfaces.

The heat transfer correlation developed for the exterior surface of one of the vertical walls (equation (11)) which should be comparable to that of an isolated large vertical heated surface is also plotted in Fig. 13, and this was found to be reasonably close to most of those reported in the literature for isolated surfaces but only within a moderate temperature difference (roughly up to 15°C). Due to the significant difference between the index of this correlation (0.14) and the index of most of those reported in the literature for isolated surfaces (1/4 or 1/3), large discrepancies are expected at large temperature differences. The large difference in the indices may be due to the much lower temperature difference covered by this study (up to 5 K) compared with that covered by the studies using isolated surfaces where temperature differences of up to 600°C were used. It should be noted, however, that such very large temperature differences are not common in building thermal applications. It is not known whether the correlation of the present study (equation (11)) can be accurately applied to isolated surfaces with a very high air to wall temperature difference. However, for isolated surfaces with a moderate air to wall temperature difference, equation (11) is recommended.

The correlation developed for the exterior surface of the flat roof of the test cell (which should be comparable to that of a large horizontal heated plate facing upward) is compared in Fig. 14 with some of those reported in the literature. It can be seen from the figure that the correlation (equation (12)) lies between that reported by Min *et al.* [2] (which is obtained from a study in real-sized enclosures) and that reported for the turbulent regime by McAdams [11], ASHRAE [12] and Alamdari and Hammond [13] (which is based on experiments on isolated surfaces). This is consistent with the result from the Min *et al.* [2] study which indicated that the heat transfer from a heated floor to the room air is the same as that for a small horizontal heated plate facing upward. It should be noted, however, that depending on the physical mounting of the isolated surface, the two cases may not always be comparable.

The two correlations developed for the heat transfer between the room air and the ceiling (equations (4) and (7)) are compared in Fig. 15 with the only one found in the literature to cover such a configuration, namely Min *et al.* [2] (which was obtained from studies in a room with a heated floor panel). Both equations were found to yield lower values of the heat transfer coefficient.

5.4. *Temperature stratification and local variations in the h values inside the test cell*

Beside the measurements of the average heat transfer coefficient on the different elements of the test cell, measurements of local heat transfer coefficients as well as local temperatures in the test cell were also made. The importance of the local values comes from the fact that local heat transfer coefficient forms the thermal link between the room thermostat and the room

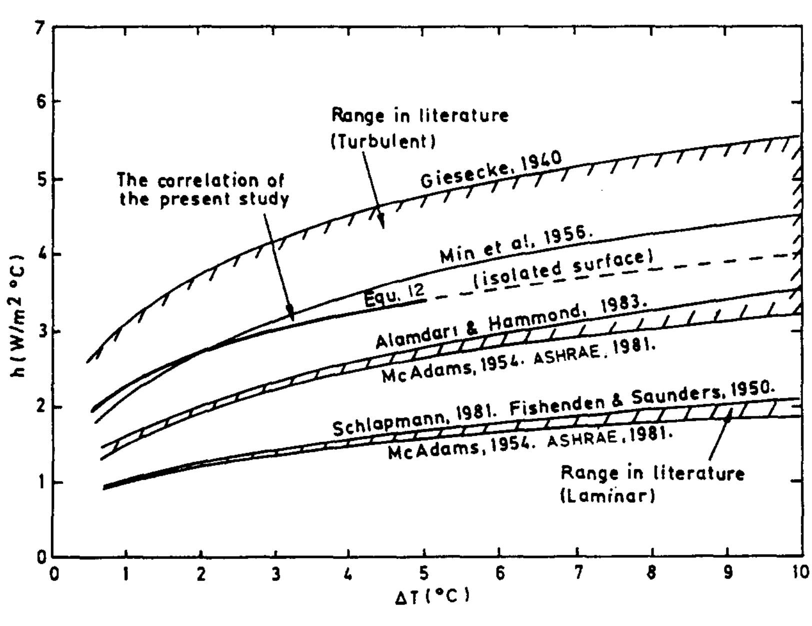


FIG. 14. Comparisons between the correlation of the present study and those reported in the literature for heated surfaces facing up.

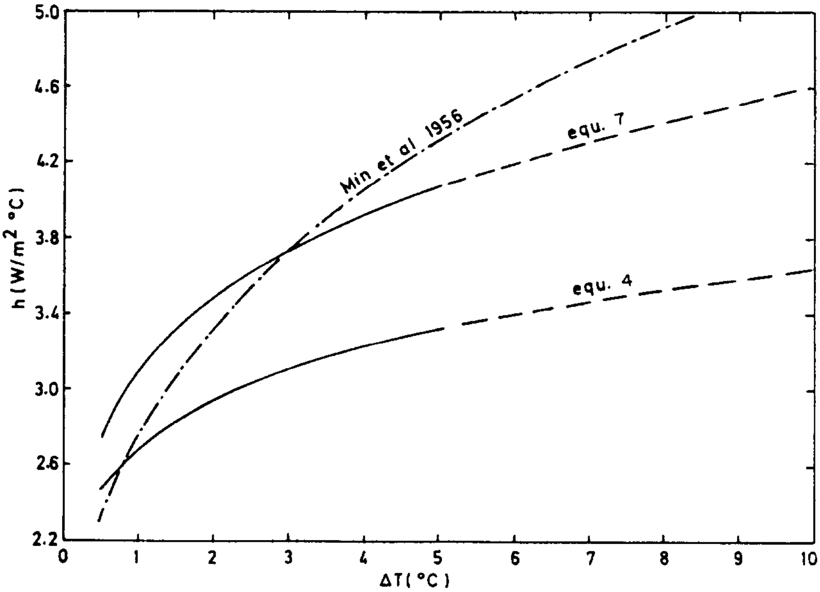


FIG. 15. Comparison between the correlations of the present study and that reported by Min *et al.* [2] for heat transfer from the room air to the ceiling.

air. The room thermostat itself may sense the local air temperature rather than the average temperature in the enclosure, especially in an enclosure with poor air circulation. In practice, therefore, the local values have an important influence on room temperature control and hence on the thermal comfort.

The data of the 'heated vertical wall' configuration (but not that of the heated wall itself) are chosen to demonstrate the local temperature variations. Where there is more than one temperature measurement at a certain level, the local temperature was considered to be the average value. Figure 16 shows the local air

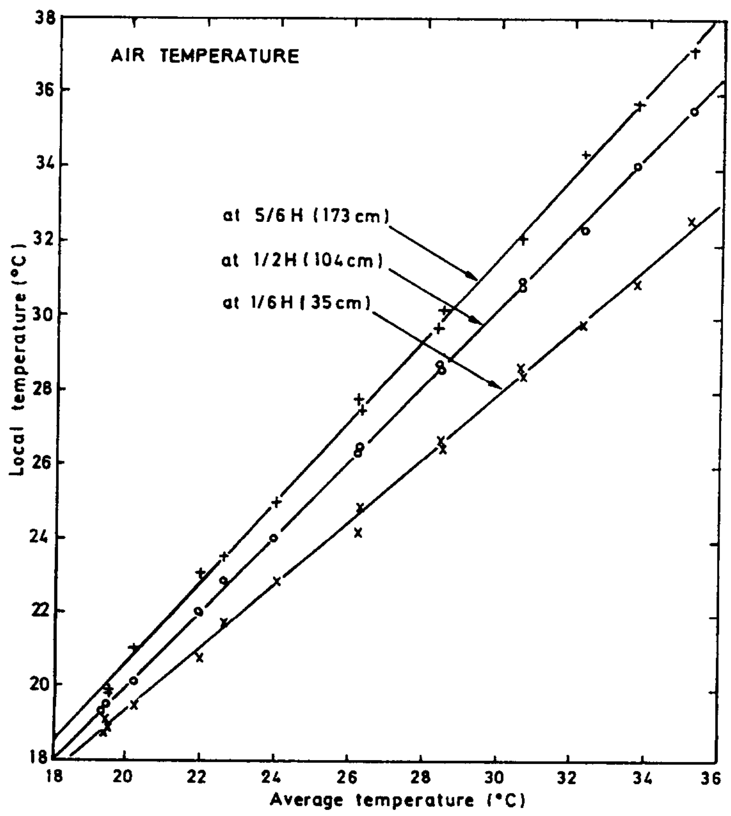


FIG. 16. Average vs local air temperature at three different levels near the test wall.

temperature at three different levels as measured near the 'test wall', namely the middle of the wall and at an equal distance above and below the middle, and how these values deviate from the average air temperature in the enclosure. As expected, the local air temperature increases as the distance from the floor increases. At the typical comfort temperature of around 20°C the stratification (averaged over the total height of the wall) was found to be 0.94°C per metre, which is consistent with the 1°C per metre quoted in the literature. (See for example Hammond [14].) It can also be noticed that the rate of change in the local air temperature with the height is not the same for each half of the wall. Also, as the average air temperature in the test cell increases, the temperature stratification becomes more severe. The stratification in the surface temperature of the wall was found to be almost identical to that of the air. The local heat transfer coefficient was found to vary by an average of $\pm 10\%$ from the average value over the wall area.

6. CONCLUSIONS

The data reported in this study were obtained in a real-sized test cell measuring $2.95 \times 2.35 \times 2.08$ m (length × width × height). The tests covered nine different configurations. These were a cell heated by a fan heater, a cell with a heated floor, with a heated vertical wall, and with a heated edge; a cell heated by a radiator placed at two different locations in the cell, a cell with a window (without curtains) heated by a radiator placed at two different locations in the cell, and finally a cell with a simulated forced convection on one of the interior vertical surfaces. All the tests were carried out under a controlled steady-state condition with the test cell tightly sealed, empty and unlit. Under these conditions, the following conclusions may be drawn:

(1) A total of 36 correlations were developed from the different configurations mentioned above. These covered the free convection heat transfer between the vertical and horizontal surfaces of a real-sized enclosure and the adjacent air. The data from the similar correlations were combined together in order to obtain new and more general correlations which can be applied to more than one configuration. The ten new correlations of Table 2 were found to be valid for the cases shown in that table which represent some of the most widely used heating configurations in buildings.

(2) The heat transfer coefficient on the interior surfaces of the vertical walls of the test cell were found to be higher by up to a factor of 1.7 from those which are currently used in building thermal models. The discrepancy rises up to a factor of 2 for the fan heater configuration. Nevertheless, a reasonable agreement was found between the data of the present study and those reported by Min *et al.* [2] for this case.

(3) Natural convection data given by other investigators for small heated horizontal surfaces facing upward were found to be in a good agreement with the correlation found in this study for the exterior surface of the horizontal roof.

(4) The heat transfer correlation developed by this study for the exterior surface of one of the vertical walls was found to be in a reasonable agreement with those reported in the literature for isolated vertical surfaces but only within a moderate air to wall temperature difference (roughly up to 15°C).

(5) The heat transfer coefficient in the enclosure, and hence the heat losses, can be reduced by minimizing the vertical area of the heating device.

(6) Compared with an empty enclosure with a radiator placed under a window (without curtains), the heat losses from the window and the ceiling can be reduced by moving the radiator away from the window.

(7) The heat transfer coefficient on a wall close to a radiator was found to be higher (by about 14% for an air to wall temperature difference of 5°C) than those which occur on the rest of the vertical walls of the enclosure.

(8) The heat transfer coefficient on a single-glazed window of an enclosure is at least three times higher than those which occur on the other opaque elements of the enclosure such as vertical walls and ceiling. Also, the heat transfer coefficient on the window is much less temperature difference dependent compared with that of the vertical walls (an index of roughly 0.2 s compared with 0.25 s, respectively).

(9) At the typical comfort temperature of around 20°C in the cell, the stratification in the air temperature near the wall and in the wall temperature for the free convection configurations (averaged over the total height of the wall) was found to be around 1°C per metre.

(10) Local heat transfer coefficients on a vertical wall of a real-sized enclosure were found to deviate by an average value of $\pm 10\%$ from the average heat transfer coefficient on the wall.

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VALIDATION DES COEFFICIENTS DE TRANSFERT THERMIQUE SUR DES SURFACES INTERNES DE BATIMENT A PARTIR D'UNE CELLULE D'ESSAI DE TAILLE REELLE

Résumé—.On conduit des expériences pour déterminer les coefficients de transfert thermique sur des surfaces internes de bâtiment (parois verticales, plafonds et vitrages) en utilisant une cellule d'essai de taille réelle 2,95 × 2,35 × 2,08 m (longueur × largeur × hauteur). Un total de 142 essais, chacun durant 24 heures environ, correspond à des conditions permanentes contrôlées pour couvrir neuf configurations de chauffage plus courantes. Dix nouvelles formules sont données pour être utilisées par les bureaux d'études. Les formules diffèrent jusqu'à un facteur deux de celles qui sont couramment utilisées dans les calculs de thermique du bâtiment.

BESTIMMUNG DER WARMEÜBERGANGSKOEFFIZIENTEN AN OBERFLÄCHEN IN GEBÄUDEN MIT HILFE EINER LABORTESTZELLE IM ORIGINALMASSTAB

Zusammenfassung—Zur Bestimmung des Wärmeübergangskoeffizienten an Oberflächen in Gebäuden (senkrechte Wände, Decken und Verglasungen) werden experimentelle Laboruntersuchungen an einer Testzelle im Originalmaßstab mit den Abmessungen 2,95 × 2,35 × 2,08 m (Länge × Breite × Höhe) durchgeführt. Insgesamt wurden 142 Versuche, von denen jeder ungefähr 24 Stunden dauerte, unter stationären Bedingungen ausgeführt. Die Versuchsbedingungen entsprachen neun meistverwendeten Heizkörperanordnungen. Als Ergebnis wurden 10 neue Korrelationen für den Wärmeübergang entwickelt und in benutzerfreundlicher Form dargestellt. Es zeigt sich, daß die Gleichungen um bis zu 100% von denjenigen abweichen, die heute bei der Simulation des thermischen Verhaltens von Gebäuden verwendet werden.

ОПРЕДЕЛЕНИЕ КОЭФФИЦИЕНТОВ ТЕПЛОПЕРЕНОСА НА ВНУТРЕННИХ ПОВЕРХНОСТЯХ ЗДАНИЙ С ИСПОЛЬЗОВАНИЕМ ЭКСПЕРИМЕНТАЛЬНОГО ЭЛЕМЕНТА НАТУРАЛЬНОЙ ВЕЛИЧИНЫ, НАХОДЯЩЕГОСЯ ВНУТРИ ПОМЕЩЕНИЯ

Аннотация—Экспериментально исследуются коэффициенты теплопереноса на внутренних поверхностях зданий (таких как вертикальные стенки, потолки и застекленные поверхности) с использованием тестового элемента в натуральную величину с размерами 2,95 × 2,35 × 2,08 м (длина × ширина × высота), находящегося внутри помещения. Броведено 142 опыта длительностью около 24 часов кажый при контроле условий стационарности для девыти наиболее употребительных конфигураций в зданиях. Десять новых обобщенных соотношений выведены и представлены в виде, удобном для применения в тепловом моделировании сооружений. Эти соотношения, как было обнаружено, отличаются до двух раз от используемых в настоящее время в тепловых моделях зданий.