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BY

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LARGE-SCALE WALL HEAT-FLOW MEASURING APPARATUS†

By K. R. SOLVASON*, OTTAWA, CANADA

THE Division of Building Research of Canada's National Research Council has recently completed construction of an apparatus to measure steady-state and periodic heat flow through 8-ft square wall sections at its Building Research Centre in Ottawa. The apparatus (Figs. 1 and 2) consists of two 8- by 8- by 4-ft boxes open on one side, between which the test wall is placed. One box is maintained at the desired constant cold side temperature from −35 F to +50 F for steady-state tests or varied according to some predetermined cycle for periodic heat flow tests. The other box is electrically heated to maintain a constant warm side temperature of from 65 F to 75 F. The heat transmission coefficient for the wall specimen is calculated from the measured electrical input and temperatures.

In the design of the apparatus an attempt was made to overcome the limitations of the following methods for determining heat transmission coefficients: (*a*) calculations using predetermined thermal conductivities or conductances of the various components and air films; (*b*) the ASTM guarded hot box method; and (*c*) the use of heat flow meters.

LIMITATIONS OF EXISTING METHODS

Many wall combinations are difficult to assess by means of calculation for the following reasons:

1. The only reliable thermal conductivities available for most materials are those obtained by hot plate tests on dry materials which may be considerably different for actual operating conditions.

2. Surface conductances are not necessarily constant over the whole of a wall surface and the average conductance will depend on the temperature difference between the wall surface and air, and that between the wall surface and its surroundings.

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† This paper is a contribution from the Division of Building Research and is published with the approval of the Director.

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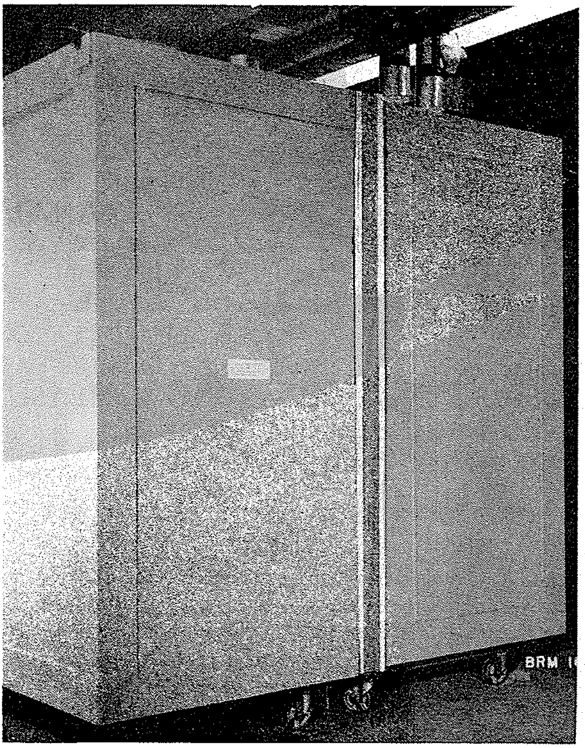

FIG. 1. . . .VIEW OF APPARATUS

3. Most wall combinations contain heat flow paths of differing conductance, the effect of which is difficult to assess without resorting to a rigorous mathematical treatment or to analogue techniques, because of lateral heat flow in some of the components.

The ASTM guarded hot box[1] method consists essentially of placing an electrically heated metering box over the center portion of the warm side of the test wall. Surrounding the metering box is a larger guard box. The temperatures in the 2 boxes are adjusted so that there is no temperature difference (and hence no heat flow) across the walls of the metering box. The conductance is calculated from the electrical input to the metering box and the temperature difference across the wall.

The main disadvantages of this method are: (*a*) The metering box interferes with the convection over the test wall, so that forced convection must be resorted to and this may give film coefficients different from those occurring in practice. It is difficult to produce equal coefficients for the metering area and the guard area, so that lateral heat transfer may occur from the measuring area to the guard area. (*b*) The metering box placed over the central portion of the test wall measures only the heat flow into that portion but it has been shown by G. O. Handegord and N. B. Hutcheon[2] that this is not necessarily the average heat flow for the whole test

[1] Exponent numerals refer to References.

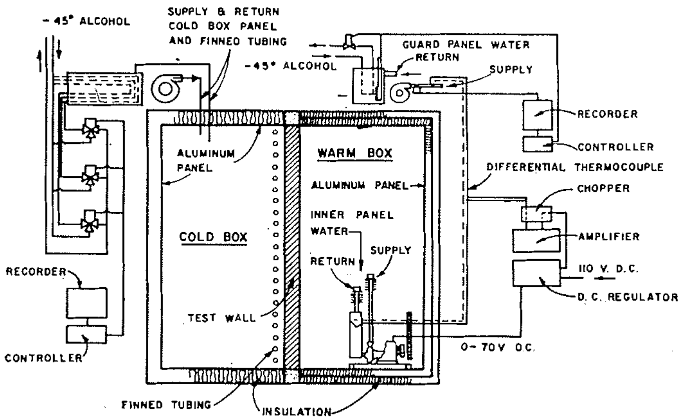

FIG. 2. . . .SCHEMATIC CROSS SECTION OF APPARATUS

wall, particularly for walls containing vertical air spaces; blocking of air spaces in the test area may change the conductance substantially. (*c*) Radiation exchange is indefinite and it is difficult to produce the same effect on both the metering area and the guard area of the test wall. Differences in the radiant exchange from the inner and outer surfaces of the metering box may require that different air temperatures be provided in the test box and the guard box in order to maintain zero heat flow across the test box walls and this may lead to lateral heat transfer in the test wall. (*d*) In many cases the metering box will not cover a representative complete module of the test wall.

The use of heat flow meters of the multiple differential thermocouple type is considered inadequate for the measurement of heat transmission coefficients for the following reasons:

1. The heat flow must be integrated over the whole wall surface and an accurate weighted average is difficult to obtain on walls where large variations in heat flow occur.

2. The meter, if placed on the wall surface, probably interferes with the convective transfer.

3. The thermal resistance of the meter itself reduces the heat flow in the area covered by the meter. Although corrections for this effect can be calculated, for some cases, they are probably not reliable when applied to walls that have surfaces of high conductivity material, since heat transfer in the plane of the wall surface permits part of the heat to bypass the meter.

4. Good contact between the meter and wall surface is often difficult to achieve and air spaces between the meter and wall surface will increase the thermal resistance through the meter-wall combination; heat flows in the plane of the meter to the points of contact are possible so that, if the thermopile element is localized in, say the center of the meter, the heat flow might bypass the element or the heat flow through it might be exaggerated, depending on where contact occurs.

CRITERIA FOR DESIGN OF THE APPARATUS

With these considerations in mind the heat flow apparatus was constructed to meet the following requirements. *First*, it was designed to meter the heat flow into the whole of representative sections of building walls, and *second*, to expose the

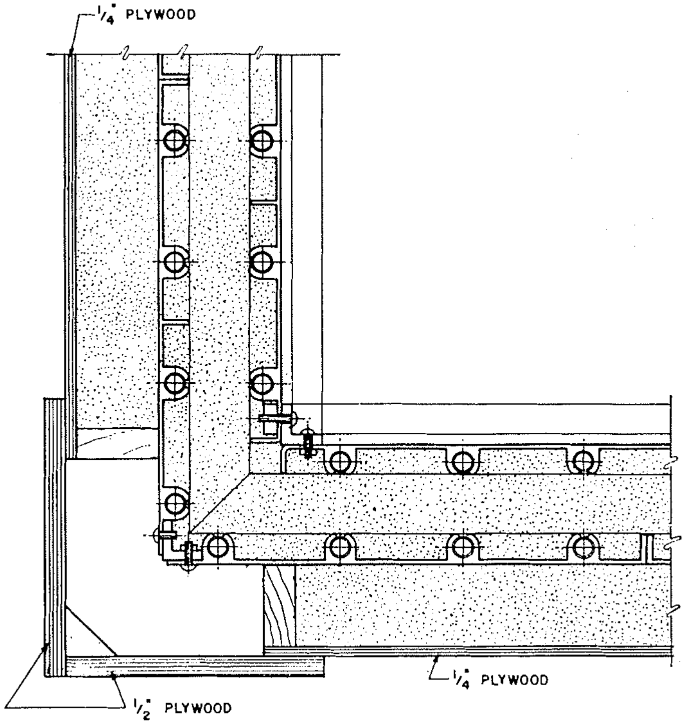

FIG. 3. . . .DETAILS OF WARM BOX WALLS

test wall to controlled temperature surroundings as well as to air at controlled temperature. *Third*, it was built to operate without forced air circulation over the warm side of the test wall in order to produce warm-side film coefficients approaching those occurring in practice. *Fourth*, it operates with the cold-side temperature either constant or varying according to some predetermined cycle.

DESCRIPTION OF APPARATUS

The apparatus consists essentially of a cold box and a guarded hot box combination, with an 8-ft square measuring area. This size was chosen in order to accommodate walls with air spaces 8-ft high and with modules up to 4-ft wide. It was decided to limit the depth of the box to 4 ft for practical reasons. This was considered large enough to produce natural convection effects, approaching those occurring in practice.

The warm box was constructed with a liner of aluminum panels (Fig. 3) similar to those used to line the ASHAE environmental rooms. These panels, which were developed for use in radiant heating and cooling applications, consist of aluminum extrusions with holders for copper tubing, through which liquid is circulated. The inside of the box was painted to increase the emissivity of the panels to about 0.9. A second set of panels was installed outside of the liner and separated from it by thermal insulation. When the outer panel is maintained at the same temperature as the inner panel, it forms a guard to prevent any heat transfer across the walls of the box. Insulation (2-in. mineral wool batts) and a plywood covering were applied to the outside of the guard panel.

A water reservoir, pump, and pump motor for circulating water through the tubes of the inner panel are located inside the box. The pump is belt driven by a series-wound d-c motor. The voltage-current characteristics of this type of motor permit the energy input to be varied from 25 watt (85 Btu per hr) to 500 watt (1700 Btu per hr) by regulating the voltage from 15 to 90 volt. Direct-current power was considered easier to control and to measure accurately than a-c. The electrical input to the pump motor is normally sufficient to provide all of the heat requirements; thus the energy imparted to the water by the pump serves to heat the panel and the energy loss from the motor and drive serves to heat the air directly. Air heaters and water heaters are also provided so that the ratio of air heat to panel heat may be varied if desired.

Twelve parallel water circuits (each circuit supplying 4 by 4 ft of panel) are used to supply the panels in order to permit the use of a large quantity of water without excessive pump head and pump power. At the maximum input of 1700 Btu per hr, about 20 gpm of water are circulated. Since only about 60 percent or less of the total power input will be supplied to the water, the drop in temperature passing through the panel is only about 0.085 F. The ratio of water quantity to input increases as the total input decreases, so that the temperature drop through the panel is always less than 0.085 F. It can also be shown that the panel to water temperature difference and also the temperature variation over the panel surface is quite small, not more than 0.1 F.

The water reservoir is incorporated into a shield in order to prevent the test wall or the box walls from *seeing* the pump motor, which will operate at some temperature higher than the panel.

A second reservoir and pump located outside of the warm box supplies water to the outer guard panel. The reservoir contains a cooling coil and an electric heater to control the water temperature.

The cold box is simply a well insulated box with an aluminum panel liner and a sheet metal exterior vapor barrier covering. The panel liner serves a three-fold purpose: (*a*) to provide controlled temperature surroundings to which the test wall is exposed; (*b*) to provide some convective heat exchange area; and (*c*) to intercept heat gains to the box itself and thus reduce the heat exchange area required inside the box. Additional heat exchange area (312 sq ft surface area of finned tubing) is provided to bring the air temperature as close as possible to the panel temperature. Liquid (1/3 water, 1/3 ethylene glycol and 1/3 methyl alcohol) at the desired temperature is circulated through the tubes of the panel liner and the finned tubing. This liquid is cooled in a shell and tube heat exchanger by a low temperature liquid (−45 F methyl alcohol) from a central low temperature chiller serving other laboratories in the Centre. The heat exchanger contains eight 3/8-in. outside diam copper coils 20 ft long. The control valves are arranged so that the cold liquid can be circulated through 2, 4, 6, or all 8 coils depending on the temperature difference desired.

CONTROLS AND MEASURING INSTRUMENTS

The control and measuring instruments for the apparatus were selected in order to limit any specific error to as low a value as practicable, or to a maximum of one percent at the minimum heat flow and temperature difference. A heat flow of 200 Btu per hr and an overall temperature difference of 30 F were selected as a design basis. Errors in the determination of an overall heat transmission coefficient may arise from errors in measurement, in control of temperatures and power input, and from heat leakage.

Measurement of Temperatures: Temperature measurements are made with 30-gauge copper-constantan thermocouples and an electronic self-balancing temperature indicator. Checks of the indicator and thermocouple wire indicate that an accuracy of ± 0.05 F can be obtained. The temperature difference measurements are thus accurate to about ± 0.1 F deg. The percentage error is then about ± 0.33 percent at an overall difference of 30 F.

Measurement and Control of the Heat Input, and Warm Box Temperatures: Warm box control system is required to adjust the input voltage to the warm box to a stable value such that the heat input equals the wall transmission within 2 Btu per hr or one percent, and to maintain the guard panel at a temperature such as to prevent a heat transfer of more than 2 Btu per hr across the box walls. The panel liner of the warm box and the water have a large thermal capacity, about 330 Btu per F deg. The maximum permissible temperature change is, therefore, only 0.006 F deg. per hr in order to limit heat storage to 2 Btu per hr.

The insulation between the inner and guard panels provides a calculated conductance of approximately 0.125 Btu per (hr) (sq ft) (F deg) or a total heat flow of 24 Btu per (hr) (F deg). Thus for precision of one percent, the average temperature difference must be controlled to about ± 0.08 F deg. A moderate ripple in the outer panel temperature can be tolerated, provided the average temperature is within the ± 0.08 F deg, since the insulation between the inner and outer panel will provide sufficient damping to prevent any appreciable heat flow at the inner panel.

The control system is shown in block form in Fig. 1. A thermocouple actuated electronic recorder with a 3-action (proportional + reset + rate) controller is used to control the guard panel temperature. This controller regulates a solenoid cooling valve and electric heater with manual switches, so that heating and cooling can be supplied continuously on, continuously off, or pulsed on-off by the controller. The d-c input to the inner panel pump is supplied by a d-c power supply[3], the output voltage of which is regulated from 0 to 70 volt by the amplified unbalance from a differential thermocouple to maintain equal inner and guard panel temperatures. The differential thermocouple signal is first converted to a square wave by a mechanical chopper, amplified by a high gain low level d-c amplifier and rectified by the chopper. The higher voltage d-c signal is then used to regulate the output voltage of the power supply.

This system has been used with only moderate success. Slight fluctuations in the guard panel temperature produce large fluctuations in the inner panel power. The effect of rapid fluctuations has been eliminated successfully by applying thermal damping to the guard panel thermocouple junction. Lower frequency temperature swings produce a similar swing in inside power. This effect, which is not serious for steady-state tests but unsuitable for periodic tests, could be eliminated by converting the present thermocouple recorder-controller to a shorter span and more accurate resistance-element-type.

Some additional difficulty has, however, been experienced with the chopper-amplifier system. The chopper point adjustment is difficult to maintain over long periods and this, together with stray inductive pickup, produces fluctuations in the power and errors difficult to assess. The present power supply, in addition, is not of sufficient capacity for the maximum inputs required. The control and power system is therefore being replaced with equipment which has recently become available. A magnetic amplifier of 500-watt maximum capacity is being procured for a power supply. The magnetic amplifier regulated by the present recorder (converted to a resistance type) and a current adjusting controller will be used to control the inner panel temperature. The recorder sensitivity will be approximately 0.001 F deg and the combination is expected to control to 0.002 to 0.003 F deg. The error due to the capacity effect of the box is thus expected to be no more than 0.5 percent at the minimum heat flow.

The guard panel temperature will be controlled by an on-off electronic temperature controller which has a sensitivity of 0.001 F deg and is expected to control temperatures to well within ± 0.03 to 0.04 F deg of the inner panel. The unbalance error is then expected not to exceed about 0.5 percent.

The input voltage and current are measured by a millivolt recorder connected across appropriate resistors. The maximum recorder error, with calibration, can be limited to less than ± 0.25 percent of full scale so that by sizing the resistors to give larger than half-scale deflections this error should always be less than 0.5 percent. The error in voltage times current is less than 1.0 percent.

Control of Cold Box Temperatures: No special problems were encountered in the control of the cold box temperature. Here a thermocouple-actuated electronic recorder with a 3-action temperature controller is used to control 1, 2, or 3 solenoid valves which regulate the flow of cold liquid through the heat exchanger. The number of valves used is selected manually, depending on the temperature, to make ¼, ½, ¾ or all of the heat exchanger area effective. With this combination, the cold-side temperature can be held constant to about ± 0.1 F deg or about 0.33 percent of the temperature difference.

The controller is now equipped so that an approximate sine wave variation in temperature can be produced with amplitudes of 5 to 15 F deg and periods of 6, 12 or 24 hr. A program controller may be added later with which any desired wave form can be produced.

If all the measurement and control errors were additive, the total error is expected to be less than 2.0 percent. Operation to date indicates that heat transmission coefficient values can be reproduced within 1.0 percent. Tests to assess measurement errors experimentally will be conducted in the near future.

EDGE HEAT LEAKAGE

Heat leakage at the edge of the test wall is another source of error. The apparatus is located in a laboratory where the temperature is maintained close to the usual warm-box operating temperature. Heat leakage will, therefore, tend to occur through the edge of the test wall into the cold box. The edges of the test wall will usually be surrounded with low conductivity material ($k = 0.25$) slightly thicker than the test wall and some 5½ in. wide (the width of the box edges) as shown in Fig. 4. Aluminum angles attached to the cold box panel and extending 2 in. over the box edge are installed to extend the low temperature plane outside of the test wall area. A similar angle is attached to the guard box panel so that the heat loss for this portion will be supplied by the guard panel rather than the inner panel.

The additional thickness of insulation (x in Fig. 4) should be such that its thermal resistance approximately equals the air-film resistance, so that the temperature in the plane of the wall surface will equal wall surface temperature. In many cases, the arrangement of the edges of the test walls may be such that good edge guarding

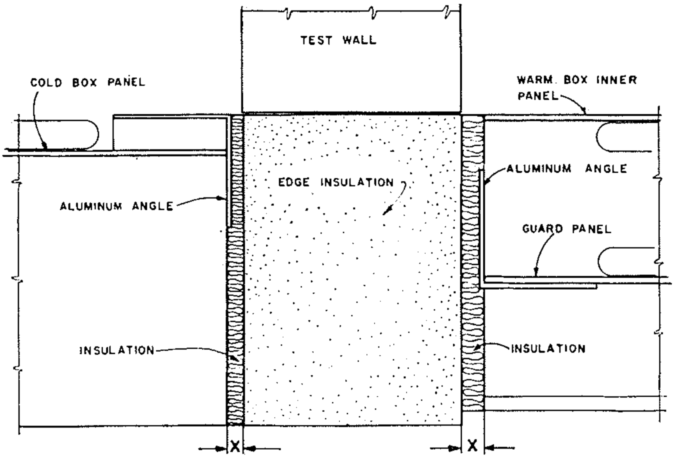

Fig. 4. . . .Arrangement at Test Wall Edges

is very difficult. It may therefore be necessary to use test sections smaller than 8 ft square and use a filler of known thermal properties.

It would be quite difficult to calculate the error introduced by edge leakage, except perhaps by one of the relaxation or analogue methods. Tests conducted to date indicate no significant temperature gradients on the test wall surfaces near the edges, so that this error can be assumed to be rather small. Further tests to assess edge effects will be carried out in the near future.

Operating Characteristics

The total heat transfer at the warmside surface of the test wall will be the sum of the net radiant exchange from the box panels to the test wall plus the convective exchange from the air to the test wall.[4] This can be expressed as:

$$q_t = q_r + q_c = F_c F_e \times 0.173 \left[\left(\frac{T_p}{100} \right)^4 - \left(\frac{T_w}{100} \right) \right] + x(t_a - t_w)^y \quad . \quad . \quad (1)$$

The radiant and convective components are usually accounted for by coefficients h_r and h_c respectively, such that

$$q_t = h_r(t_p - t_w) + h_c(t_a - t_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The usual surface conductance coefficient, f_i, is the sum of h_r and h_c, hence,

$$q_t = f_i(t_i - t_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If the average air temperature (t_a) in the apparatus is not equal to the panel temperature, t_p, an equivalent temperature t_i, can be calculated from Equations 1, 2 and 3 which would produce the same heat flow as the actual t_p and t_a.
Thus,

$$t_i = \frac{h_r t_p + h_c t_a}{h_r + h_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The form factor F_c for the test wall and box can be shown to be equal to 1.0, and the emissivity factor F_e to equal $e_p e_w$ where e_p and e_w are the emissivities of the box walls and the test wall surface respectively.

If e_p and e_w are known, the factors for convective heat flow x and y in Equation 1 can be evaluated by measurements of panel temperature, t_p, average test wall temperature, t_w, and average air temperature, t_a, at several values of heat transmission, q_t. For the range of conditions expected, these factors are not expected to vary so that they need be evaluated for one wall only. In any later tests, if the average wall surface temperature can be measured, it will be possible to calculate the wall surface emissivity. For those walls where surface temperature variations make the assessment of the average difficult, it will be possible to calculate the average temperature if the emissivity e_w is known. The cold-side heat exchange can be treated similarly.

Thus the surface conductance can be accurately established, the average surface temperatures defined, and the conductance of the wall itself (without surface conductances) can be calculated. Overall conductances for any other surface conductances can then be calculated.

NOMENCLATURE

q_t = total heat flow rate, Btu per (sq ft) (hr).
q_r = net radiant exchange, Btu per (sq ft) (hr).
q_c = convective exchange, Btu per (sq ft) (hr).
F_c = the form factor for radiant exchange between the box panels and the test wall.
F_e = emissivity factor.
T_p = box panel temperature, Fahrenheit absolute.
T_w = average test wall temperature, Fahrenheit absolute.
t_a = average air temperature, Fahrenheit.
t_p = box panel temperature, Fahrenheit.
t_w = average wall surface temperature, Fahrenheit.
t_i = temperature, Fahrenheit, equivalent to t_a and t_w.
x = constant.
y = constant.
h_r = surface coefficient for radiant heat transfer Btu per (hr) (sq ft) (F deg).
h_c = surface coefficient for convective heat transfer Btu per (hr) (sq ft) (F deg).
f_i = combined surface coefficient Btu per (hr) (sq ft) (F deg).

The warm side surface conductance, f_i, resulting in the apparatus is expected to be quite close to the still air conductances that result in practice, if a wall is exposed to air and surroundings at nearly equal temperatures. This coefficient is not a constant but rather a function of the temperature level, the temperature difference, and the wall emissivity.

The cold-side surface conductance coefficients in the apparatus are expected to be much lower than will be desired in most cases. Forced circulation either by fans or induced by jets will be provided to increase the conductance as required.

CONCLUSION

This apparatus is expected to provide an accurate and realistic means for obtaining steady-state heat flows or conductances through built-up wall sections, windows, and doors. The sample size used in the apparatus is considered large enough so that the results, particularly for inside surface conductances, will be representative of much present-day construction.

The test wall is exposed to controlled temperature surroundings as well as to controlled temperature air. The convective and radiant components of the surface conductances can thus be evaluated.

The apparatus also provides a means for assessing the transient response of wall sections. In this respect the effect of moisture movement induced by cycling temperature can be investigated.

The apparatus can also be adapted to general calorimetry tests such as the lag characteristics and maximum air-conditioning loads from appliances. Tests for water vapor and air transfer can also be carried out without any major additions or alterations.

ACKNOWLEDGMENTS

The author is indebted to N. B. Hutcheon, assistant director of the Division of Building Research, and to A. G. Wilson, head of the Building Services Section, for their many helpful suggestions in the design of the apparatus and in the preparation of this paper, and to H. L. Egan, laboratory assistant, who performed most of the construction work.

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DISCUSSION

HARRY BUCHBERG, Los Angeles, Calif.: The author has presented an excellent paper on the design and development of a very well conceived and useful piece of apparatus. As pointed out, many different studies can be made with its use. Of particular interest would be the determination of transfer functions in order to check the validity of calculations that are based on a knowledge of the properties of the materials that go into the structure.

As another comment the impression was gained that the author does not have very much confidence in the use of heat meters. His attention might be drawn to the work of Professors Gier and Dunkle at the University of California in the determination of wall conductances using heat meters, in which they solved at least many of the difficulties mentioned by the author.

J. G. MACORMACK, New York, N. Y.: Does the equipment permit orientation of direction of heat-flow? Is it possible to run the equipment in a horizontal position and get the heat-flow downward?

AUTHOR'S CLOSURE: Gier and Dunkle's paper is familiar and use has been made of quite a number of their meters and this is one of the reasons for the remarks made in the paper.

With minor modification the equipment can be turned over and used for floor tests or it can be turned over the other way and used for roof and ceiling heat-flow. This would require some modification in the circulation system, but it is just a matter of getting the pump sitting in the right location.

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