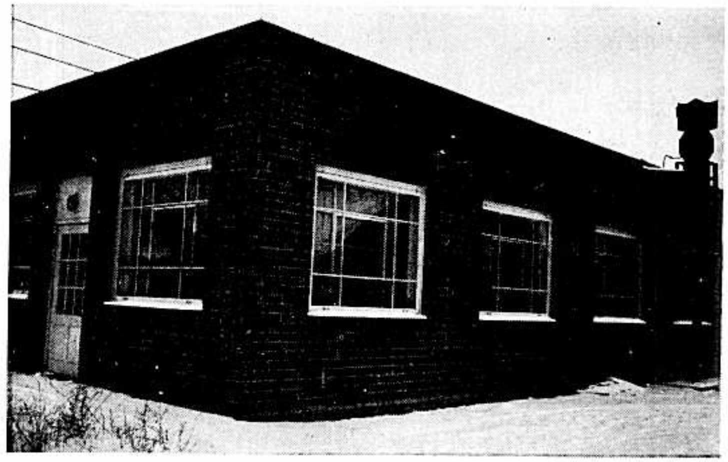
# Limitations in Heat Output from Radiant Heating Panels

by
D. L. MILLS, L. J. LATART and F. W. HUTCHINSON

Reprinted from Heating and Ventilating
January 1949

REVERE COPPER AND BRASS INCORPORATED
EXECUTIVE OFFICES, 230 Park Avenue, New York 17, N. Y.

# Tests Show Limits in Rating Floor Panels



Metallurgical Laboratory at Revere Copper and Brass Incorporated, where panel rating tests were carried out.

D. L. MILLS,* L. J. LATART,* and F. W. HUTCHINSON**

**Tests of three actual panels in service are described and conclusions drawn regarding probable output.**

IN the design of any type of heating system the first problem is that of determining the rate at which heat must be supplied to the structure, and the second problem is that of providing sufficient transfer surface to assure delivery of heat at the required rate. With radiant heating, more than with other types of heating systems, these two problems are not separable since it can be shown (1)[1] from theory that the rating of a particular panel—operating at fixed temperature in a comfortable room—will vary with the structural and ventilation characteristics of that room. Thus for design conditions corresponding to an outside air temperature of 30F and a floor panel surface temperature of 80F, analysis will show (2) that the rating of the panel will vary with the type of structure and the amount of ventilation from a low of 34.5 Btu per (hr) (sq ft) of panel surface to a high of 38.5 Btu per (hr) (sq ft); a rating change in excess of 10%. For correct design it is thus evident that a fixed panel rating cannot be used with accuracy; some of the published design procedures, including those in reference (2), take account of the variation in rating and apply it in evaluating the required size of panel.

In spite of the above disadvantages it is frequently desirable in practice to keep in mind some average value of a rating which can be used as a rough check or for making preliminary estimates of required panel size. Many and widely varying recommendations have been made concerning the ratings of floor-type radiant panels, but very little data are available in the published literature to support or to substantiate the recommended values. For floor panels operating at 85F design temperature in a room with 70F air and 70F mean radiant temperature, claims have been made of ratings as high as 90 to 100 Btu per (hr) (sq ft), but—insofar as the authors have been able to determine—there are no published records of the field tests or the laboratory experiments which justify these extraordinarily high ratings. For the conditions stated above, a panel rating of 90 Btu per (hr) (sq ft) would require that the combined equivalent film coefficient for transfer from floor panel to its surroundings be of the order of $90/(85-70) = 6$ Btu per (hr) (sq ft) (F); such a value might be anticipated if the air velocity *within the room* were of the order of 15 miles per hour, but for normal air movement in an occupied space such a claim is fantastically at variance with expectancy as based on published laboratory data.

The great majority of recommendations for floor panel ratings are based on calculation from the experimentally determined convection coefficients presented in the 1938 paper (3) of Wilkes and Peterson. These investigators used a relatively large flat surface in their tests and conducted the work under conditions of no air movement. Based on their results the convection coefficients which have been widely used in panel heating analysis and design are 0.4 Btu per (hr) (sq ft) (F) for a ceiling panel, 0.7 Btu per (hr) (sq ft) (F) for a wall panel, and 1.1 Btu per (hr) (sq ft) (F) for a floor panel. Radiant transfer is not a function of surface position (except insofar as the angle factors may vary in an actual installation) hence the equivalent coefficient for radiation will be the same for wall, floor, or ceiling panels. In the usual range of temperatures an average value of the equivalent radiant film coefficient is unity, hence the combined equiva-

*Research Department, Revere Copper and Brass Incorporated, Rome, N. Y.
**Professor of Mechanical Engineering, University of California, Berkeley, Calif.
[1]Numbers in parentheses correspond to references listed at the end of this article.

Part of test room No. 1. Note workbench enclosed to floor and instrument on floor in foreground

lent film coefficients for ceiling, wall, and floor panels are 1.4, 1.7, and 2.1 Btu per (hr) (sq ft) (F), respectively. For *any* inside surface, irrespective of its orientation, the ASHVE *Guide* uses 1.65 Btu per (hr) (sq ft) (F) as the combined film coefficient, and it is this value which has been used in calculating the many hundreds of $U$ values that appear in the *Guide.*

The importance of accuracy in the inside film coefficient applies to all types of heating systems, but it is interesting to note that if the selected value is too small the error will provide a factor of safety in rating a heating panel, but will be non-conservative with respect to load determination. Thus if the film coefficients of the order of 6.0 which have been claimed for some floor panels were found to be correct it would follow that a wall which the *Guide* now lists as having a $U$ of 0.25 would actually have a $U$ of approximately 0.28, hence would lose 12% more heat than present calculations show. On the other hand, a floor panel rated by use of the 2.1 coefficient would obviously be conservative (by almost 300%!) as compared with one rated on the basis of 6.0. From these considerations it is evident that need exists for a check on the values of film coefficients which are actually occurring on installed floor-type radiant panel heating systems.

## Experimental Set-Up

In order to check actual performance against theoretical expectancy, individually controlled floor panels in each of three rooms in an occupied building were tested. Before discussing the test conditions, attention should be specifically called to the very important fact that the results of tests of this kind are entirely independent of the method used to liberate heat within the structure of the panel. Whether air, steam, water, or electricity provides the energy and whether copper tube or iron or steel pipe is used, the results should be the same since consideraton is limited to just three things: (1) The amount of heat delivered to the panel; (2) The average surface temperature of the panel; (3) The average air temperature within the room. Likewise, neither the diameter nor the spacing of tubes should have any effect on the results providing only that the surface of the panel is heated to a reasonable degree of uniformity.

The three systems on which experiments were conducted all used hot water in ¾-in. copper tube (⅞-in. O. D.) spaced on 9-in. centers. In system number one the depth from panel surface to the top of the tube was ½ in., whereas in systems number two and three the corresponding depth was 1 in. and 2 in. respectively. All coils were installed in uncovered 4-in. concrete floors poured over gravel fill. The three systems exist in rooms located within the same building and detailed descriptions of the entire heating plant are available in a series of articles (4) which were published at the time the structure was erected. The building is 25 ft $\times$ 60 ft and houses the metallurgical section of a research department. The three rooms in which the present tests were conducted are all in regular use and are occupied by workers who had no connection with, nor interest in, the heating research, hence were primarily, and rightly, interested in their own comfort. Air temperatures maintained within the rooms were selected by the occupants and may therefore reflect personal idiosyncracies. Thus in every respect the three systems are actual, operating, field installations of floor-type radiant heating and test results from them should therefore be indicative of what can be expected from any other practical, full-scale, installation.

The problem of obtaining laboratory accuracy on field tests is always one of great importance. In the present case a solution was extremely simple since the field tests were anticipated at the time the building was erected and provision was made then for measurement of flow rates and of various temperatures. Thermocouples were embedded in the various surfaces and a 16-point recorder was used to provide a continuous record for an extended period preceding each test; in this way it could be ascertained with finality that steady state conditions existed during the periods from which data have been selected for analysis.

## Assumptions and Calculations

The total energy supplied to each system was determined by recording the flow rate to that system, $W$ pounds per hour, and the temperature of the water entering the coils, $t_e$, and leaving the coils, $t_l$. The rating of the panel $Q_p$ in Btu per (hr) (sq ft), including losses from the rear is then,

$$Q_p = W(t_e - t_l)/A_p \qquad (1)$$

where $A_p$ is the panel area.

A true panel rating, however, would be based only on energy actually supplied to the occupied space, hence $Q_p$ should be corrected for losses from the rear of the panel to the ground. Further, in most actual rooms some sections of the floor are likely to be covered with built-in cabinets or other obstructions while other areas may be partially covered. If the influence of obstructed areas were not taken into account the resultant over-all rating would obviously be lower than could be expected from a fully exposed panel; since the intent in this paper is to investigate the possibility of realizing the large ratings that have sometimes been claimed, it is considered conservative to fully account for obstructed areas. To do this—as well as to account for rear losses—some assumptions must be made. Thus for purposes of analysis in this paper it is assumed that:

(1) Losses to the ground are 10% of the rate of energy dissipation from the heating surface of the panel to unobstructed surroundings *in these particular cases.*

(2) Panel surface which is covered by furniture or cabinets that extend all the way to the floor loses, to the room, 10% as much energy as does unobstructed panel area.

(3) Panel surface which is covered by furniture that extends to within 2 in. of the floor loses 20% as much energy to the room as does unobstructed panel area.

(4) Panel surface under ordinary furniture, as chairs, tables, and unenclosed workbenches loses energy to the room at the same rate as unobstructed panel area.

Now let $R$ denote the corrected rating, Btu per (hr) (sq ft), of unobstructed floor panel surface and note that:

$1.1R$ = Total energy loss (1.0 for loss to room + 0.1 for loss to ground) of unit area of unobstructed panel.

$0.2R$ = Total energy loss of unit area of covered (assumption 2) panel.

$0.3R$ = Total energy loss of unit area of unobstructed (assumption 3) panel.

Then if unobstructed, covered, and obstructed areas are taken respectively as $A_u$, $A_c$, and $A_o$ and if the total panel area is $A_p$ it follows that,

$$(1.1RA_u + 0.2RA_c + 0.3RA_o)/A_p = Q_p \qquad (2)$$

Combining equations 1 and 2,

$$R(1.1A_u + 0.2A_c + 0.3A_o) = W(t_e - t_l) \qquad (3)$$

or

$$R = W(t_e - t_l)/(1.1A_u + 0.2A_c + 0.3A_o) \qquad (4)$$

Equation (4) gives the corrected panel rating for conditions of the particular test. To permit generalization it will be advantageous to determine the combined equivalent film coefficient from $R$ and from a knowledge of the average surface temperature of the panel, $t_s$, and the air temperature within the room, $t_a$:

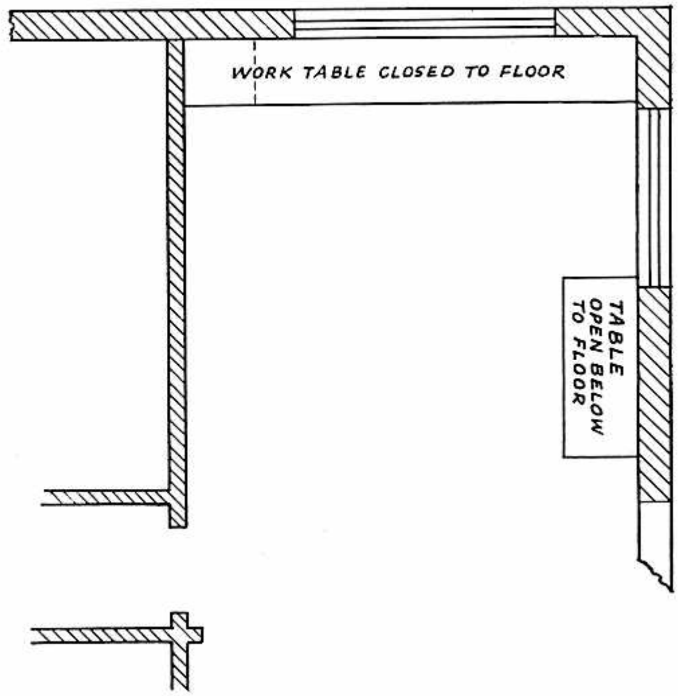
$$h = R/(t_s - t_a) \qquad (5)$$

## Experimental Results

*System 1.* This system, using ½-in. depth of bury is installed in a room having a total floor area of 220 sq ft. The heating panel consists of a single sinuous coil of ¾-in. copper tube on 9-in. centers; 14 rows wide by 11 ft long over the return bends. Panel area, $A_p$, is 118 sq ft with approximately 154 linear feet of tube. The area of the floor slab in which the panel is located is 144 sq ft. On one side is a worktable with cabinets under it that extend to the floor and cover 8 sq ft of panel. There is no other obstructed panel area so:

$$A_u = 110 \quad A_c = 8 \quad A_p = 118$$

At the time of the test steady state was known to exist




Floor plan of test room No. 1.

and the temperature of the panel surface was determined at the center of each quadrant and at the center of the panel itself; the resultant average surface temperature is then,

$$t_s = (81 + 77 + 78 + 81 + 80)/5 = 79.4F$$

Room air temperature, $t_a$, was taken 48 in. above the floor and found to be 68F. Flow rate was measured in gallons per minute and converted to pounds per hour giving W = 491. Entering and leaving water temperatures were determined by means of calibrated thermocouples located in thermometer wells giving: $t_e = 97.5F$, $t_l = 89F$.

Substituting the above experimental data into equation (4),

$$R = 491(97.5 - 89)/(1.1 \times 110 + 0.2 \times 8) = 34.0$$

and by substititution into equation (5),

$$h = 34.0/(79.4 - 68) = 2.98 \text{ Btu per (hr) (sq ft) (F)}$$

*System 2.* This system, using 1-in. depth of bury, is installed in a room having a total floor area of 144 sq ft. The heating panel consists of a single sinuous coil of ¾-in. copper tube on 9-in. centers, 14 rows wide by 11 ft long. Panel area is 118 sq ft with approximately 154 ft of tube. Covered area under a partition and under large worktable legs amounts to 7 sq ft, whereas obstructed area under cabinets that come to within 2 in. of the floor, is 5 sq ft; thus:

$$A_u = 106; \ A_c = 7; \ A_o = 5$$

Steady state existed at the time of the test and the panel surface temperature was measured at six points giving,

$$t_s = (85 + 82 + 86 + 85 + 85 + 85)/6 = 84.7F$$

Other experimental values were:

$$t_a = 76; \ t_e = 97.5; \ t_l = 91; \ W = 491$$

Then from equation (4),

$$R = 491(97.5 - 91)/(1.1 \times 106 + 0.2 \times 7 + 0.3 \times 5) = 26.7$$

and: $h = 26.7/(84.7 - 76) = 3.06$

*System 3.* The third system, using 2-in. depth of bury, is installed in a room having a total floor area of 482 sq ft. There are two coils with combined panel area of 214.5 sq ft and with approximately 99 sq ft of heating surface covered by worktables and sinks which are over cabinets that extend down to the floor; thus:

$$A_u = 115.5; \; A_c = 99$$

Due to changing weather conditions prior to the time of the test on this system it was not possible to realize an exact condition of steady state. Data reported here were obtained from three sets of readings, each set taken at the end of successive 45-minute intervals so the averaged values represent a combined test period of approximately two hours; maximum variation of any particular temperature throughout the test period was 1.5F. The surface temperature of the panel over each of the heating coils was so nearly the same that a single value of $t_s$ has been taken as valid for the panel as a whole,

$$t_s = (88.0 + 88.0 + 87.0) + (88.0 + 88.0 + 87.0) + (87.0 + 86.5 + 86.0) + (85.0 + 85.0 + 84.5) + (88.0 + 88.5 + 88.0) + (88.0 + 88.0 + 87.0) + (88.0 + 88.0 + 87.0) + (88.5 + 88.5 + 88.0) + (89.0 + 89.0 + 89.0)/27 = 87.5F$$

where each parenthesis includes the three test readings of a single thermocouple.

The two heating coils, identified as 1 and 2, received water as follows:

For coil 1

$W = 375.5$ lbs/hr

$$t_e = (98.5 + 96.5 + 98.0)/3 = 97.7F$$

$$t_l = (94.5 + 93.0 + 93.0)/3 = 93.5F$$

For coil 2

$W = 650.3$ lbs/hr

$$t_e = (98.0 + 98.0 + 98.5)/3 = 98.2F$$

$$t_l = (96.0 + 95.5 + 95.5)/3 = 95.7F$$

The air temperature was the same for both regions of the room and averages to:

$$t_a = (78.5 + 78.0 + 77.0)/3 = 77.8F$$

Then substituting into equation (4),

$$R = 375.5(97.7 - 93.5) + 650.3(98.2 - 95.7)/1.1 \times 115.5 + 0.2 \times 99 = 21.8$$

and

$$h = 21.8/(87.5 - 77.8) = 2.25$$

## Discussion of Results

The film coefficients resulting from these three tests are intended for qualitative rather than quantitative consideration. Since the object of this investigation was to explore the possibility of realizing very high ratings from floor panels the assumptions which underlie the calculations have credited a maximum of energy transfer to unit area of unobstructed panel. Thus, upward loss from covered panel areas has been taken as 10% of rating, but since such covered areas usually occur under workbenches and cabinets that are located along exterior walls it is highly probable that the actual upward loss exceeds the assumed value.

Similarly, the panels tested in all three of these systems did not occupy 100% of floor area, hence there was appreciable loss from the panel boundaries by conduction to the adjacent unheated concrete slab. This loss was not considered, hence the energy actually lost by conduction has been credited to surface dissipation and has increased the calculated value of the film coefficient. Finally, the tests were conducted during the first few months after the structure had been constructed and there is every reason to believe that losses from the rear of the panel to the ground were in excess of the assumed value of 10%. Actually, ground loss undergoes a long-term transience which does not appreciably affect the short-term steady state test, but such losses will be abnormally great during the first half of the first heating season.

All of the above factors lead to an expectation that the film coefficients calculated from these three tests are somewhat higher than the values which can actually be obtained from a floor-type heating panel.

## Summary

Based largely on the laboratory work of Wilkes and Peterson many existing design procedures for radiant heating use a combined radiation and convection film coefficient of 2.1 Btu per (hr) (sq ft) (F) temperature difference between the surface of a floor panel and the temperature of the room air at the breathing level. For panels designed for 85F maximum surface temperature in a room at 70F, the above coefficient gives a maximum expected rating of the order of 30 Btu per (hr) (sq ft). Frequent claims have been made that floor ratings of the order of 90 Btu per (hr) (sq ft) have been obtained in practice, but—to the authors' knowledge—no substantiating test data have been published.

This paper reports tests of three systems under controlled experimental conditions. Necessary assumptions concerning rear losses, end losses, and reduced surface loss due to obstructions have in each case favored higher net rating so it is believed that the ratings which have been obtained are unconservatively high. The results are intended only for qualitative use, but they appear to establish—for the three typical field systems which were tested—that the claimed ratings of 90 Btu per (hr) (sq ft) are over 200% of the actual ratings obtained experimentally from these three installations. The authors conclude that extreme caution should be used in using floor panel ratings (for 85F design temperature) that exceed 40 Btu per (hr) (sq ft).

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(3) Wilkes, G. B., and Peterson, C. M. F., Radiation and Convection from Surfaces in Various Positions, *ASHVE Transactions*, Vol. 44, 1938.

(4) Mills, D. L., and LaTart, L. J., Radiant Heat With Copper Tubing: (1) Experiment in Practice; (2) Fabrication of Heating Coils; (3) Embedding Coils in Radiant Heating Panels, *Heating and Ventilating*, Vol. 44, Nos. 10, 11, and 12, October, November, and December, 1947.

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| | | | |
|---|---|---|---|
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