



Radiant heating design and testing
by Leo Classen

A THESIS submitted to the Graduate Faculty In partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering at Montana State College
Montana State University
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Abstract:

Radiant heating has been used In this locality for at least six years. Some of the early Installations were designed with a very limited knowledge of this type of heating. The success of these first Installations Indicates that radiant heating is versatile and that a simple design procedure should be possible. Many designs have been published, but most of these attack the problem from a mathematical viewpoint. Since none of these design procedures have been accepted by the average builder, The Engineering Experiment Station set up a research project to gather Information which would be useful In the application of radiant heating.

Chapter one presents the first step In this research.

The radiant heating laboratory was designed to represent an average residence heated by a floor panel. It served as a proving ground for the panel testing equipment. It also proved valuable as a laboratory for testing and comparing control systems and the products of various manufacturers which are on the market for panel Installations.

Chapter two deals with the application of the panel test to the Wessel residence. This test Indicates a Possible output of 52.7 Btu per sq ft hr at a panel surface temperature of 88° F. based on a 12 day test. The effect of this panel design and edge Insulation Is presented with suggestions for designing future Installations.

Chapter three Is the development and application of a design procedure for radiant heating with a floor panel. This procedure Is the result of research on radiant heating articles published by various authorities as well as the incorporation of information gained In the radiant heating laboratory and the Wessel residence test.

Chapter four, presents Information on the use of various floor covering materials with a floor panel. This is primarily a comparison test which shows the effect of using the various available floor coverings.

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ABSTRACT

Radiant heating has been used in this locality for at least six years. Some of the early installations were designed with a very limited knowledge of this type of heating. The success of these first installations indicates that radiant heating is versatile and that a simple design procedure should be possible. Many designs have been published, but most of these attack the problem from a mathematical viewpoint. Since none of these design procedures have been accepted by the average builder, The Engineering Experiment Station set up a research project to gather information which would be useful in the application of radiant heating.

Chapter one presents the first step in this research. The radiant heating laboratory was designed to represent an average residence heated by a floor panel. It served as a proving ground for the panel testing equipment. It also proved valuable as a laboratory for testing and comparing control systems and the products of various manufacturers which are on the market for panel installations.

Chapter two deals with the application of the panel test to the Wessel residence. This test indicates a possible output of 52.7 Btu per sq ft hr at a panel surface temperature of 88° F. based on a 12 day test. The effect of this panel design and edge insulation is presented with suggestions for designing future installations.

Chapter three is the development and application of a design procedure for radiant heating with a floor panel. This procedure is the result of research on radiant heating articles published by various authorities as well as the incorporation of information gained in the radiant heating laboratory and the Wessel residence test.

Chapter four, presents information on the use of various floor covering materials with a floor panel. This is primarily a comparison test which shows the effect of using the various available floor coverings.

CHAPTER I

INTRODUCTION

Panel heating has been a highly controversial subject for the past ten years. Most manufacturing companies, concerned with finding new fields for their products and many colleges have set up experiments to answer some of the many questions that baffle the individual considering such an installation. At the present time there are a number of methods available for designing a panel heating system. However a survey made of 30 installations in Montana showed that most of them were designed by employing the impressions given by some of the published methods and the experience of the designer or installer. Since most installations are residences and do not involve an expenditure that would warrant the employment of a heating consultant, the design is left to the installer who is in most cases a plumber. A survey of the plumbers in this area left the impression that most design procedures they studied were to long and involved, thus consuming more of their time than could be economically justified by the installation. As a result a rule of thumb was employed as it is in hot air and hot water systems. Most of these rules of thumb are a chosen figure of heat output per unit area used at the individual's discretion. A common figure used was 40 Btu per square foot of floor area. This figure seemed to yield good results as is born out by the number of installations still in operation. However, the figure started going up first to 45 and 50,

then to 80 and 100 Btu per square foot, thus again making it impossible to choose the best figure.

After a considerable amount of Research and study of the available data, no simple clear cut design procedure could be developed, so it was decided to set up a method for testing a panel to make available test data that would be helpful in making up a design procedure.

Means were not available for the construction of an ideal experimental residence, however after a cost analysis of the situation it was found that means were available for the construction of a small test bungalow which would serve as a proving ground for the equipment needed to test a panel as well as a testing laboratory for panel heating controls. This consisted of housing units donated by the college housing administration. See figures 1 and 2.

Design of the Radiant Heating Test Laboratory

Preliminary discussion revealed that two units would be necessary, one to house the boilers and testing instruments while the other served as a radiant heated room, utilizing a flow panel. The floor panel was designed as three separate units, so that the losses from each could be isolated and a comparison made of the edge and ground losses to the useful output of the top panel.

Most available sizes of pipe and tubing have been used as coils in floor panels; however, the tendency seemed to



Figure 1 Housing Units

The radiant heated room is shown in place on the ground. At the right is the instrument room before it was moved into position.



Figure 2 Radiant Heat Laboratory

Side view of units after installation was completed. Outside control bulbs can be seen above the sign near the roof.

be the use of large tubes from 3/4 inches up. This may lead to the impression that the larger the pipe the more heat output can be expected, which would be misleading, because the panels output is limited by the highest comfortable surface temperature. After a consideration of the merits of large piping and a comparison of these merits to the additional cost involved, it was decided to use 1/2 inch copper tubing on 9 inch centers and attempt to show that comparable results could be expected. The tubing arrangement and length is as shown in figures 3 and 4.

Opinions gathered from people living in the housing units led to the conclusion that it would be impossible to heat the unit with a floor panel. However, after considering the elimination of the floor heat loss by the use of a floor panel, it was decided to try heating the unit as it was. If this could not be done, insulation was to be added until the heat loss balanced the heat output of the panel. In order to determine the approximate amount of heat required a heat loss calculation was made as follows:

Calculation of areas:

$$\text{East Wall Area} = 6' \times 20' = 120 - 5.97 (\text{glass}) = \underline{114.03 \text{ sq ft}}$$

$$\text{Ceiling Area} = 20' \times 7' = 140 - 2 (\text{vents}) = \underline{138 \text{ sq ft}}$$

$$\begin{aligned} \text{North and South End Areas} &= (6' \times 7') \times 2 = 84 - 2.3 (\text{glass}) \\ &= \underline{81.7 \text{ sq ft}} \end{aligned}$$

Glass Area = 3 windows 19" x 15" = 5.97 sq ft

1 window 30" x 11" = 2.3 sq ft

Total Glass Area = 8.25 sq ft

Calculation of U factors using conductivities (k) and conductances (c) as given in A.S.H.V.E. Guide of 1950:

U for wall construction

5/16" plywood inside = $1/c_1 = 2.56$

1" Hair Felt between layers of paper = $k_1 = .25$ $1/k_1 = 4.00$

3/8" Pressed paper outside = $k_2 = .33$ $3/8/k_2 = .878$

Film or surface resistance inside = $1/f_1 = 1/1.65 = .605$

$f_1 = 1.65$ was chosen because of the low emissivity of the aluminum paint on the outside of the unit. The factor for average construction is usually 6.

$$R_t = 1/f_1 - 1/c - 1/k_1 - 1/4/k_2 - 1/f_o$$
$$.605 - 2.56 - 4.00 - .878 - .333$$

$$R_t = .121 = \underline{8.376}$$

$$U_{\text{wall}} = 1/R_t = 1/8.376 = \underline{.120}$$

U_c ceiling coefficient

1/4" pressed paper inside $1/4/k_1 = .757$

1 1/2" insulator hair felt $1\ 1/2/k_2 = 6.00$

3/8" pressed paper outside $3/8/k_1 = .878$

asphalt roofing $1/c_2 = .28$

inside surface resistance $1/f_1 = .605$

outside surface resistance $1/f_o = .333$

$$R_t = \underline{8.953}$$

$$U_c = 1/R_t = 1/8.953 = .112$$

U_e for ends

U_e for North and South ends is the same as U for the ceiling except for the asphalt roofing.

$$R_t = 8.953 - 1/c_2$$

$$8.953 - .28 = 8.673$$

$$U_e = 1/8.673 = \underline{.114}$$

U_g for glass

$$\text{single glass } U_g = \underline{1.13}$$

U_{rv} for roof vents

The high conductivity of the steel plate causes the U factor to be a function of the inside and outside film coefficients. Thus the resistance of the steel can be considered negligible.

$$R_t = 1/f_1 - 1/f_0$$

$$R_t = .605 - .333$$

$$U_e = 1/R_t = \underline{1.06}$$

Wall heat loss

$$Q_w = U_w A_w t_i - t_o$$

Using an inside temperature of 70° F. and an outside temperature of - 20° F. $t_i - t_o$ is equal to 90° F.

$$Q = .120 \times 114 \times 90$$

$$= \underline{1230 \text{ Btu/hr}}$$

Ceiling Heat Loss

$$\begin{aligned} Q_c &= .112 \times 138 \times 90 \\ &= \underline{1390 \text{ Btu/hr}} \end{aligned}$$

North and South Ends Heat Loss

$$\begin{aligned} Q_e &= .114 \times 81.7 \times 90 \\ &= \underline{838 \text{ Btu/hr}} \end{aligned}$$

Glass Heat Loss

$$\begin{aligned} Q_g &= 1.13 \times 8.25 \times 90 \\ &= \underline{839 \text{ Btu/hr}} \end{aligned}$$

Roof vents Heat Loss

$$\begin{aligned} Q_{rv} &= 1.06 \times 2 \times 90 \\ &= \underline{191 \text{ Btu/hr}} \end{aligned}$$

Infiltration Heat Loss

Total Volume = 20' x 6' x 7' = 840 cubic ft.

Infiltration will be calculated by the air change method, assuming 1/2 air change per hour. This calculation will be valid only for still weather conditions, because the windows are very poorly fitted and a wind would considerably increase the number of air changes.

$$\begin{aligned} Q_{ac} &= .018 \times 840/2 \times 90 \\ &= \underline{680 \text{ Btu/hr}} \end{aligned}$$

Total Heat Loss

$$\begin{aligned} Q_t &= 1230 + 1390 + 838 + 839 + 191 + 680 \\ &= \underline{5186 \text{ Btu/hr}} \end{aligned}$$

The area available for a floor panel was 119 sq ft. The heat loss divided by this is the heat output per sq ft required at an outside temperature of - 20° F.

$$\frac{5186 \text{ Btu}}{119 \text{ sq ft}} = 43.5 \text{ Btu/ sq ft}$$

CONSTRUCTION

The housing units were originally one room equipped to serve as a kitchen, dining room, living room and bedrooms. To be useful, a complete stripping was necessary leaving only the outer shells.

A heavy wood frame was constructed under one unit to support the walls making it possible to remove the floor and form the panel at ground level. The panel foundation is similar to that used by various firms installing radiant panels in this area, consisting of crushed aggregate.

The floor unit consists of 3 sections. First a guard ring surrounding the room, 4 1/2 inches wide and 7 5/8 inches deep formed on the aggregate. Next a bottom slab 3 5/8 inches thick 6 feet wide and 19 3/4 feet long. See figure 3 showing the South end and figure 4 the north end. The separating board seen in figures 3 and 4 was left in place after the concrete hardened, in this way serving as insulation between the guard ring and bottom and top slabs. A 3/4 inch sheet of asphalt impregnated sheeting was placed on the bottom slab to insulate it from the top slab which is 19 3/4 feet long, 6

feet wide and 4 inches thick. The insulating sheathing and top coil arrangement can be seen in figure 5 which is the South end and Figure 6 the North end of the panel.

A strip of insulation was installed above the guard ring to prevent a heat transfer from it to the room. Figure 7 is a cross section of the floor construction. The amount of tubing used in each panel was, top panel 165 feet, bottom panel 160 feet and 105 feet in the guard ring. Figure 8 shows the piping arrangement used to facilitate switching from one control system to another.

Selection of Panel Testing Equipment

The choice of testing equipment turned out to be a major problem since tests of this type are new and no standard equipment is available. The problem was divided into two fields; one of temperature measurement and the other fluid flow measurement.

The most important temperatures are those of the panel supply and return water. It is also desirable to have the temperatures of the effective panel environment such as panel surface, room temperature and outside temperature. A satisfactory instrument must be multiple recording, accurate and portable. No instrument on the market met these requirements. The best choice was a Foxboro Dynalog which records 6 points from 50 - 200° F. and is accurate to 1/4 of 1%. The porta-



Figure 3 South End of Bottom Panel

1/2" tubing on 9" centers can be seen at the right. Guard ring coils are visible in the center.



Figure 4 North End Bottom Panel

Wood sleepers which were used to position the tubes are visible. Insulation under the panel is crushed aggregate.



Figure 5 South End Top Panel

1/2" tubing on 9" centers, coil at far left is the supply run which goes around the outside of the panel.



Figure 6 North End Top Panel

Wood sleepers 1" high were used to position the tubing at a uniform distance in the panel. These rest on the 3/4" insulation which was placed on the bottom panel to restrict heat exchange between panels.

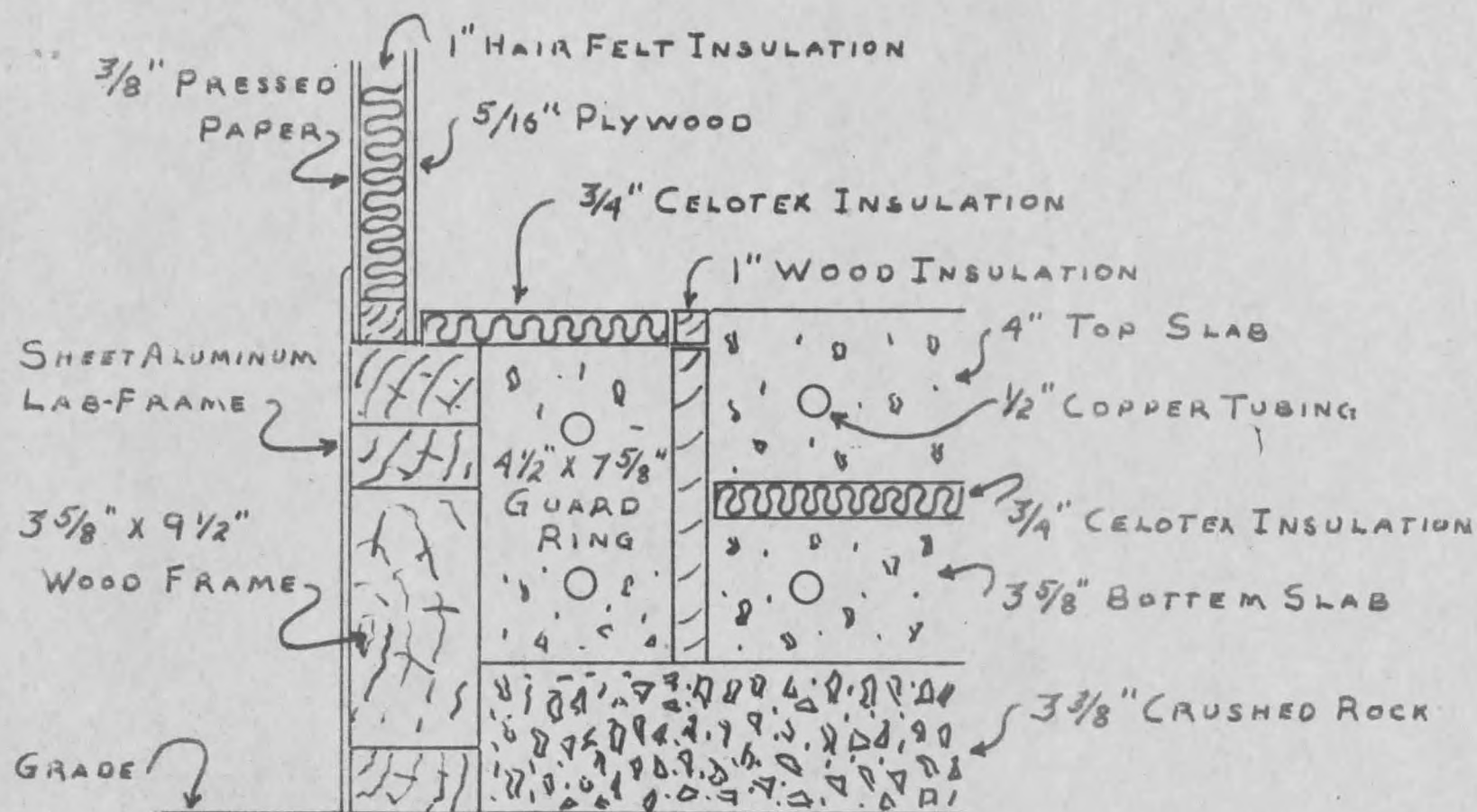


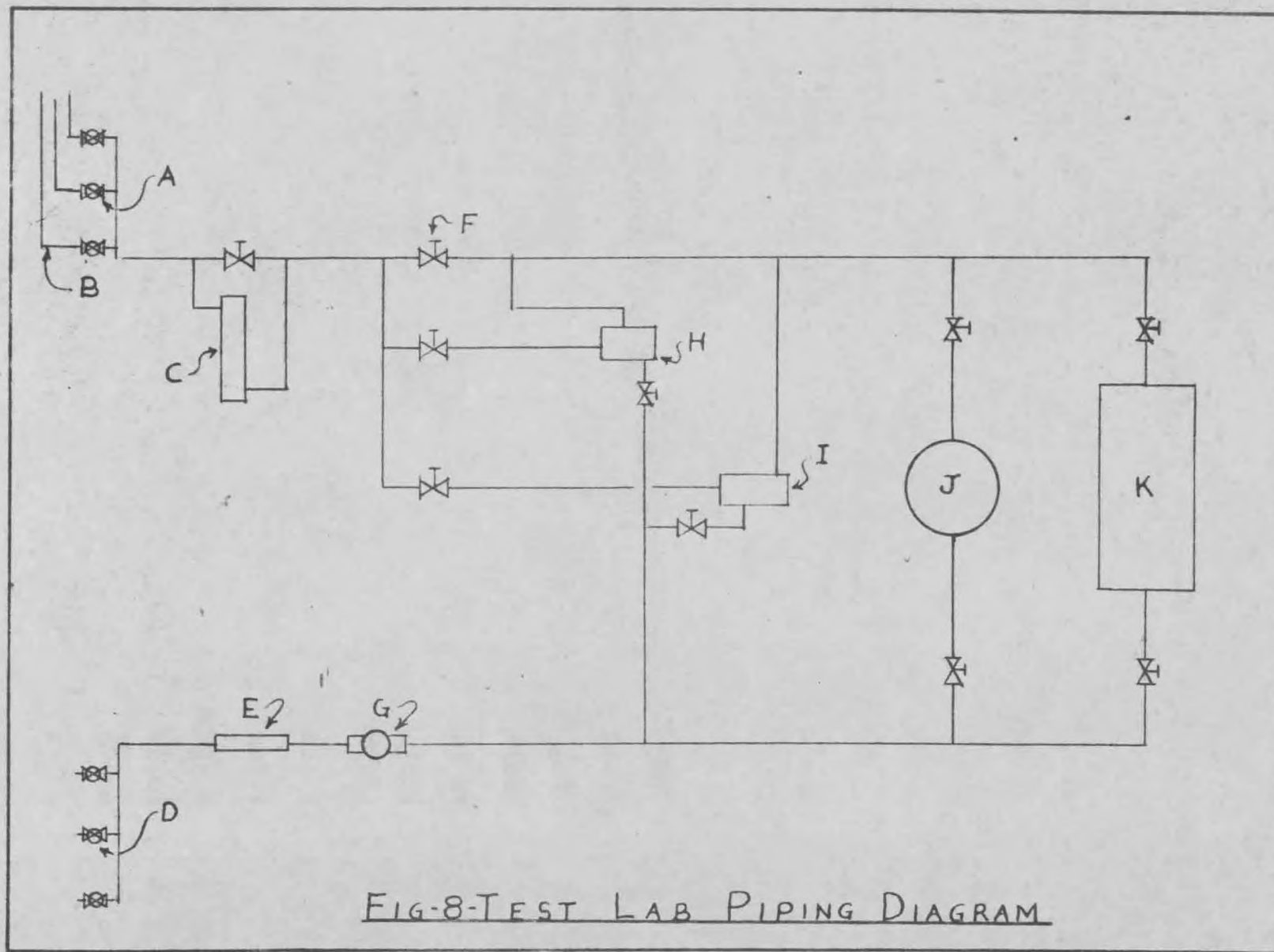
FIG-7-PANEL DETAIL OF TEST LAB

bility of this instrument was questionable and on arrival was found possible but not practical, the reason being the extremely fragile surface bulbs which are now improved. The six points on the recorder were used for supply water, top, bottom and guard ring return water, panel surface and room temperature. A separate recorder made by the Auto Lite Corporation was used for outside temperatures because a lower range was required.

The basic requirement in a flow measuring device is that it adds nothing to the heating system, which may vary the usual operating conditions. An ideal way would be to use a piece of equipment already in the system. The first trial was the circulator, because it operated on a curve with flow in G.P.M. versus developed head in feet of water. Thus by recording the head developed and referring to a curve the flow could be obtained. Operating characteristics curves for various circulators were obtained from the manufactures and checked in the laboratory. These tests showed that the curves were for an average pump construction and could not be used because each pump has its own characteristics. An attempt was made to plot a curve for each pump; however results showed that the accuracy obtained from such curves was not sufficient for a good heat flow calculation.

Flow measurement using a 90° elbow has proven successful for pipe lines from one inch up, but no data could be found on

the use of this method for small piping. It is conceivable that small iron elbows could not be used because of their rough interior finish. However, copper elbows are considerably more smooth so an experiment was set up to evaluate their usefulness. Tests on a one inch copper elbow proved successful so smaller sizes were also tested. All the elbows tested, of a given size, followed practically the same curve indicating that individual calibration was not necessary. A recording differential pressure meter made it possible to record flows over a period of time. These records showed that no fluctuation occurred as long as conditions remained constant. Three one half inch elbows were installed in the test laboratory to measure the flow through each section of the panel see Fig. 8. A Fisher and Porter indicating flow meter was installed in series with these and comparisons showed that they agreed within 3% at 90 gallons per hour. Figure 8 which follows is the piping diagram used to facilitate switching from one control system to another.



Description of Equipment in Piping Diagram Figure 8

A - Ohio Brass Balancing Valves

These were installed to test their ability to control the flow through the panel and their use as a shut-off valve.

B - 1/2" Flow Measuring Elbows

A differential pressure recording meter was used with these to give the flow during a test.

C - 3/4" Fisher and Porter Flow Meter

This meter has a capacity of 180 gallons per hr. It was used for spot checks of the recording flow meter during tests.

D - Taco Balancing Valves

The flow through the various panels was controlled by these to give approximately the same return water temperatures.

E- Water Strainer

It is advisable to install one of these in all systems because it prevents the circulation of foreign particles which may damage the pump or controls.

F - 1" Gate Valves

Gate valves were used throughout the system to prevent excessive head loss.

G - 1" B & G Circulator

The pump was located in an accessible spot so other models

can be conveniently tested and compared.

H - Automatic 3-way Mixing Valve

Sarcothern valve which mixes boiler and return water with regard to the outdoor and indoor temperature.

I - Manually Controlled 3-way Mixing Valve

Tacomatic valve employing a thermostat which is adjusted to fit the system.

J - Gold Top Boiler

110,000 Btu input packaged boiler, originally equipped with controls and circulator, but later altered to fit laboratory pumpers.

K - American Standard Boiler

125,000 Btu input cast iron sectional boiler including limit control and automatic gas valve.

GUIDE FOR CALCULATIONS USED IN TABLE I

Column A

Gives the time and date of heating cycle.

Column B - Average Outside Temperature ° F.

This temperature was obtained by a 24 hour recorder mounted on the laboratory wall. Temperatures were read directly and in cases of sudden changes were read for each hour and averaged for the day.

Column C - Water Supply Temperature ° F.

A surface temperature element was fastened to the supply line and then covered with about 1" of glass wool. The temperatures were recorded for a 24 hour period and averaged for the heating cycle.

Columns D, F, and H - Return Temperatures ° F.

These are the bottom, top and guard ring return temperatures and were taken in the same way as the supply temperature in Column C.

Columns E, G, and I - Change in Water Temperature.

These are the water temperature drops in each of the panels as calculated by subtracting D, F and H from C.

Columns J, K, L - Flow # per hour.

The flow through each of the panels was measured by a half inch calibrated elbow on each circuit. A header was constructed of 1/8" pipe so that the recording flow meter could be switched from one circuit to another at

any time during the heating cycle.

Columns M, N and O - Input Btu/hr.

These are the total heat input to the panel calculated from the flow and temperature drop through each panel.

Column F - Heat Input Time Hours.

This is the length of each heating cycle taken from the temperature recording charts.

Columns Q, R, and S - Input Btu/sq ft hr.

These are the heat inputs per unit area of the panel found by dividing columns M, N, and O by the exposed area of the panel. These areas are 119 sq ft per top and bottom panels and 55.5 sq ft for the guard ring, the guard ring area includes side and bottom exposure of the ring.

Column T - Average Room Temperature.

This is the room air temperature as recorded by the Foxboro recorder. The room temperature element is located on the West or inside wall of the heated room.

Column U - Average Panel Surface Temperature.

This is the average panel surface temperature of the top panel. The location of surface element was determined by taking an average of the temperatures of each sq ft of the panel and finding a spot of the panel that approached this calculated average.

TABLE I HEAT LABORATORY DATA SHEET

A	B	C	D	E
Time & Day of Test	Average Outside T.	Water Supply T.	Return T. Top Panel	T. Top Panel
3:00pm Jan.19	17°	121°	107°	14°
9:00am Jan.20				
9:00am Jan.20	20°	121°	106°	15°
9:00am Jan.21				
9:30am Jan.21	32°	120°	106°	14°
9:30am Jan.22				
8:00pm Jan.22	13°	122°	104°	18°
8:00am Jan.23				
8:30am Jan.23	25°	123°	104°	19°
4:00pm Jan.23				
8:30am Jan.24	35°	124°	100	14°
7:30pm Jan.24				
9:00pm Jan.24	39°	124°	106°	18°
8:00am Jan.25				
7:00pm Jan.26	-10°	122°	110°	12°
8:00am Jan.27				
5:30pm Jan.28	-28°	128°	114°	14°
8:30am Jan.29				
10:00pm Jan.29	-22°	126°	114°	12°
8:00 am Jan.30				
7:00pm Jan.30	-13°	118°	107°	11°
8:00am Jan.31				
7:00pm Jan.31	-18°	122°	109°	13°
2:00am Feb. 1				
Average	7.5°	122.7°	107°	15.1°

TABLE I HEAT LABORATORY DATA SHEET (cont'd)

A	F	G	H	I
Time & Day of Test	Return Bottom Panel	T Bottom Panel	Return Guard T.	T Guard Ring
3:00pm Jan.19 9:00am Jan.20	109°	12°	106°	15°
9:00am Jan.20 9:00am Jan.21	109°	12°	107°	14°
9:30am Jan.21 8:30am Jan.22	109°	11°	107°	13°
8:00pm Jan.22 8:00am Jan.23	106°	16°	106°	16°
8:30am Jan.23 4:00pm Jan.23	109°	14°	109°	14°
8:30am Jan.24 7:30pm Jan.24	102°	12°	110°	14°
9:00pm Jan.24 8:00am Jan.24	106°	18°	110°	14°
7:00pm Jan.26 8:00am Jan.27	111°	11°	111°	11°
5:30pm Jan.28 8:30am Jan.29	117°	11°	111°	14°
10: pm Jan.29 8:00am Jan.30	115°	11°	112°	20°
7:00pm Jan.30 8:00am Jan.31	109°	9°	106°	12°
7:00pm Jan.31 2:00am Feb. 1	110°	12°	109°	13°
Average	109.2°	13.2°	109°	14°

TABLE I HEAT LABORATORY DATA SHEET (cont'd)

A	J	K	L	M	N	O
Time & Day of Test	Flow #/hr Top Bottom Guard			Btu input per hr top	Btu input /hr bottom	Btu input hr/guard
3:00pm Jan.19	190	197	98	2660	2360	1470
9:00am Jan.20						
9:00am Jan.20	197	197	134	2950	2360	1870
9:00am Jan.21						
9:30am Jan.21	197	190	134	2760	2090	1740
8:30am Jan.22						
8:00pm Jan.22	190	176	134	3430	2820	2140
8:00am Jan.23						
8:30am Jan.23	190	176	134	3620	2460	1870
4:00pm Jan.23						
8:30am Jan.24	190	176	134	2660	2110	1870
7:30pm Jan.24						
9:00pm Jan.24	190	176	134	3430	3170	1870
8:00am Jan.25						
7:00pm Jan.26	320	278	190	3840	3060	2090
8:00am Jan.27						
5:30pm Jan.28	320	278	190	4470	3060	2660
8:30am Jan.29						
5:30pm Jan.29	317	246	183	3800	2710	2380
10: pm Jan.29						
7:00pm Jan.30	317	246	183	3490	2210	2200
8:00am Jan.31						
7:00pm Jan.31	317	246	183	4120	2950	2380
2:00am Feb. 1						
Average	244	224	151	3650	2796	2150

TABLE I HEAT LABORATORY DATA SHEET (cont'd)

A	P	Q	R
Time & Day of Test	Heat Input Time	Input 119sqft Btu/sq ft Top hr	Input 119sqft Btu/sq ft Bottom hr
3:00pm Jan.19	18 hrs	22.4	19.8
~~9:00am Jan.20~~			
9:00am Jan.20	24 hrs	24.8	19.8
9:00am Jan.21			
9:30am Jan.21	23 hrs	23.2	17.5
8:30am Jan.22			
8:00pm Jan.22	12 hrs	28.8	23.7
8:00am Jan.23			
8:30am Jan.23	7.5 hrs	30.4	20.6
4:00pm Jan.23			
8:30am Jan.24	11 hrs	22.4	17.7
7:30pm Jan.24			
9:00pm Jan.24	13 hrs	28.8	26.6
8:00am Jan.25			
7:00pm Jan.26	13 hrs	32.2	25.6
8:00am Jan.27			
5:30pm Jan.28	15 hrs	37.6	25.6
8:30am Jan.29			
10:00pm Jan.29	10 hrs	24	18.5
8:00 am Jan.30			
7:00pm Jan.30	13 hrs	29.3	18.5
8:00am Jan.31			
7:00pm Jan.31	7 hrs	34.6	24.8
2:00am Feb. 1			
Average	12.8 hrs	28.3	21.9

TABLE I HEAT LABORATORY DATA SHEET (cont'd)

A	S	T	U
Time & Day of Test	Input 55.5 Btu/sq ft Guard hr	Average Room T.	Average Slab Surface T.
3:00pm Jan.19	26.4	77°	95.5°
9:00am Jan.20			
9:00am Jan.20	33.7	74°	95.5°
9:00am Jan.21			
9:30am Jan.21	31.4	74°	93.3°
8:30am Jan.22			
8:00pm Jan.22	38.6	70°	93°
8:00am Jan.23			
8:30am Jan.23	33.7	72°	93°
4:00pm Jan.23			
8:30am Jan.24	33.7	70°	88°
7:30pm Jan.24			
9:00pm Jan.24	33.7	78°	97°
8:00am Jan.25			
7:00pm Jan.26	37.6	68°	88°
8:00am Jan.27			
5:30pm Jan.28	48	64°	92°
8:30am Jan.29			
10:00pm Jan.29			
8:00 am Jan.30	29.8	69°	92°
7:00pm Jan.30	39.6	68°	91°
8:00am Jan.31			
7:00pm Jan.31	42.9	67°	91°
2:00am Feb. 1			
Average	36.7	70.8°	92.3°

DISCUSSION OF RESULTS IN TABLE I

The test instruments were received and installed during the summer and fall of 1950. Preliminary tests were made in November and December of 1950. During these tests various control methods were tried in an attempt to find the one that would best regulate the supply water. The one found to supply water with the least variation was a Taco-matic 3-way mixing valve which could be manually controlled. These tests also showed that the water temperature drop through the panel was so small that it could not be accurately read from the recording charts. This was analyzed as a peculiar characteristic, of a panel too large for the required output, which would not be present in a residence where the heat loss would require a full utilization of the panel. The temperature drop could have been increased by the addition of more panel area, however, this was considered impractical at the time. As a solution the flow was throttling to the point where a readable temperature differential was present. Throttling a circuit is never desirable, because it prevents the full utilization of the tubing, but it is a valuable remedy in cases where too much tubing has been installed, and the panel becomes too warm for comfort.

In the case of the heat laboratory, throttling was not employed to limit the heat output of the panel. A calcula-

tion of the Reynolds number for this reduced flow showed that velocity was considerably above that of laminar flow so no reduction in heat transfer from the water to the tube was expected.

Columns M, N and O of Table I show that the combined input of the guard ring and the bottom panel were greater than the input to the top panel. These losses are greatly exaggerated because of the relatively small ground area covered by the laboratory panel; however, they show that heat can be lost in this way and that precautions should be taken to prevent it. These precautions must be effective but economically justified. Over a period of years the cost of the best preventive measures can be justified by the savings in fuel. The drawback here is the desire of the builder to get by as cheaply as possible and not worry too much about the annual heating cost. The value of the construction used is directly related to the insulation value of the material. From this it is evident that a subfloor of Zonolite concrete is much better than one of crushed rock but the cost is also greater.

Column S, the input to the guard ring, shows that the loss per sq ft hr is greater than that of the bottom panel. This would indicate that the prevention of this loss should be the primary concern. The reason for this greater loss

per unit area is; the guard ring is exposed to the outside air temperature which is naturally more severe than the ground temperatures. A survey of about 30 installations in Montana showed that the most common type of edge insulation used was a 1/2" thick strip of Celotex between the foundation wall and the floor panel. A few of these installations had no edge insulation and were not satisfactory because the outer edge of the panel was uncomfortably cold. A strip of insulation 1" thick extended to the depth of two feet below the top of the panel would cut the edge loss considerably.

Column Q, the input to the top per sq ft hr, has an average of 28.3 Btu per sq ft hr for an average panel surface temperature, column U of 92.3°. This does not compare with the result of the Wessel residence test which is 52.1 Btu per sq ft hr for a panel surface temperature of 88°. Since the Wessel residence is much larger and better represents the conditions which will surround the average panel, the results from it will be considered more accurate. The reasons why the test laboratory output did not match the Wessel residence results are not definitely established, however there are a number of possibilities. First, the interior of the laboratory is finished with a high gloss enamel, a finish of this type is neither a good radiator nor a good receiver of radiant heat waves. From this viewpoint

the panel would have to be at a much higher temperature to transmit the same amount of radiant heat. Second, the exterior of the laboratory has an aluminum finish which would cut its radiant heat loss and so make it possible to keep the laboratory at 70° with less heat input than was calculated to the heat losses. Third, the laboratory walls contain a small glass area so the mean effective temperature is higher than that of the Wessel residence for similar outside conditions. The effect of a higher mean effective temperature is it limits the amount of heat transferred by radiation. This is evident from the basic formula for radiant heat exchange which states that the amount of heat exchanged is directly proportional to a constant times the difference of the radiant and receiving body temperatures to the fourth power. Fourth, the bottom panel might have been transferring some heat through the insulation to the top panel. This would be possible if the heat loss from the bottom panel was restricted and the panel temperature approached the circulating water temperature.

The first and second possibilities can be eliminated in future tests by using a rough finish with dark paint. The third will require some additional temperature recording devices to record the temperatures of all the surfaces facing the panel and in this way make available a mean effective

temperature. The last possibility would require a means of measuring the average temperature of the bottom panel. This could be done by drilling through the top slab and inserting temperature bulbs into the bottom panel.

The use of the testing laboratory as a proving ground for the test procedure was successful because it presented conditions similar to those of a panel in the average residence with advantages such as accessibility at all times and convenient location. Equally successful was its use as a control testing laboratory. The size and characteristics of the panel had no effect on the results of the control tests since the controls were compared on their ability to maintain a given room temperature. In this laboratory they all shared the same advantages or disadvantages.

Some possibilities for future tests in the floor panel test laboratory are:

1. A test showing the effect of different interior finishes on the panel's output.
2. A test using different types of floor coverings and paints on the panel to show their effect on panel output.
3. A test showing the effect of the mean effective temperature on the panel's output. This could be done by installing wall and ceiling panels that

could be varied in temperatures. A number of thermocouples should be installed on these panels to make available surface temperature data for the calculation of the MET.

4. A test showing the effect of air temperature on the panel's output.

CHAPTER II

PANEL TEST ON WESSEL RESIDENCE

The data obtained on a particular installation can not be set up as a criterion for all installations, because each heating system has its own peculiar characteristics which are impossible to duplicate. The reasonable approach would be to gather useful information on a number of installations, analyze this data and draw a conclusion based on the points agreement.

The Wessel residence can be considered as an average sized modern home, however, from the heating system standpoint it is experimental, because it is one of the first installations using 3/8" tubing with 9" OC spacing in a floor panel, see figures 9 and 10. The reason for using smaller tubes was to attempt to show that the output from these could be favorably compared to that of larger tubes. Another phase of this test will compare the heat supplied to the residence with the results of a regular heat loss calculation. The heat loss calculations for the Wessel residence are as follows:

Calculation of U coefficients

U Outside walls

Resistance

Lap Siding

$$1/c = .78$$

1" Fir Sheathing (with building paper) $1/c = 1.16$

Alfal Insulation

Two air spaces faced on one side

with Reflective Insulation

$$1/c = 1.4 \times 2 = 2.8$$



Figure 9 Living Room of Wessel Residence

Showing 3/8" coils on 9" centers. Rocks were put under the coils to position them. These were later removed because they presented a hazard to the tubes.



Figure 10 Study and Bedroom of Wessel Residence

The coil shown at the upper right is 12 inches from the outside wall. Coils are lying on 6 to 8 inches of crushed aggregate.

One air space with reflective insulation
on both sides.

$$1/c = 2.17$$

Inside and outside film coefficients

$$1/f_1 = 1/1.65 = .601$$

R Total

$$1/f_o = 1/6 \quad \frac{.166}{6.346}$$

$$U_w = 1/R_t = 1/6.346 = \underline{\underline{.158}}$$

U_c Ceiling

Resistance

4-ply built up roofing

$$1/c = .28$$

1" Fir Sheathing

$$1/c = .98$$

Alfal Insulation (same as walls)

$$1/c = 4.97$$

Montana board

$$1/c = 1.51$$

Film coefficients

$$1/f_1 = .166$$

$$1/f_o = \underline{\underline{.160}}$$

R Total

$$= 8.516$$

$$U_c = 1/8.516 = \underline{\underline{.177}}$$

$$U \text{ for Double Glass} = .45$$

$$U \text{ for Single Glass} = 1.13$$

$$U \text{ for Doors} = .52$$

Total areas from building plans.

$$\text{Outside Walls} = 711 \text{ sq ft}$$

$$\text{Ceiling} = 935 \text{ sq ft}$$

$$\text{Double Glass} = 179 \text{ sq ft}$$

$$\text{Single Glass} = 79 \text{ sq ft}$$

$$\text{Doors} = 42 \text{ sq ft}$$

The heat loss for an outside temperature of -20° F and an outside temperature of 70° F using the heat transfer formula is as follows:

$$Q = U A (T_1 - T_0)$$

$$Q = U A [70 - (-20)]$$

$$Q = U A \times 90$$

$$Q_{\text{wall}} = .158 \times 711 \times 90 = \underline{10120 \text{ Btu/hr}}$$

$$Q_{\text{Ceiling}} = .117 \times 935 \times 90 = \underline{9,850 \text{ Btu/hr}}$$

$$Q_{\text{Double Glass}} = .45 \times 179 \times 90 = \underline{7,160 \text{ Btu/hr}}$$

$$Q_{\text{Single Glass}} = 1.13 \times 79 \times 90 = \underline{8,040 \text{ Btu/hr}}$$

$$Q_{\text{Doors}} = .52 \times 42 \times 90 = \underline{1970 \text{ Btu/hr}}$$

Q infiltration

One air change will be used because of the fire place and luvers.

$$\text{Volume} = 935 \times 8.5 = 7940 \text{ cu ft}$$

$$Q_1 = .018 \times 7940 \times 90$$

$$= \underline{12,800 \text{ Btu/hr}}$$

$$\text{Total heat loss} = Q_t = \underline{50,040}$$

Assuming a 15% edge and bottom loss the total heat input required at -20° would be

$$Q_t + \text{losses}$$

$$50040 + (.15 \times 50040)$$

$$50040 + 7560 = \underline{57600 \text{ Btu/hr}}$$

The edge and bottom losses do not effect the exposed

panel output so the heat required from the top of the panel at -20° F. will be 50,040 Btu per hour.

The panel test procedure as developed in the radiant heat laboratory was applied to the Wessel residence. Since this test is a measure of the total heat input to the panel it is evident that some relationship must be set up for obtaining the heat output from the top of the exposed panel. The total heat input is utilized in three ways, these are: Heat output from the exposed panel, heat lost through the edge and bottom of the panel and heat stored during the heating cycle. The amount of heat stored can be calculated directly using the mass of the panel and the temperature increase of the panel during the heating cycle. The heat lost through the bottom and edge of the panel and the heat output from the panel top must be divided by an assumed ratio. This assumption will be eliminated in future tests by a heat meter which will be used to find the heat output from the top of the panel.

The results of this test are presented in Table II. The guide for the calculations and the testing procedure follows.

Test Procedure and Guide for
Calculations Used in Table II

Column A - Date of Test.

The test started at 5 :00 P.M. February 6 1951 and was calculated for 24 hour periods thereafter.

Column B - Average outside temperature ° F.

A 24 hour temperature recorder was used to record the outside temperature during tests. In some cases the average temperature was obtained by Visual inspection. However, in most cases it was necessary to pick an average temperature for each hour and then average the 24 temperatures for a daily average temperature.

Column C - Average supply temperature ° F.

The supply water temperature as recorded on the daily charts varied about 30° during each heating cycle, due to burner operation. Although, this is undesirable, no practical way was found to supply water at a constant temperature. The variations formed a definite pattern so a mean supply temperature was chosen for each hour and an average was calculated for the heating cycle.

Column D - Average return temperatures varied much less than the supply temperatures and could be read from the chart with little difficulty.

Column E - Water flow of pounds per hour.

A recording flow meter chart supplied a chart number

representing a differential head across the flow elbow. This number multiplied by the elbow constant (.605 for a 1" elbow) equals the flow in G.P.M. The density of the water was corrected for temperatures by using an average between the supply and return temperatures.

Column F - Total input to panel per hour Btu.

This is the supply water temperature "C", less "D" return water temperature, times "E" flow in pounds per hour.

Column G - Length of Heating Cycle hours.

The cycle length was determined from the flow charts and is the total time of circulator operation or heat input.

Column H - Heat input per cycle in Btu.

This is the panel output per hour "F" times the length of heating cycle.

Column I - Heat Input per day in Btu.

This is a summation of the cycle heat inputs during the 24 hour period.

Column J - Panel Surface temperature T_1 ° F.

This is the temperature at the beginning of the heating cycle determined from the temperature chart and by a surface temperature element. The location of this element was chosen by using a surface type pyrometer to take readings on different parts of the panel and picking a representative location.

Column K - Panel Surface temperature T_2 ° F.

This is the temperature of the panel at the end of the

heating cycle obtained in the same way as "J".

Column L.- T Panel Surface ° F.

The change in panel surface temperature is the difference between "J" and "K".

Column M - Heat stored cycle Btu.

To find the amount of heat stored during a heating cycle, a series of thermocouples must be buried at various distances under the slab. The exact position of these thermocouples and their temperature would give the amount of soil affected by the cycle. This was not done in the Wessel Residence. To compensate for this heat storage, an assumption that the heat storage was confined to the slab, was made. Tests¹ conducted during the 1948-49 winter season by the Society of Heating and Ventilating Engineers, showed that for cycles up to 5 hours, the temperature increase at the bottom of the slab insulation was only 1° F., but would be greater for longer cycles. The volume of concrete used in heat storage was calculated by using the entire floor area. Although some areas of the floor do not contain tubes, the high conductivity of the concrete causes the total area to become a reservoir of heat.

Concrete mass calculations follow:

Living room $(15 \times 19.5) + (1.5 \times 4.5) = 299$ sq ft

Kitchen & dining room $(18.5 \times 10) + (5 \times 8) = 225$ sq ft

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Utility room $(14.5 \times 7.5) + (5 \times 2.5) + (2 \times 4) = 129.3$ sq ft

Bedroom and Study $(23.5 \times 11) + (8.5 \times 5) = 301$ sq ft

Total floor area $299 + 225 + 129 + 301 = 954$ sq ft

Volume of concrete 954 sq ft $\times 1/3$ ft thick = 318 cu ft

Weight of concrete 318×137 #/ cu ft = 43,600 #

Heat capacity $43600 \times .25$ specific heat = 10,900 Btu/°F.

The heat stored for a given cycle is the heat capacity times the temperature rise of the panel during the cycle, Column K minus Column L.

Column N - Heat stored Btu/hr.

This is the heat stored per cycle "M" divided by "G" the cycle length.

Column O - Heat stored Btu/hr ft².

This is "N" divided by the total panel area 954 sq ft.

Column P - Input to panel Btu/hr sq ft.

This is the total input to panel per hour "F" divided by the total area of the panel 954 sq ft.

Column Q - Total output Btu/hr sq ft.

This is the total panel input "P" less "O" the heat stored per sq ft per hour.

Column R - Bottom and Edge Losses Btu/hr sq ft.

An assumption had to be made for the amount of heat

lost through the edges and bottom of the slab. Laboratory tests¹ have shown that these losses depend on the amount of edge insulation under and around the panel. In some cases where no edge insulation was used the losses ran as 30% of the panel output. The Wessel residence has a 1/2" Celotex edge insulation, a sheet of Celotex extending 4 feet inside the foundation, and 6 to 8 inches of crushed rock under the panel. After considering this it was assumed that 15% of the total panel output would be edge and bottom losses.

Column S - Useful Output of Panel Btu/sq ft hr.

This is the total panel output "Q" less the bottom and edge losses "R". This is on the basis of the total floor area.

Column T - Exposed Panel Output in Btu/sq ft hr.

Part of the panel is covered with built in cabinets and closets which will not allow a free transfer of heat. Convective and radiant heat transfer is considerably reduced in these areas leaving conduction as the panels' only means of heat transfer. The amount of heat conducted along the wood cabinets and finally given off by radiation and convection would be almost impossible to measure. Since these areas are limited in the amount of heat they can give off, a ratio was set up between the total panel area 954 sq ft and the

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exposed panel area 796 sq ft. This ratio, 1.19, times "S" the output on the total panel basis will give the output per sq ft of exposed panel.

Column U - Panel Input day basis Btu/hr.

This output was calculated by assuming that all the heat supplied during a 24 hour period was given off during that day. A possible error would be in the amount of heat carried from one day to the next by a difference in temperature times the thermal capacity of the building. This output is "I" the input per day divided by 24.

Column V - Panel Output day basis Btu/hr sq ft.

This is the total panel output "U" divided by the exposed panel area 796 sq ft. This should not be considered as the maximum output possible from this panel because the heating cycle added heat for only a portion of the day while the panel gave off heat at a reduced rate until all the stored heat had been dissipated.

Column W - Exposed Panel Output day basis Btu/hr.

This is the total panel output per hour "U" less the assumed edge and bottom losses 15%.

Column X - Calculated Heat Loss.

The procedure used for changing the heat loss calculation to the temperature during a test was as follows:

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$$\begin{aligned} Q_t \text{ at } +20^\circ &= Q_t \text{ at } -20^\circ \times \frac{(T_1 - T_o) + 20^\circ}{(T_1 - T_o) - 20^\circ} \\ &= 50,040 \times \frac{(t_1 - t_o)2}{(t_1 - t_o)1} \\ &= 50,040 \times \frac{(70 - 20)}{(70 - -20)} \\ &= 50,040 \times 5/9 \\ &= \underline{27,800 \text{ Btu/hr}} \end{aligned}$$

Edge and bottom losses were not included in this calculation because it was to be used for comparison with "W".

Column Y - Temperature drop of circulating water.

This is the difference between "C" and "D".

Column Z - Degree days.

A calculation of degree days was made using "B" the average outside temperature during the test. The heat input during this test and the degree days make it possible to calculate the heat input required per degree day for this residence. A seasonal heating cost for a given number of degree days can be calculated using this information.

AA - Input per foot of Tubing Btu/hr.

This is the input as measured from the charts (Column "F") divided by the total length of 3/8 inch tube used in the panel.

BB - Heat Stored per foot of tubing Btu/hr ft.

This is the total heat stored per hour (Column "N") div-

ided by the total $3/8$ inch tube length.

CC - Output per foot of tubing Btu/hr ft.

This is columns AA and BB the total output from the top and bottom of the total panel area.

DD - Exposed panel output per sq ft of tubing Btu/ft hr

This is column CC times the ratio of the total panel area to the exposed panel area.

EE - Edge and Bottom losses Btu/hr ft.

These are 15% of the exposed panel output DD.

FF - Useful output per ft. of tubing Btu/hr.

This is the output from the exposed panel area (DD) less the edge and bottom losses 15%

GG - Panel Surface Temperature ° F.

This is the average panel surface temperature during the heating cycle calculated from Column "J" and "K".

HH - Indoor Air Temperature ° F.

This is taken from the temperature reading chart and is the average temperature for the heating cycle.

II - MET is the mean effective temperature from the design chart figure 13 by using the exposed panel output T and the panel surface temperature GG.

TABLE II TEST DATA FROM WESSEL RESIDENCE

A	B	C	D	E	F
Day of Test	Average Outside Temp. °F.	Average Supply Temp. °F.	Average Return Temp. °F.	Water Flow lb. per hr.	Total Input to Panel Btu/hr.
1st day Feb. 6-7	29°	138°	109°	2210	64,050
2nd day Feb. 7-8	41°	136°	106°	2845	85,500
3rd day Feb. 8-9	46°	135° 135° 136°	102° 104° 108°	2210 2210 2210	72,800 68,500 61,800
4th day Feb. 9-10	55°	136°	104°	2210	70,700
5th day Feb. 10-11	45°	136°	108°	2210	61,800
6th day Feb. 11-12	46°	130° 136° 136°	101° 106° 108°	2210 2210 2210	64,100 66,300 61,800
7th day Feb. 12-13	25°	140°	112°	2210	61,800
8th day Feb. 13-14	25°	150°	112°	2520	95,500
9th day Feb. 14-15	31°	150°	112°	2520	95,500
10th day Feb. 16-17	21°	130°	110°	2820	56,400
11th day Feb. 17-18	27°	122° 123° 115°	97° 102° 100°	2845 2820 2730	71,200 59,300 68,500
12th day Feb. 18-19	26°	115°	100°	2730	41,000
Average Values	34.8°	133°	105°	2441	66,610

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	G	H	I	J	K
Day of Test	Length of Heating Cycle hrs.	Heat Input per Cycle in Btu.	Heat Input per day in Btu.	Panel Surface T ₁ ° F.	Panel Surface T ₂ ° F.
1st day Feb.6-7	5.33	341,387	341,387	90°	104°
2nd day Feb.7-8	4.00	342,000	342,000	82°	92°
3rd day Feb.8-9	5.17 3.42 3.58	376,376 234,270 221,244	831,890	79° 84° 88°	90° 90° 94°
4th day Feb.9-10	3.83	270,781	270,781	81°	90°
5th day Feb.10-11	8.00	494,400	494,400	80°	95°
6th day Feb.11-12	2.00 2.08 5.17	128,200 137,904 319,506	585,611	83° 85° 84°	86° 90° 93°
7th day Feb.12-13	12.42	767,556	767,556	85°	98°
8th day Feb.13-14	8.00	764,000	764,000	83°	100°
9th day Feb.14-15	5.33	509,015	509,015	82°	96°
10th day Feb.16-17	17.42	982,488	982,488	76°	96°
11th day Feb.17-18	.75 7.75 1.67	53,400 459,575 68,500	581,470	85° 84° 81°	86° 92° 83°
12th day Feb.18-19	16.08	659,280	659,280	83°	92°
Average Values	6.22	414,340	621,500	83°	93°

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	L	M	N	O
Day of Test	T Panel Surface °F.	Heat Stored Cycle Btu.	Heat Stored Btu/hr.	Heat Stored Btu/hr ft
1st day Feb. 6-7	14°	152,600	28,630	30
2nd day Feb. 7-8	10°	109,000	27,250	28.6
3rd day Feb. 8-9	11° 16° 8°	119,900 65,400 87,200	23,190 19,120 24,360	24.3 20.0 25.5
4th day Feb. 9-10	9°	98,000	25,610	26.8
5th day Feb. 10-11	15°	163,500	20,440	21.4
6th day Feb. 11-12	3° 4° 9°	32,700 43,600 98,000	16,350 20,960 18,970	17.1 21.9 19.8
7th day Feb. 12-13	13°	141,700	11,410	12.0
8th day Feb. 13-14	17°	185,300	23,160	24.2
9th day Feb. 14-15	14°	152,600	28,630	30
10th day Feb. 16-17	20°	218,000	12,510	13.10
11th day Feb. 17-18	1° 8° 2°	10,900 87,200 21,800	14,530 11,250 13,050	15.2 12.0 13.1
12th day Feb. 18-19	9°	98,000	6,100	6.4
Average	10°	104,740	16,840	17.7

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	P	Q	R	S
Day of Test	Input to Panel Btu/hr ft ²	Total Output Btu/hr ft ²	Bottom & Edge Losses Btu/hr ft ²	Useful Panel Output Btu/hr ft ²
1st day Feb.6-7	67.2	37.2	5.6	31.6
2nd day Feb.7-8	89.6	60.9	9.1	51.8
3rd day Feb.8-9	75.5 71.2 65.0	51.2 51.2 39.5	7.6 7.6 4.9	43.6 43.6 34.6
4th day Feb.9-10	74.2	47.4	7.1	40.3
5th day Feb.10-11	65.0	43.6	6.5	37.1
6th day Feb.11-12	67.2 69.6 65.0	50.1 47.7 45.2	7.5 7.2 6.8	42.6 40.5 38.4
7th day Feb.12-13	65.0	53.0	8.0	45.0
8th day Feb.13-14	100.0	75.8	11.4	64.4
9th day Feb.14-15	100.0	70.0	10.5	59.5
10th day Feb.16-17	59.1	46.0	6.9	39.1
11th day Feb.17-18	79.9 62.1 43.0	64.7 50.0 29.9	9.6 7.5 4.5	55.1 42.5 25.4
12th day Feb.18-19	43.0	36.6	5.5	31.1
Average	69.8	52.1	7.8	44.3

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	T	U	V	W
Day of Test	Exposed Panel Output in Btu/sq ft hr	Panel Input Day Basis Btu/hr	Panel Output Day Basis Btu/hr ft ²	Exposed Panel Day Basis Btu/hr
1st day Feb. 6-7	37.6	14,200	17.8	15.1
2nd day Feb. 7-8	61.6	14,230	17.8	15.1
3rd day Feb. 8-9	51.8 51.8 41.2	34,600	43.5	47.0
4th day Feb. 9-10	48.0	11,200	14.1	12.0
5th day Feb. 10-11	44.1	20,600	25.9	22.0
6th day Feb. 11-12	50.6 48.2 45.7	24,400	30.7	26.1
7th day Feb. 12-13	53.5	31,900	40.0	34.0
8th day Feb. 13-14	76.6	31,800	40.0	34.0
9th day Feb. 14-15	70.8	21,200	26.6	22.4
10th day Feb. 16-17	46.5	40,000	50.2	42.7
11th day Feb. 17-18	66.0 50.5 30.4	24,250	30.4	25.9
12th day Feb. 18-19	37.0	27,400	34.4	29.1
Average	52.7	25,890	32.5	27.6

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	X	Y	Z
Day of Test	Calculated Heat Loss Btu/hr	Temp. Drop of Circulating Water	Degree Days
1st day Feb. 6-7	22,800	29°	36°
2nd day Feb. 7-8	16,200	30°	24°
3rd day Feb. 8-9	13,350	33° 31° 28°	19°
4th day Feb. 9-10	8,340	32°	10°
5th day Feb. 10-11	13,900	28°	20°
6th day Feb. 11-12	13,300	29° 30° 28°	19°
7th day Feb. 12-13	25,000	28°	40°
8th day Feb. 13-14	25,000	38°	40°
9th day Feb. 14-15	22,800	38°	34°
10th day Feb. 16-17	27,200	20°	44°
11th day Feb. 17-18	24,000	25° 21° 15°	38°
12th day Feb. 18-19	24,000	15°	39°
Average	19,691	28	Total 363

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	AA	BB	CC
Day of Test	Input/ft Tubing Btu/hr	Stored Btu/hr ft Tubing	Tubing Output Btu/ft hr Total Panel
1st day Feb.6-7	51.9	24.6	27.3
2nd day Feb.7-8	73.4	23.4	50.0
3rd day Feb.8-9	62.5 58.7 53.0	19.8 16.4 20.9	42.7 42.3 32.1
4th day Feb.9-10	60.6	22.0	38.6
5th day Feb.10-11	53.0	17.5	35.5
6th day Feb.11-12	55.0 56.9 53.0	14.0 18.0 16.3	41.0 38.9 36.7
7th day Feb.12-13	53.0	9.8	42.2
8th day Feb.13-14	82.0	19.9	62.1
9th day Feb.14-15	82.0	24.6	57.4
10th day Feb.16-17	48.4	10.7	37.7
11th day Feb.17-18	45.8 50.8 58.7	12.5 9.7 11.2	33.3 41.1 47.5
12th day Feb.18-19	35.2	5.2	30.0
Average Values	57.6	14.5	43.1

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	DD	EE	FF
Day of Test	Tubing Output Btu/hr ft Exposed Panel	Edge and Bottom Losses Btu/hr ft	Tubing Useful Output Btu/hr ft
1st day Feb.6-7	32.7	4.9	27.8
2nd day Feb.7-8	59.9	9.0	50.9
3rd day Feb.8-9	51.1 50.7 38.4	7.7 7.6 5.8	43.4 43.1 32.6
4th day Feb.9-10	46.2	7.1	39.1
5th day Feb.10-11	42.5	6.4	36.1
6th day Feb.11-12	49.1 46.5 44.0	7.4 7.0 6.6	41.7 39.5 37.4
7th day Feb.12-13	50.5	7.6	42.9
8th day Feb.13-14	74.5	11.2	63.3
9th day Feb.14-15	68.8	10.3	58.5
10th day Feb.16-17	45.2	6.8	38.4
11th day Feb. 17-18	39.8 49.2 56.9	6.0 7.3 8.5	33.8 41.9 48.5
12th day Feb.18-19	35.9	5.4	30.5
Average Values	51.7	7.7	44.0

TABLE II TEST DATA FROM WESSEL RESIDENCE (cont'd)

A	GG	HH	II
Day of Test	Panel Surface Temp. ° F.	Indoor Air Temp.	MET from Figure
1st day Feb.6-7	97°	69°	78°
2nd day Feb.7-8	87°	70°	58°
3rd day Feb.8-9	84.5° 87° 91°	68° 68° 70°	60° 61° 70°
4th day Feb.9-10	85.5°	69°	65°
5th day Feb.10-11	87.5°	68°	65°
6th day Feb.11-12	84.5° 87.5° 88.5°	69° 69° 69°	61° 64° 66°
7th day Feb.12-13	91.5°	70°	67°
8th day Feb.13-14	91.5°	68°	57°
9th day Feb.14-15	89°	69°	56°
10th day Feb.16-17	86°	67°	64°
11th day Feb.17-18	85.5° 88° 82°	68° 69° 64°	54° 63° 66°
12th day Feb.18-19	87.5°	69°	67°
Average Values	88°	68.5°	63.5°

Discussion of Results in Table II
for Wessel Residence

The installation of the flow meter and the temperature recorder, see figure 12, took about 3 hours. It might have taken a little longer if the flow elbow had not been previously installed. The only alteration required on the system was the addition of a flow check valve to prevent counter flow during the off periods. The baseboard radiation was turned off so the heat input to the residence was by the floor panel only.

Column "B", the average outside temperature, shows that the temperature range was from 21° to 55° . This is not the usual winter condition but the test was calculated on the heating cycle basis so the outside temperature has little affect other than causing a longer heating cycle. This shows up clearly when comparing columns "B" and "G" for the fourth and tenth day of the test.

A comparison of the average of column "U", the input to the panel in Btu per hr as calculated from the total heat input for the day, and column "X", the calculated heat loss per hour, shows that the average input was about 6000 Btu greater than the calculated heat loss. The ground losses do not make up for all of this difference, which would be about 25 percent. Part of it was caused by opening doors and windows when the house was overheated.

The solar heat through the glass, see figure 11, and the heat stored in the panel by the heating cycle during the night caused the room temperature to be uncomfortable during the morning hours. Part of the heat stored can be eliminated by a cycling thermostat which would anticipate the drop in heat loss in the morning by a shorter heating cycle. Column "C" shows the capacity for storage of the panel, this is one of the undesirable conditions in panel heating usually referred to as heat lag. Heat lag is undesirable during the beginning and end of the heating cycle because it causes a slow response to the heating demands. One solution to heat lag is the use of a light panel made possible by using a number of small reinforced concrete panels.

Heat storage during the heating cycle shows that all the heat output of the coil is not given off by the panel. From this it is evident that there is something limiting the amount of heat given off. This limit is called the panel surface coefficient and has the units of Btu per sq ft hr $^{\circ}$ F. The last dimension of the coefficients, $^{\circ}$ F, refers to a difference in temperature between the panel surface and the mean effective temperature of the room. This indicates that the output is varied by the panel's surface temperature and the NET which is a function of the outside temperature and the transmission coefficient of the walls and windows.



Figure 11 Wessel Residence

This view shows the South exposure consisting largely of thermopane windows. The overhang was designed to eliminate solar heat in the summer and utilize it in the winter.



Figure 12 Test Instruments

The boiler is partially visible at the left. The temperature recorder is in the center and the flow recorder on the right.

Column "Z" degree days and the sum of the heat input for the test, Column "I", will yield the heating cost for an average season as follows:

$$\begin{array}{rcl} \text{Total heat input for 12 days} & = & 7,458,000 \\ \text{Degree days in 12 days} & = & 363 \\ \text{Heat input required per degree day} & = & \frac{7,458,000}{363} \\ & = & \underline{20,700 \text{ Btu}} \end{array}$$

The approximate number of degree days per year for this locality from ASHVE Guide = 7930

$$\begin{array}{rcl} \text{Heat input required per year} & = & 20,700 \times 7,930 \\ & & \underline{164,000,000} \end{array}$$

Gallons of oil require using a heating value of

$$135,000 \text{ Btu/Gal. and } 80\% \text{ efficiency } \frac{164,000,000}{108,000} = 1520 \text{ Gal.}$$

$$\text{Cost using 16 cents per gallon} = \underline{\$243.00} \text{ per season}$$

Columns "Y", "C" and "D" give the data on the supply and return water data. In some cases the temperature drop is as high as 38°. Considering the mild temperatures involved, this is high and might become a problem in cold weather. The effect of a high temperature drop is that it causes the last part of the panel to be at a lower temperature than the supply end and so reduces the heat output of the last part. Some remedies for a high temperature drop are: the use of a larger circulator, the use of closer tube spacing and the use of shorter circuits.

The Wessel residence utilized a $3/4$ " circulator, 9 $\frac{1}{2}$ OC spacing and 170' circuits. Since the panel can not be changed the only correction that can be made is the use of a larger circulator.

The averages of columns "T", "J" and "X" are the most important results of the test because they show what can be expected of $3/8$ " tubing on 9" centers. These will be used in the design procedure which follows in the next section.

Columns "AA" through "FF" are the results of this test presented on the ft length of tubing basis. These have no direct bearing on this test but can be used for comparison of tube sizes and spacings in future tests. A comparison was made of the temperature gradients for the 9" spacing of $3/8$ " tubes at the Wessel residence and the $1/2$ " tubes 9" OC in the heat laboratory. This difference in temperature of the panel directly above the tube and that half way between the tubes was around 3° in both cases. This would indicate that $3/8$ " tubing performed as well as $1/2$ " tubing under operating conditions. The only advantage in using $1/2$ " tubing would be a faster pick-up which in most cases would not warrant the extra cost.

The tubing layout of the Wessel residence is yielding satisfactory results except for the cold portion around the edge of the panel. This was caused by not running the first

coil close enough to the foundation wall to take care of the edge losses. A check on the panel surface temperatures at the outer edge of the living room showed that the temperature 15" from the outer wall was 90° while the temperature at the outside wall was 60°. This would indicate that the 1/2" of Celotex insulation used between the panel and foundation wall was inadequate.

CHAPTER III

RADIANT HEATING DESIGN

The most important consideration in the design of a panel system is the desired surface temperature. Almost all types of structures can be heated by a radiant panel if no restrictions are placed on the surface temperature. Most authorities on panel heating agree that the panel surface should not be above 85° and that the maximum output per sq ft of panel at this temperature is 35 Btu. This statement may be true but should not be construed as a limit for panel heating. It is evident from the theory of radiant heat transfer that the amount of heat transferred by radiation depends primarily on the temperature of the radiant and receiving bodies. The best possible way to show the amount of heat transferred would be a curve with the unit heat transfer versus the temperature of the receiving body while the radiant body remains constant. Since all the heat transferred from a radiant panel is not by radiation alone but a combination of radiant and convective heat transfer, the curve used for such a panel must take this into consideration. The best chart to date giving this information is the Graphical Method, first proposed from calculations by Giesecke¹. This was done with two curves, one showing the heat delivered by radiation for a mean effective temperature (MET) and a panel surface temp-

1. How to Figure Total Heat Delivered by a Panel, pp. 484
AP & AC August, 1940

erature and the other is the heat delivered by convection and conduction for a difference between panel and air temperature. The product of these outputs for a given panel surface temperature and mean effective temperature was plotted as shown in figure 13 by R. G. Vanderweil¹, later an experiment² was set up by Vanderweil and R. S. Leigh³ by which they proved that the output of a panel can be predicted, within 10%, by using this chart. This experiment consisted of a 2 sq ft heated panel which was placed at various positions throughout the test room. The room was equipped with numerous thermocouples fixed on the wall, ceiling and glass areas, thus making available temperatures for the calculation of a MET. The heat output of the panel was calculated from the electrical input and an assumed bottom loss of 3%.

The test results from the Wessel residence were compared to this chart on the basis of mean effective temperature. The chart MET was read off the chart using the exposed panel output Col. I, Table I and the panel surface temperature GG, Table I.

No data was taken in the Wessel test for the calculation of a MET. However, an assumption was made that the MET is 2° lower than the room temperature. The basic consideration for

1. Heating and Ventilating October, 1945, pp. 102, "Heat Flow in Panel Heated Rooms".
2. Measurement of Heat Output of a sq ft Panel, Feb. 1948 p. 107
3. Project Engineer for Chase Brass and Copper.

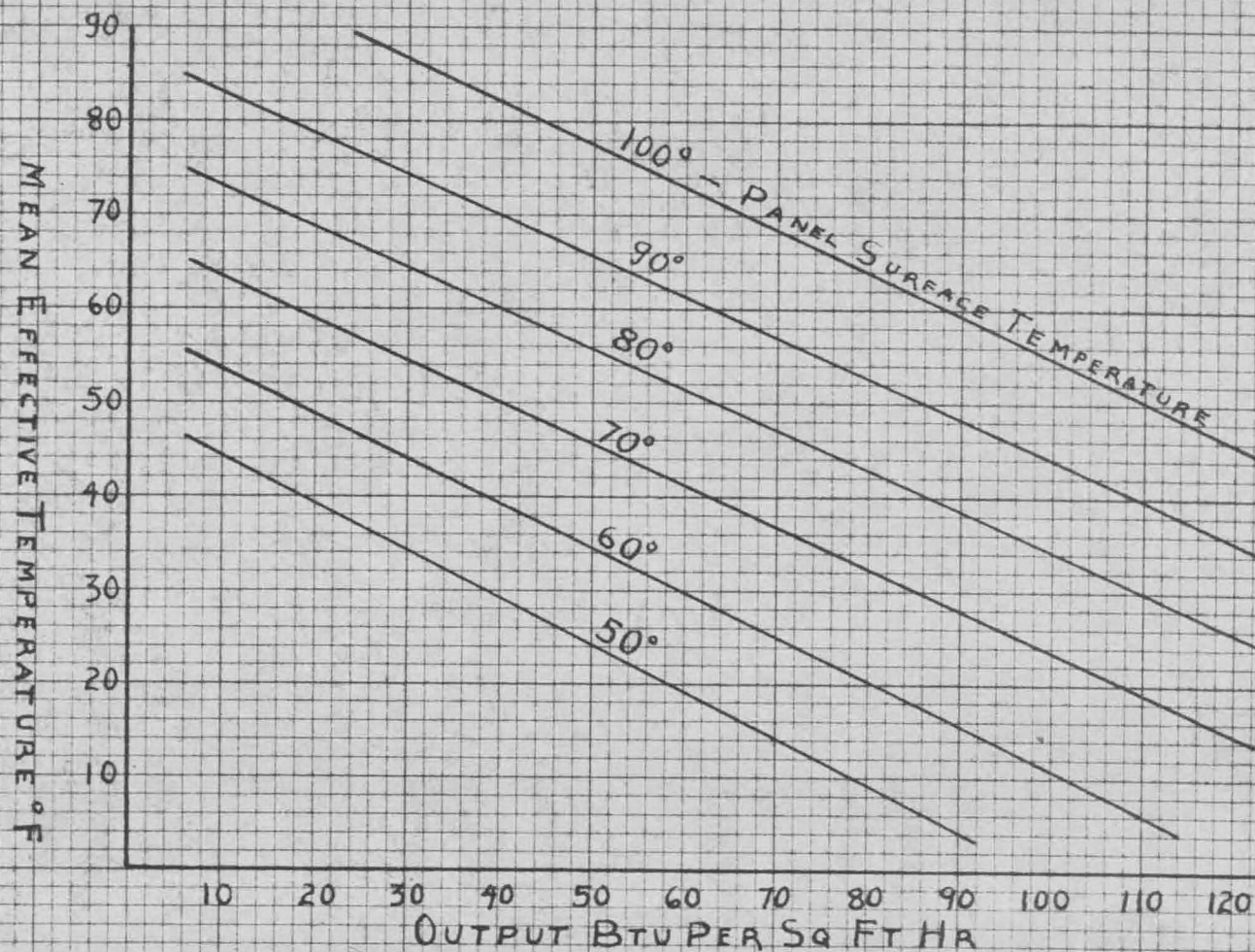


FIG-13-DESIGN CHART

this assumption was the large glass area and unfinished interior surfaces. Inside surface temperatures of the glass areas were taken at an outside temperature of $+10^{\circ}$. These fell below the scale of the pyrometer which was 60° . The comparison of MET is as follows:

The average MET from the chart is 63.5

The average MET assumed is the average of
Column HH less 2° ($68.5 - 2$) = 66.5°

The difference is 3° or $\frac{3}{68.5} = \underline{\underline{4.7\%}}$

The next design step is the choice of tube size and spacing to give this output. Before a spacing is chosen certain limits must be considered. First, the maximum spacing which is limited by the allowable temperature gradient between coils. For a design utilizing a panel surface temperature over 90° this gradient must be held at a minimum to eliminate hot spots and assure a uniform high output. The minimum spacing used must be a balance between the cost, temperature gradient and panel response desired.

Material from a preliminary survey of local installations was consulted to help decide on the more desirable spacing. This survey was conducted at an outside temperature of $+15^{\circ}$ to -23° F. A surface type pyrometer was used to read the temperatures from the top of one coil to the top of the next, also supply and return water temperatures.

The survey results are:

Wessel Residence

3/8" copper tubing 9" OC. Temperature outside +15° F.

Readings were taken of concrete surface temperature in the center of the living room.

95° over the tubes to 92° between tubes.

Supply temperature 136° and return temperatures 108°

Swingle Residence

1" copper tubing 12" and 6" OC. Temperature outside -15°F.

82° to 93° for 12" OC spacing concrete surface

94° to 96° for 6" OC spacing asphalt tile surface with

120° supply water and 104° return temperature.

Brockman Residence

1/2" copper tubing 8 1/2" OC. Temperature outside -15° F.

84° to 78° on wood floor over concrete panel

83° to 74° over tile finished floor

Supply water temperature 128° return temperature 108°.

Benson Residence

Outside temperature -23°

1/2" tubes 6" to 9" OC. between finished and sub floor

tubes imbedded in concrete fill.

83° to 78° for 9" OC. spacing

88° to 85° for 6" OC. spacing.

Supply water temperature 145° and return temperature 133°

Helburn Residence

1" W.I. pipe 9" OC. Temperature outside -15° .

89° to 92° center of living room asphalt tile covering

Boiler supply water temperature 116° and return 109°

Iverson Residence

1/2" copper 8" OC. Temperature outside -15°

102° to 97° center of living room asphalt tile floor covering.

Supply water temperature 158° , return 97°

An attempt was made to plot the results of this survey on a curve to show the mean water temperature for a given spacing and surface temperature. However, it was found that insufficient data was available and not enough points could be located to draw a decisive conclusion. The common error in this survey was caused by not knowing the length of time the heating system had been operating when the readings were taken. The only way to obtain this data would be to set up a prolonged test on each residence similar to that conducted on the Wessel residence. The results from this 12 day test fell in a pattern sufficient to give the 9" OC. spacing line in figure 14.

Since the ratio of surface to mean water temperatures was established by the curve it was a simple matter to calculate curves for other tube spacings. The assumption is that the temperature changes are proportional to the change

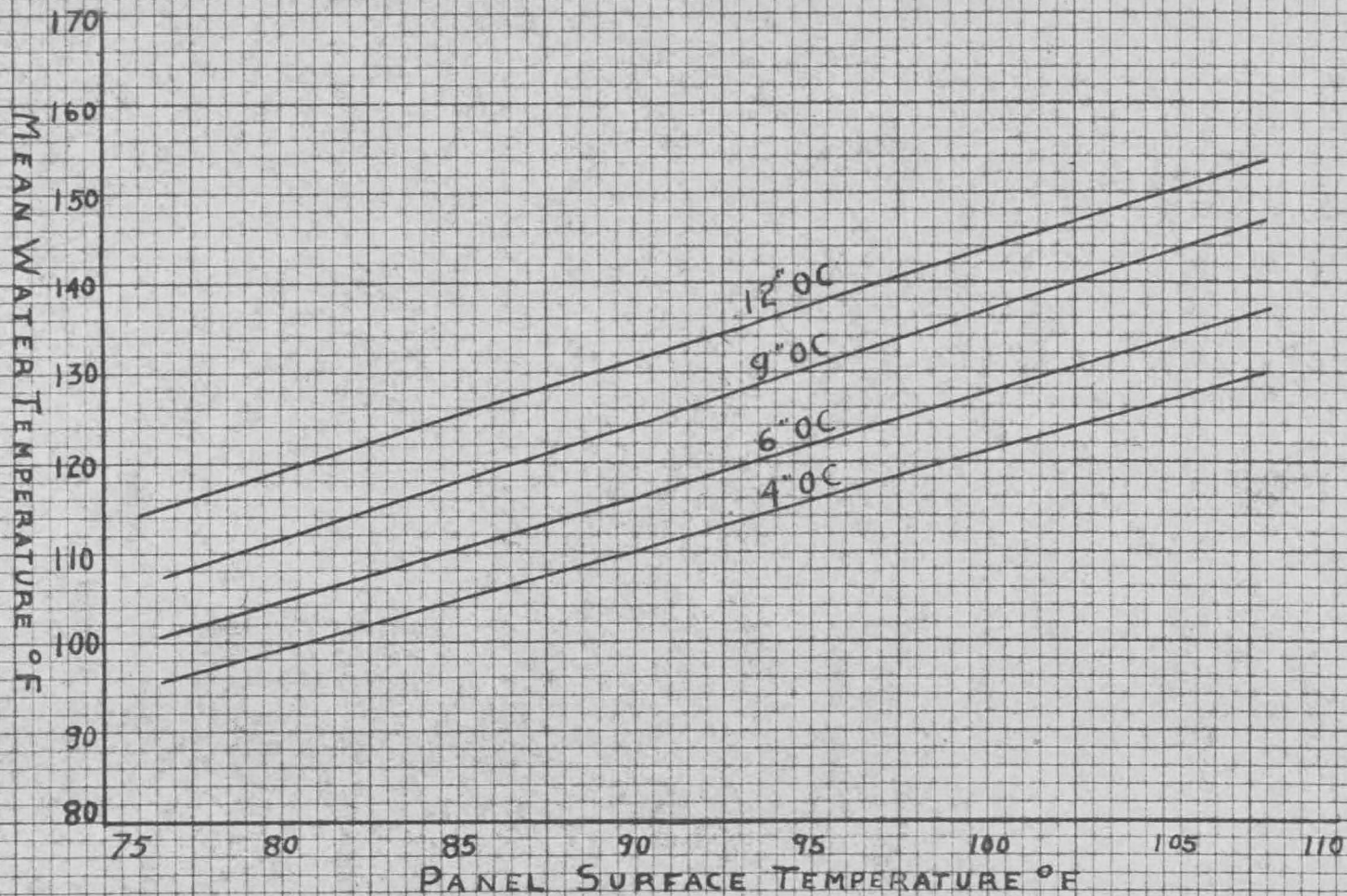


FIG-14-DESIGN CHART

in mean distance of heat travel.

The calculations for additional curves are as follows:

Using the basic heat transfer formula

$$Q = U A (T_s - T_w)$$

Where Q is the rate of heat transfer per unit area.

U is the coefficient of heat transfer for the mean distance of heat travel through the concrete.

T_s is the average surface temperature of the panel

T_w is the mean temperature of the calculating water.

From the Wessel Panel

$$T_s = 88^\circ$$

$$T_w = 119^\circ$$

The mean distance used for calculating the U for a spacing was found by taking an average of the distance from the tube wall to the panel surface and the tube wall to the center line of the tube spacing.

For 9" OC. spacing and a depth of bury of 2" this is

$$\frac{2" + 9/2"}{2} = \frac{2 + 4.5}{2} = \underline{3.25 \text{ inches}}$$

$$U = \frac{1}{x/k}$$

Using a K = 10 Btu per sq ft hr ° F. per inch of thickness for average concrete

$$U = \frac{1}{3.25/10} = \frac{10}{3.25} = \underline{3.08} \text{ Btu/hr sq ft } ^\circ \text{ F.}$$

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So $Q_1 = 3.08 A_1 (119 - 88^\circ)$
 $= 102 A_1$

and $Q_2 = U_2 A_2 (T_w - T_s^\circ)$

Assuming

$$A_1 = A_2$$

$$T_s = T_{s2}$$

and setting $Q_1 = Q_2$ the equation becomes

$$102 = U_2 (T_w - 88^\circ)$$

U_2 for 6" OC. spacing

$$102 = 4 (T_w - 88^\circ)$$

$$\text{Mean distance} = x = \frac{2 + 6/2}{2}$$

$$T_w = \frac{102}{4} - 88^\circ$$

$$x = 5/2 = 2.5$$

$$T_w = \underline{\underline{113.5^\circ}}$$

$$U_2 = \frac{1}{2.5/10} = \frac{10}{2.5} = \underline{\underline{4}}$$

For surface temperatures other than 88° , Q_1 or the left hand side of the combined equation changes in proportion to the new temperature ratio.

For 82° on the 9" OC. curve the water temperature is 113°

so, $Q_1 = 3.08 \times (113 - 82^\circ)$
 $= 95.5$

and, $T_w = \frac{95.5}{U_2} - 82^\circ$

Where U_2 is calculated for the desired tube spacing and T_w is the water temperature at the spacing if the surface temperature is 82° .

for $T_s = 95^\circ$
 $Q_1 = 3.08 \times (130 - 95^\circ)$
 $= 108$

$$T_w = \frac{108}{U_2} + 95^\circ$$

for $T_s = 105^\circ$
 $Q_1 = 3.08 (140 - 105)$
 $= 108$

$$T_w = \frac{108}{U_2} + 105$$

These four surface temperatures locate the curve with respect to the water temperature and were used for locating 4, 6, and 12" spacing curves on figure 14.

The chart figure 14 was calculated for a 3/8" tube buried 2" deep but can be applied for 1/2" and 3/4" tubing because the tube size seems to have little effect. The basis for this is the limit set on the amount of heat transferred by panel surface coefficient. From this viewpoint no matter how much heat is supplied to the panel only a portion as shown in figure 13 can be given off. This was proved by the amount of heat stored in the Wessel test.

The curves in figure 14 were limited to a 12" OC. spacing. A larger spacing should not be used because it would cause a considerable temperature gradient. Spacing smaller than 4" is hard to obtain and useless because the temperature

is practically uniform across this spacing.

Room temperature can be substituted for the mean effective temperature on chart figure 14 if the room has a small glass area and is well insulated. Temperature readings taken on the walls of panel heated rooms show that the surface temperature is very close and in some cases higher than the air temperature. This is caused by the radiant effect of the panel which heats the walls by long heat waves. Poorly insulated walls and large glass areas do not receive enough heat by radiation to raise the surface temperature so have a considerable effect in lowering the mean effective temperature.

Application of Design Charts to Helburn Residence

This residence is perhaps the greatest challenger of radiant heating in this locality because of the large glass area used. (see figures 15 and 16) The total panel area is 2100 sq ft, but the built-in closets and cabinets reduce this to an estimated 1750 sq ft. A heat loss of 83,264 Btu/hr requires an output of 47.5 Btu per sq ft of the exposed panel area.

Step 1. Given a desired room temperature of 70° and a maximum panel surface temperature of 92° at -20° the maximum output would be found as follows:

Since the amount of glass area will effect the mean effective temperature and the living room has a greater portion of glass than other parts of the house, a MET will be calculated for it. An acceptable way to calculate a MET is to take the sum of all the areas facing the panel multiplied by their surface temperature and divide it by the total of the individual areas. To do this the inside surface temperature must be calculated. This is possible since the temperature at any point in the path of heat flow can be calculated using a ratio of the resistances to that point by the following formula¹:

1. 1950 ASHVE Guide pp. 142



Figure 15 Helburn Residence

The northwest exposure shows the glass area used in the study, bathroom and bedrooms.



Figure 16 Helburn Residence

The southwest exposure shows the glass area in the living room, foreground and the utility room right rear.

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$$\frac{R_1}{R_2} = \frac{T_1 - T_x}{T_1 - T_o}$$

where R_1 = inside film resistance $1/1.65 = .606$
 R_2 = total resistance as used in the heat loss calculations.

T_1 = inside design temperature = 70°

T_o = outside design temperature = -20°

T_x = surface temperature to be found

T_x for double glass

$$U = .45 = 1/R_2$$

$$R_2 = 2.2$$

$$\frac{.606}{2.2} = \frac{70 - T_x}{90}$$

$$T_x = 70 - \frac{.606 \times 90}{2.2}$$

$$70 - 24.8$$

$$T_x = \underline{45.2^\circ} \text{ window surface temperature}$$

T_x for wall type "A"

$$R_2 = R_t = 20.503$$

$$R_t = 1/f_1 = .606$$

$$\frac{.606}{20.5} = \frac{70 - T_x}{90}$$

$$T_x = 70 - \frac{.606 \times 90}{20.5}$$

$$T_x = 70 - 2.66$$

$$T_x = \underline{67.30} \text{ wall "A" surface temperature}$$

T_x for ceiling

$$R_t = 7.876$$

$$T_x = 70 - \frac{.606 \times 90}{7.876}$$

$$= 70 - 6.93$$

$$T_x = \underline{63.1^{\circ}} \text{ ceiling surface temperature}$$

T_x for inside walls

$$T_x \text{ surface} = T_{\text{room}}$$

$$= \underline{70^{\circ}}$$

Since wall type "A", double glass, ceiling and inside wall make up the major portion of the total room surface facing panel, the door and other small areas will have little effect on the MET.

$$\text{Glass area} \quad 355 \text{ sq ft} \times 45.2^{\circ} = 15,100$$

$$\text{Ceiling} \quad 616 \text{ sq ft} \times 73.1^{\circ} = 45,000$$

$$\text{Wall type "A"} \quad 397 \text{ sq ft} \times 67.3^{\circ} = 26,700$$

$$\text{Inside wall} \quad 322 \text{ sq ft} \times 70^{\circ} = \underline{22,500}$$

$$\text{Total area} \quad 1670 \text{ sq ft} \quad \underline{109,300}$$

$$\frac{109,300}{1670} = \underline{66.50 \text{ MET}}$$

From the foregoing it is evident that this calculation would become tedious if performed for every room in the house. The general procedure would be to calculate it only for critical areas employing an unusually large portion of glass. A simpler method would be to assume a MET considering the glass area used. The limits of this assumption would be a Met within $+3^{\circ}$ to -5° of the air temperature depending on the amount of glass area. Using a MET of 66.5° and a maximum panel surface temperature of 92° , the chart figure 13 yields an expected output of 52 Btu/sq ft hr.

Step II. The output from Step I has set the panel surface temperature at 92° . This surface temperature can be obtained by a number of pipe sizes or spacings; however, after a study and consideration of the Wessel residence, three decisive factors become evident. First, a close tube spacing should be used to obtain a uniform surface temperature and a fast pick up after the sun sets and does not add solar heat. Next, the use of short coils, 170 feet or less in critical areas, causes a higher velocity on greater heat output to the slab which also increases the pick up. Finally, 4 inch spacing should be used under large windows and outside walls. Justification of this concentrated tube spacing is from the following view points. First, 4"OC. spacing concentrates the heat on these outside areas which raises the temperature and

output. In this case, 6" and 4" OC. spacing were chosen and used in the same coil. The effect of this can be read from figures 13 and 14. For a mean water temperature of 118° the portion of 6" spacing will have surface temperature of 92° giving 53 Btu/sq ft hr while that of 4" spacing will be at a temperature of 99° giving 66 Btu/sq ft hr. These outside areas are occupied for shorter periods of time so the high temperature is less objectionable than it would be if the entire panel were at that temperature. Second, tests¹ have shown that the greatest heat loss is at the panel edge and drops appreciably beyond 2 feet from the outer edge. By keeping this area hot no heat is transferred to it from the central areas of the panel. This may result in a slightly higher edge loss; however, this is compensated for by the elimination of the usual cold area around the edge of the panel. Finally, a hot area next to the outside wall will counteract the cold convective current set up by the cold window surface which usually travels along the floor until it is sufficiently heated to rise and mix with the room air.

Results for the panel test laboratory showed that a considerable amount of heat can be lost through the edge and bottom of the panel. It does not necessarily follow that a

loss of this magnitude should be expected in this installa-
1. See Heat Losses From Concrete Floor Panels by ASHVE Laboratory, Vol.23 No.1 Heating, Piping and Air Conditioning.

tion. However, since this loss is worth considering it was deemed advisable to use the best available edge insulation, 2 inches of foam glass.

The large glass area in the Wessel residence added some solar heat which could not be utilized. A similar problem was anticipated in the Helburn residence so a split system, see figure 18, will be installed to make use of the available solar heat. The bedrooms and north side, figure 15, will be on one system controlled by a thermostat in the master bedroom. The other system will heat the south and west sides of the residence, figure 16, controlled by the thermostat in the living room. It was reasoned that this design would allow the solar heat, on sunny days, to heat the south and west rooms while the bedrooms and north side were heated by the boiler. The additional cost of the split system is \$100 or 3% of the total heating system cost. Aside from saving fuel, this additional cost is justified by a standby system in case one system fails, the faster pick-up with the two pumps and better control over bedroom temperatures.

Panel Design For Helburn Residence

The first step in heating design is the heat loss calculations. These are as follows:

Calculation coefficients.

Type "A" wall		R
10" logs	$K = .90 = 10/90$	= 11.10
Alfoil Insulation		= 7.69
3/4" wood paneling	$= 3/4 / .80$	= .94
$1/f_o = 1/6$		= .167
$1/f_1 = 1/1.65$		= <u>.606</u>
	R_t	= 20.503
$U = 1/20.5 = \underline{\underline{.0488}}$		

Type "B" wall		R
8" concrete	$K = 12 = 8/12$	= .665
Alfoil Insulation	$= 1/c$	= 7.69
1/2" sheet rock	$= 1/c$	= .35
$1/f_o$	$= 1/6$	= .167
$1/f_1$	$= 1/1.65$	= <u>.606</u>
	R_t	= 9.478
$U = 1/9.478 = \underline{\underline{.105}}$		

Type "C" wall R
same as "B" except partially underground

Type "D" wall
3/4" plywood outside = $1/c$ = .75

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1/2" plywood inside = 1/c	=	.47
Alfoil Insulation	=	7.67
1/f _o	=	.167
1/f _i	=	<u>.606</u>
R _t	=	9.663

$$U = 1/9.663 = \underline{\underline{.103}}$$

Combined Roof and Ceiling Coefficients

Roof

	R
6" soil K = 8.5 = 6/8.5	= .706
4 ply roof = 1/c	= .28
ship lap fir or pine = 1/c	= .98
1/f _o = 1/10	= .10
1/f _a = 1/2.2	= <u>.45</u>
R _{tl}	= 2.516

$$U = 1/2.516 = \underline{\underline{.398}}$$

Ceiling

	R
Alfoil = 1/c	= 5.88
3/4" wood panel	= .94
1/f _a = 1/2.2	= .45
1/f _i = 1/1.65	= <u>.606</u>
R _{t2}	= 7.876

$$R_t = R_1 + R_2 = 10.392$$

$$U_t = 1/10.4 = \underline{\underline{.096}}$$

No edge or bottom losses were included in these calculations because they do not affect the design procedure. They will be taken care of by increasing the supply water temperature to maintain the designed surface temperature.

Table III Heat Loss Data For Helburn Residence

	Area or Volume	Coeffi- cient U	Temp. ° F.	Heat Loss Btu/hr	Totals Btu/hr
<u>Living room</u>					
Walls "A"	397	.049	90°	1750	
Single glass	24	1.13	90°	2440	
Double glass	335	.45	90°	13600	
Ceiling	616	.096	90°	5320	
Door & storm door	20	.32	90°	595	
Infiltration lac	5852	.018	90°	9500	33,195
<u>Study</u>					
Walls "A"	71	.049	90°	312	
Double glass	74	.45	90°	3000	
Ceiling	90	.096	90°	778	
Infilt. 3/4 ac	641	.018	90°	1040	5,130
<u>Bath #1</u>					
Walls "A"	24	.049	90°	106	
Double glass	25	.45	90°	1100	
Ceiling	78	.096	90°	674	
Infilt. 3/4 ac	438	.018	90°	710	2,590
<u>Bedroom # 1</u>					
Walls "A"	52	.049	90°	228	
Double glass	60	.45	90°	2430	
Ceiling	180	.096	90°	1560	
Infilt. 3/4 ac	1010	.018	90°	1640	5,858
<u>Bedroom # 2</u>					
Walls "A"	18	.049	90°	79	
Double glass	20	.45	90°	810	
Ceiling	120	.096	90°	1040	
Infilt. 3/4 ac	675	.018	90°	1090	3,019
<u>Bedroom # 3</u>					
Walls "A"	180	.049	90°	790	
Double glass	57	.45	90°	2310	

Table III, cont'd.

	Area or Volume	Coeffi- cient U	Temp. ° F.	Heat Loss Btu/hr	Totals Btu/hr
Ceiling	120	.096	90°	1040	
Infiltra. 3/4 ac	675	.018	90°	1090	5,230
<u>Rear Storeroom</u>					
Walls "A"	30	.049	90°	132	
Walls "B"	17	.105	90°	161	
Wall "C" above grade	36	.105	90°	340	
Wall "C" under ground	50	.105	40°	210	
Ceiling	72	.096	90°	622	
Infilt. 3/4 ac	360	.018	90°	584	2,049
<u>Bath # 2</u>					
Walls "C" above	25	.105	90°	232	
Double glass	3	.45	90°	121	
Ceiling	77	.096	90°	665	
Infilt. 3/4 ac	386	.018	90°	625	
Wall "C" below	36	.105	40°	151	1,794
<u>Dining and Entry</u>					
Wall "B"	32	.105	90°	302	
Double glass	27	.45	90°	1095	
Door & Storm door	20	.32	90°	575	
Ceiling	171	.096	90°	1480	
Infilt. 1 ac	1496	.018	90°	2420	5,872
<u>Kitchen and Utility</u>					
Wall "D"	11	.103	90°	102	
Wall "E"	44	flower pot			
Double window	75	.45	90°	3040	
Single window	24	1.13	90°	2440	
Door & Storm door	20	.32	90°	575	
Ceiling	429	.096	90°	3710	
Infilt. 1 ac	3217	.018	90°	5200	15,067
<u>Halls</u>					
Ceiling	216	.096	90°	1870	
Infilt. 1 ac	970	.018	90°	1570	3,440
<u>Walls exposed to unheated rooms</u>					
Walls 28 x 8, assum- ing plain partition, & unheated rooms at 40°	224	.37	30°	2020	2,020
Total Heat Loss =					83,264 Btu/hr

Tubing Layout For Helburn Residence

The general procedure used for calculating the amount and spacing of the coils, see figure 17, is as follows:

Living room

Heat loss 33.195 from Table III

Heat output, see figure 13, for a 92° surface temperature is 52 Btu/sq ft hr.

Panel area required = $\frac{33.195}{52} = 637$ sq ft

Available panel area, by measurement, from the floor plan excluding a small area around the fire place and the flower pot in the northwest corner = 600 sq ft

Since the required panel area is larger than the available panel area, a section of the panel must be designed as explained in Step II.

The amount of high temperature panel required was assumed to be a section 2 ft wide around the outside walls of the room. This assumption gives

2' x 52' or 104 sq ft at 90° or 66 Btu/hr sq ft

Leaving $600 - 104 = 496$ sq ft at 92° or 52 Btu/hr sq ft

The combined output will be, see figures 13 and 14,

$$104 \times 66 = 6850 \text{ Btu/hr}$$

$$496 \times 52 = \underline{26800}$$

$$\text{Total} = \underline{\underline{33.650 \text{ Btu/hr}}}$$

The amount of 3/8" tubing required is:

$$104 \text{ sq ft} \times 3 \text{ ft tubing per sq ft} = 312$$

$$496 \text{ sq ft} \times 2 \text{ ft tubing per sq ft} = \underline{992}$$

$$\text{Total } 3/8" \text{ tubing used in living room} = \underline{\underline{1304}}$$

Study

$$\text{Heat loss} = 5130$$

$$\text{Panel area required} = \frac{5130}{52} = 99 \text{ sq ft}$$

$$\text{Panel area available} = 90 \text{ sq ft}$$

Assume

$$34 \text{ sq ft } 4" \text{ OC. at } 90^\circ \text{ or } 66 \text{ Btu/sq ft hr} = 2240$$

$$56 \text{ sq ft } 6" \text{ OC. at } 92^\circ \text{ or } 52 \text{ Btu/sq ft hr} = \underline{2910}$$

$$\text{Total} = \underline{\underline{5130 \text{ Btu/hr}}}$$

$$\text{Tubing required} = (34 \times 3) + (56 \times 2) = 224 \text{ ft}$$

Bath # 1

$$\text{Heat loss} = \underline{2590 \text{ Btu}}$$

$$\text{Panel required} = \frac{2590}{52} = 49.7 \text{ sq ft}$$

$$\text{Available panel area} = 78 \text{ sq ft}$$

A bath room should be slightly warmer than the rest of the house so 60 sq ft of the floor area was used at 6" spacing or 52 Btu.

$$\text{Total output} = 60 \times 52 = \underline{3120 \text{ Btu/hr}}$$

$$\text{Tubing required} = 60 \times 2 = \underline{120 \text{ ft}}$$

Bedroom # 1

$$\text{Heat loss} = 5858 \text{ Btu/hr}$$

Available panel area = 145 sq ft

Assume 25 sq ft with 6" spacing at 52 Btu/sq ft hr

Finish panel with 9" spacing which gives a surface temperature of 85° from chart figure 14 and output of 41 Btu/sq ft hr. (From chart figure 13)

Total output = $(25 \times 52) + (120 \times 41) = \underline{6100 \text{ Btu/hr}}$

Tubing required = $(25 \times 2) + (120 \times 1.33) = \underline{210 \text{ ft}}$

Bedroom # 2

Heat loss = 3019 Btu/hr

Available panel area = 84 sq ft

Assume 9 sq ft at 6" OC. = 52 Btu/sq ft hr

40 sq ft at 9" OC. = 41 Btu/sq ft hr

35 sq ft at 12" OC. = 24 Btu/sq ft hr

Total output $(9 \times 52) + (40 \times 41) + (35 \times 24) = \underline{2948 \text{ Btu/hr}}$

Tubing required $(9 \times 2) + (40 \times 1.33) + (35 \times 1) = \underline{106 \text{ ft}}$

Bedroom # 3

Heat loss = 5230 Btu

Available panel area = 100 sq ft

Assume 20 sq ft, 4" OC., 99° = 66 Btu/sq ft hr

80 sq ft, 6" OC., 92° = 52 Btu/sq ft hr

Total output = $(20 \times 66) + (80 \times 52) = \underline{5480 \text{ Btu}}$

Tubing required = $(20 \times 3) + (80 \times 2) = \underline{220 \text{ ft}}$

Rear Storeroom

Heat loss = 2049 Btu/hr

Available panel area = 39 sq ft
Assume 85 sq ft, at 12" OC, 79° = 24 Btu/sq ft hr
Total output = 85 x 24 = 2040 Btu/hr
Tubing required = 85 x 1 = 85 ft

Bath # 2

Heat loss = 1794 Btu/hr
Available panel area = 39 sq ft
Assume 39 sq ft at 6" OC, 92° = 52 Btu/sq ft hr
Total output = 39 x 52 = 2030 Btu/hr
Tubing required = 39 x 2 = 78 ft

Dining entry

Heat loss = 5872 Btu/hr
Available panel area = 130 sq ft
Assume 50 sq ft at 6" OC, 92° = 52 Btu/sq ft hr
Assume 80 sq ft at 9" OC, 85° = 41 Btu/sq ft hr
Total output (80 x 41)-(50 x 52) = 5880 Btu/hr
Tubing required (80 x 1.33)-(50 x 2) = 206 ft

Kitchen and Utility

Heat loss = 15,067 Btu/hr
Assume 20 sq ft at 4" OC, 99° = 66 Btu/sq ft hr
30 sq ft at 12" OC, 79° = 24 Btu/sq ft hr
260 sq ft at 6" OC, 92° = 52 Btu/sq ft hr
Total (20 x 60)-(30 x 24)-(260 x 2) = 15540 Btu/hr
Tubing required(20 x 34)-(30 x 1)-(260 x 2) = 610 ft

Halls

No heat will be supplied to the halls since the supply and return lines were placed in them and will sufficiently heat them.

Specifications

I

A - Installation All tubes shall be cut using a copper tubing cutter and reamed to the original inside diameter, sized with the appropriate tubing sizer, cleaned with steel wool coated with soldering flux and soldered with 95-5 or 50-50 wire type solder. Care should be exercised in not moving the soldered fitting until the solder has set.

All coils in the living room shall be positioned to within 1" of the panel top at the time the concrete is poured. All other coils shall be positioned not more than 2" and not less than 1" from the panel surface.

B - All coil bends of 6" or less in diameter shall be made by using a spring type tubing bender. No coupling shall be allowed within 1 foot of any bend.

C - If any tube should be kinked on installation no attempt shall be made to straighten it. The kinked portion shall be removed and a new piece soldered in its place.

D - Tube Connectors All tube connectors within a coil shall be sweat joints using a tubing coupling. No swedged joints shall be allowed.

E - Crossing Tubes If it is necessary to cross any tube, it shall be done with a very gradual bend allowing a 1/2" space between the tubes.

II

Header Construction Each distribution header shall be of sweat type construction using 1" hard drawn type M or type L tubing. All reductions from header size to circuit size shall be made with reducing tees.

III

A - Copper Tube Copper tubing for flow coils shall have an outside diameter of .50" and a wall thickness of not less than .028 or greater than .032 .

B - Main Supply Lines shall be of 3/4" and 1" soft copper tubing equivalent to type K chase tubing.

IV

A-Gate Valves shall be equivalent to Ohio brass No. 25 line valve.

B - Check Valves shall be equivalent to Ohio brass No. 2 line swing check valve.

C - Circuit Stop Valves shall be equivalent to No. 6 Ohio brass globe stop valves.

D - Water Supply Valve Valve on line from domestic water supply shall be equivalent to No.2 Ohio brass line valve.

E - Balancing Valves The flow through the circuits shall be controlled by two shut off valves on opposite ends of the circuit.

V

A - Expansion Tank shall be equivalent to a number 15, fif-

teen gallon Bell and Gassett tank.

B - Hydro Flow Valve shall be equivalent to a B & G Hydro Flow air charger and tank drainer.

VI

Circulators 1" and 2" shall be equivalent to Bell and Gassett circulators.

VII

Pressure Relief Valves shall be equivalent to 3/4" simplex with a maximum setting of 30 pounds.

VIII

Thermostats and Relays Each circulator shall be controlled by a low voltage thermostat and relay equivalent to a Minneapolis-Honeywell Model TM 11A and S-10.

Aquastat and stack control shall be equivalent to a Minneapolis-Honeywell aquastat N. L444 and stack switch Minneapolis-Honeywell No. RA 117A.

IX

Edge Insulation The outside edge of the heated panel shall be insulated from the foundation wall by a 2" foam glass or its equivalent.

X

Circuit Venting Procedure The procedure for venting air from the system shall be: first, close all circuit valves and the two valves on the boiler supply and return lines; second, open the drain cock on the return header; third, open the

drain cock on the supply header allowing domestic water to enter system under full pressure; fourth, open one circuit allowing a high velocity stream of water to flow through the circuit until a solid stream of water is seen at the return cock; fifth, close this circuit and progress in a similar manner through the system. To fill boiler first close all circuit lines, second, open supply and return valves to boiler and fill system to specified boiler pressure.

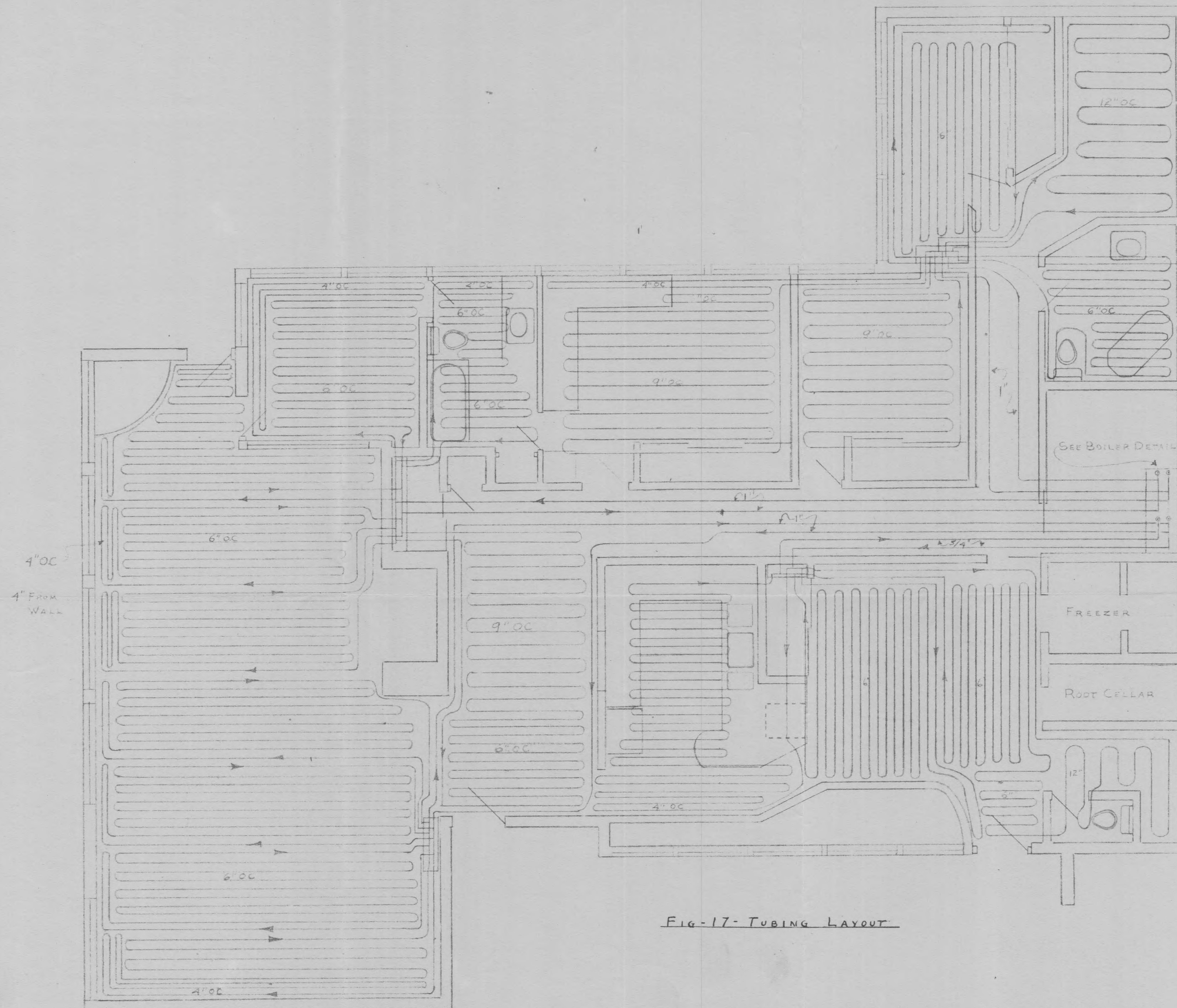


FIG-17- TUBING LAYOUT

RADIANT HEATING
 HELBURN
 RESIDENCE
 BOZEMAN MONTANA
 FEB 19, 1951 By LEO GLASSEN

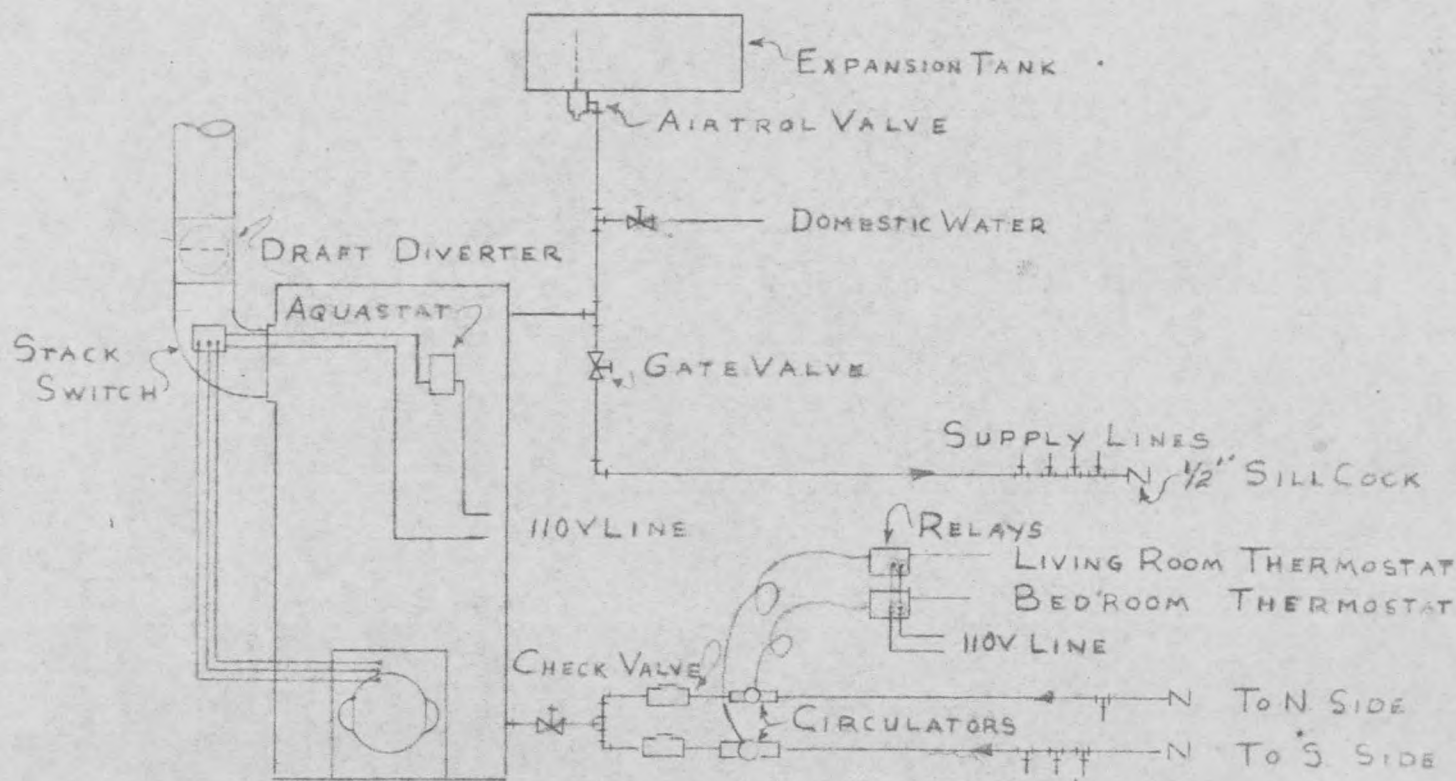


FIG-18- BOILER PIPING DETAIL

CHAPTER IV

FLOOR COVERING TEST PANEL

Since most floor panels are made of concrete, there is the problem of covering the concrete to give a more pleasant and better working surface. The first question is, what type of covering should be used. All materials used have some insulating effect, which is undesirable because it causes a greater heat lag and a greater heat loss downward. The problem was how to set up a test that would show up the advantages of the different types of coverings on the market.

As a solution it was decided to build a small insulated test panel as shown in figure 19. Water was used as a medium of heat exchange between the heating element and the panel. A small circulating pump was installed to circulate the water. However, preliminary tests showed that the circulator added an unmeasurable amount of heat to the water being pumped, so could not be used. Experimentation with the cover of the test panel removed showed a considerable amount of convective circulation caused by the heating element. Further tests using thermometers, see figure 32, on the test panel surface showed that the surface temperatures were within 1° F. of uniformity so no other means of circulation was employed. The temperature of the water in the panel was controlled by a thermostatic switch, sensitive to plus of minus 1° F. Heat was supplied to the panel by a 100 watt heating element. The amount of

heat supplied was measured by a Watt hour meter, in Watt hours which was converted to Btu, see figure 33.

The primary purpose of this test panel is its use as a comparison device rather than an accurate model showing exact heat output for a unit area of surface. As a comparison device it is possible to eliminate the high degree of accuracy that would be required to prove the theoretical performance of a radiant panel. Some of the possible errors are the use of glass thermometers and surface type pyrometer accurate to plus or minus 1° F.; the impossibility of controlling the room temperature better than plus or minus 2° F.; a mean effective temperature that varied from plus to minus 2° F., and a Watt hour meter accurate to .25 percent.

In order to have a good comparison of heat restriction or heat emission between two types of flooring the laboratory conditions should be the same. Since this was impossible, each sample was tested for four days, then the results of each test were plotted with water and surface temperature versus Btu transmitted as shown in figures 20 to 31.

By using these curves a comparison of the heat emitted can be made at any temperature within the usual floor panel surface temperature range. The insulating effect of the floor coverings can be compared at a chosen heat output by subtracting the surface temperature from the water tempera-

ture at that output value.

The data used for plotting these curves is given in Table IV, comparison results are given in Table V for a surface temperature of 90° and a heat output of 45 Btu per sq ft hr.

No. 1, the 4" glass wool covering, is not a sample and was used only to determine the edge and bottom losses as plotted in figure 20.

The guide for Table IV is as follows:

Column A Sample Number

Arbitrary numbers were assigned to each sample for the test; also given in this column is a description of the sample.

Column B Watt Hour Meter Reading

This is the Watt hour consumption of the 100 watt heating element for each test.

Column C Test Length Hours

Each sample was tested for about one day at a given temperature. Shorter tests were tried, but results were not consistent for the first six hours. This was probably caused by the variable heat input to the insulation around the panel until an equilibrium was established.

Column D Heat Input Btu/hr

Is the Watt hour input "B" divided by the length of test "C" times the conversion factor 3.1413 Btu per Watt hr.

Column E Panel Output Sides and Bottom

Is the heat loss through the bottom and sides of the panel as determined from tests and picked off curve, figure 20 for each water temperature.

Column F Panel Output Top

Is the difference between heat input "D" and the losses "E".

Column G Panel Output Btu/sq ft hr

Is the output from the panel top "F" divided by the exposed panel area 3 sq ft.

Column H Water Temperature ° F.

Is the temperature of the water inside the test panel as measured by a thermometer mounted in the panel. This temperature was recorded throughout the day at varied intervals. The temperature was maintained by a thermo-switch.

Column I Panel Surface Temperature ° F.

The panel surface temperature was measured by 4 glass thermometers fixed to the sample's surface with a small piece of modeling clay. The average of the four readings was recorded for each test.

Column J Room Temperature ° F.

The room temperature was recorded for each 24 hour period by recording thermometer mounted at the panel level. The test room was heated by a steam radiator controlled by a thermostatic radiator valve.

Table IV Floor Test Panel Data

A	B	C	D
Description	Meter Watts per hour	Test Length hours	Heat Input Btu/hr
No.(1) Fig. 20	320	23.33	47.
4" glass	339	24.00	48.
wool cover	380	23.50	56.
	450	17.75	86.
No.(2) Fig. 21			
Plastic coated	725	20.04	112
cork	881	22.17	136
	1448	23.58	209
No.(3) Fig. 22	713	16.50	147
Asphalt tile	543	16.00	115
	2202	24.40	302
No.(4) Fig. 23	2038	24.00	288
Inlaid linoleum	1485	22.30	198
(gray)	1052	24.30	148
No.(5) Fig. 24	1041	23.80	149
Felt back	1271	22.80	180
Inlaid linoleum	1550	20.90	252
No.(6) Fig. 25	1500	20.30	252
Grease proof	1240	25.40	166
Asphalt tile	690	20.50	115
No.(7) Fig. 26	1466	21.20	236
Black	977	23.30	143
Inlaid linoleum	872	24.70	120
	1870	51.45	123
No.(8) Fig. 27	1365	22.00	212
Sandran plastic	1250	23.50	179
coated linoleum	970	23.75	134
No.(9) Fig. 28	1925	23.75	276
Flexa-chrome	230	5.33	173
plastic tile	990	24.42	138
No.(10) Fig.29	1660	21.08	265
Rubber tile	3250	48.75	227
	935	18.83	169
	1530	21.83	242

Table IV Floor Test Panel Data (cont'd)

A	E	F	G
Description	Panel output side & bottom Btu/hr	Panel output top Btu/hr	Panel output Btu/sq ft hr
No.(1) Fig.20 4" glass wool cover	29. 30. 35. 52.	18. 18. 22. 34.	6. 6. 7. 11.
No.(2) Fig.21 Plastic coated cork	16 25 47	96 119 162	32 37 54
No.(3) Fig.22 Asphalt tile	29 23 54	118 92 248	39 31 83
No.(4) Fig.23 Inlaid linoleum (gray)	55 38 20	233 160 128	78 53 43
No.(5) Fig.24 Felt back Inlaid linoleum	21 34 54	118 146 198	49 49 66
No.(6) Fig.25 Grease proof Asphalt tile	51 36 18	201 130 97	67 43 32
No.(7) Fig.26 Black Inlaid linoleum	45 20 15 15	191 123 105 108	64 41 35 36
No.(8) Fig.27 Sandran plastic coated linoleum	27 22 11	185 157 123	62 52 41
No.(9) Fig.28 Flexa-chrome Plastic tile	50 30 18	226 143 120	75 47 40
No.(10) Fig.29 Rubber tile	39 31 21 35	226 196 148 207	75 65 49 69

Table IV Floor Test Panel Data (cont'd)

A	H	I	J
Description	Water Temp. ° F.	Panel Surface Temp. ° F.	Room Temp. ° F.
No.(1) Fig.20 4" glass wool cover	99° 100° 102° 112°		70° 70° 70° 70°
No.(2) Fig.21 Plastic coated cork	92° 97° 109°	86° 97° 99°	69° 71° 71°
No.(3) Fig.22 Asphalt tile	99° 96° 113°	95° 92° 107°	71° 71° 71°
No.(4) Fig.23 Inlaid linoleum (gray)	113° 102° 97°	106° 96° 91°	71° 71° 70°
No.(5) Fig.24 Felt back Inlaid linoleum	96° 102° 111°	90° 96° 105°	70° 71° 71°
No.(6) Fig.25 Grease proof Asphalt tile	113° 102° 93°	108° 97° 89°	71° 71° 71°
No.(7) Fig.26 Black Inlaid linoleum	108° 94° 91° 91°	102° 90° 87° 87°	70° 70° 70° 70°
No.(8) Fig.27 Sandran plastic coated linoleum	98° 95° 89°	95° 91° 85°	70° 70° 71°
No.(9) Fig.28 Flexa-chrome Plastic tile	112° 99° 92°	108° 94° 88°	69° 70° 70°
No.(10) Fig.29 Rubber tile	106° 101° 94°	100° 96° 90°	68° 70° 70°

Table IV Floor Test Panel Data (cont'd)

A	B	C	D
Description	Meter Watts per hour	Test Length hours	Heat Input Btu/hr
No.(11)Fig.30	1320	20.66	218
Rug without	1150	20.00	196
pad	1090	22.33	167
No.(12)Fig.31	1250	23.42	182
Rug with pad	770	19.25	136
	730	23.00	109

A	E	F	G
Description	Panel output side & bottom Btu/hr	Panel output top Btu/hr	Panel output Btu/sq ft hr
No.(11)Fig.30	55	163	54
Rug without	50	146	48
pad	37	130	43
No.(12)Fig.31	59	123	41
Rug with pad	40	96	32
	32	77	26

A	H	I	J
Description	Water Temp. ° F.	Panel Surface Temp. ° F.	Room Temp. ° F.
No.(11)Fig.30	113°	92°	70°
Rug without	110°	88°	69°
pad	104°	87°	70°
No.(12)Fig.31	115°	87°	69°
Rug with pad	106°	82°	70°
	101°	80°	69°

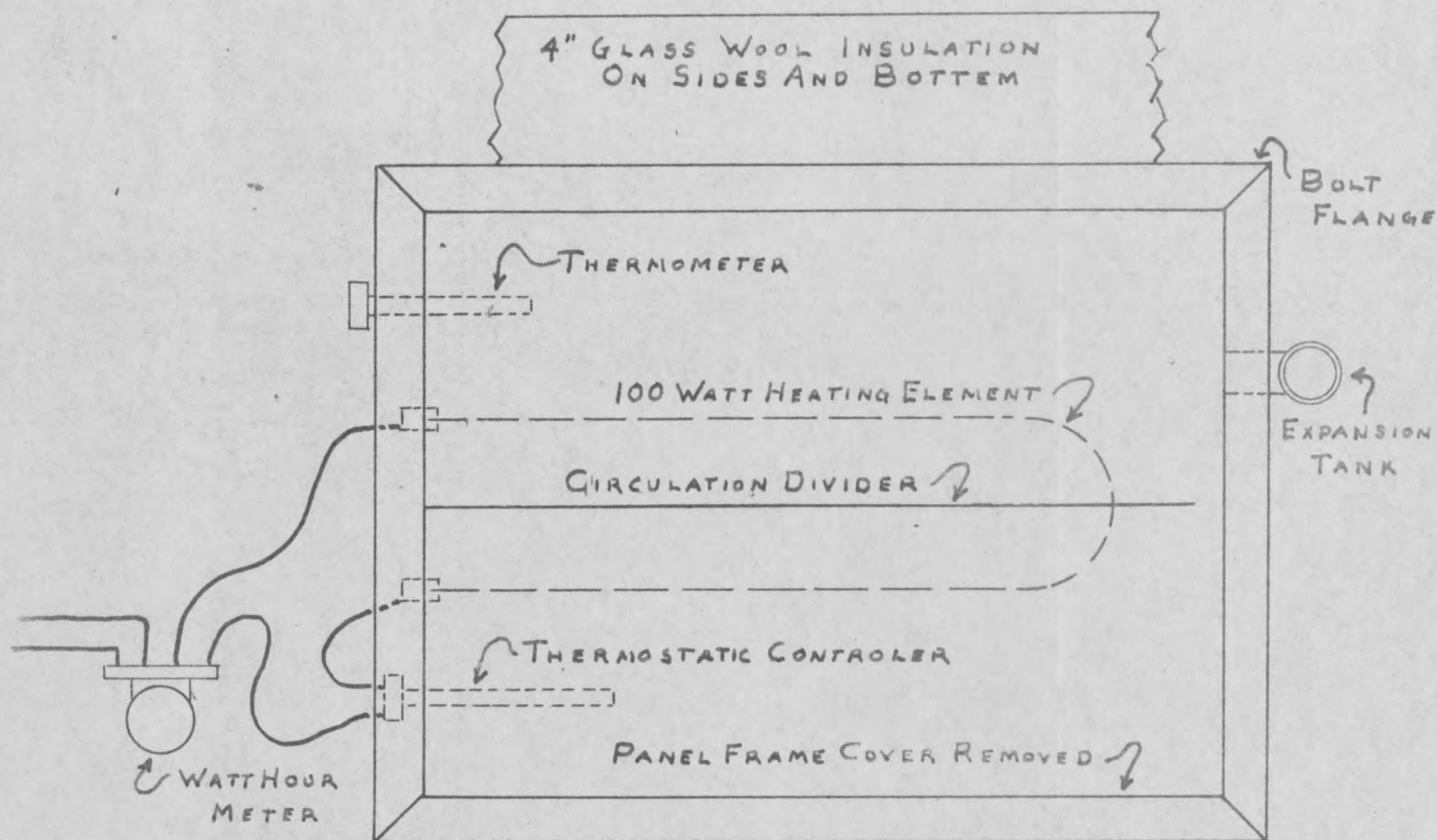


FIG 19-FLOOR COVERING TEST PANEL

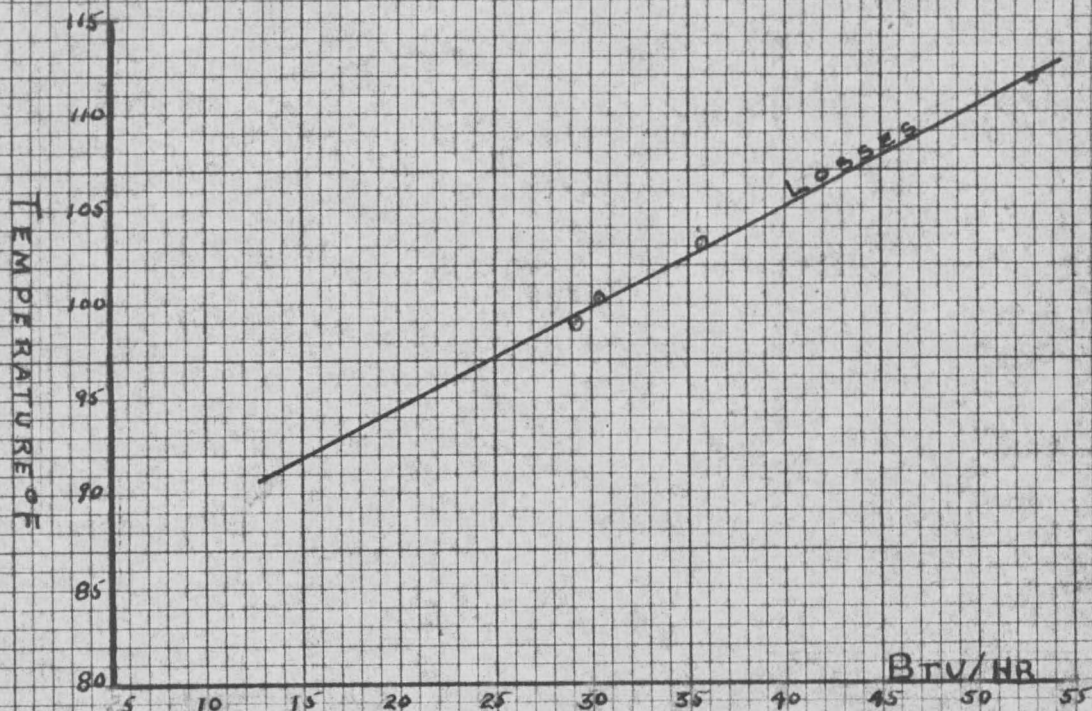


FIG 20 PANEL BOTTEM ANDEGE LOSS

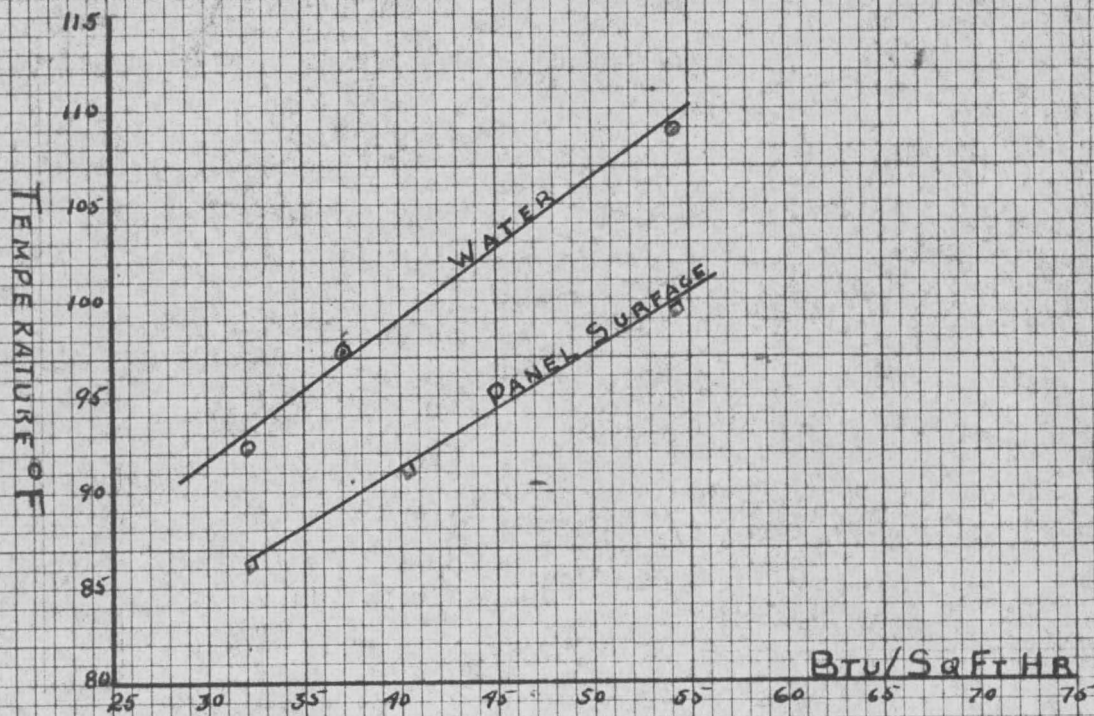


FIG 21 PLASTIC COATED CORK

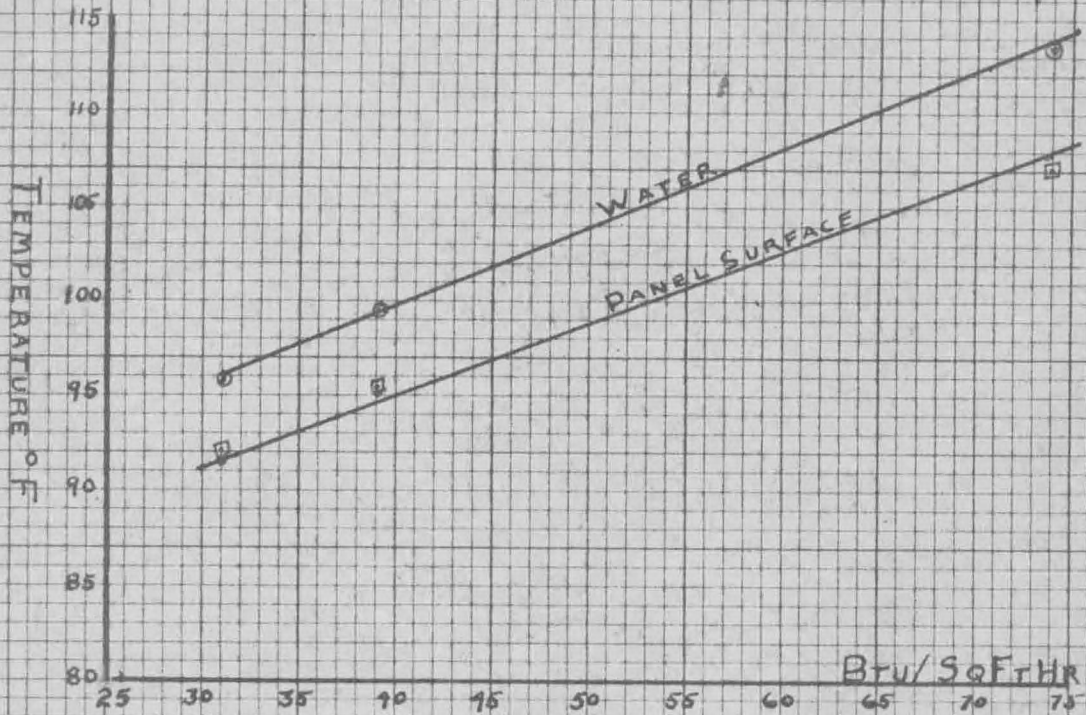


FIG 22 ASPHALT TILE

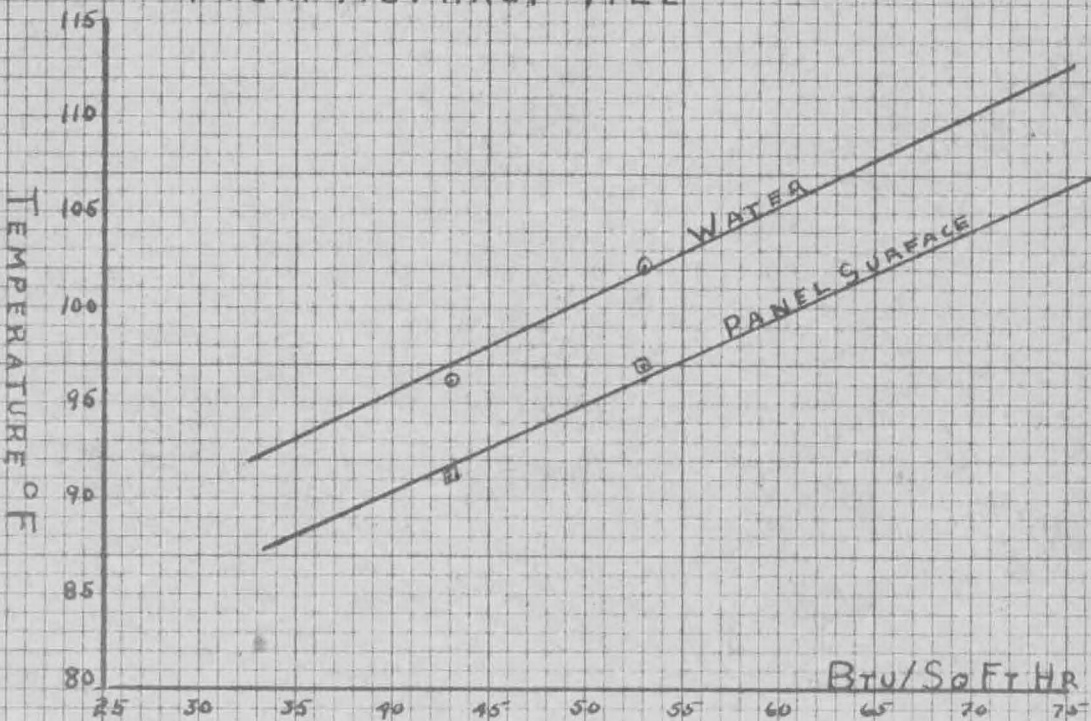


FIG 23 INLAID LINOLEUM GRAY

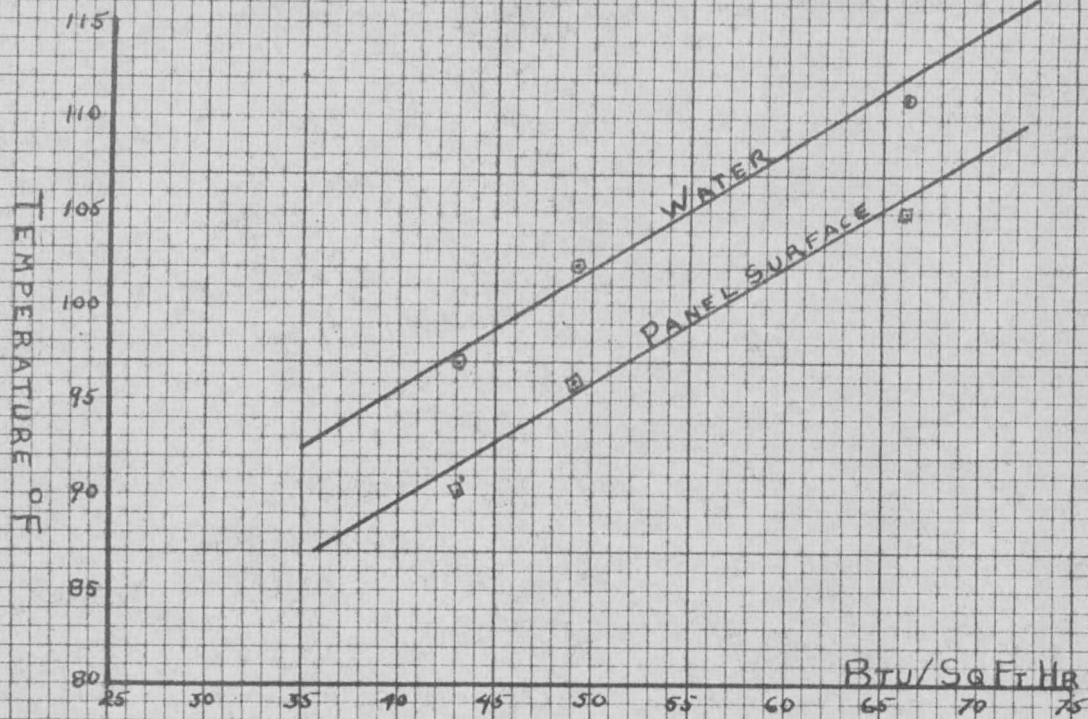


FIG 24 FELT BACK INLAYED

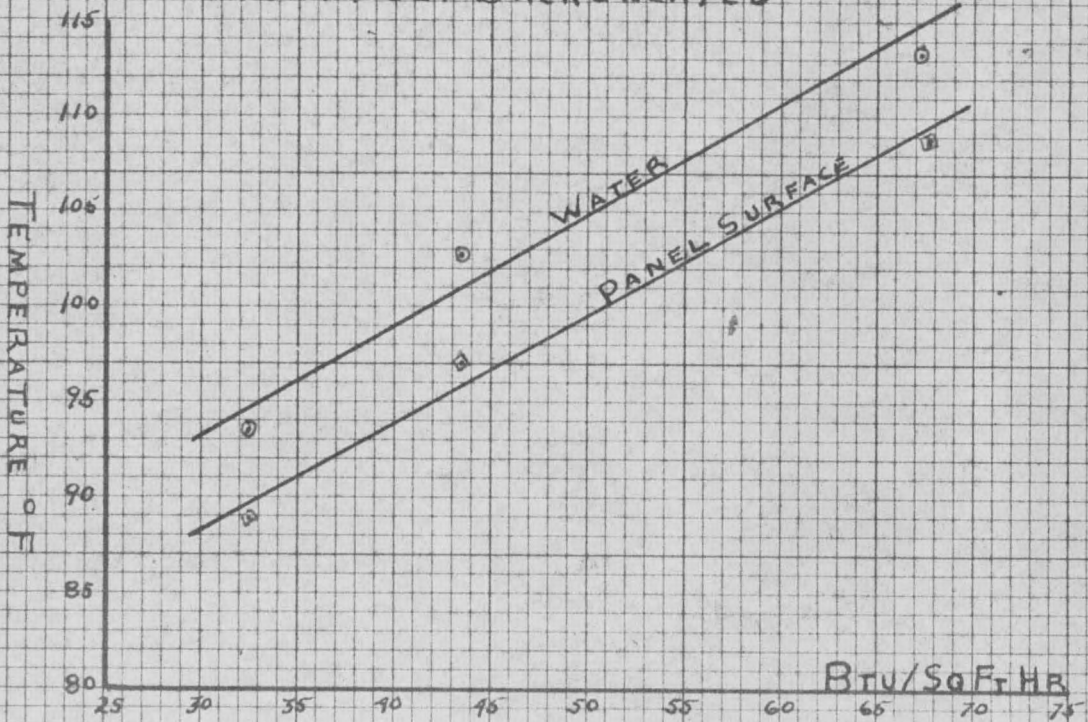


FIG 25 GREASE-PROOF TILE

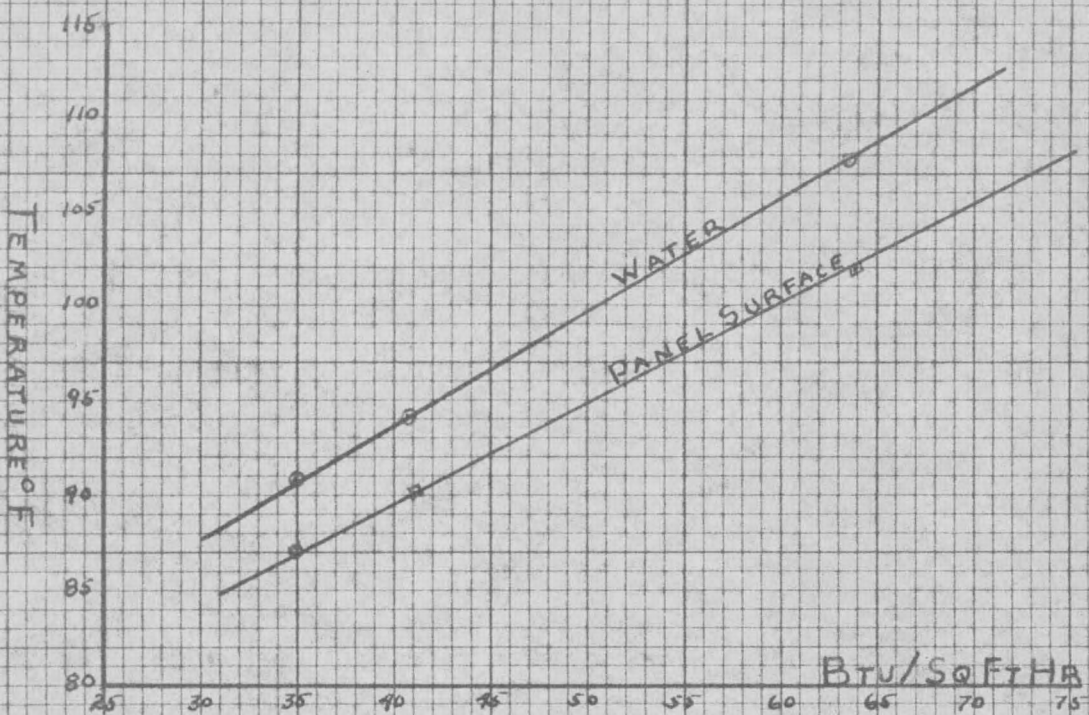


FIG 26 BLACK INLAYED LINOLEUM

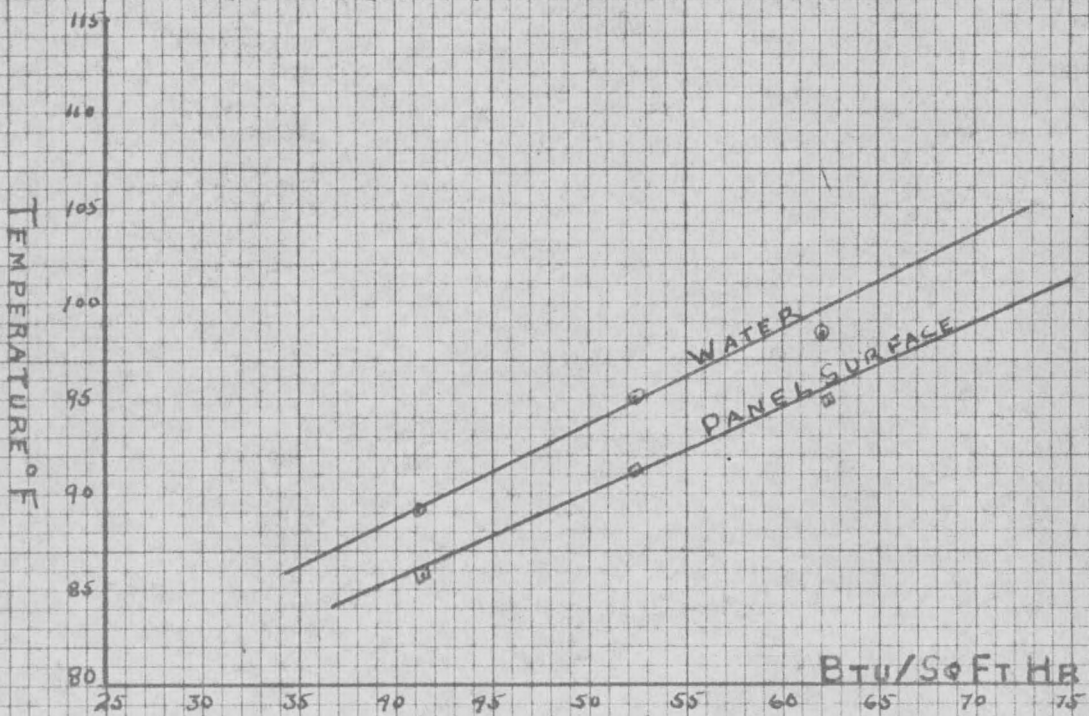


FIG 27 SANDRAN LINOLEUM

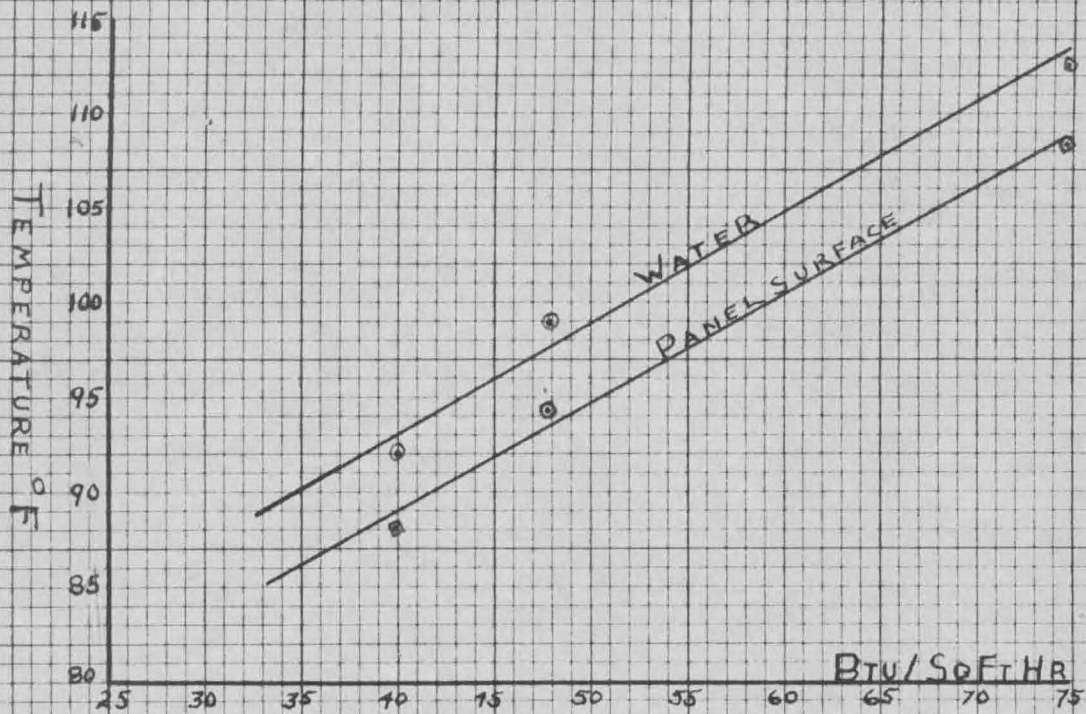


FIG 28 FLEXACHROME PLASTIC TILE

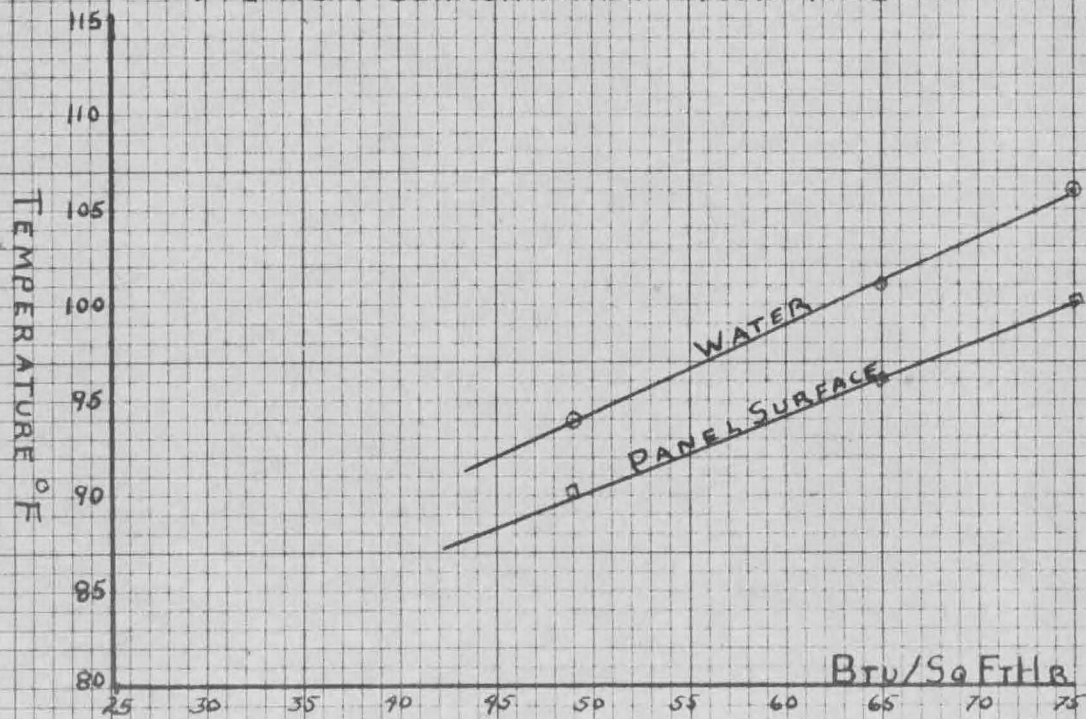


FIG 29 RUBBER TILE

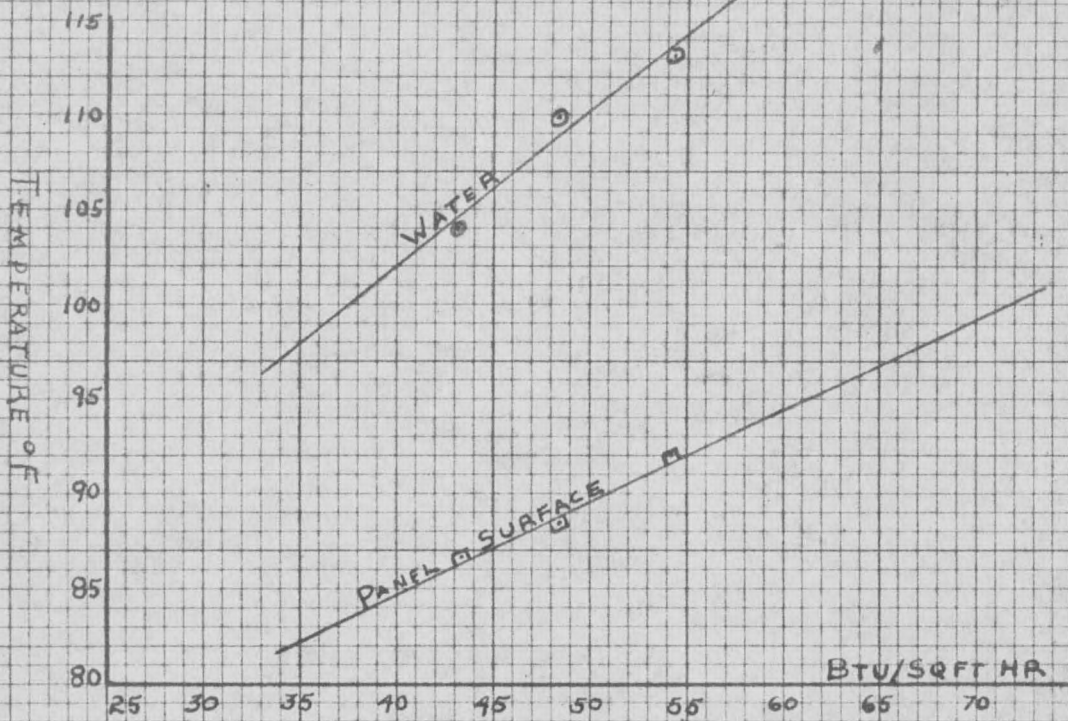


FIG 30 RUG WITHOUT PAD

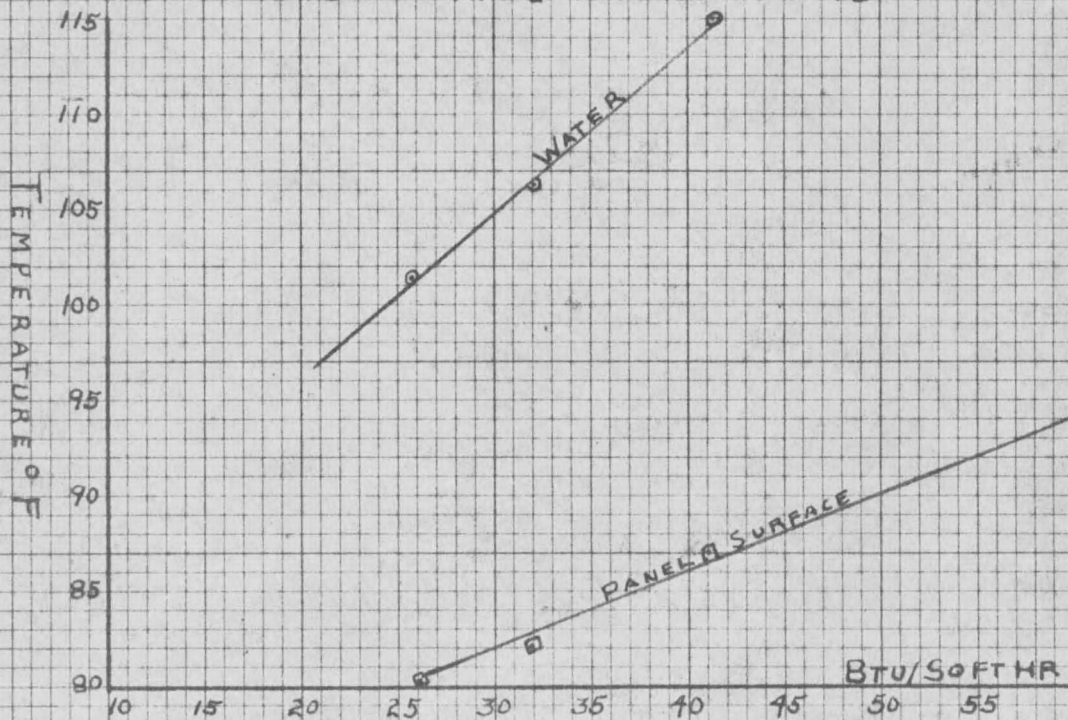


FIG 31 RUG WITH PAD

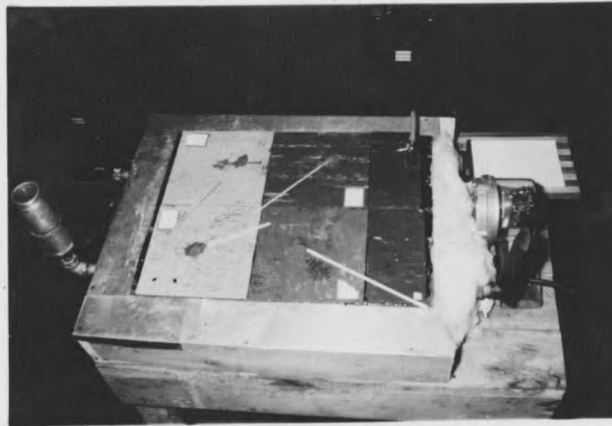


Figure 32 Top View Floor Covering Test Panel

The glass thermometers used to measure surface temperatures are visible also the air vent (top right) and expansion tank, lower left. The 4" of glass wool is covered with sheet aluminum.



Figure 33 End View Floor Covering Test Panel

At the left is an immersion type thermometer, Watt hour meter, center, and thermostatic switch at the right. The heating element leads, not visible, are located in the center of the panel. The surface type pyrometer used to check the thermometers is not shown.

Table V Comparison of Floor Covering Samples

Column A			Column B		
Sample Description	Output with 95° Surface Temp. Btu/sq ft hr		Sample Description	Temp. drop, water to exposed sample surface °F at an output of 45 Btu/sq ft hr.	
Fig. No.			Fig. No.		
31 12	62		27 8		3
29 10	62		29 10		4
30 11	61.1		28 9		4
27 8	61		26 7		4.5
28 9	51		22 3		5
26 7	50		23 4		5.5
23 4	50		25 6		5.5
24 5	49		24 5		6
21 2	46		21 2		9
25 6	42		30 11		19
22 3	40		31 12		30

Discussion of Floor Covering Test Results

Column A, Table V compares the outputs of the samples at a surface temperature of 95°F. by listing them in the order of their emissive value. The effect of a rough surface on radiation is clearly shown by the output of the rug sample 11 and 12. The general trend of the output for samples 2 through 10 shows that the best output is for a soft covering like

rubber, sample 10, and diminishes to the lowest output for sample 3, a hard asphalt tile.

Column B, Table V shows the insulating effect of the various floor covering samples. The temperature drop through a floor covering is important in cases where the water temperature can not be increased at the boiler and some other method of increasing the panel output must be found. Sample 12, which is a rug pad, showed a drop of 30° , while the temperature drop through sample 8, a plastic coated linoleum, was 3° . If a panel is so constructed that it requires 180° supply water for a surface temperature of 90° , it is evident that a 30° drop could not be allowed and one of the more conductive coverings must be chosen. In most cases where the supply water is between 130° and 150° the choice of a floor covering is not a problem. In cases where one coil serves two rooms and it is desired to cover the panel in one of these rooms with a rug, the insulating effect of the rug must be considered. If a rug is used in this case it will be found that there is very little heat output from it, while the other part of the panel has a high output.

The effect of a floor covering should be considered in the design of a system. One of the best ways to provide for use of any floor covering is to heat each room on a separate circuit equipped with balancing valves. If this is done the

boiler can be adjusted to supply water hot enough for the rooms is throttled to the point where it gives the required surface temperature.

The choice of a floor covering on the basis of this test would be one that provides the highest output for a given surface temperature, Column A Table V. Rubber tile is the best choice of the hard finish coverings on the basis of a high output and a low temperature drop.

A rug and pad are good on the basis of a high output and cushioning effect for high temperatures, but must be provided for in the panel design.

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Radiant heat design and testing.

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