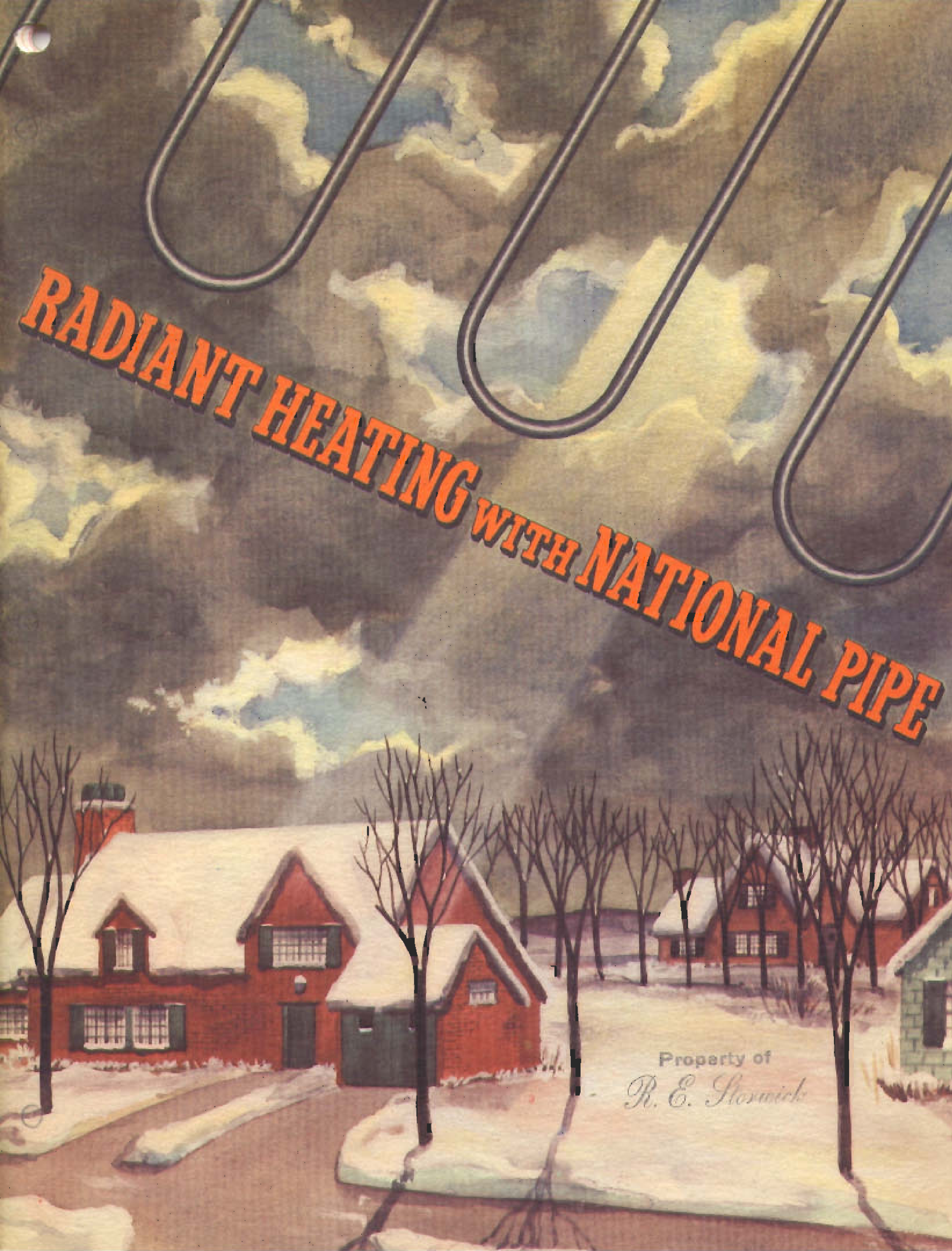
RADIANT HEATING WITH NATIONAL PIPE



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R. E. Storwick

RADIANT HEATING

WITH NATIONAL PIPE

BULLETIN Nº 19

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UNITED STATES STEEL

Radiant Heating

PROBABLY NO INNOVATION in the building industry within recent years has created such broad and enthusiastic interest as has the subject of radiant or panel heating. Certainly it has set in motion a series of technical discussions and popular articles in the business and daily press which have given it near, if not top ranking, in news of the building industry. Architects, engineers, contractors, and prospective home owners alike have shown a keen interest in the possibilities of applying this much publicized method of heating in building plans now in hand or under consideration.

Radiant heating, however, is not a new or an untried idea. For over a decade it was successfully used in Europe before it came into prominence in the United States. Contrary to some popular opinion, there is really nothing mysterious or spectacular about a radiant system. Fundamentally, it is simply another method of using the long established mediums of either hot water or steam for heating purposes. It differs from the conventional systems mainly in that instead of using one or more radiators in a room, pipe coils are placed in the ceiling, floor, or walls to warm the surfaces and thus bring the temperature to a comfortable degree for the occupants.

Generally speaking, radiant heating is adaptable wherever conventional systems have been employed, i.e., in homes, schools, churches, office and public buildings, while certain special advantages are emphasized by its proponents for industrial buildings, shops, garages, and large areas where maintenance of uniform temperature is desirable, as well as minimum loss of usable space. Another large scale use of radiant systems is for the removal and prevention of snow and ice from airport runways, driveways, sidewalks, etc.

Both in this country and abroad some of the radiant heating installations have been made in quite large projects—for example, the Royal Liver Building in Liverpool. This sizable building contains one thousand rooms and the radiant heating system is said to be both efficient and economical in operation.

With the resumption of an active building program in this country advocates of radiant heating confidently predict this modern method will be extensively employed in both large and small structures of many different types. The home building program especially has already stimulated a wide and growing interest in the subject and will continue to receive considerable attention in numerous articles in various publications and in forum discussions in association or society meetings.

National Tube Company has been fully cognizant of the growing interest in this type of heating. Extensive study and investigation of all phases of the piping service have been and will continue to be made to the end that prospective users of radiant heating systems may have the benefit of the widest and most extensive experience available in all kinds of pipe problems, including radiant heating. The actual design of radiant systems, however, is properly the function of professional heating engineers. Their knowledge and experience in heating problems in general can be readily applied to any contemplated radiant installation. In matters pertaining to the piping proper, National Tube Company will gladly extend the fullest possible cooperation without charge or obligation.

Photographs of radiant heating and snow melting installations shown herein are for the purpose of illustrating the wide application of these systems in different types of structures throughout the country, and are not intended as an endorsement of the design, layout, or method of installation. In some instances photographs were taken at incompleted stages of the work, and may not represent the finished or actual practice followed in the final work done.

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What Is Radiant Heat?

RADIANT ENERGY

No better example of the phenomenon of radiation can be found than that of the sun in its energizing and heating of the earth upon which we live. It imparts to all living things radiant energy in the many forms necessary to a continuance of life.

The manner in which this energy is radiated from the sun is not definitely known, but it is believed that it reaches us through the medium of the ether, and in the form of wave motions of varying intensity and length. Many of us have thrown a stone into a quiet pool of water, and fascinated, we watched the first energy wave motion develop in the form of an ever widening circle, to be followed by many others at regular intervals. While wave motion from the sun is not apparent to the naked eye, the manner in which energy is transmitted by the sun is similar to the wave motion of the water. We must understand, however, that the energy radiated to us from the sun takes many forms, and each of these forms is identified by its wave length, which, to go back to our analogy, would be the distance from one circular wave to the one following. Energy waves of one length produce light with resultant color effects, as seen in a rainbow following a shower. This bow of light has all the colors of a spectrum, brought about by dispersion and diffraction of light waves in the drops of water falling through the air.

Some short energy waves possess that property which can effect chemical changes, as evidenced by that coat of tan which is sometimes sought after during summer months; and in a less agreeable way, by the familiar souring of milk left too long on the kitchen doorstep. Energy of the longer wave lengths is utilized in connection with electronics, radio transmission, and radar.

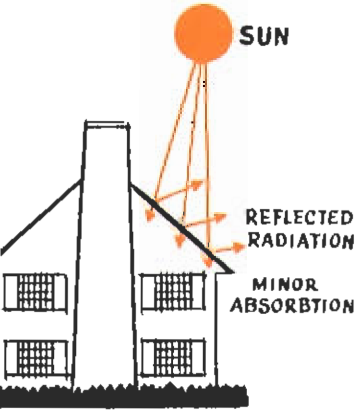
Many other interesting effects are produced by radiation of energy from the sun but for the purpose of this discussion we are concerned only with that form and wave length of energy which is converted into heat upon contact with the earth's surface.

In discussing radiant heat and its manifestations, we must first dismiss from our minds the thought that either the open air around us, or that contained in a house or building, is heated by radiant energy waves. The air is not directly heated in this way, but does, in passing over surfaces which have absorbed radiant energy and converted it into heat, carry off a small part by convection.

HOW DOES RADIANT ENERGY BECOME HEAT?

Radiant heat energy waves strike all objects in their paths, and are either absorbed, or reflected to other objects.

If we were able to see with the eye of a microscope powerful enough and were to watch the effect of the impact between the waves of radiant energy transmitted from either the sun or a radiant heat panel on the plaster or woodwork in a room, we would witness an extremely interesting occurrence. We would observe that the infinitesimally small particles of plaster or wood called "molecules" were in a highly agitated state, vibrating in all directions at terrific speed, and striking one another an enormous number of blows per second. This has come about as a result of impact of the radiant energy waves on those molecules nearest the surface of the plaster or wood, and the impact has been transmitted in chain fashion to other molecules throughout the material. This impact of the molecules, one upon another, creates heat in the plaster or wood just as effectively as does the impact of a steel hammer on some steel object.



An excellent example of the heat producing energy transmitted by radiating waves from the sun is found in the simple experiment of holding a magnifying glass over a piece of paper at a distance which permits the parallel energy transmitting rays of the sun to enter one side of the convex glass and converge upon the paper on the opposite side at the focal point, Figure 1. The concentration of radiant energy is sufficient to heat the paper until it ignites and burns.

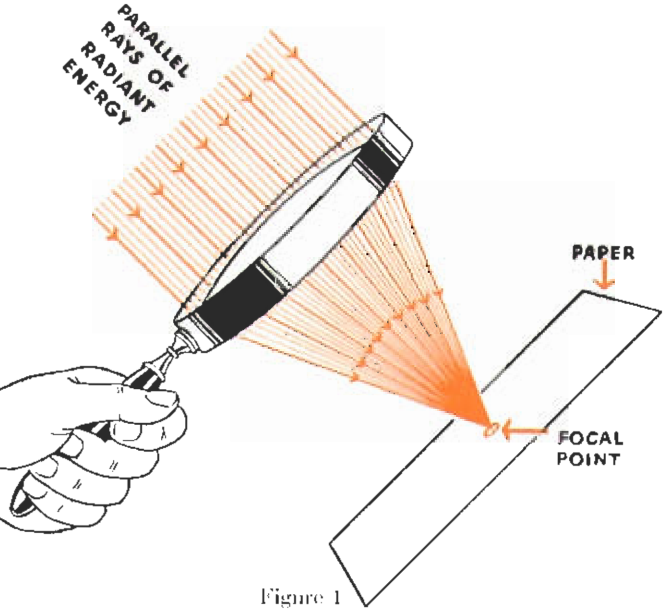


Figure 1

The energy or heat-creating waves from the sun, or from any other heating surface, are not all absorbed through the surfaces of the bodies upon which they impinge. Some strike bright metallic objects and the greater proportion of these are immediately reflected without being absorbed. Others reaching dull, dark-colored, lusterless substances are absorbed to a much greater extent. Thus, a substance which reflects heat well is a poor absorber of heat.

BRITISH THERMAL UNIT (B.T.U.) THE MEASURE OF HEAT

The scientist, Dr. Joule, determined the relation between heat and energy by recourse to a well known experiment, Figure 2. He suspended a weight, which in its free descent rotated the drum and paddles, causing a churning action in the water.

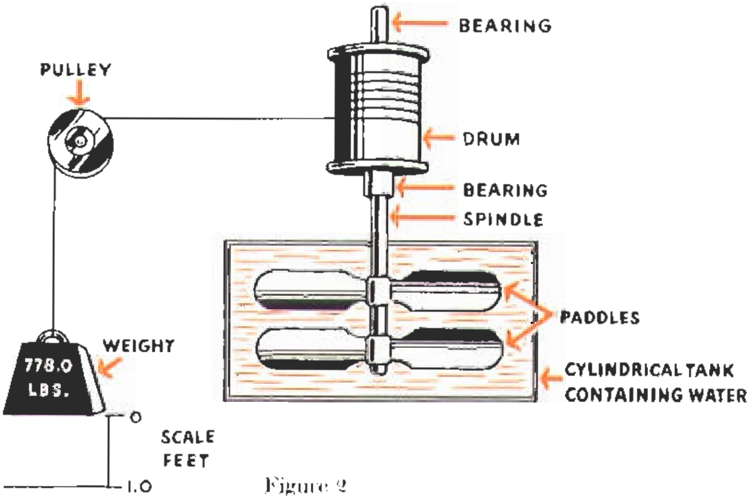


Figure 2

By this experiment Dr. Joule determined that when the weight of 778 pounds had fallen one foot, representing 778 foot-pounds of work done or ENERGY EXPENDED, the temperature of the water had increased by one degree Fahrenheit. He thus related heat, energy, and motion.

The result of this experiment is the basis for the "measure" of heat which is described as the British Thermal Unit (B.T.U.) and defined as "the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit."

HEAT-ENERGY RADIATION, EVERYDAY OCCURRENCE

Many of us are familiar with the popular demonstration of the effect of the sun's radiant energy found in the illustrations of many of our magazines. These depict skiing enthusiasts enjoying this winter sport clad only in bathing suits, with the air temperature near freezing point. They are quite comfortable. This is possible because the velocity of the air is low, and the amount of heat carried off from the body by air convection currents is adequately offset by radiation of heat creating energy waves to the body from the sun. The snow aids considerably in creating this feeling of bodily comfort by reflecting much of this energy to the body surface of the skier.

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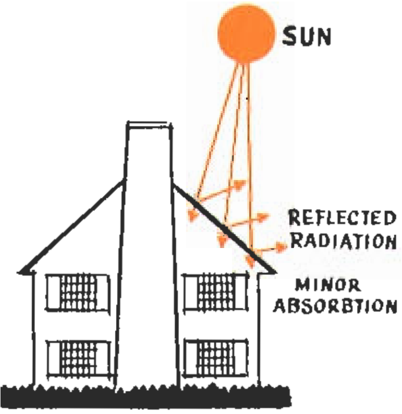

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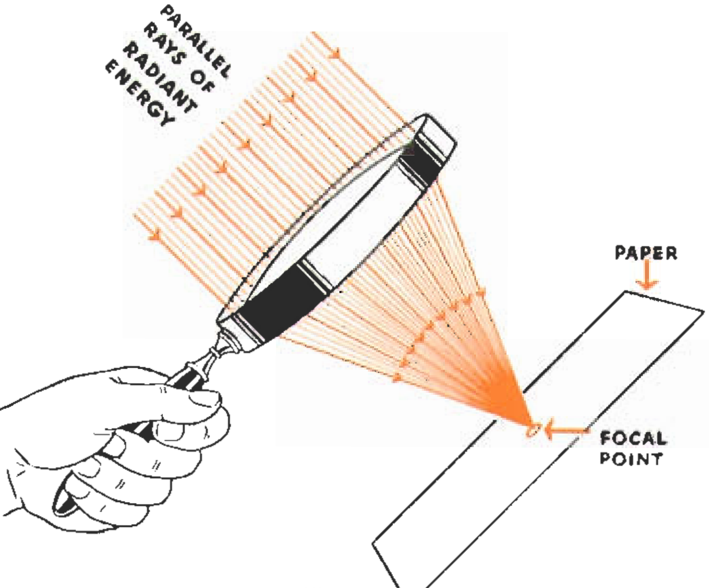

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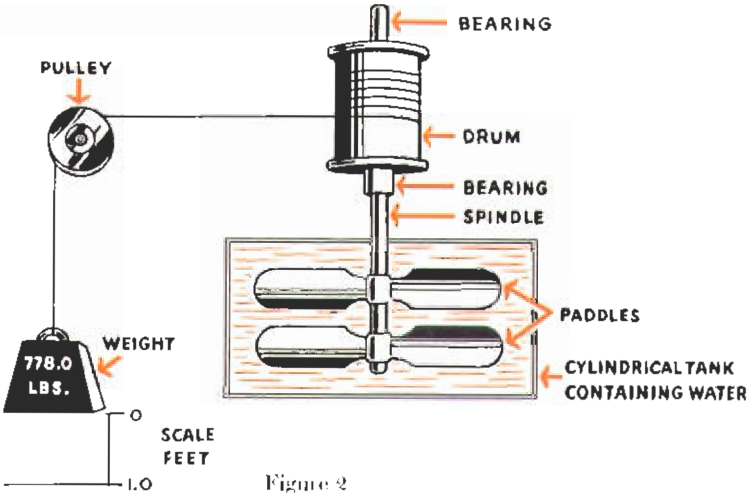

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Human Comfort, the Basis for Design of Radiant Heating

THE HUMAN BODY is a most remarkable heating unit, particularly when we consider the abuse to which it is subjected. We feed it fuel as we do a furnace in the form of food and drink. By a process called "metabolism" this fuel is converted into bodily energy and heat.

In the medical profession the heat measuring-unit is the "calorie," used to describe the heat and energy building value of various foods when absorbed by the human body, thus the term "calorific value of food."

Human beings require a constant replenishment of "calorie" or, similarly expressed, B.T.U. bearing foods to replace that heat or energy which is dissipated by physical exertion, by loss to surrounding bodies of lower temperature through the action of air convection currents passing over the body, and by evaporation. Science has established that the average human being, normally engaged in the daily activities, may lose heat at the rate of 400 B.T.U.'s per hour to surroundings at about 70° F. temperature, and in still air. Of this amount, 300 to 320 B.T.U.'s per hour may be lost by radiation and convection from the body surfaces, with the remainder being lost by evaporation of moisture from the lungs and the body surface, Figure 3.

Skiing enthusiasts enjoy this winter sport clad in light attire with the air temperature near freezing point. Comfort is enjoyed because of radiation of heat creating energy waves to the body from the sun. The snow aids considerably in creating this feeling of bodily comfort by reflecting much of this energy to the body surface of the skier.

We have observed by the illustration of the skier in a bathing suit or light attire how it is possible for human beings to be comfortable, provided the heat lost by convection at low temperature is balanced by the radiation of heat to the body at higher temperature. Similarly, in heating a room, we may obtain a condition of comfort by a proper balance of wall surface temperature and room air temperature.

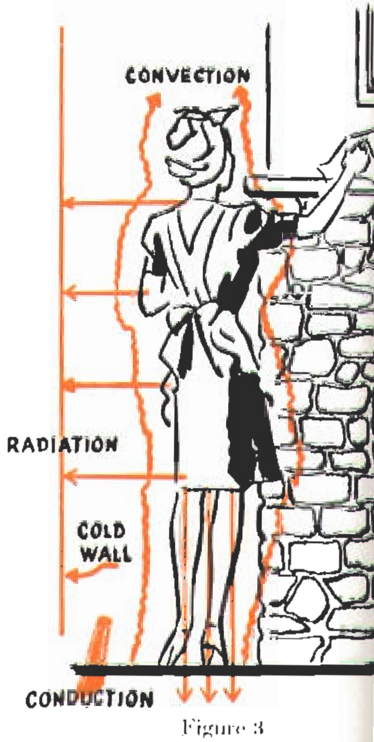

Figure 3

Dr. Yaglou, as a result of an investigation conducted at Harvard University with three male adults clothed in three-piece suits and at rest during the tests, found that conditions of comfort were obtained with the following three sets of temperatures* for which the bodily heat loss by radiation and convection is given:

Mean Radiant Temperature (M.R.T.) Deg. F.	Air Temperature Deg. F.	Heat Loss Radiation & Convection B.T.U./Hour
71	71	291
79	63	321
85	59	303

*Courtesy "Heating, Piping and Air Conditioning" and F. E. Giesecke, Consulting Engineer.

It is seen from the table that we can obtain a condition of bodily comfort with several combinations of room surface temperature and room air temperature. It is necessary only that the heat carried off by air convection currents be adequately offset by reduction in the amount of heat lost by radiation to the room surfaces and thence by conduction through the building materials.

To do this we must heat the room walls, windows, doors, and floor, to a mean surface temperature such that these surfaces will reradiate to the body the greater part of the heat which it gives off by radiation.

The surface temperature of the average person varies between 80° F. and 83° F. If, then, the mean surface temperature of the room, or as it is technically termed, the mean radiant temperature, is established for design purposes at 75° Fahrenheit, the heat loss from the human body to these same surfaces by radiation will be small.

The chart,* Figure 4A, shows that approximately 50 B.T.U.'s per hour are lost from the body by radiation, when the average body surface temperature is assumed to be 80° Fahrenheit. This leaves (320-50) = 270 B.T.U.'s per hour, which we are permitted to lose by convection before a feeling of discomfort is felt.

Referring to Figure 4B we find that the equivalent air temperature for this heat loss is 65° Fahrenheit, and that theoretically the air temperature may reach this value before discomfort is felt. In actual practice however, as in the heating of a home, the air temperature and the mean radiant temperature may approach equilibrium, due to contact of the air with the heating surface, and we are more likely to obtain an air temperature differing by only a few degrees or so from the mean radiant temperature. For a comfort temperature of 70° F. we may expect the mean radiant temperature to be 72° F. and the air temperature 68° F.

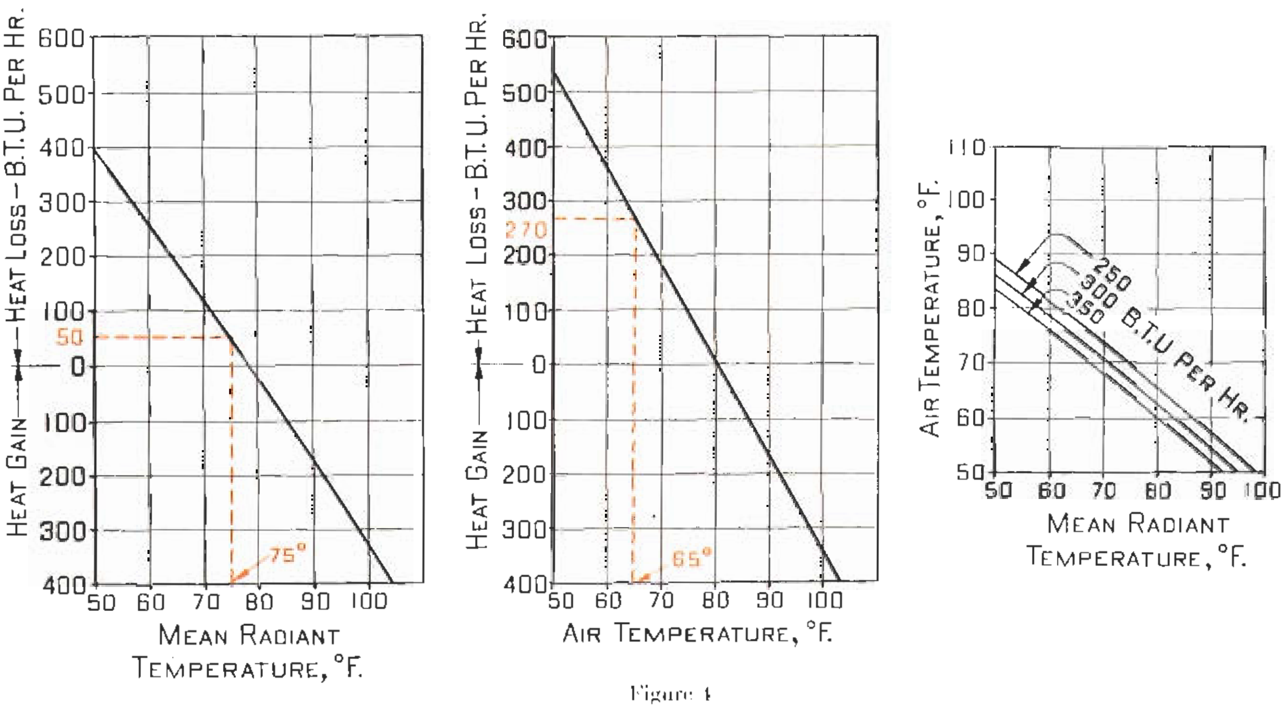

Figure 4

A—Heat loss or gain by radiation for various mean radiant temperatures, in Btu per hr. per person, according to the formula $Q = 15 \times 0.156 [(540/100)^4 - (T_r/100)^4]$, where 15 sq. ft. is the area of the body available for radiation, 0.156 is the radiation factor for the body, 540 is the absolute mean surface temperature of body (460 + 80), T_r is the absolute mean radiant temperature of the room, and Q is amount of heat, Btu per hr. per person.

B—Heat loss or gain by convection for various air temperatures, in Btu per hr. per person, according to the formula $Q = 18 \times 1 (80 - t_a)$, where Q is Btu per hr. per person, 18 sq. ft. is the area of the body exposed to convection air currents, 1 Btu per hr. per sq. ft. per ° F. temperature difference is the film coefficient, 80° F. is the mean surface temperature of the body, and t_a is the temperature of the air in the room in degrees Fahrenheit.

C—Relationship between air temperature and mean radiant temperature for combined convection and radiation heat losses of 250, 300, or 350 Btu per hr. per person, as indicated on curves.

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Some Temperature Studies in Radiant Heated Rooms†

HAVING CONSIDERED THE VARIOUS POINTS relating to heat requirements and how the heat can be applied, we are now in a better position to consider what are the requisites of a good system of heating and ventilating. Dr. Leonard Hill, and also the Industrial Fatigue Research Board of England, laid it down that a good system should provide an air temperature at the foot level equal to that at the head level, if not greater. There should be a fair degree of air movement and the air should not smell stuffy and unpleasant. This compares with nature's provisions for heat supply where we find that rays from the sun, with the long waves reflected from the earth and surrounding objects, warm the lower strata of moist air from our feet upward and give that ideal condition which our bodies require. The nearer we approach these conditions, the more closely we attain the ideal method of heating.

We have seen that, if an installation is to give the required degree of comfort and meet the physiological requirements of the body, a large percentage of the heat must be supplied as thermal radiations, and the relative humidity of the air should be maintained somewhere between 50% and 60%, preferably the latter.

Most of the heat should be given off by thermal radiations, with the remainder as convected heat.

Ordinary radiators may give off as little as 10% to 20% of their heat by thermal radiation and the remainder as convected heat, while with concealed heaters, unit heaters, and warm-air systems we get no thermal radiation whatever, except the secondary action from the furniture. Since all the objects in a room are at a lower temperature than the surrounding air, and consequently at a lower temperature than our bodies, we get no supply of energy from these sources.

Figure 5 illustrates diagrammatically the conditions we generally obtain with a concealed heater or a warm-air system. Assume that steam is turned on and a stream of warm air is introduced into the room from the grille or from the top of the concealed heaters. From concealed heaters the temperature of the air may be 130° F. to 150° F., although I have actually measured the air temperature leaving the grilles as high as 190° F.

With a warm-air system the inlet temperature may be as high as 180° F. to 200° F. This warm air is not only detrimental to the system, but, having passed over a high temperature surface, it has become polluted, for when the air passes over a surface at high temperature the dust is broken down chemically and ammoniacal vapors given off. With steam radiators we get similar results, but the air leaving

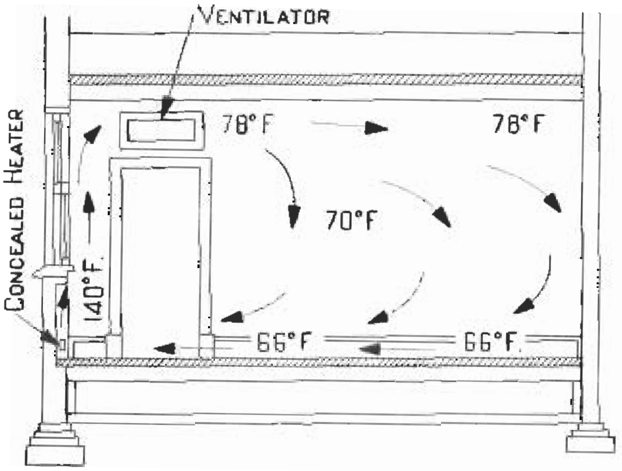

Figure 5. Temperature distribution in a room heated with a concealed heater or a warm-air system.

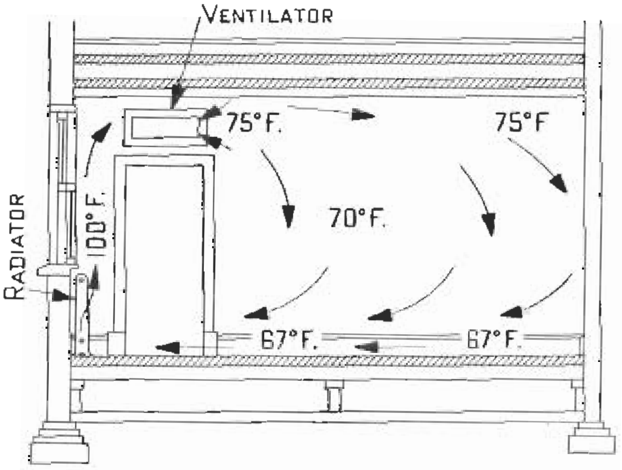

Figure 6. Normal temperature distribution with a steam radiator. Average observations in rooms of 9 to 10 ft. in height.

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†By T. Napier Adlam, Consulting Engineer, Member of Institution of Heating and Ventilating Engineers, Great Britain. Reprinted from "Heating and Ventilating" by courtesy of the publisher, and of the author, Mr. Adlam.

the top of the radiator will be 90° F. to 110° F., depending on the room temperature.

In either case the warm air naturally rises to the ceiling and will remain there, giving up part of its heat to the cold ceiling, and should there be any outlet at the high level the warmest air will escape before being of further use. As more warm air rises from the source of supply, the air at the ceiling, which does not escape, chills gradually. This continues until we get a series of layers at different temperatures. In other words, we have a temperature gradient from ceiling to floor.

The steepness of this temperature gradient will depend on the temperature of the air rising from the source, the heat loss from the room, and the quantity of air which is circulating.

TEMPERATURE DIFFERENCE OF 6° F. TO 14° F. NOT UNUSUAL WITH CONVECTED HEAT

When using steam pipes and radiators it is usual to get from 6° F. to 8° F. difference in temperature between the floor and the ceiling. With warm air or concealed heater I have found a difference of 10° F. to 14° F. to be quite common. Figure 6 illustrates diagrammatically the normal conditions met with in a room heated by steam radiators. These are average observations in rooms from 9 ft. to 10 ft. high. For higher rooms the temperature at the ceiling will be correspondingly higher.

Figure 7 shows average recorded temperatures at various heights for heating with radiators and convectors, and it will be clearly seen that the high temperature gradient means greater loss of heat.

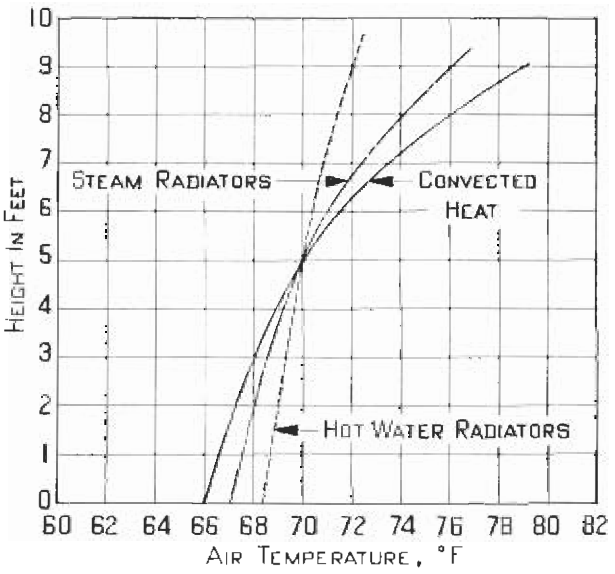

Figure 7. Average recorded temperatures at various heights for heating with radiators and convectors.

We find a warm stratum of air for breathing and for the head, with a cooler temperature for the feet, just the opposite to that required.

I have taken observations in many of the high buildings in this country and find that air is constantly passing through the ventilators over the doors into the corridors or passages at a temperature five or more degrees above the temperature of the air at the breathing line. This chimney effect in tall buildings is a bugbear to all heating engineers and architects because of the difficulty of being able to overcome its influence. The graduation of heating surfaces is only a partial remedy, for it does not hold good under all weather conditions.

Compare this with a building heated with thermal radiations where the air temperature at the high level varies only slightly from that at the breathing line. The result, as can be seen by Figure 11, would be an almost constant temperature throughout the building.

MEAN SURFACE TEMPERATURE DEPENDS ON CHARACTER OF SURFACE

It has been computed that if the mean surface temperature all over a room is about 60° F. the room feels warm, regardless of the air temperature. This, however, depends on the character of the surface. If the surface is covered with tinfoil or a highly polished metal surface, the results would be infinitely superior from a heating point of view than it would be if we had a dull black surface or even the usual papered surfaces.

Much has been written as to the best method of producing these heat rays, the relative virtues of ascending and descending rays, and the best temperature at which the surface should be maintained, but invariably it will be found that the respective advantages are illuminated according to the particular system advocated. Undoubtedly they all have advantages, and it is by correct discrimination that heating engineers can choose the best method for the particular problem in hand.

At present we will deal with each method diagrammatically, and later each system in use will be explained in detail. In Figures 8, 9, and 10 we have indicated a room heated by thermal radiations with rays emitted from a heated floor, ceiling, and walls respectively.

Figure 8 shows a heated floor of a room made with any material ordinarily used for flooring, except material which is likely to become plastic with heat, such as wood blocks bedded in pitch, etc. Wood, stone, marble, concrete, or other composite material may be used, and carpets may be laid on the floor without interference with the heat. In fact a carpet adds greatly to the comfort, for a heated floor covered with a thick carpet has given to the writer the best impression of real comfort of any heated room yet tried.

The required surface temperature of the floor varies with the kind of surface used and also with the exposure of the room. For instance, in testing out various materials I have found when trying white marble, which was to be used for the floor of Liverpool Cathedral, that with the polished white surface of the marble I required a surface temperature of 11° F. above the air temperature to give off a certain quantity of heat. I could obtain the same results with a surface temperature of 8° F. above the air temperature when the marble was covered with a thin coating of lamp black.

HIGHER TEMPERATURES DESIRED IN THIS COUNTRY THAN IN ENGLAND

I should explain that in England it is found, generally speaking, that while 60° F. is a suitable air temperature with radiators and pipes, an equal feeling of comfort is obtainable at 56° F. to 58° F. with thermal radiations. In this country, however, it is desirable to have a higher room temperature than in England for several reasons.

In the first instance, people in this country evidently wear lighter undergarments and therefore rely more on artificial heat. Too, in this country they do less outdoor exercise and consequently the physiological heat generator will not function so readily as with people in England. The air, without doubt, is dryer, and consequently a higher temperature compensates for this.

Many years of living in a higher artificial temperature has had its effect upon the system, and as through the ages and process of evolution environment has changed life and custom, so I think in this country the metabolism of the average American is now demanding higher temperature to make up for the conditions to be met.

During cold weather I have as a test condition been in my office and worked in perfect comfort while my colleagues had to resort to their overcoats to keep themselves sufficiently warm, which proves that it is not so much the climate, but the gradual acquisition by continual use. I keep careful records and find that I require one or two degrees higher temperature than when I first came to this country.

Therefore, in dealing with the application of thermal radiations I am taking a basic temperature of 61° F., as I find that this temperature with a relative humidity of 50 gives to my friends here a very real sense of comfort.

Referring to Figure 8, it can be seen that we get from the heated floor a stream of thermal radiations passing upward over the whole area, or from that portion of the floor which we choose to heat.

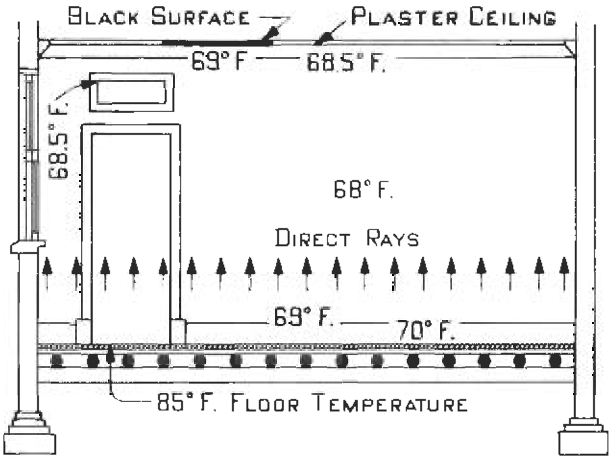

Figure 8. Room with heated floor made of ordinary flooring material, showing temperature distribution.

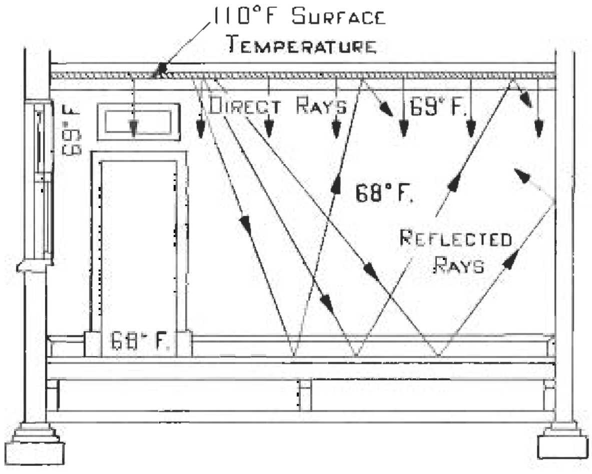

Figure 9. Room warmed by a heated ceiling, showing the manner in which radiant rays are reflected.

"The temperature of floors in Radiant Heating Systems is in the temperature range of 75° F. to 85° F. and the health of children using floors to play on is thus safeguarded" (see page 28).

WOOD FLOOR TEMPERATURE TO BE 16° F.-18° F. ABOVE INDOOR AIR TEMPERATURE IN SEVERE WEATHER

For maintaining the conditions stated before with an ordinary wood floor I find a surface temperature of 16° F. to 18° F. above the air temperature is necessary in extreme weather conditions. In milder weather 12° F. will suffice. This, however, is with an abnormal amount of glass exposure and with an exposed flat roof in addition. The range of temperatures recorded is indicated in Figure 8 and also in Figure 11.

It is interesting to note that in Figure 8 the temperature 4 inches above the floor is but slightly higher than that at the breathing line, and so also is the air temperature 1 inch below the black portion of the ceiling. A portion of the ceiling was purposely made black to study the effect. The black surface absorbs the rays received from the floor and its temperature is raised slightly above the air temperature. Now the molecules of air in direct contact with the warmer surface receive heat by conduction, and immediately the molecules are set into vibration and rebound from the surface for a short distance, depending on the impulse received.

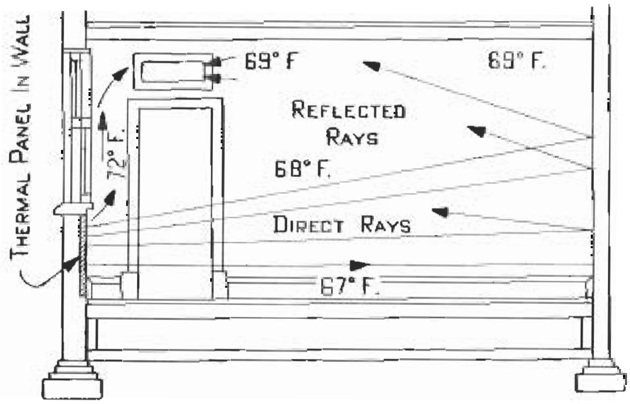

Figure 10. Room heated with thermal radiations from heated surface on side walls.

RADIOMETER DEMONSTRATES ABSORPTION BY BLACK SURFACE

We may have a demonstration of this effect any time we stop at an optical store or a jeweler's, where a radiometer is on display. This instrument is usually constructed with two glass bulbs one above the other. In each of these bulbs there are four platinum vanes mounted on a light framework, which is pivoted on a needle point. One side of each vane is highly polished and the opposite side is coated with lamp black. The glass vessels are exhausted so that the air is very rarified and offers little resistance to movement. When a stream of rays impinges on the vanes they revolve so that the polished surfaces take the lead in the direction of the rotation. Energy is absorbed by the black, and reflected by the polished surfaces. The blackened surface naturally rises in temperature and the residual air is heated and, in terms of the kinetic theory, the air molecules striking the hot surface rebound with an augmented velocity. This reactive force on the black side causes the vanes to revolve.

With floors and ceilings, which are fixed, we get the effect of fixed surfaces, but with a continuation of discharging warm molecules of air driven away from the surfaces to a distance sufficient to absorb the energy. Hence we get a slightly higher air temperature near all heating surfaces, for a distance varying with the force given to these molecules.

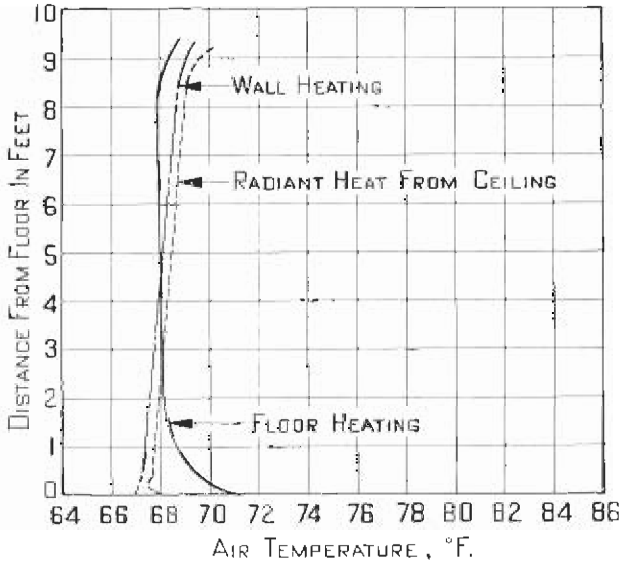

Figure 11. Comparison of results with radiant heating from ceiling as compared with wall and floor radiant heating.

In Figure 9 we have a room warmed by a heated ceiling. The rays descend from the ceiling, and if not intercepted by furniture or other obstacles will impinge on the floor and have the same effect on the floor as the heated floor, Figure 8, has on the ceiling.

The disadvantage with this method is that anyone sitting at a desk or table will have his legs and feet screened from the rays, while the head and shoulders will receive the full shower of rays direct from the ceiling.

RADIANT HEAT FROM SIDE WALLS HAS SOME ADVANTAGES OVER OTHER METHODS

Figure 10 shows a room heated with thermal radiations given off from a heated surface placed on the side walls. When so placed we get a larger amount of convected heat than we do when either the floor or ceiling heating is used. On the other hand, we shall find in dealing with the details of the schemes that this method holds many virtues which the others do not.

The amount of convection obtained from the side wall surfaces amounts to about 40% to 50% of the total heat given off. The heat rays are given off horizontally, but with all unpolished surfaces we get an irregular surface, which has the property of sending out rays from all its facets. This means that rays are emitted at all angles, and with the reflected rays from other surfaces the room is filled with a shower of rays from all directions. A disadvantage of this method is that a large piece of furniture placed in front of the heated surface will annul its effect, but we should no more think of putting an article of furniture against the heating panel than we should of placing it so as to cover up the window or the door.

If we know the heating panel should be placed in a certain position and left exposed, why not have the courage of our convictions and say it must be so, in the same way as we would a window or an electric light. We can, of course, place the heating panel under the windows and, if sufficient surface can be installed, this makes a very admirable place, for the convected heat given off is well able to deal with the exposure of the window. The rays impinging on the opposite wall and those reflected will warm up the whole room, but care naturally must be taken to obtain full advantage of all the reflected rays, for by so doing great economy is effected.

RADIANT HEATING APPROVED BY BRITISH INDUSTRIAL FATIGUE RESEARCH BOARD

With the small amount of convected heat given off, we obtain a current of warm air spreading itself over the ceiling which takes care of the heat loss through the ceiling, as it is apparent that with vertical radiant surfaces we do not get the rays impinging on the ceiling as we do in the other two systems. Speaking as a whole, however, it is the considered judgment of the British Industrial Fatigue Research Board that the thermal radiation method of heating gives a much more even temperature than does heating with radiators or with warm air.

This is even more true in this country than in England, for I find here, with the more extreme conditions and the different methods of construction, a greater variation in temperatures throughout the room with radiator heating than is the case in England.

With very large rooms with a high exposure factor it would add considerably to the comfort to have a combination of floor and wall heating. It is invariably found that the occupants in such a place will complain of cold feet even though the air is overheated. This is due to the screening effect of all heat, either radiant or convected, the cold floor, and no doubt a slow current of cool air moving over the whole floor surface. If the floor was raised to a temperature of say 6° F. to 8° F. above the air temperature and the additional heat added by wall panels, the effect would be ideal.

Estimating Heat Losses From Homes

Basic to the design of all heating systems is the determination of heat loss from the building.

Heat is lost continuously by conduction through, and radiation from, the walls, windows, roofs, and doors of the home or other building being heated.

The amount of heat lost depends upon the extent to which the building has been insulated and the difference between the desired room temperature and the temperature outside the building.

The measure of heat loss from a building or from the room of a building is obtained by the following formula:

$$H = AU\ (t_i - t_o)$$

Where H = Heat loss in B.T.U.'s per square foot per hour through a wall, floor, roof, door, or other part of a structure.

A = Area of the surface through which heat is lost. Square feet.

U = Transmission coefficient, or the heat loss in B.T.U.'s per hour per degree difference between inside and outside temperatures, through the combination of building materials from which, for example, the outside walls are to be built.

t_i – Inside room temperature. °F.

t_o – Outside air temperature. °F.

The heat loss coefficient U has been established for many combinations of building materials, and may be found on pages 49 to 53 of this Bulletin. For example, the value of U for a wall, consisting of wood siding or clapboard and 1-inch wood sheathing on the outside, and having ½-inch plaster on wood lath on the inside, is given as 0.24 B.T.U.'s per square foot per degree difference in temperature (page 51).

If the desired room temperature is 70° F. and the lowest outside air temperature in the region in which the building is to be erected is assumed as zero degree F., the total heat loss per square foot of building surface will be

$$(70°\text{ F.} - 0°\text{ F.})\ 0.24\text{ B.T.U.'s/hour.}$$

A total of 4000 feet of NATIONAL Steel Pipe was used for the radiant heating system in the residence of Jack Pahl, eastern San Francisco Bay area, California.

Interior view of the Jack Pahl residence, showing ceiling coils being covered with plaster.

To determine the total heat loss from a room requires only that the entire surface areas for other combinations of materials be multiplied by their heat loss rate in B.T.U.'s per hour.

Where the values of U for the combinations of materials which the plans call for in the walls, roofs, etc. are not given in the Heat Transmission Tables, they may be calculated by the equation

$$U = \frac{1}{\frac{1}{f_1} + \frac{1}{f_2} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_n}{k_n}}$$

Where f_1 = coefficient of heat transfer between the inside wall surface and the still inside air of the room.

f_2 = coefficient of heat transfer between the outside wall surface and the moving outside air.

f_1 and f_2 are expressed in B.T.U. per hour per square foot of surface per degree Fahrenheit temperature difference between air and wall surfaces.

x_1, x_2, and x_3 = thicknesses in inches of material composing the wall, ceiling, etc.

k_1, k_2, and k_3 = coefficient of thermal conductivity of the materials expressed in B.T.U. per square foot of surface area per inch of thickness per degree Fahrenheit difference in temperature per hour.

When air spaces exist between materials, the formula must be modified to accommodate the factor a which defines the heat transmitted across the air space between materials, expressed in B.T.U. per square foot or surface area per hour per degree Fahrenheit difference in temperature.

The formula then becomes:

$$U = \frac{1}{\frac{1}{f_1} + \frac{1}{f_2} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{a_1} + \frac{1}{a_2}}$$

It becomes apparent that any change in the number of layers of material or in the air spaces between materials affects the number of factors k, or a in the formula. Values for these factors may be found on page 49.

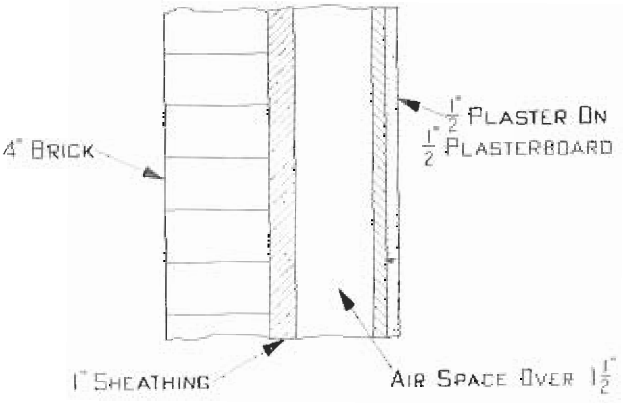

Figure 12

EXAMPLES OF THE USE OF FORMULAE

What is the coefficient of transmission air to air for the wall of a house as illustrated in Figure 12, assuming an outside wind velocity of 15 miles per hour? Use values of heat conductivity given on page 49.

$$U = \frac{1}{\frac{1}{f_2} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{a_1} + \frac{x_3}{k_3} + \frac{x_4}{k_4} + \frac{1}{f_1}}$$

$$= \frac{1}{\frac{1}{6.0} + \frac{4.0}{5.0} + \frac{1}{0.80} + \frac{1}{1.1} + \frac{1}{2.82} + \frac{0.5}{3.3} + \frac{1}{1.6}}$$

$$= \frac{1}{0.166 + 0.8 + 1.250 + 0.909 + 0.354 + 0.151 + 0.625}$$

$$= \frac{1}{4.255} = .23$$

EXAMPLE

What is the heat loss through the wall of a room of such material, assuming the area of the wall to be 8 feet high by 22 feet long, the inside temperature 70° F., the outside design temperature zero degree Fahrenheit?

$$H = AU\ (t - to)$$
$$= 8 \times 22 \times 0.23\ (70 - 0)$$
$$= 2831 \text{ B.T.U.'s per hour.}$$

ESTIMATING HEAT LOSS FROM A ROOM

The total heat loss from a room is equal to that lost through walls, floor, ceiling, doors, and windows plus that due to infiltration of cold air.

EXAMPLE

What is the heat loss from the living room of a bungalow type house, without basement, which is to be heated by radiant heat from a concrete floor panel? See Figure 13.

DESIGN DATA

Outside design temperature – 0° F.
Inside living room temperature – 70° F.

Temperature of space between living room ceiling and roof assumed as mean of room and outside temperature = 35° F.

Initial heat loss to ground during period of warming up of panel may be assumed as 20 per cent of all other heat losses from room. Assume infiltration loss equal to one air change per hour for residences.

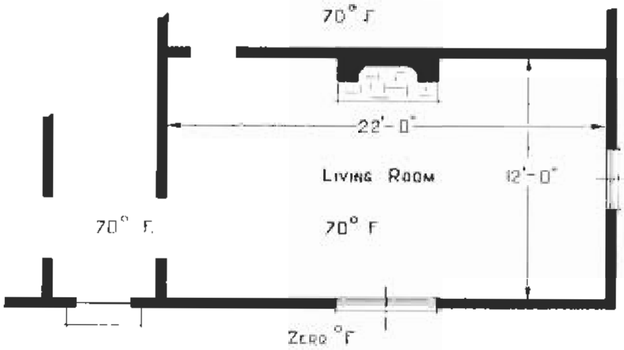

Figure 13

LIVING ROOM HEAT LOSS CALCULATIONS

Room dimensions feet	Building materials	1 Net area sq. ft.	2 Transmission coefficient U	3 Temperature difference	1×2×3 Heat loss B.T.U.'s/hr.
Exposed wall 8 × 22	As per Figure 12	155	0.23	70	2496
Window 3.5 × 6	Single—storm (See page 54)	21	0.45	70	661
Exposed wall 8 × 12	As per Figure 12	87.25	0.23	70	1405
Window 2.5 × 3.5	Single—storm (See page 54)	8.75	0.45	70	276
Ceiling 12 × 22	Item 60, page 51	264	0.066	35	610
Heat loss by air change method* = 8 × 12 × 22 × 0.018 × 70				=	2661
Heat loss, sub-total				=	8109
Initial heat loss to ground = 20 per cent of sub-total				=	1621
Total heat loss				=	9730

*The Heating, Ventilating and Air Conditioning Guide 1945 states that "An allowance of one air change per hour for all sources of air leakage for the entire volume may be considered average for a well-constructed residence."

An interesting and unusual radiant heating coil arrangement, for the attractive home of J. T. Kelley, Barrington, Illinois.

Estimating Radiant Heating Coil Requirements—Floor Panel

(UTILIZING ENTIRE FLOOR AREA FOR PANEL)

SIMPLIFIED PROCEDURE

Living Room Coils

1. Determine the total heat loss from the room. Heat Loss = 9730 B.T.U.'s per hour, (from page 15).
2. Divide the total heat loss by the entire floor area of the living room to determine the heat transfer rate in B.T.U.'s per hour per square foot of surface required from the concrete floor heating panel.
   Area of floor = 12 × 22 = 264 square feet.
   Heat transfer rate = 9730/264 = 37 B.T.U.'s per square foot of surface.
3. Make a diagrammatic layout of the coils to be used, based on recommended spacing, using 1-inch standard pipe for residences. Measure off the pipe length. See Figure 14.

Coil Spacing and †Heat Transfer Rates When Installed in Concrete Floors or Plaster Ceilings

Standard pipe size inches	Spacing inches	Heat transfer rate C
½	6 to 8	0.8
¾	9 to 12	1.0
1	12 to 16	1.2
1¼	15 to 20	1.4

†Heat transfer rate is B.T.U. per foot of pipe, and per degree difference in temperature between hot water in coils and room air.

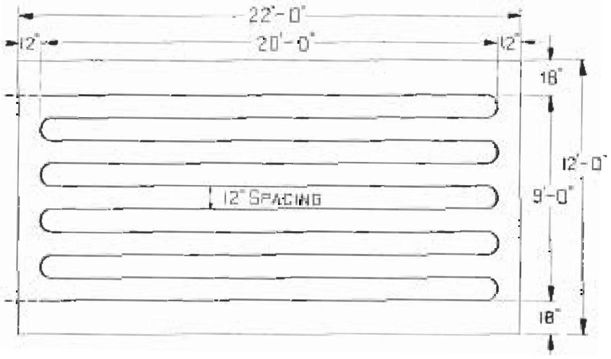

Figure 14

4. Determine the water temperature required in the coil to obtain the heat transfer rate of 37 B.T.U.'s per hour.

Required water temperature may be obtained from the formula—

$$t_w = \frac{H_t \times F_a}{C \times L} + t_r$$

Where t_w = mean water temperature in coils
t_r = room air temperature
H_t = heat transfer rate from heating panel in B.T.U.'s per hour per square foot of panel surface
F_a = floor panel area
C = a constant (see accompanying table of coil spacing and heat transfer rates) for 1-inch standard pipe $C = 1.2$
L = length of the heating coil in feet

For this example mean water temperature is

$$t_w = \frac{37 \times 264}{1.2 \times 205} + 70 = 40 + 70 = 110° \text{ F.}$$

The water temperature in the hot water line to the coils should therefore be 110 + 5 = 115° F. and in the return line to the boiler 110 − 5 = 105° F. The temperature difference as related to quantity flow provides a check on the total heat input.

5. Estimate the quantity of water to be pumped through the coils to give the required heat transfer per foot. This is governed by the water temperature drop desired in the radiant heating coils.
   Customary practice is to design for a temperature drop of 10° F. to 20° F., and in this example 10° F. is used.
   The quantity of water to be pumped through the coil is given by—

$$Q = \frac{H}{180\, t_d}$$

The radiant heating system in this modern Sales and Service building of Ernest Burwell, Inc., Spartanburg, S.C. assures maximum comfort for personnel in the service department, and a pleasant atmosphere for prospective customers in the display room.

Where Q = quantity of water in gallons per minute

H = total heat loss from room B.T.U.'s per hour

t_d = desired water temperature drop

$$Q = \frac{9730}{480 \times 10} = 2.0 \text{ gallons per minute}$$

6. Determine the friction loss in the coils for the required flow capacity. Referring to Chart, Figure 15, the friction loss per 100 feet of 1-inch standard pipe is 0.35 feet.

   Length of straight pipe in the coil is 191 feet.
   Number of bends is 9.

   Assuming that the coil bends offer frictional resistance corresponding to straight pipe equal in length to 25 times the pipe diameter, the total equivalent length of pipe is—

$$\text{Length of straight pipe} + \frac{\text{No. of bends} \times 25 \times \text{pipe diameter}}{12}$$

$$= 191 + \frac{9 \times 25 \times 1.0}{12} = 219 \text{ lineal feet}$$

   Total friction loss in the coils is $2.19 \times 0.35 = 0.77$ (approximately one foot).

   After the total heat loss for the house has been determined in the manner outlined, and the total friction loss through all coils, valves, and fittings determined, the type and capacity of circulating pump to be used can be determined.

TESTING COILS

All coils after being fabricated and welded on the job should be subject to an air or hydrostatic test pressure of 250 pounds per square inch for a period of twelve hours, or to a test pressure and for a testing period, stipulated by the heating engineer.

USE OF CHART, FIGURE 16, TO ESTIMATE RADIANT HEATING COIL REQUIREMENTS —FLOOR PANEL

1. Determine heat loss from room.
   Heat loss = 9730 B.T.U.'s per hour, from page 15.
2. Assume a water temperature for the radiant heating coils of 110° F. (previous example).
3. Obtain the difference between water and room air temperatures, using water temperature from previous example.
   Temperature difference is $(110 - 70) = 40°$ F.
4. Assume 1-inch standard pipe size, and using the estimated heat loss of 9730 B.T.U.'s and the temperature difference of 40° F., read off the length of pipe required for the coil, 205 feet.

NOTE:—The Chart, Figure 16, is based on a heat transmission value for residences of relatively low ceilings as encountered in usual construction of 3.5 B.T.U.'s per square foot of external pipe surface, per degree Fahrenheit temperature difference (td) water to air. The maximum desirable water temperature is approximately 130° F.

RESISTANCE OF VALVES AND FITTINGS TO FLOW OF FLUIDS*

When the flow of a fluid in a pipe line is altered by some obstruction, such as a valve or fitting, the velocity is changed, turbulence is magnified, and a drop in pressure results. This pressure drop may be insignificant in long lines where it is very small in comparison to the total drop, but when the line is short, the pressure drop through valves and fittings becomes a major item in the total pressure drop value.

It has been shown by previous investigators that the drop in pressure through valves, fittings, etc., is some constant multiplied by the velocity head, $\frac{V^2}{2g}$. Therefore,

$$H_1 = k\frac{V^2}{2g},$$

where H_1 = loss of head in feet

k = coefficient (values given in table on page 18)

V = velocity of water, feet per second

$2g = 64.4$

Radiant heating panel coils in position ready for concrete pouring. 3000 feet of 1-inch National Pipe—Union Mortuary, Bountiful, Utah.

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*Courtesy—Crane Co.

Coefficient k

Type	k	Authority
Globe valve	10.0	Crane tests
Angle valve	5.0	Crane tests
Close return bend	2.2	
Standard tee	1.8	Giesecke & Badgett
Standard elbow	.9	Giesecke & Badgett
Medium sweep elbow	.75	Crane tests
Long sweep elbow	.60	Bulletin No. 2712—University of Texas
45° elbow	.42	Bulletin No. 2712—University of Texas
Gate valve (fully open)	.19	Bulletin No. 252—University of Wisconsin
¼ closed	1.15	Bulletin No. 252—University of Wisconsin
½ closed	5.6	Bulletin No. 252—University of Wisconsin
¾ closed	24.0	Bulletin No. 252—University of Wisconsin
Borda entrance	.83	"Hydraulics" Daugherty
Sudden enlargement:		
d/D = ¼	.92	"Hydraulics" Daugherty
d/D = ½	.56	"Hydraulics" Daugherty
d/D = ¾	.19	"Hydraulics" Daugherty
Ordinary entrance	.50	"Hydraulics" Daugherty
Sudden contraction:		
d/D = ¼	.42	"Hydraulics" Daugherty
d/D = ½	.33	"Hydraulics" Daugherty
d/D = ¾	.19	"Hydraulics" Daugherty

Courtesy—Crane Co.

Warehouse and office building of J. E. Dilworth Company, Memphis, Tennessee, including driveway of adjoining customers' parking area, not visible, equipped with radiant heating and snow melting systems respectively. An outstanding example of the full use of this modern method of heating and snow removal.

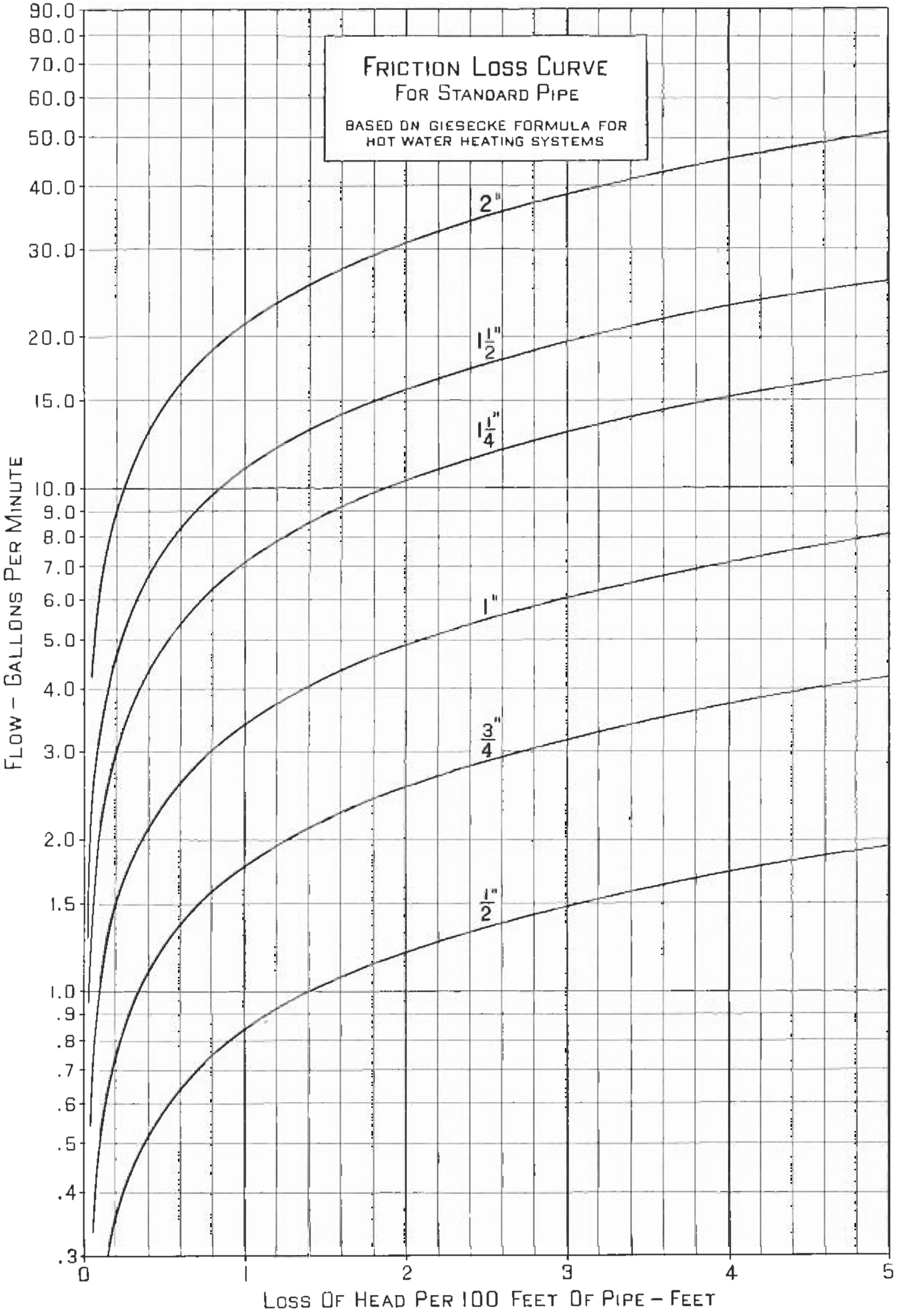

FRICTION LOSS CURVE
FOR STANDARD PIPE
BASED ON GIESECKE FORMULA FOR
HOT WATER HEATING SYSTEMS
2"
1 1/2"
1 1/4"
1"
3/4"
1/2"
FLOW – GALLONS PER MINUTE
90.0
80.0
70.0
60.0
50.0
40.0
30.0
20.0
15.0
10.0
9.0
8.0
7.0
6.0
5.0
4.0
3.0
2.0
1.5
1.0
.9
.8
.7
.6
.5
.4
.3
0
1
2
3
4
5
LOSS OF HEAD PER 100 FEET OF PIPE – FEET

Figure 15

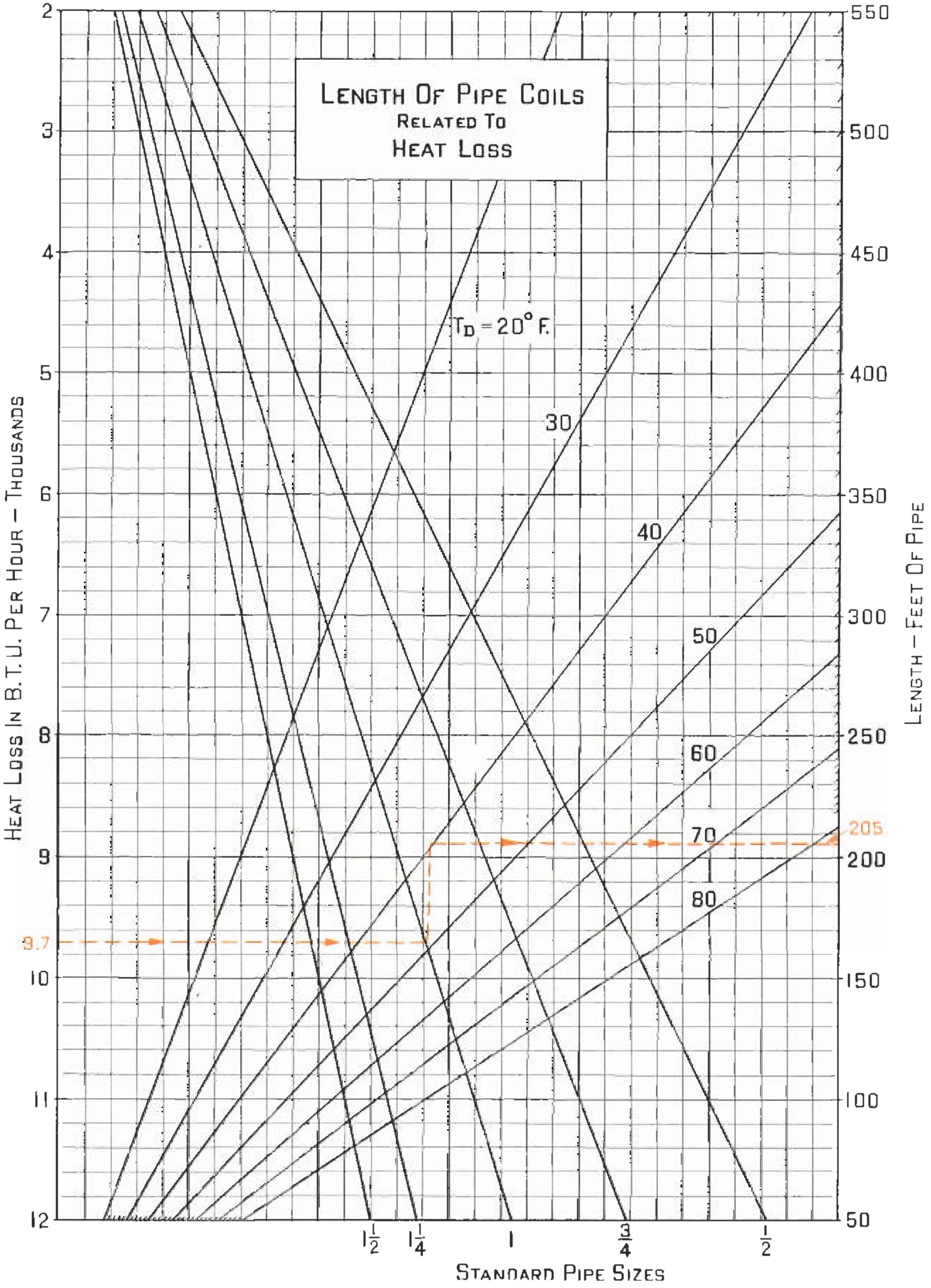

LENGTH OF PIPE COILS
RELATED TO
HEAT LOSS
TD = 20° F.
30
40
50
60
70
80
HEAT LOSS IN B.T.U. PER HOUR – THOUSANDS
2
3
4
5
6
7
8
9
9.7
10
11
12
LENGTH – FEET OF PIPE
550
500
450
400
350
300
250
205
200
150
100
50
1½
1¼
1
¾
½
STANDARD PIPE SIZES

Figure 16

Estimating Heating Coil Requirements†—Ceiling Panel

for Living Room With Wood Floors Over Basement*

(BASED ON THE MRT METHOD)

THERE ARE VARIOUS METHODS of installing pipe coils, and several of these are shown on page 31. In the following example the coils are assumed as installed in the living room ceiling, similar to Figures 1 and 2, page 34, and page 2.

1. Establish the desirable comfort or operative temperature. Radiant Heating Comfort Chart, Figure 18, page 23, shows the operative temperature for houses to be 70° F.

This operative temperature may be obtained by many combinations of the mean radiant temperature (MRT) and air temperature, as the Chart shows. For example, an operative temperature of 70° F. may be obtained with an MRT of 75° F., and an air temperature of 65° F. However, as has been stated previously, the mean radiant and air temperatures will tend to approach equilibrium in houses, offices, and similar structures. For this example 72° F. MRT and 68° F. air are therefore selected.

2. Tabulate the heat loss for a living room of the same dimensions, coefficients, etc., as shown on page 15 but with hardwood floor on yellow pine subflooring on joists (see table below).

3. Establish the surface temperature of the ceiling panel at 100° F.

4. Determine the mean radiant temperature (MRT) for all inside surfaces, excluding the ceiling, as follows:

$$\frac{\text{Transmission coefficient } U \times \text{temperature difference inside room to outside air}}{1.65} = \text{difference between inside surface and operative temperature}$$

or for exposed walls $= \frac{0.23 \times 70}{1.65} = 10°$ F.

70° F. – 10° F. = 60° F. inside wall surface temperature

or for windows $= \frac{0.45 \times 70}{1.65} = 20°$ F.

70° F. – 20° F. = 50° F. inside window surface temperature

Surfaces of inner walls of room are assumed to be the same as the operative temperature = 70° F.

Surface	Net area square feet	Surface temperature °F.	Net area × surface temperature
Exposed wall	155	60	9300
Exposed wall	87.25	60	5235
Inside wall	176	70	12320
Inside wall	96	70	6720
Windows	29.75	50	1487
Floor	264	70	18480
Totals	808		53542

53542 ÷ 808 = 66° F. MRT of inside surfaces excluding ceiling. The temperature of the ceiling outside the panel area does not affect this MRT value for estimating purposes.

5. Determine the heat delivered by radiation for a panel temperature of 100° F., and an MRT of 66° F. From Chart, Figure 19, this is equal to 34 B.T.U.'s per hour per square foot of panel surface.

6. Determine the heat delivered by convection for 100° F. panel surface and 68° F. air temperature. The difference in temperature is 100° F. – 68° F. = 32° F. From Chart, Figure 20, the rate of heat delivered by convection is 0.52 B.T.U. per square foot per hour per degree F. or 0.52×32° = 16.6

	Dimensions feet	Net area square feet	Transmission coefficient *U*	Temperature difference °F.	Heat loss B.T.U.'s/hr.
Exposed wall	‡ 8×22	155	.23	70	2496
Exposed wall	‡ 8×12	87.25	.23	70	1405
Window	3.5×6	21	.45	70	662
Window	2.5×3.5	8.75	.45	70	276
*Floor	12×22	264	.34	5	449
Air change	Total vol. = 2112 cu. ft. × .018 × 70° F.				2661
				Total heat loss	7949 B.T.U.'s/hr.

---

†Based on data from "Radiant Heating and Cooling" by F. E. Giesecke, Consulting Engineer, and published in "Heating, Piping, and Air Conditioning," June to October, 1940.

*The basement beneath living room is a heated game room. Recommended operative temperature 65° F. Source of heat loss coefficients for window and ceiling given in Table, page 15.

‡Excluding windows.

B.T.U.'s per hour per square foot of panel surface.

7. Add (5) and (6) or 34+16.6=50.6 B.T.U.'s per hour of radiated and convected heat per square foot of panel surface.

8. Determine panel area required.

$$\frac{\text{Total heat loss}}{\text{Heat transfer rate}}=\frac{7949}{50.6}=157 \text{ square feet.}$$

A panel** 16 feet long by 10 feet wide meets this area requirement.

9. Assume a pipe size and spacing for coils, say 1-inch standard pipe on 12-inch (1-foot) centers. See Figure 17.

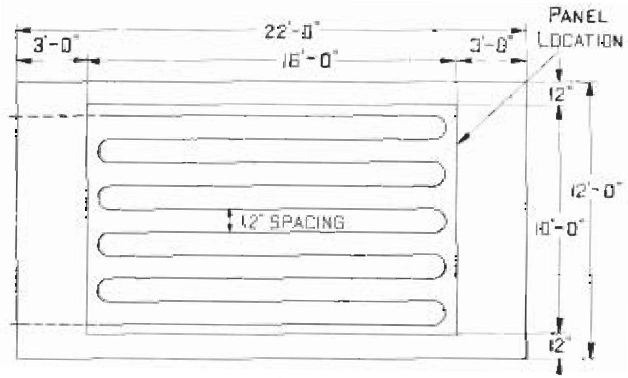


Figure 17

10. Find the length of pipe required for the panel by dividing the area (8) by the spacing selected in feet (9).

$$\frac{\text{Area}}{\text{Spacing}}=\frac{16\times 10}{1}=160 \text{ feet}$$

11. Assume that the insulation above the panel is such that 10 per cent of the heat delivered flows upward and 90 per cent downward. Divide the total heat loss (2) by 0.90.

$$\frac{7949}{0.90}=8830 \text{ B.T.U.'s per hour to be delivered}$$

by panel.

12. Divide the heat to be delivered by the panel by the length of coil:

$$\frac{8830}{160}=55 \text{ B.T.U.'s per linear foot of pipe}$$

13. Divide the total heat transfer rate per linear foot of pipe (12) by the tabulated heat transfer rate per foot (page 16) for 1-inch pipe to obtain the difference between room air and required water temperature.

$$\frac{55}{1.2}=46° \text{ F. temperature difference, water to air.}$$

The average water temperature required in the coils to deliver the required heat should thus be equal to room air temperature plus temperature difference, water to room air:

$$68°+46.8°=114.8° \text{ F.}$$

14. Determine the MRT of all room surfaces and check against required design MRT of 72° F.

$$\frac{(\text{Panel area}\times\text{panel temperature})+(\text{Total area all surfaces}-\text{panel area}\times\text{MRT of step 4})}{\text{Total area all surfaces}}=$$

$$\frac{(160\times 100° \text{ F.})+(912\times 66° \text{ F.})}{1072}=\frac{76192}{1072}=71.1° \text{ F.}$$

MRT which is satisfactory.

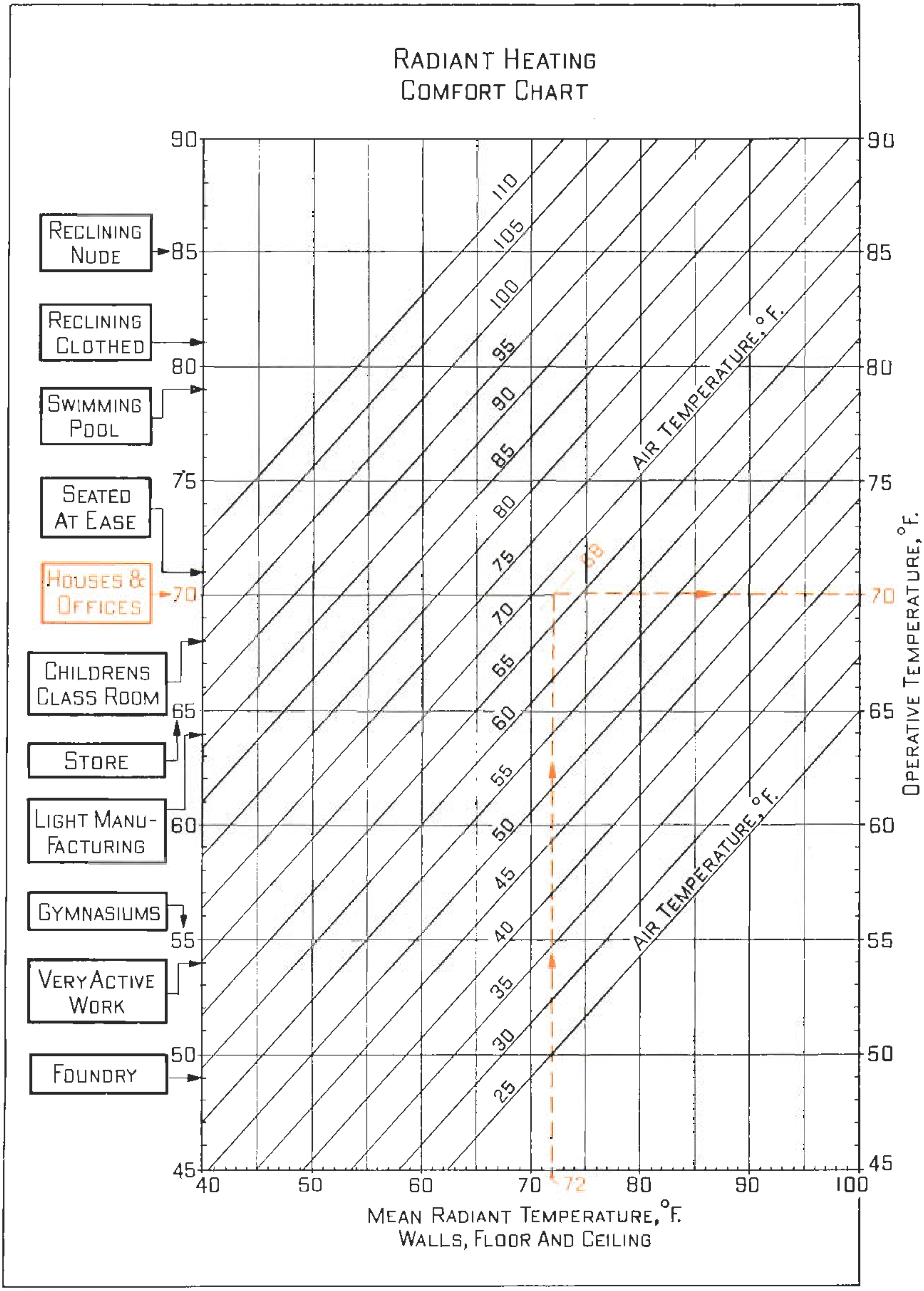
By use of emissivity coefficients for surfaces radiating to absolute zero, a more accurate analysis gives 72.2° F. for a final MRT. Should 72° F. MRT and 68° F. air not produce satisfactory conditions, the ceiling temperature can easily be adjusted as required by changing the temperature of the circulating water.

15. The method of determining water flow capacity and friction loss is outlined on pages 16 and 17.

The Kuykendall Chevrolet Company garage, in Lubbock, Te is skillfully designed with two important things in mind: efficiency for low operating cost; maximum comfort for mechan personnel. Radiant heating is effectively used for these resul

---

**Some design engineers utilize the full ceiling areas as a heating panel.



*Courtesy "Heating and Ventilating."

Figure 18

HEAT DELIVERED BY RADIATION
TO ROOM BY PANEL*

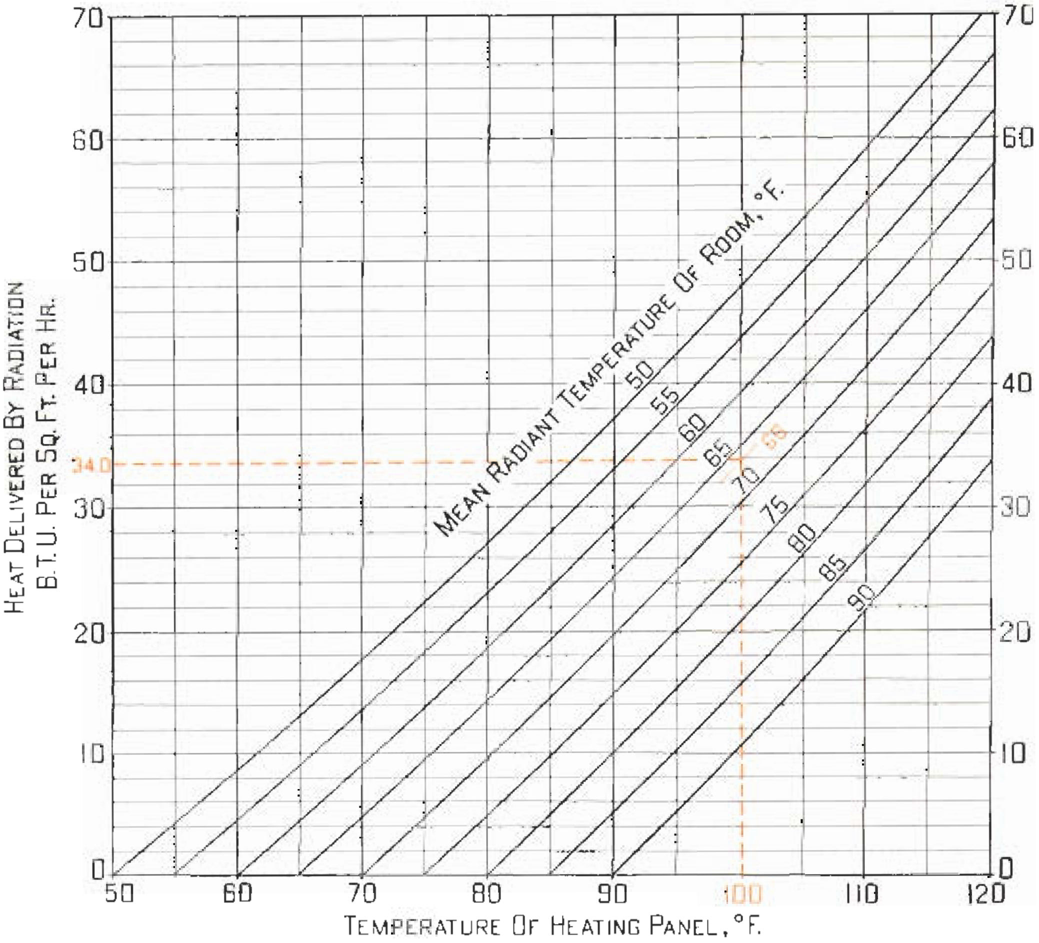


Figure 19

HEAT TRANSMISSION FROM PANEL TO AIR
BY CONVECTION AND CONDUCTION*

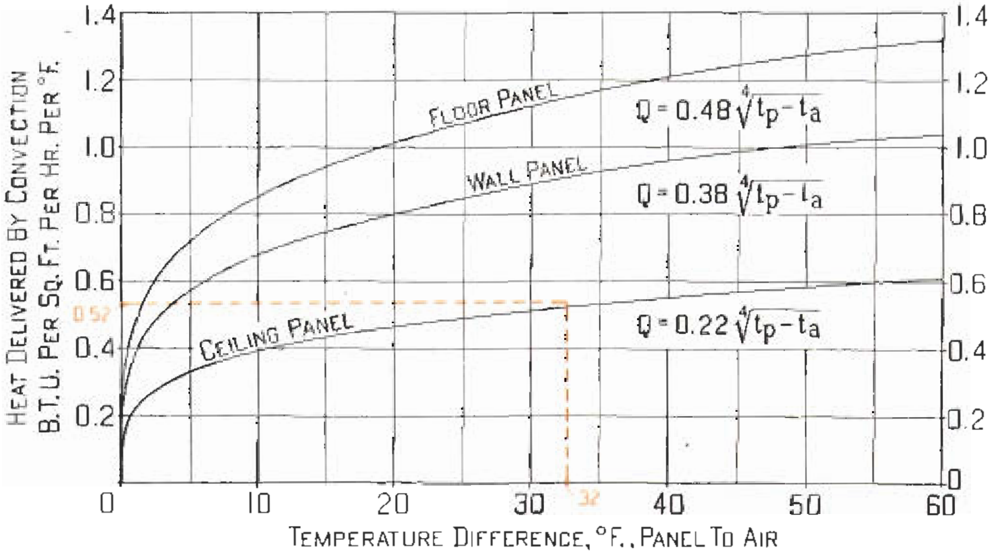


Figure 20

*Courtesy "Heating, Piping and Air Conditioning" and F. E. Giesecke, Consulting Engineer.

Heat Losses Through Floors of Basementless Buildings†

THERE HAS LONG BEEN SOME QUESTION regarding the accuracy of estimates of heat loss through the floors of basementless buildings. Heat loss coefficients in standard tables are for air-to-air; in the case of floors laid on the ground the air-to-air coefficient is probably in error. In addition, the temperature of the ground below the floor is exceedingly difficult to estimate.

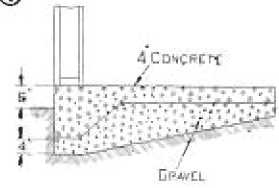
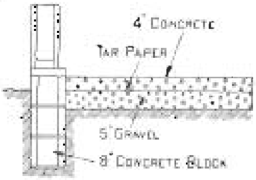
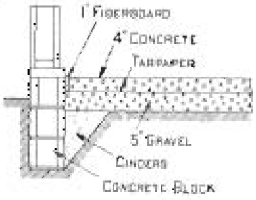
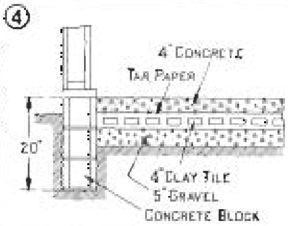
Nevertheless, the estimating problem exists, especially in the field of low cost housing, and has recently received attention by the National Bureau of Standards, whose findings have been reported in one of the Department of Commerce Building Materials and Structures booklets.* The study was of concrete floors and wood floors laid over crawl spaces as well as on the ground.

The results showed that the heat loss of the floors laid on the ground was decreased by insulating the edges; that the heat loss through the center of such floors is relatively small when the enclosing structure is continuously heated; that the edge loss for a wood floor laid over a crawl space is small; and that the edge loss for an insulated concrete floor laid over a crawl space was considerable. The floors tested are illustrated in types 1 to 8 in the accompanying drawings and on page 27.

For determining the heat loss through these floors, the general plan adopted was to provide a heavily insulated structure above a specimen of each kind of floor and to observe the amount of heat, supplied in the form of electric energy, necessary to maintain a temperature of 70° F. within the structure during cold weather. Although the walls and ceiling of the structure were heavily insulated, some heat loss through them was inevitable. To correct the data for this condition, tests were made during which the floors themselves were so insulated that the heat loss through the structure could be measured.

FLOORS ON GROUND

The investigators concluded that the heat loss of a floor laid on the ground is not proportional to the temperature difference between the air inside and the air outside of the house at any given instant. The floor heat loss appears to be dependent upon the temperature of the ground at some region beneath the surface, and this, in turn, depends upon the average temperature of the air above the ground and the amount of heat received by the ground from the sun and the amount of heat loss from it by radiation or otherwise during some period prior to the observation.

TYPE OF FLOOR	VALUE OF "F" IN B.T.U. PER HOUR PER FOOT OF EDGE PER DEGREE F.
① 	0.81
② 	0.69
③ 	0.55
④ 	0.75

Four types of floor laid on ground. Loss factor "F" takes into account the exposed edge.

For each of the floors, 1, 2, 3, and 4, the observed heat loss was divided by the length of the exposed

---

†From "Heating and Ventilating," by courtesy of the publisher.

*Report, BMS103, "Measurement of Heat Losses from Slab Floors," by Richard S. Dill, W. C. Robinson, and H. E. Robinson. Available from the Superintendent of Documents, 10 cents.

edge, as defined on page 25, and the result entered in the tables as "heat loss per foot of exposed edge." From this, three factors were derived, one of which is particularly useful.

The heat loss in B.T.U.'s per hour per linear foot of exposed edge was divided by the average temperature difference observed during each observation period between the air inside the structure and the air outside, to yield a factor "F" which takes into account the exposed edge.

For estimating design heat losses from slab floors on the ground, the investigators propose three formulas, of which the following is suggested by them as probably being the most adaptable:

$$Q = LF\ (T_i - T_g)$$

where

Q = heat loss through floor, B.T.U.'s per hour,
L = length of floor edge adjacent to exposed wall of building, feet,
F = heat loss factor, B.T.U.'s per degree temperature difference,
T_i = inside design temperature, F,
T_g = average outside temperature for week preceding instant for which estimate is to be made, F.

The last term (T_g) is the only one for which data are not readily available. The report did not include a table of this factor for various localities and, unfortunately, as defined, the data could be confusing. To simplify the matter and to be on the safe side, it seems that T_g could be defined as

T_g – average outside temperature for the week preceding the coldest temperature of record, F.

Such data are simple to compile and have been obtained by questionnaire: see Table 1.

The following example will indicate the application of the equation.

EXAMPLE

A 30 × 28-foot residence near New York City is built on a slab of 4 inches of concrete laid on 5 inches of gravel. What will be the floor heat loss?

TABLE 1

VALUES OF T_g FOR 51 CITIES FOR ESTIMATING HEAT LOSSES THROUGH FLOORS OF BASEMENTLESS BUILDINGS

Albany, N. Y.	5	Houston, Tex.	43	Portland, Me.	12
Atlanta, Ga.	20	Indianapolis, Ind.	16	Portland, Ore.	20
Baltimore, Md.	21	Kansas City, Mo.	2	Rochester, N. Y.	6
Birmingham, Ala.	19	Lincoln, Neb.	0	St. Louis, Mo.	15
Boston, Mass.	8	Little Rock, Ark.	13	Salt Lake City, Utah	9
Buffalo, N. Y.	6	Los Angeles, Calif.	51	San Diego, Calif.	49
Chicago, Ill.	5	Louisville, Ky.	25	San Francisco, Calif.	45
Cincinnati, Ohio	17	Memphis, Tenn.	12	Savannah, Ga.	38
Cleveland, Ohio	16	Milwaukee, Wisc.	1	Scranton, Pa.	8
Dallas, Tex.	26	Minneapolis, Minn.	4	Seattle, Wash.	29
Denver, Colo.	8	Nashville, Tenn.	14	Spokane, Wash.	1
Detroit, Mich.	14	New Haven, Conn.	9	Syracuse, N. Y.	9
Duluth, Minn.	5	New Orleans, La.	32	Topeka, Kansas	2
Fort Wayne, Ind.	13	New York, N. Y.	14	Trenton, N. J.	12
Grand Rapids, Mich.	8	Oklahoma City, Okla.	9	Utica, N. Y.	4
Harrisburg, Pa.	9	Philadelphia, Pa.	18	Washington, D. C.	14
Hartford, Conn.	16	Pittsburgh, Pa.	12	Wichita, Kansas	0

The Vivian Webb Ch
el at the Webb Scho
Claremont, Califor
radiant heated w
NATIONAL St
Pipe.

A floor panel of
inch NATIONAL S
Pipe for the Viv
Webb Chapel.

SOLUTION

The slab is similar to floor type 2, for which $F = 0.69$. The exposed edge L will add up to $(30+30+28+28) = 116$ feet. Presumably the inside temperature would be 70° F., and from the table, T_a for New York = 14° F., so that

$Q = 116 \times 0.69\ (70-14) = 1180$ B.T.U.'s per hour.

The data are probably sufficiently accurate basis for most estimating purposes because the floor heat loss is likely to be small compared to other losses. However, the data are incomplete in that they do not cover the cases of frozen ground and of snow-blanketed ground. To supply data, it would be necessary to repeat the tests in a colder climate.

No reason is apparent why the data are not applicable for regions where the average outdoor temperature does not remain continuously below freezing for more than a day or so, except that snow, which is an insulator, may decrease floor heat loss.

The data indicate that insulating the floor at the edge is beneficial both in saving heat and in reducing lateral temperature gradients across the floor.

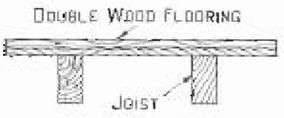
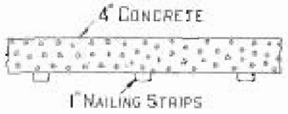
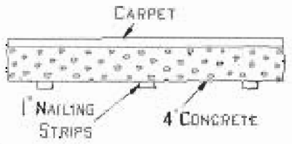
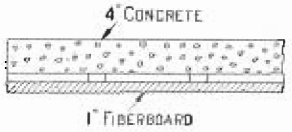
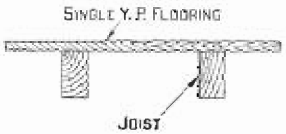
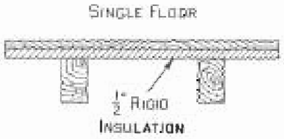
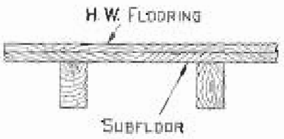
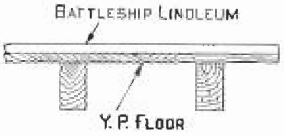
FLOORS OVER CRAWL SPACE

Floors tested by the Bureau over crawl spaces are illustrated in types 5 to 8, inclusive. This type of floor has a conventional heat loss coefficient (air-to-air). The authors' results checked computed values of U as follows:

Floor Type	U Value	
	Observed	Computed
5	0.24	0.27
6	0.46	0.50
7	0.34	0.38
8	0.30	0.17

Summing up, the tests on floors laid over crawl spaces indicated that the factors contained in handbooks are suitable for estimating heat losses through such floors, except in the case of a floor which is heavily insulated on the underside. In this case, the edge loss increases largely in comparison to the total loss through the floor, and this may result in an underestimate unless it is taken into consideration. The number of factors involved indicate that heat losses through floors laid over crawl spaces should be computed on the basis of an estimated crawl-space temperature. For a continuously ventilated crawl space, the temperature should be assumed to be the same as the outdoor temperature.

Since the usual U values were found to be reasonably accurate for floors over crawl spaces, the accompanying table gives U values for four other floors (9 to 12, inclusive) *not tested* by the Bureau. These values are from H & V's Reference Data 73-74, and are included for convenience.

TYPE OF FLOOR	VALUES OF "U" FLOORS OVER CRAWL SPACES
⑤ 	0.24
⑥ 	0.46
⑦ 	0.34
⑧ 	0.30
⑨ 	0.45
⑩ 	0.27
⑪ 	0.34
⑫ 	0.34

Floors over crawl spaces. Conventional loss coefficient U is used.

What Heating Engineers Claim for Radiant Heating

1. ECONOMY OF OPERATION

With good controls, savings in operating costs of as much as 30 per cent and higher are reported for radiant heating systems over the conventional heating systems.

2. EVEN DISTRIBUTION OF HEAT

Heating coils are designed so that they effectively transfer heat through the entire ceiling or floor surface, resulting in low, yet adequate, floor and ceiling surface temperatures and in a more uniform distribution of heat in the room.

3. GREATER WALL SURFACE AVAILABLE

There are no visible parts in a radiant heating system, heating coils being buried beneath the floor, or in the space between ceiling and the floor above. Greater wall space is thus available for placing of furniture and for decorative effects.

4. INCREASED FEELING OF COMFORT

The air in a room heated by radiation does not dry out as does air passing through or in contact with a high temperature heating unit. It is fresh, moderate in temperature, and gives the room occupant a more alert feeling. There is complete lack of stuffiness, as commonly found in overheated air.

5. CLEANLINESS

Because there are no air currents set up, dust is not carried and deposited on room surfaces, furniture, and draperies.

6. FLOOR UTILITY

The temperature of floors in radiant heating systems are in the temperature range of 75° Fahrenheit to 85° Fahrenheit and the health of children using floors to play on is thus safeguarded.

7. BETTER ROOM TEMPERATURE CONTROL

There are no drafts or hot and cold spots in a room heated by a radiant system. Because of the uniform distribution of heat from the entire floor or ceiling surface, the temperature is practically uniform at all levels and in all parts of the room.

8. AUTOMATICALLY CONTROLLED

A radiant heating system is automatically controlled by thermostat, and, once in operation, requires little or no attention.

9. NOISELESS IN OPERATION

Because the only moving parts in a radiant heating system are contained in the circulating pump, the system is noiseless and the room occupant is unaware of its operation.

10. PSYCHOLOGICAL EFFECT

The application of heat from the floor seems to produce a psychological relaxation on the part of workmen assembling equipment, particularly equipment which includes the handling of small parts requiring a good deal of concentration. This more restful and comfortable feeling develops a better mental attitude which reduces the amount of errors and increases the quality of the work.

Heiland Research Corporation, Denver, Colorado, used 6000 feet of 1-inch NATIONAL Steel Pipe for radiant heating.

Radiant Heating Question Box*

Q. 1. ***What size pipe should be used for a radiant heating job?***

A. Pipe coils for radiant heating are generally constructed with ½, ¾, 1, or 1¼-inch pipe.

In selecting a particular pipe size for a radiant heating job, the designer is governed by a proper balance between the cost of such a system and efficiency in operation. He must consider the required heat transfer from the panel to the room or space to be heated, and this depends upon the heat lost through the walls, windows, and other surfaces of the building. Generally speaking, he must decide which pipe size at stated center to center distance will give the desired heat transfer from the radiant heating coils to the space to be heated at the lowest cost per foot of pipe or per coil.

The Heating, Ventilating and Air Conditioning Guide 1945 states that, when hot water pipes are embedded in concrete slabs or attached to plastered surfaces, their rate of heat emission may be assumed as:

0.8 B.T.U. per hour per linear foot for ½-inch pipe when spaced on 6-inch centers
1.0 B.T.U. per hour per linear foot for ¾-inch pipe when spaced on 9-inch centers
1.2 B.T.U. per hour per linear foot for 1-inch pipe when spaced on 12-inch centers

for each degree difference in temperature between coil water temperature and air temperature.

This information is given as a general guide for the designer, and the values will vary depending on depth of pipe below surface of panel, and on materials from which panel is constructed. Additional experience and research will develop more definite and complete data. Actual installation tests show these values to be very conservative.

Another factor in determining pipe size is the quantity of hot water which must be distributed through the coils. We may obtain the necessary heat transfer from coils to room with a certain pipe size and spacing, but we must also determine whether or not the friction loss developed in distributing the necessary quantity of water is excessive, for this influences the pump size.

Q. 2. ***What is the approximate cost of a radiant heating system?***

A. Mr. Raymond Viner Hall, prominent architect, gives the following as the cost of several radiant heating installations made some years ago and the percentage of the total cost of the home:

Job	Total cost	Radiant heating	Per cent	Fuel
A	$5,200	$405	9.5	Gas
B	5,200	375	7.2	Coal
C	6,500	525	8.1	Gas
D	7,000	675	9.6	Oil
E	8,500	575	6.8	Gas

After reviewing the cost of twenty radiant heating jobs installed during 1940 and 1941, Mr. Hall states "Even those whose mania is operational costs need not apologize for the first cost of the floor type heating system."

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*Based on inquiries received by National Tube Company.

Section of Mulvihill Motor Company building in Grand Rapids, Michigan, where 6000 feet of 1¼-inch National Pipe was used for the radiant heating system.

The cost of a complete system varied in the year 1941 from 72 dollars to 136 dollars per room, excluding baths and attached garages, and depending on the type of fuel, controls, etc.

These systems have represented from 6.8 per cent to 9.6 per cent of the total construction cost.

Q. 3. *Would a radiant heating job cost more than a hot air job?*

A. A radiant heating installation might cost more than a hot air gravity heating system but could easily cost less than a system equipped with blowers, but this higher cost is offset by less fuel consumption and better comfort conditions for the occupants of the building.

There are no hot and cold air grilles to take up space and mar the appearance of walls, or to cut down on available wall or floor space for placing of furniture. Where grilles are placed at baseboard level the entire wall space above may be useless in placing furniture, and in the case of expensive pieces like pianos and radios, the forced hot air entering a room influences their location.

The cost of installation of radiant heat to the prospective user is not to be measured only in terms of dollars and cents.

Q. 4. *Would it cost more or less to operate a radiant heating system than other types, and by how much?*

A. Experience has shown that a well-designed and installed radiant heating system is much more economical to operate than other systems.

Savings of 30 per cent and higher over operating costs of other systems are recorded, where good controls are used.

Q. 5. *Would a radiant heating installation have to be a welded job?*

A. Either threaded and coupled joints or welded joints may be used in fabricating coils; however, welded joints are used almost exclusively and are the preferred method. In either case the system should be subject to a hydrostatic test pressure of not less than 5 times the intended operating pressure.

Q. 6. *In a building without any basement how would the pipe be laid?*

A. The coils would, if designed for a floor installation, be placed in the floor structure approximately as illustrated on page 34.

If concrete is used, general practice is to set the fabricated coils on a gravel base with the pipe raised sufficiently above the gravel to pour at least 1 inch, and preferably 2 inches, of concrete under the pipe, as shown in the illustration. The concrete is then poured over the coils. Various modifications of this method are in use, some of which require placing of insulation beneath the coils, and others laying the coils on previously set concrete rather than laying over gravel. These modifications have a twofold purpose; one, obtaining greater transfer of heat from the coils upward into the space to be heated, and corresponding reduction of heat loss to the ground; two, preventing ground water from seeping up and around the bottom of the coils.

Q. 7. *In a building having a basement, what would be the procedure for installation?*

A. Assuming that the basement is to be used frequently by the occupants, and is to contain a game room or workshop, it would be desirable to design for a floor panel. In existing buildings, wall coils may be used in the basement. The space above the basement may be heated by coils installed in the floor or ceiling above. Where pipe is to be so installed, it should be run above the joists for the floor or under the joists if installed in the ceiling. In either case, the pipe should be securely attached to the joists.

Q. 8. *Would much heat be lost to the ground through the concrete slab?*

A. In a concrete slab heating panel there is initial heat loss to the ground when the unit is first placed in operation. This however occurs mainly during the warming up period, and as the soil becomes warm, less and less heat is lost until when the system is in full operation heat loss to the ground is a small percentage, provided a good layer of broken stone or other insulating material is placed below the concrete floor.

Q. 9. *Is much heat transmitted to the basement from coils in the floor above?*

A. The heat transmitted to the basement from coils built into the floor of the surface above would depend upon the method of installation and particularly on insulating value of materials beneath the coils. The heat transfer to the basement depends also upon the temperature differential between the basement and that of the room above. Since the basement temperature will generally be lower than that of the room above, an allowance should be made for the loss from the rooms above through the floor, and in the usual manner based on the "heat loss coefficients" for the materials used in the floor.

Q. 10. *How does the heat transfer property of wrought steel pipe in radiant heating systems compare with that of wrought iron or copper pipe?*

A. For all practical purposes they are the same.

When coils are set on sand or gravel and a concrete slab poured over them, or where they are embedded in plaster, the transfer of heat from the pipe to the concrete or plaster surface is by conduction.

A noted authority on radiant heating states that "when so installed there is practically no difference in the B.T.U. transfer rate from the concrete or plaster surface, for wrought steel, wrought iron, or copper pipe."

Q. 11. ***How does the coefficient of expansion of wrought steel pipe compare with that of wrought iron pipe?***

A. The mean coefficient of linear expansion for wrought steel pipe in the temperature range of 32° F. to 392° F. is equal to 0.0000068 compared with 0.0000072 for wrought iron. Therefore, if a radiant heating coil whose over-all length is 40 feet is installed at 40° F. and after the concrete is poured and set, heated by the passage of water to 140° F., it will, if constructed of iron pipe, expand 0.32 inches as compared with 0.31 inches for steel pipe. The coefficient of linear expansion of concrete depends to some extent on the concrete mix. For a 1: 1½: 3 concrete mix, the coefficient generally used is the same as that for steel or 0.0000068. In other words, for practical purposes these coefficients are the same.

Q. 12. ***Is corrosion a factor in radiant heating systems?***

A. Corrosion is an inconsequential factor in a radiant heating system correctly designed, and can be disregarded in selection of piping materials. Radiant heating systems differ basically from present conventional heating systems in only one important respect, namely, "the method of heat transfer." Both systems use the same heating medium—hot water or steam; both are assembled from the same equipment—boiler, piping, and controls; both depend on the same fuels—gas, coal, or oil. In other words, the character of service performed, particularly with respect to the piping, is identical. Both are closed circuits. Once in operation, the same water is recirculated over and over again. It is a matter of common knowledge that the small amount of dissolved oxygen entering a system when first filled, and in the occasional make-up, is quickly absorbed and of inconsequential effect on the piping and boiler in any closed circuit. Since corrosion does not take place unless there is a continuous supply of oxygen, it is obvious that this factor is of no consequence whatever in radiant systems, regardless of the kind of pipe used. Throughout the country and over a long period of years in hundreds of thousands, even millions, of buildings of all types, including homes, office buildings, schools, and public buildings, etc. conventional closed circuit hot water systems have been used with a remarkable record of freedom from trouble caused by corrosion. During this period numerous buildings torn down or remodeled revealed that the piping removed from the heating system, including both steel and iron, was in a good state of preservation and outlasted the serviceable life of the building by a wide margin. Therefore, since the functional service of pipe in a radiant system is identical with that of a regular pipe and radiator system, the factor of corrosion can be regarded as an item of no consequence. Since the introduction of radiant systems in this country, various kinds of pipe have been employed, but the predominant tonnage has been of wrought steel and

E. C. Hall Company Building at Tigard, Oregon, radiant heated with NATIONAL Steel Pipe.

wrought iron. Both of these materials have served with equal satisfaction and substantiate the previous records developed through many actual service tests and investigations of the relative behavior of steel and iron in hot water service.

Similarly with respect to external corrosion, when pipe is buried in concrete slabs, external corrosion is not of any greater consequence than in using steel reinforcing bars.

When installed in wood construction floors or ceilings, the temperature of the pipe coils being higher than that of the surrounding air, condensation is not a factor and corrosion does not occur.

Q. 13. ***How do the bending properties of wrought steel pipe compare with those of wrought iron pipe?***

A. All tables which list the shortest radius to which pipe may be bent, indicate a shorter permissible radius for wrought steel pipe than for wrought iron pipe. However, both steel and iron pipe may be bent to the radii generally required in radiant heating coils without affecting the strength of the pipe material adversely.

Q. 14. ***Having a broadloom carpet laid wall to wall over a standard type rug pad, all directly on the concrete heating slab, what would be the effect on the heat transfer?***

A. Mr. Elmo Hall, prominent heating engineer, has investigated this matter and his findings are as follows:

"This question seems to worry many engineers who know that rugs, furniture, files, and other equipment will be placed on or adjacent to their floor heating panels. Imagine, then, the author's surprise to find the entire floor space of a hangar heated by floor panels covered for several months with insulating wooden floor panels designed as a base for buildings used in arctic operations. These wooden panels, 4′×8′ in size, consist of a 2×4 frame covered with $\frac{7}{8}$" matched flooring on one side, $\frac{1}{2}$" laminated panels on the other with the space between the 2×4 joists packed with loose insulating material. In the hangar in question, the concrete floor is 4 inches high in the center of the hangar and the wooden panels touched the concrete only at that point being wedged up toward the outside walls to bring them level, thus providing another insulating air space.

The Arsenal Junior High School, Pittsburgh, Pa., open-air rooms radiant heated.

"It seems impossible to devise a more severe insulating test for a radiant floor panel, yet the thermostats located above the insulating panels showed the same temperature as before the insulating panels were placed on the floor with a rise in concrete temperature from 72° F. to 86° F. and with the *top* of the insulated panels the same temperature as the floor had been previously. The panels were in place two months before the author found them and, while there must have been a time lag in again bringing the building up to design temperature, several of the personnel working in the space who were questioned could shed no light on the lag and stated that if such lag occurred, they had not been aware of it. Since witnessing this accidental test, the author will never again worry over a few rugs or pictures placed on or in front of a heating panel."

Q. 15. ***How is aluminum foil used to prevent heat loss?***

A. Regarding the use of aluminum for insulating purposes, the following is quoted from a report of the U. S. Bureau of Standards Letter Circular LC535:

"Since the principles involved in the use of aluminum foil or other bright metal sheet as thermal insulation are not generally understood, a brief discussion will be given here. Aluminum foil is used to increase the insulating value of air spaces by reducing heat transfer by radiation. It is of value only in conjunction with air spaces, and has no value when placed in continuous contact with solid material on both sides, except in so far as it may act as a building paper in preventing air leakage.

"Clean metallic surfaces in general are good reflectors and poor emitters of radiant heat. Since a large proportion of the heat transfer across air spaces bounded by non-metallic materials takes place by radiation, the use of aluminum as one or both boundaries of a space will materially reduce the heat transfer across the space. It will be evident that the insulating effect does not depend on the thickness of the metallic foil, while the insulating value of ordinary types of insulating materials depends mainly on their thickness. The insulating value of air spaces bounded on one or both interior surfaces with aluminum foil increases with increasing width of space up to about 3/4-inch width. Spaces wider than about 3/4-inch have substantially the same insulating value, regardless of width.

"While there is limited information as to the permanence of the reflective surfaces of aluminum under various conditions of use, such information as is available indicates that under normal conditions the reflectivity is likely to be reasonably permanent. Installations are reported where no appreciable deterioration of the aluminum has occurred over a considerable period of years. Thin layers of dust readily visible to the eye do not cause any very serious lowering in the reflecting power. If aluminum is wetted over considerable periods of time, there is possibility of corrosion, particularly if the water is alkaline. The appearance of the surface is not a reliable guide as to its reflectivity for radiant heat, and foil which appears dark or discolored may have lost little in insulating value if the surface film is thin.

"The use of lacquer to resist possible corrosion under severe conditions of use reduces the reflecting power to some extent. The effect of a very thin coat of lacquer is small, but relatively thick lacquers, even though they are almost invisible to the eye, may seriously reduce the effectiveness of the foil.

"The effect of reduced reflectivity on heat transfer across an air space is less marked the narrower the space, since heat transfer by conduction and convection plays a more important role than radiation in the case of narrow air spaces.

"Aluminum foil is also applied in a crumpled form so that it is self-spacing. If two or three crumpled sheets are hung in the air space of a frame wall, there is so little contact between the sheets that the insulating values are essentially the same as those given for the spaced sheets."

Q. 16. *Are floors of radiant heated rooms uncomfortably warm?*

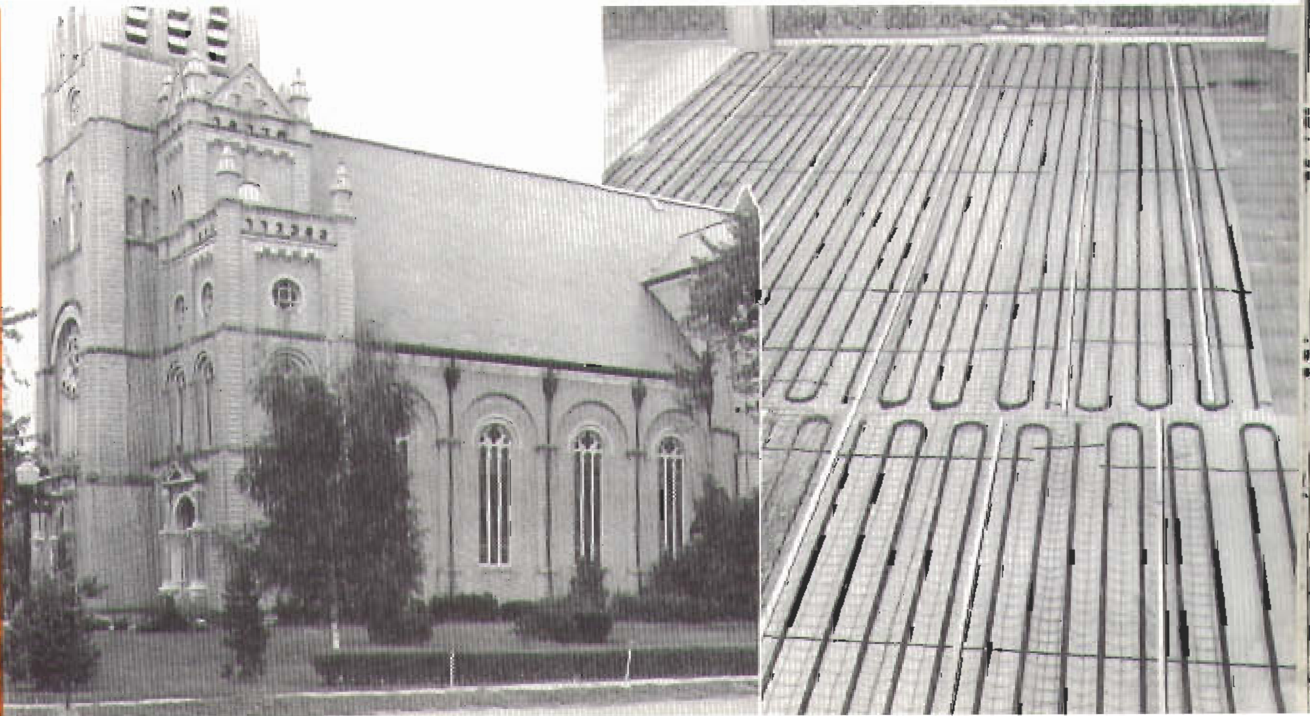
A. No. The surface temperature of a properly designed floor type radiant heating installation is between 75 and 85 degrees, and experience has shown that when designed for this range of temperatures, comfortable conditions are found to exist.

The Heating, Ventilating and Air Conditioning Guide 1945 lists the following as desirable heating surface temperatures:

Highest Safe Surface Temperatures for Heating Panels

Type of panel	Surface temperature degree F.
Plastered Ceiling (Pipe embedded)	115
Plastered Walls (Pipe embedded)	120
Floor, any method	90
Floor, Border, and Aisles	120

Low surface temperature radiation is recommended regardless of the heating medium employed.



St. John's Church in Delphos, Ohio, is a good example of a hard-to-heat building made comfortable with radiant heating. The auditorium in this beautiful old stone church is 135 feet x 65 feet. The vaulted ceiling is 60 feet high. Here was a natural for radiant heating.

Methods of Floor and Ceiling Installation of Radiant Heating Coils

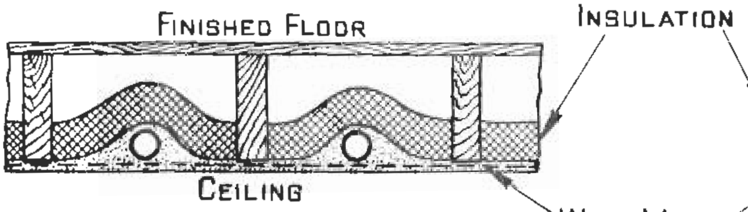


Figure 1. The radiant heating coil is laid on lath which is attached to the joists in the usual manner. Rock wool or any other loose insulating material is then laid over the pipe between joists.

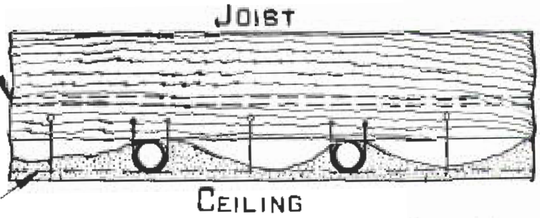


Figure 2. The coil is hung below the joists with the metal mesh lathing wired to the pipe. Plaster is then applied to the lath and covers the pipe as indicated.

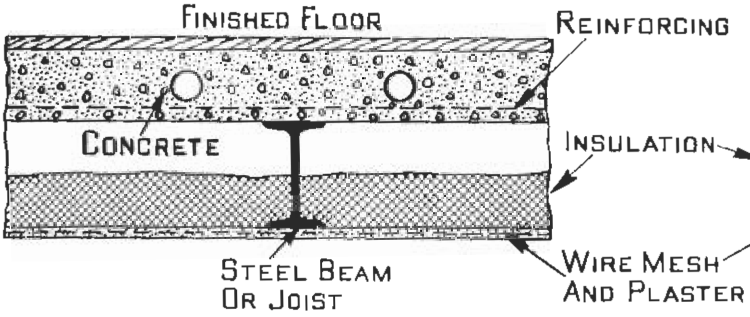


Figure 3. Radiant heating coil is shown embedded in concrete floor panel over reinforcing supported by steel beam or joist.

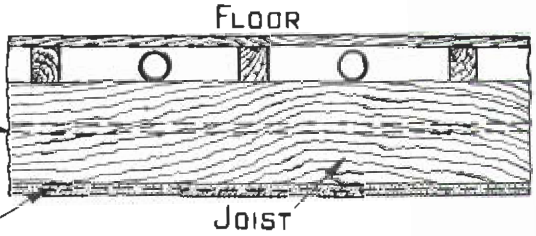


Figure 4. The coil is laid on top of joists with wood flooring, supported by sleepers, laid over coil. It is necessary in this case to use some type of insulating board between the joists as shown. This board is supported by stops nailed to the joists.

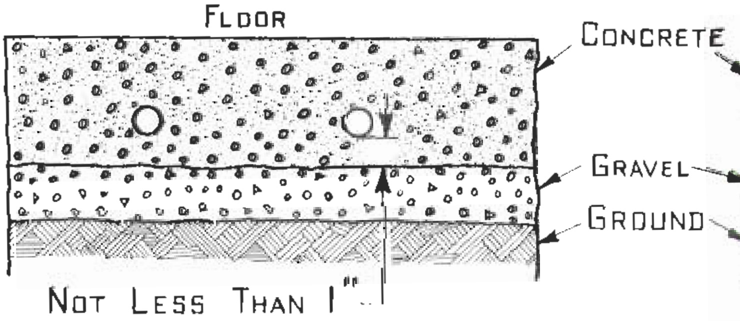


Figure 5. Radiant heating coil is shown embedded in concrete which has been poured on a bed of packed gravel.

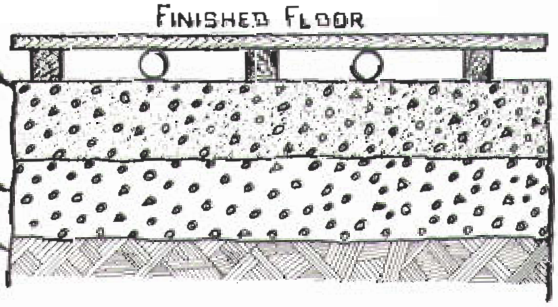
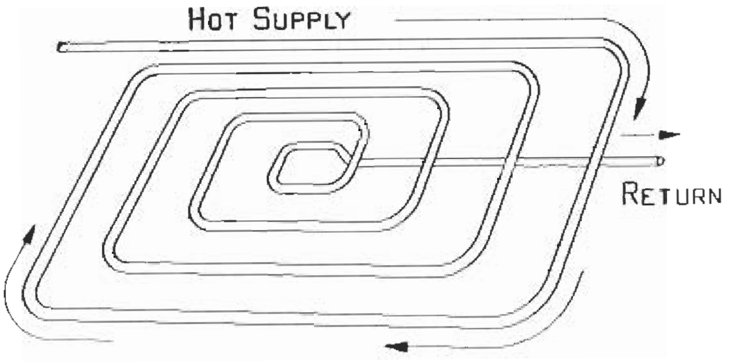


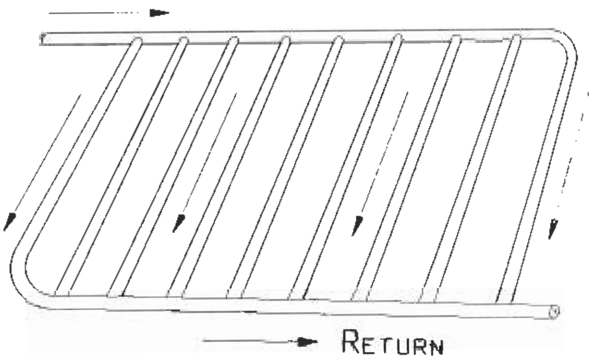
Figure 6. Coil has been laid on top of concrete, and wood flooring, supported by sleepers, laid over the coil.

One of several ne
buildings of Dres
Manufacturing Di
sion, Bradford, Pa
radiant heated. A
unusual feature of this
installation was the
use of plain end pip
couplings of their ow
manufacture, for join
ing a considerable pro
portion of the piping

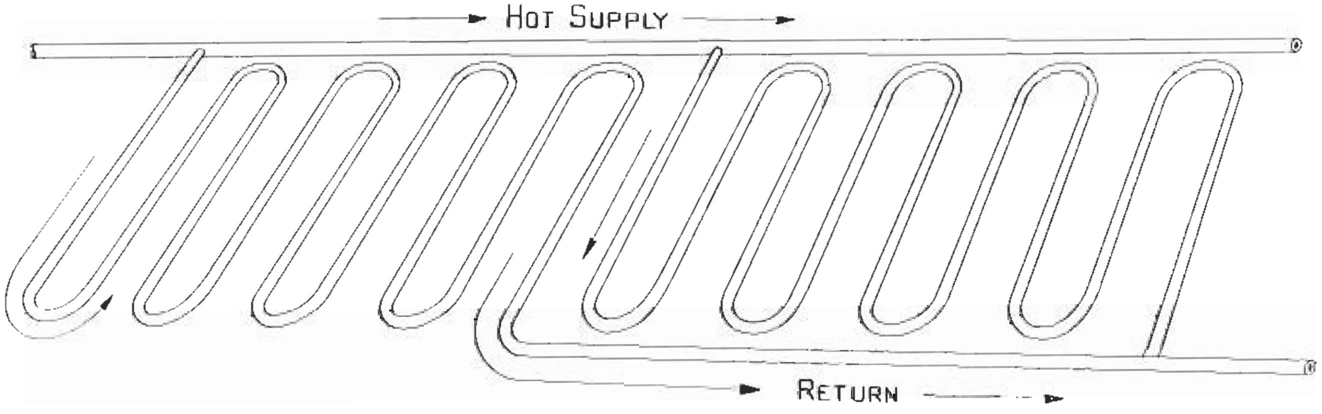
Typical Coil Patterns



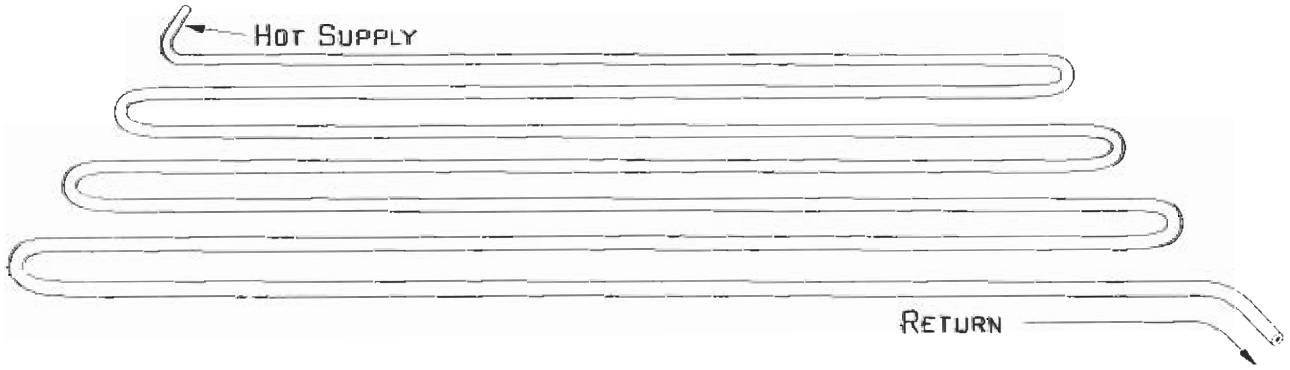
SQUARE COIL
HOT LINES ON ROOM EXTERIOR
COOLER IN CENTER



GRID COIL
MINIMUM BENDING REQUIRED



CONTINUOUS COILS IN PARALLEL
LOW FRICTION HEAD LOSS



CONTINUOUS COIL
MINIMUM WELDING REQUIRED

Snow Melting

THE OLD "SHOVEL BRIGADE" is fast losing its place as a method of disposing of snow. Today, the modern and more efficient Snow Melting System is being increasingly used throughout snowfall areas of the United States.

Residential sidewalks, private driveways, roadways into industrial plants, sidewalks before theater ticket offices, church entrances, train platforms, bus terminals, playgrounds—all lend themselves to this method of snow removal.

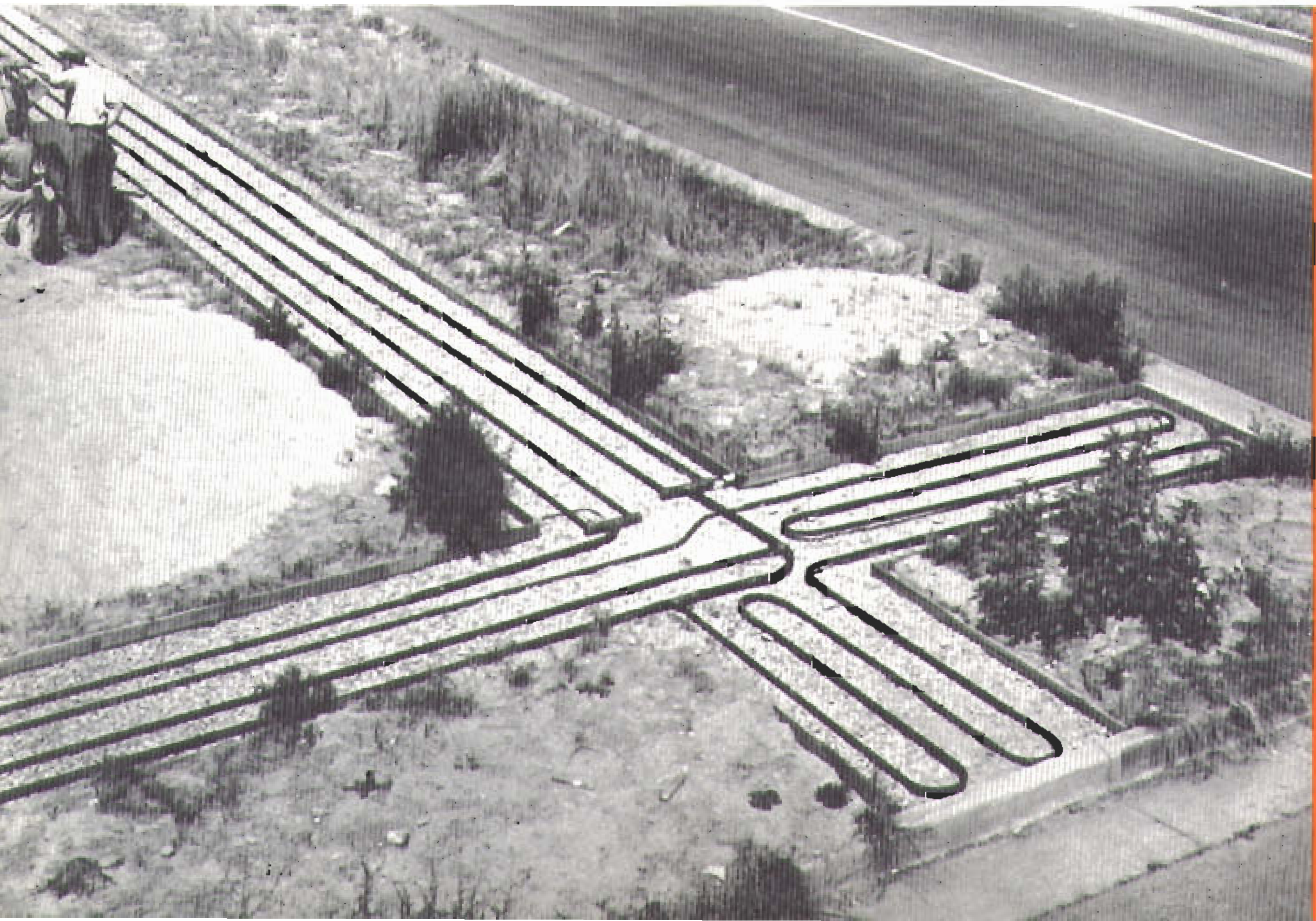
Many of our large office buildings and department stores now have steel pipe coils, similar in pattern to those used in radiant heating systems, installed in their sidewalks for snow melting purposes, and the results, both from an economic and psychological standpoint, are highly satisfactory.

ADVANTAGES OF SNOW REMOVAL SYSTEMS

If you are one of those fortunate home owners who has a snow melting system, you already know that "grand and glorious" feeling when you take that early morning peek out the window to size up the day's weather and find that a white blanket of snow lies everywhere except on your sidewalk and driveway. Or if the weather crossed up the predictions and the snow unexpectedly covers everything, it's a simple matter of moving a thermostat and your snow removal system starts to work.

Or picture the sidewalk in front of a department store during shopping hours; it is snowing heavily, yet no snow appears on the sidewalk. The melting system is at work. Shoppers are free to view the merchandise displayed under safe and agreeable conditions, even though surrounding areas are covered with snow and ice, and the hazards they create. Customer interest in the merchandise displayed is bound to be greater—there are no interrupting thoughts of discomfiture and quick departure.

A snow melting system frees the surface of all snow, eliminating the possibility of a dangerous spot or film of ice which often remains when other snow removal methods are used—a potential accident hazard and attendant claims of lawsuits.



Sun Age Homes, Denver, Colorado, radiant heating for interior and snow melting system for driveway and sidewalks.*

---

*Trade Mark, Reg.

Consider also the multiple advantages of such a system for the sidewalks and entrances of large office buildings, hotels, theatres, and public buildings in general. Snow-free approach surfaces eliminate the unsightly conditions so often seen on snowy days when pedestrians cannot help carrying in wet and dirty snow to discolor floors and carpets, plus the maintenance cost of repeated cleanings during the day. Bus terminals, railway stations, and similar places where large numbers of persons frequently congregate are particularly advantageous locations of snow melting systems, both outside and inside the structure. Another promising use of these systems is to provide comfort and safety for both attendants and customers at gas service stations. A clean, safe, attractive station invites patronage and creates good will.

Again, the driveways and loading platform approaches of industrial plants can be protected against delays in product deliveries and possible injury to personnel by the installation of a snow melting system, while at the same time eliminating the need for snow removal equipment and its attendant maintenance and operating costs.

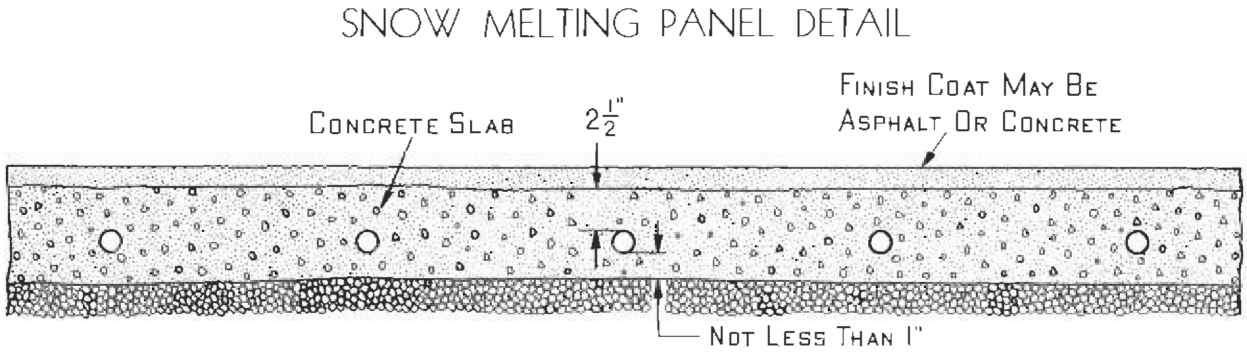


Figure 21. Total depth of concrete and bed of crushed rock or sand to be in accordance with installation requirements. Spacing ¾-inch pipe on 12-inch centers. 1-inch pipe on 16-inch centers.

DESIGN OF A SNOW MELTING SYSTEM

In designing a snow melting system the heat input required is governed by several fundamental factors. For instance, the amount of heat lost to the ground—and by the edge—varies with soil conditions below the melting system. There is also a considerable fluctuation in the amount of heat lost from the melting surface to the air, being greatest when snow is falling lightly and air temperatures are well below freezing, and least at freezing temperatures when sleet turning to snow packs on the pavement or driveway surface. Where the more severe temperatures are encountered, heat losses may be reduced by the use of suitable insulation beneath the slab in which the coils are imbedded. In narrow walks, placing the pipe as far away from the edges as practicable will also reduce the total losses, since it is at the edges that the loss is greatest.

Concerning the quantity of heat required, experience has shown that standard butt-weld steel pipe installed in panels in the concrete as shown in Figure 21 below will be sufficient to melt any fall of snow, provided the boiler supplying the system is adequately sized.

The illustration offered is intended as an example of accepted practice to meet the conditions mentioned. Variations naturally exist between different sectional locations and the circumstances surrounding a specific installation. The designing engineer will adjust the details in the specifications for the job to make the most efficient and economical system. In general, care should be used to see that the pipe is completely surrounded by the concrete. This all-round contact assures immediate and more efficient transmission of heat to the slab and quicker action on the snow. It is particularly desirable to have at least 1-inch and preferably 2 inches of fine concrete mix under the pipe to prevent even occasional surges of ground water from coming into contact with the pipe. A coarse, dry mix may prevent the soft or semiliquid part of the concrete from completely bonding with the pipe—a most important factor from several standpoints. The pipe should never be laid directly on the ground or in contact with cinders or other acid-creating materials.

There are two conditions encountered in snow removal; first, where the snow is melted as it falls, which would be particularly desirable on sidewalks and entrances to public buildings; and, second, where snow has fallen and packed while the system is idle.

It is far more desirable to have the system in continuous operation when the temperature approaches freezing and weather predictions are for snow or sleet. During this period the circulating medium should be maintained at a temperature between 60 and 80 F. Under such conditions the heat loss from the melting surface would be about 50 B.T.U.'s per hour per square foot of surface.

The benefits derived from having the system in operation before snow falls would be the immediate melting of snow when it does fall, thus preventing hazardous freezing conditions and more economical removal than for heavy-packed snow.

The temperature of the heating medium can be lowered and heat preserved when snow is not predicted.

The heating medium should be a mixture of antifreeze and water, such as 50% ethylene glycol and 50% water.

ESTIMATING HEAT LOSS

The chart, Figure 22, Page 39, gives the heat required to melt 1-inch of snow per square foot of surface and at various air temperatures. To this must be added the designer's estimate of losses to ground and atmosphere. The weight of snow at various temperatures is given on the curve on page 12.

ESTIMATING HEAT REQUIREMENTS FOR SNOW MELTING SYSTEMS

The heat required to melt snow as it falls is given by equation (a).

$$\text{(a)} \quad H = \frac{rf}{e}$$

where H = Btu/hr

r = Rate of snowfall, lb./hr.

f = Heat of fusion, Btu/lb.

e = Slab efficiency

(The sensible heat required to raise the snow temperature to 32 F has been neglected.) The heat of fusion for ice is 144 Btu/lb. so equation (a) becomes

$$\text{(b)} \quad H = \frac{144r}{e}$$

At the present time there are little data for slab efficiency. It is known, however, that heat can be

Snow melting coils under this outdoor terrace of Home for Convalescents keep the terrace free of snow and comfortable for longer periods.

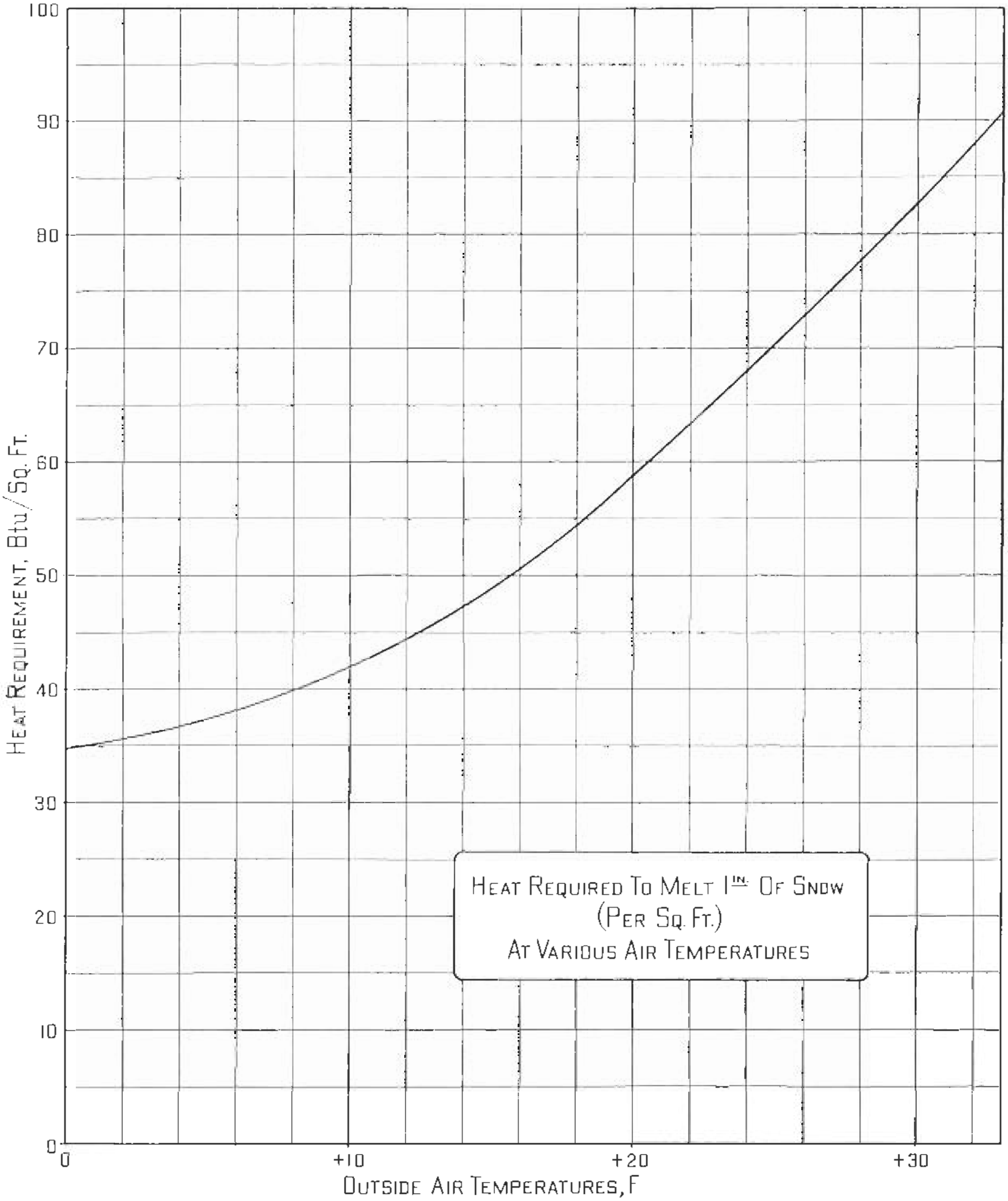


Figure 22

Figure 22 is given for a quick check of the heat required to melt snow; no allowance has been made for edge or back losses. Example—If air temperature is +10 F and 1 inch of snow covers the slab, 42 Btu/sq. ft. will be required to clear the slab.

lost to the ground—from the edges and back—and to the air above the slab. Some of this convection loss may be regained, but this regain is also unknown. Generally slab efficiency is taken as 0.5 for uninsulated slabs, and increased as the insulation is increased.

The values for r in equation (b) can be determined from accompanying table, Rate of Snowfall. This is the first publication of this data in the manner shown in the table and, therefore, some explanation is given.

Basically, this is a table of frequency distribution. The entrys are the *number of observations* taken at six-hour intervals in which the specific amounts of precipitation were measured. For example, the observer measured a fall of 0.00—0.24 inches of water equivalent, 2052 times in Albany. The total readings in the table were taken between November 15th and February 15th of each year during 1940—1949 inclusive. In the cities where less than 3720 observations were made, the full 10-year period was not completed.

The term "water equivalent" is rarely used in the heating industry, but it is the only unit used by the U. S. Weather Bureau to measure snowfall. Actually water equivalent is an excellent unit for determining snow melting heat requirements because it is a unit of weight. One inch of water equivalent equals $\frac{62.4}{12} = 5.2$ lbs. sq. ft. of water at 32 F. In other words, regardless of the density of snow, water equivalent measures the heat content (or lack of it) in the snow. It is possible to change equation (b) so that it will contain inches of water equivalent instead of lbs/hr.

$$\text{(c)} \quad H = 144 \times \frac{5.2i}{e}$$

$$= \frac{750i}{e}$$

where H = Heat output, Btu/hr. sq. ft.

i = Rate of snowfall, inches of water equivalent per hr.

e = Slab efficiency.

In selecting the proper rate of snowfall, i, it should be remembered that record snow storms often paralyze a city; therefore, the snow melting system in a business area may not be used. Generally, it will be permissible to have the system under-designed for one storm a year, i.e. designed to take care of approximately 98% of all conditions encountered. It is not economical to have a system of such capacity that it would be operated at full load only once in several years. Since the table is based on a 10-year period, it may be permissible to ignore the 10 top rates in the table. Take Albany for an example; by dropping the six highest records (columns 4 and 5—combined—6), the design rate would seem to be 0.49 inches of water equivalent per six hours. If the snow fell uniformly, $i = \frac{0.49}{6}$ = 0.08 inches of water equivalent per hour. If, though, the snow fell in flurries, in fact all in one hour, the rate would be higher than 0.08. By examining the record falls for Albany, Buffalo, and New York, shown in lower table, page 41, one can readily see that flurries raise the average rate of fall, but they do last for several hours. It seems reasonable to assume that the rates shown in the Frequency Distribution table represent a three-hour instead of a six-hour fall. For Albany then, equation (c) would be

$$H = \frac{750}{e} \times \frac{0.49}{3} \text{ (for Albany),}$$

for an uninsulated slab in Albany

$$H = \frac{750}{0.5} \times \frac{0.49}{3} = 250 \text{ (approx.) Btu/hr./sq. ft.}$$

RATE OF SNOWFALL[1] (From U. S. Weather Bureau Data)

In Inches of Water Equivalent

City	Frequency Distribution[2]				Total Readings Taken	Heaviest Fall[3] in 24 Hours (For entire history of Weather Bureau) Inches of Snow
	0.00 0.24	0.25 0.49	0.50 0.74	0.75 0.99		
Albany, N.Y.	2052	29	5	1	3720	30.4
Asheville, N.C.	463	5	1	0	3536	15.8
Billings, Mont.	1640	4	0	0	3532	16.6
Bismarck, N.D.	2838	0	0	0	3720	12.0
Boise, Idaho	1300	3	0	0	3720	13.0
Boston, Mass.	1323	11	4	2	3720	16.5
Buffalo, N.Y.	1871	23	3	1	3720	24.3
Burlington, Vt.	2390	9	0	0	3720	24.2

RATE OF SNOWFALL[1] (From U. S. Weather Bureau Data)—*Concluded*
In Inches of Water Equivalent

City	Frequency Distribution[2]				Total Readings Taken	Heaviest Fall[3] in 24 Hours (For entire history of Weather Bureau) Inches of Snow
	0.00–0.24	0.25–0.49	0.50–0.74	0.75–0.99		
Caribou, Maine	1363	19	1	0	1672	17.4
Chicago, Ill.	1498	3	0	1	2976	14.9
Cincinnati, Ohio	1045	3	0	0	3720	11.0
Cleveland, Ohio	1569	2	0	0	3720	17.4
Columbus, Ohio	1351	4	1	0	3720	11.9
Denver, Colorado	1207	4	0	0	3720	23.0
Detroit, Michigan	1830	5	2	0	3720	24.5
Evansville, Ind.	916	5	1	1	3720	20.0
Hartford, Conn.	1514	14	9	3	3720	19.0
Kansas City, Mo.	1189	12	2	1	3720	25.0
Madison, Wisconsin	2370	5	2	0	3720	12.9
Minneapolis, Minn.	2703	7	0	0	3720	16.2
New York, N.Y.[4]						25.8
Oklahoma City, Okla.	613	8	4	0	3720	11.3
Omaha, Nebraska	1795	8	1	0	3720	16.4
Philadelphia, Pa.	891	10	2	1	3720	21.0
Pittsburgh, Pa.	1365	6	2	0	3720	20.1
Portland, Maine	2054	33	4	1	3720	23.8
St. Louis, Mo.	1088	5	0	1	3720	20.4
Salt Lake City, Utah	1482	5	0	0	3720	15.3
Spokane, Wash.	1545	11	1	0	3720	10.6
Washington, D.C.	533	7	2	1	3348	25.0

[1] Rate of snowfall, as shown in this table, refers to inches of water equivalent per six hours. The records were obtained at 1:30 A.M., 7:30 A.M., 1:30 P.M., and 7:30 P.M. from November 15th to February 15th during the years 1940 to 1949 inclusive. (For some cities such as Asheville, N.C., where the total readings were less than 3720, the readings are not for the full 10 years.) The difference between the sum of the frequencies and the total readings is the number of times the maximum temperature in the six-hour period was above freezing.

The units for the rate of snowfall, as shown in the column headings for frequency distribution are inches of water equivalent per six hours. In other words, the observer melted the precipitation and measured the depth of water. For Albany, only once was the water 0.75 inches or more in depth after a six-hour fall. If, however, the records had included the Albany storm of December, 1915, the observer would have measured 1.33 inches on his 1:30 P.M. reading.

[2] Frequency Distribution, as shown in this table, refers to the number of observations at six-hour intervals in which the amounts of precipitation were measured when the maximum air temperature in the period was below freezing. The purpose of imposing this temperature restriction on the data is that *any* precipitation, rain, snow, or sleet that falls during below-freezing temperatures will require melting. On the other hand, any precipitation falling at above-freezing temperatures will melt of its own accord.

[3] Maximum snowfall in 24 hours is given as a comparison to the record storms listed for Albany, Buffalo, New York, and Pittsburgh.

[4] New York City record was not used in this tabulation since the records for that station were not comparable with those of the other stations.

HEAVIEST SNOWFALL IN 24 HOURS
(From U. S. Weather Bureau Data)

	Rate of Snowfall—Inches of Water Equivalent per Hour																									
	Hour of Storm																								Total Water Equiv.	Total Inches of Snow
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24		
Albany[†] 13 Dec. 1915 start at 9:30 A.M.	.01	T	.01	.05	.09	.27	.27	.27	.27	.16	.21	.09	.09	.09	.09	.08	.08	.08	.02	.02	.02	.02	.01	.01	2.31	24.7
Buffalo 15 Dec. 1945 start at 3:00 P.M.	.05	.01	.07	.13	.21	.21	.21	.05	T	0	0	.01	.01	.01	.01	.01	.13	.13	.13	.13	.13	.13	.20	.19	2.16	24.3
New York 26 Dec. 1947 start at 3:00 A.M.	.01	.02	.06	.09	.11	.15	.15	.20	.21	.22	.19	.16	.39*	.27*	.16*	.09*	.05*	.02	.01	.01	.01	.02	.03	.01	2.67	25.8
Pittsburgh[‡] 24 Nov. 1950 start at 8:00 A.M.	.05	.05	.05	.08	.10	.08	.03	.06	.04	.10	.12	.08	.10	.10	.12	.09	.01	.02	.02	.09	.06	.05	.17	.23	1.90	20.1

*Combination of sleet, rain, or snow (temperature 26—29F)

[†] In March, 1888 Albany had a snowfall of 30.4 inches but no hourly records are available.

[‡] 20.1 inches is Downtown office record but future records will be considered at 17.5 inches from Airport office.

FRICTION LOSS

The friction loss per 100 feet of pipe may be obtained by taking twice the friction loss shown in Figure 15, page 19. There is more hydraulic friction to a mixture of ethylene glycol and water than there is to water alone. It is sufficiently accurate for snow melting systems to assume that this added friction is twice the friction for water.

TESTING

After installation and before pouring concrete all piping should be tested to about 250 psi pressure, which should be maintained until all welds and connections have been checked for leaks.

AIR VENTING

An arrangement should be made to vent the air by installing a vent at a high point in the system just as is done in regular radiant heating installations.

CONTROL OF SYSTEM

The control of a snow melting system can be either manual or automatic. There are several automatic devices on the market, and the designer will know when they should be used and how best adapted to the particular installation.

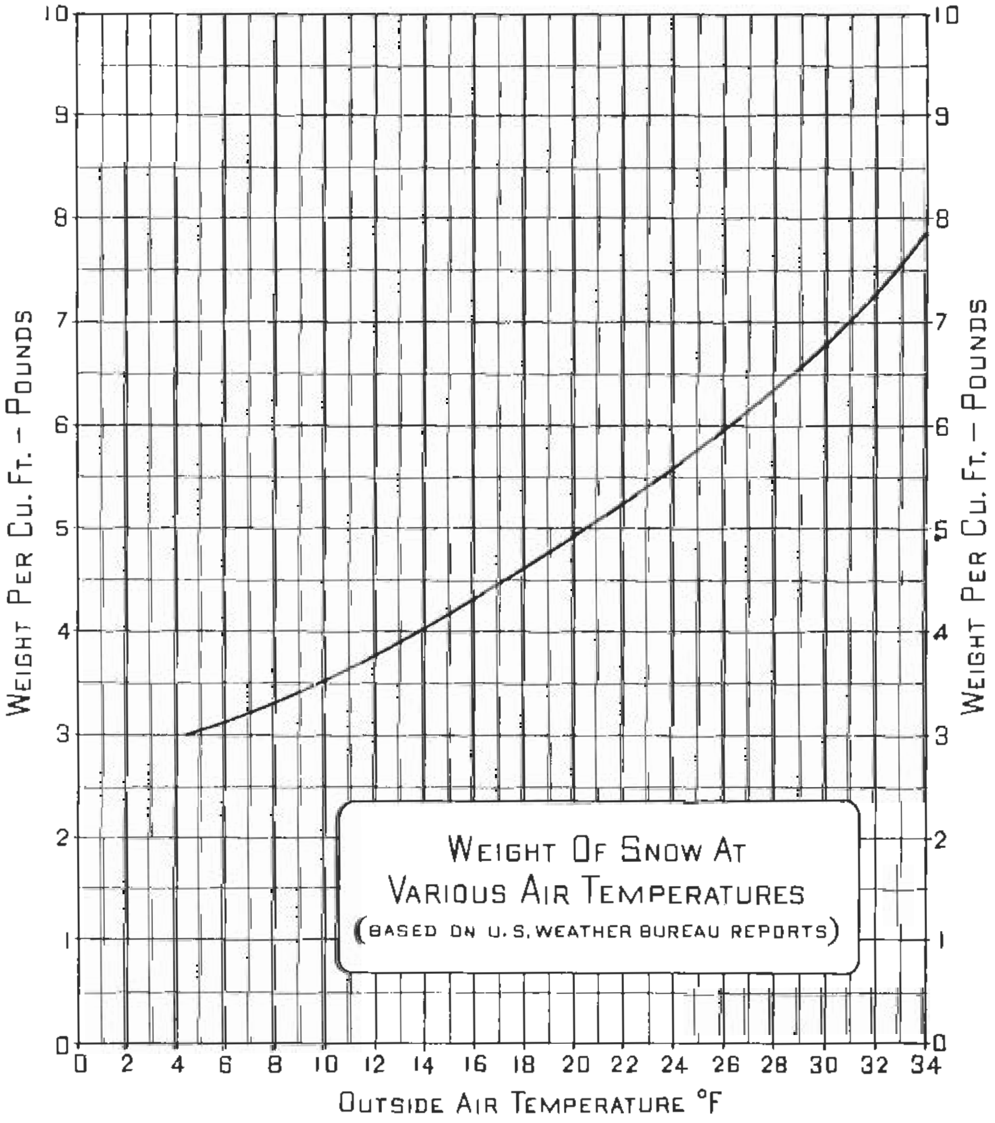
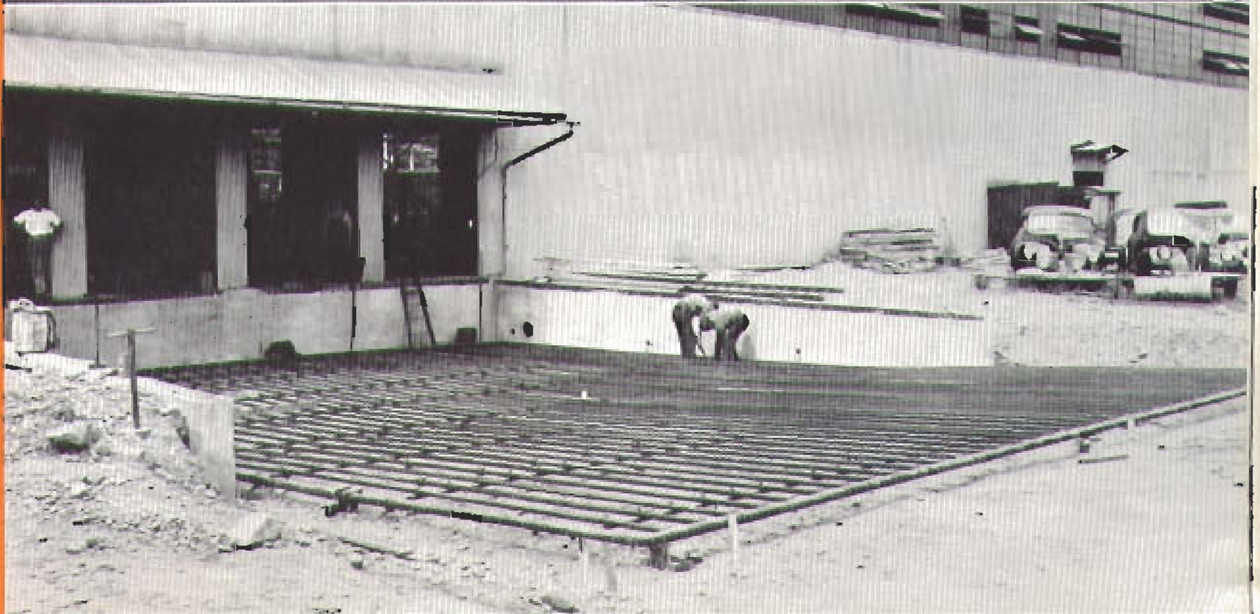


Figure 23

COMBINATION OR SEPARATE SYSTEMS

Steam heating systems are usually installed with some excess capacity above normal requirements. It is quite likely therefore that, in many instances, the capacity of present heating equipment will permit the incorporation of a heat exchanger and an additional circuit to effectively operate a snow melting and ice removal system. A separate system is advisable where the snow melting area involved is of considerable extent or where the area is of considerable distance from the heat source. These considerations apply more particularly to existing installations. Where new construction is involved, the problem is somewhat less complicated and the design engineer will readily determine whether a separate system or allowance in boiler capacity of the regular equipment will be more efficient. If an additional circuit is incorporated into the regular system for the snow melting job, this circuit should operate separately from other circuits to prevent the antifreeze solution from reaching the regular heating lines, as this would lower the efficiency of the indoor part of the system, as well as creating greater than necessary expense for the antifreeze.

This modern warehouse and operating building of The Peoples Natural Gas Company, Pittsburgh, Pa. incorporates a steel pipe radiant heating system for warmth, and protection of delicate instruments, and also a snow removal system under the sidewalks and driveway.



Snow melting system in a ramp at one of the Dresser Manufacturing Division buildings, several of which are also radiant heated.

NATIONAL Pipe for Radiant Heating and Snow Melting

Whether for use in the home, office, or industrial building, pipe for radiant heating and snow melting systems should be selected on the basis of its inherent characteristics to meet the several requirements for this particular service, and also, on its record of past performance in similar service.

The heating engineer or contractor responsible for the design and installation will rightfully demand pipe that has the necessary physical properties to make smooth, uniform bends and with a minimum of difficulty in the fabricating operation.

He will require also that the pipe used will have good welding quality to assure strong, sound welds, and to save both time and labor in welding the installation.

He will want pipe that offers a minimum of frictional resistance to assure that his design calculations will be translated into actual service performance.

Most important of all, he will want pipe that, on the basis of practical research and the infallible test of time and service, offers a life expectancy, under normal operating conditions, equal to the serviceable life of the building itself.

Obviously, if the pipe selected has been made by a concern with wide experience in the field of domestic and industrial hot water supply piping, and whose resources in materials, manufacturing equipment, and research facilities are without equal, such pipe, produced under these ideal conditions, offers the best assurance that it will meet any requirement for radiant heating service.

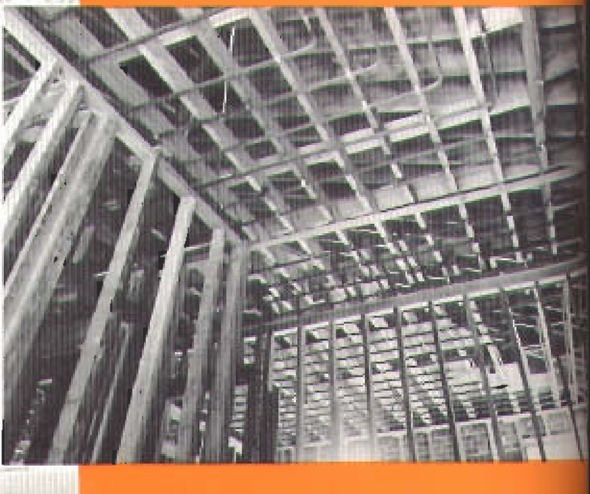
The same NATIONAL advantages that have helped establish steel pipe as the nation's standard for conventional hot water and steam heating systems are all available for radiant heating systems. You can still get the plus value of the Scale Free and Spellerizing Processes, special NATIONAL Pipe features. This means the interior surface of NATIONAL Pipe is smooth, free from mill scale, with minimum frictional resistance. It means also that even though corrosion in radiant heating is an inconsequential factor, as it is in conventional heating systems, these processes, nevertheless, give the pipe maximum corrosion resistance. It means, further, that the weld strength is increased approximately 20 per cent by the extra rolling, which helps give NATIONAL its well-known bending properties.

Still another NATIONAL advantage for radiant heating is that steel pipe is easy to weld—a factor that helps reduce cost on the job and speeds up installation. And, like steel reinforcing bars, it expands at the same rate as concrete and plaster, thus adding strength to the structure.

Whatever measure of value you use to determine the ideal in pipe for radiant heating systems, you will find NATIONAL possesses more practical points or advantages than any other pipe employed for this type of service.

Residence of Elmo Hall, Denver, Colorado. Radiant heating with NATIONAL Steel Pipe installed throughout.

Ceiling coils in same residence.



Grid-type panels of National Pipe for project of private garages in Staunton, Virginia.

PHYSICAL PROPERTIES OF NATIONAL STANDARD PIPE FOR RADIANT HEATING

Minimum Yield Strength, 30,000 pounds per square inch.

Minimum Ultimate Strength, 50,000 pounds per square inch.

The center to center spacing required in radiant heating coils is such that excellent bends may be obtained with many portable bending machines available. The table of radii to which pipe may be bent is given as a guide for those sections of a heating system other than in the coils where shorter bends are required.

Radii to which Pipe May Be Bent

Pipe size	Minimum advisable radius of bends—R	Shortest radius to which pipe can be bent
½	2½	1½
¾	3¾	1¾
1	5	2
1¼	6¼	2¼
1½	7½	2½
2	10	3

All dimensions in inches.

The radius of pipe bends preferably should equal or exceed the dimensions in the column defining minimum advisable radius.

NATIONAL STANDARD PIPE—Black and Galvanized

All weights and dimensions are nominal

Size: nominal	Weight per foot	Pipe		
	Plain end	Thickness	Diameters	
			Outside	Inside
Ins.	Lbs.	Ins.	Ins.	Ins.
⅛	.24	.068	.405	.269
¼	.42	.088	.540	.364
⅜	.57	.091	.675	.493
½	.85	.109	.840	.622
¾	1.13	.113	1.050	.824
1	1.68	.133	1.315	1.049
1¼	2.27	.140	1.660	1.380
1½	2.72	.145	1.900	1.610
2	3.65	.154	2.375	2.067
2½	5.79	.203	2.875	2.469
3	7.58	.216	3.500	3.068
3½	9.11	.226	4.000	3.548
4	10.79	.237	4.500	4.026
5	14.62	.258	5.563	5.047
6	18.97	.280	6.625	6.065

RELATIVE DISCHARGING CAPACITIES OF NATIONAL STANDARD PIPE

Pipe size	Internal diameter D	$D^{5/2}$	Pipe size								
Ins.	Ins.		1/8	1/4	3/8	1/2	3/4	1	1 1/4	1 1/2	2
1/8	.260	.037530	1.0	...	...	...	...	...	...	...	...
1/4	.364	.079938	2.1	1.0	...	...	...	...	...	...	...
3/8	.493	.17065	4.5	2.1	1.0	...	...	...	...	...	...
1/2	.622	.30512	8.1	3.8	1.8	1.0	...	...	...	...	...
3/4	.824	.61634	16	7.7	3.6	2.0	1.0	...	...	...	...
1	1.049	1.1270	30	14	6.6	3.7	1.8	1.0	...	...	...
1 1/4	1.380	2.2372	60	28	13	7.3	3.6	2.0	1.0	...	...
1 1/2	1.610	3.2890	88	41	19	11	5.3	2.9	1.5	1.0	...
2	2.067	6.1426	164	77	36	20	10	5.5	2.7	1.9	1.0
2 1/2	2.469	9.5786	255	120	56	31	16	8.5	4.3	2.9	1.6
3	3.068	16.487	439	206	97	54	27	15	7.4	5.0	2.7
3 1/2	3.548	23.711	632	297	139	78	38	21	11	7.2	3.9
4	4.026	32.523	867	407	191	107	53	29	15	9.9	5.3
5	5.047	57.225	1525	716	335	188	93	51	26	17	9.3
6	6.065	90.589	2414	1133	531	297	147	80	40	28	15

Pipe size	Internal diameter D	$D^{5/2}$	Pipe size								
Ins.	Ins.		2 1/2	3	3 1/2	4	5	6	8	10	12
2 1/2	2.469	9.5786	1.0	...	...	...	...	...	...	...	...
3	3.068	16.487	1.7	1.0	...	...	...	...	...	...	...
3 1/2	3.548	23.711	2.5	1.4	1.0	...	...	...	...	...	...
4	4.026	32.523	3.4	2.0	1.4	1.0	...	...	...	...	...
5	5.047	57.225	6.0	3.5	2.4	1.8	1.0	...	...	...	...
6	6.065	90.589	9.5	5.5	3.8	2.8	1.6	1.0	...	...	...

The figure opposite the intersection of any two sizes is the number of smaller size pipes required to equal one of the larger.
Example: How many 1-inch pipes will it take to equal the discharge of one 1¼-inch pipe?
Solution: The figure in the table opposite the intersection of these two sizes gives two 1-inch pipes.

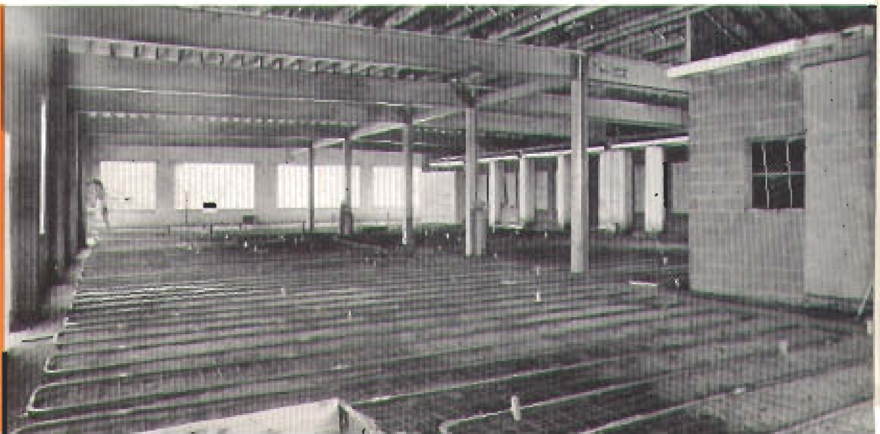
This attractive ranch-type house in Weston, Massachusetts, is further enhanced in "home appeal" by the use of modern radiant heating—floor-type panels divided into two zones.

NATIONAL STANDARD PIPE—INTERNAL PROPERTIES

Size	Diameters		Thickness inches	Circumference inches	Surface/Lineal foot	
	Outside inches	Inside inches			Square inches	Square feet
⅛	.405	.269	.068	.8451	10.141	.0704
¼	.540	.364	.088	1.1453	13.722	.0953
⅜	.675	.493	.091	1.5488	18.586	.1291
½	.840	.622	.109	1.9541	23.449	.1628
¾	1.050	.824	.113	2.5887	31.064	.2157
1	1.315	1.049	.133	3.2955	39.546	.2746
1¼	1.660	1.380	.140	4.3354	52.025	.3613
1½	1.900	1.610	.145	5.0580	60.696	.4215
2	2.375	2.067	.154	6.4937	77.924	.5411
2½	2.875	2.469	.203	7.7566	93.079	.6464
3	3.500	3.068	.216	9.6384	115.66	.8032
3½	4.000	3.548	.226	11.146	133.76	.9289
4	4.500	4.026	.237	12.648	151.78	1.0540
5	5.563	5.047	.258	15.854	190.25	1.3212
6	6.625	6.065	.280	19.054	228.65	1.5878

Size	Lineal feet of tube per square foot of surface	Transverse area square inches	Capacity per Lineal Foot			Length of tube containing one cubic foot
			Cubic inches	Cubic feet, also area in square feet	United States gallons	
⅛	14.200	.0568	.6820	.0004	.0030	2533.8
¼	10.494	.1041	1.2487	.0007	.0054	1383.8
⅜	7.7479	.1909	2.2907	.0013	.0099	754.36
½	6.1410	.3039	3.6463	.0021	.0158	473.91
¾	4.6356	.5333	6.3992	.0037	.0277	270.03
1	3.6413	.8643	10.371	.0060	.0449	166.62
1¼	2.7679	1.4957	17.949	.0104	.0777	96.275
1½	2.3725	2.0358	24.430	.0141	.1058	70.733
2	1.8480	3.3556	40.267	.0233	.1743	42.913
2½	1.5471	4.7878	57.453	.0332	.2487	30.077
3	1.2450	7.3927	88.712	.0513	.3840	19.479
3½	1.0760	9.8868	118.64	.0687	.5136	14.565
4	.9488	12.730	152.76	.0884	.6613	11.312
5	.7569	20.002	240.02	.1389	1.0391	7.1993
6	.6298	28.890	346.68	.2006	1.5008	4.9844

Installation of floor panel for plant of Perfection Pipe Nipple Company, Madison, Ohio. Coils are shown preparatory to laying concrete.

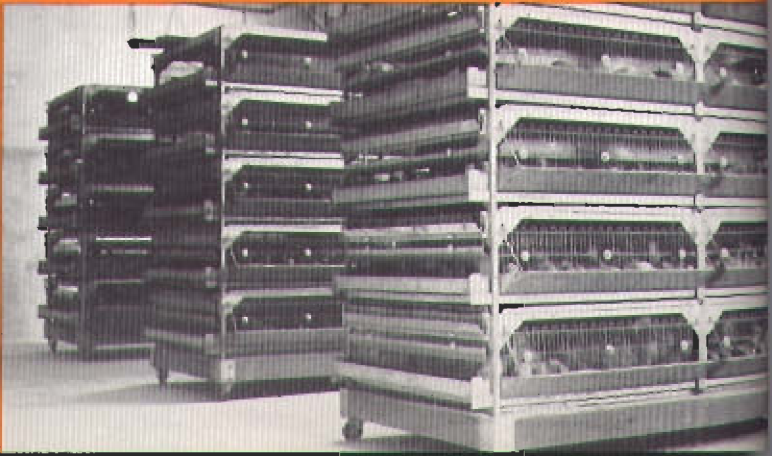


NATIONAL STANDARD PIPE—EXTERNAL PROPERTIES

External diameter	Circumference	Surface per lineal foot		Lineal feet of tube per square foot of surface	Transverse area	Volume or displacement per lineal foot		
Ins.	Ins.	Sq. ins.	Sq. ft.		Sq. ins.	Cubic inches	Cubic feet, also area in square feet	United States gallons
.405	1.2723	15.268	.1060	9.4314	.1288	1.5449	.0009	.0067
.540	1.6965	20.358	.1414	7.0736	.2290	2.7483	.0016	.0119
.675	2.1206	25.447	.1767	5.6588	.3578	4.2942	.0025	.0186
.840	2.6389	31.667	.2199	4.5473	.5542	6.6501	.0038	.0288
1.050	3.2987	39.584	.2749	3.6378	.8659	10.391	.0060	.0450
1.315	4.1312	49.574	.3443	2.9047	1.3581	16.298	.0094	.0706
1.660	5.2150	62.581	.4346	2.3010	2.1642	25.971	.0150	.1124
1.900	5.9690	71.628	.4974	2.0104	2.8353	34.023	.0197	.1473
2.375	7.4613	89.535	.6218	1.6083	4.4301	53.162	.0308	.2301
2.875	9.0321	108.38	.7527	1.3286	6.4918	77.902	.0451	.3372
3.500	10.996	131.95	.9163	1.0913	9.6211	115.45	.0668	.4998
4.000	12.566	150.80	1.0472	.9549	12.566	150.80	.0873	.6528
4.500	14.137	169.65	1.1781	.8488	15.904	190.85	.1104	.8262
5.563	17.475	209.70	1.4563	.6867	24.301	291.62	.1688	1.2624
6.625	20.813	249.76	1.7344	.5706	34.472	413.66	.2394	1.7907

Turkeys are receiving the comfortable benefits of radiant heating. A flock of turkeys at the Kroeger Turkey Farm, Lima, Ohio, thriving on the radiant rays from NATIONAL Steel Pipe coils.

Battery of hover trays at Kroeger Turkey Farm. Trays heated by hot water by means of ½-inch NATIONA Steel Pipe coils. The floor panels are 1¼-in NATIONAL Steel Pipe embedded in concrete.



Section of radiant heated chick room—Lakeview Poultry Farm and Hatchery, Thiel Brothers, Barker, New York.

CALCULATING TRANSMITTED HEAT COEFFICIENTS*

VALUES OF HEAT CONDUCTIVITY (k) AND OF CONDUCTANCE (C)† OF COMMON BUILDING MATERIALS, OF AIR SPACES (a), OF SURFACES (f), AND OF HEAT INSULATION (k).

Units are B.T.U. per hour per degree temperature difference per square foot area and for 1 in. thickness except when otherwise noted.

Material	Thickness, inches	k
Common Building Materials		
Blocks		
Cinder	8	0.62
	12	0.51
Concrete	8	1.00
	12	0.80
Bricks		
Common (clay)	1	5.0
Face (clay)	1	9.2
Glass	. .	
Cement mortar	1	12.0
Concrete	1	12.0
Gypsum fiber	1	1.66
Plaster (gypsum)	1	3.3
Plasterboard	3/8	3.73
	1/2	2.82
Plaster and wood lath	A	2.0
Roofing materials		
Built-up roofing	3/8	3.53
Composition roofing	A	6.5
Shingles		
Asbestos	A	6.0
Slate	1	10.37
Wood	A	1.28
Stone	1	12.0
Stucco	1	12.0
Tile or terrazzo	1	12.0
Tile, hollow clay	4	1.0
	6	0.64
	8	0.60
	12	0.40
	16	0.31
Tile, hollow gypsum	4	0.46
Wood lumber		
Maple	1	1.15
Yellow pine	1	0.80
Yellow pine lap siding	A	1.28
Surfaces and Air Spaces		f
Air spaces	(over 1½ in.)	**1.1**
Inside surfaces (f^{i})		**1.6**
Outside surfaces (f$_{o}$)	at 15 m.p.h.	**6.0**

Material	Density, lb. per cu. ft.	k‡
Heat Insulating Materials		
Asbestos wood	123	2.70
Balsa wood	20.0	0.58
Balsa wood	7.3	0.33
Balsam wool	2.2	0.27
Cabot's quilt	4	0.255
Celotex	13.2	0.34
Corkboard (pure)	14.0	0.34
	10.6	0.30
Dry zero	1.0	0.24
Eagle insulating wool	9.4	0.27
Fibrofelt	13.6	0.32
Glass wool	1.5	0.27
	0.85	0.25
Hairinsul (loose, 75% hair, 25% jute)	6.3	0.27
Hairinsul (loose, 50% hair, 50% jute)	6.1	0.26
Hairfelt	13.0	0.26
	11.0	0.26
Insulex or Pyrocell	30.0	1.0
	12.0	0.44
Insulite	16.9	0.34
Keystone hair	11.0	0.25
Linofelt	4.9	0.28
Lith	14.3	0.40
Maftex	16.1	0.34
Magnesia (rigid), 85% **magnesia**, 15% asbestos	19.3	0.51
Masonite	19.8	0.33
Regranulated cork	8.1	0.31
Rock cork	14.5	0.33
Rock wool	10.0	0.27
Rock wool	21.0	0.30
Sawdust	—	1.04
Shavings	—	0.71
Temlok	15.0	0.33
Thermax	24.2	0.46
Thermofelt (felted, jute and **asbestos fibers**)	10.0	0.37
Thermofelt (felted, hair and **asbestos fibers**)	7.8	0.28
Thermofill (flaked gypsum)	34.0	0.23
	19.8	0.35
Torfoleum	10.2	0.29

*Courtesy of Publisher, "Heating and Ventilating Reference Data."

"A" appearing in the column headed "Thickness, Inches" means "thickness as applied," not 1 in. thickness.

†Conductance (C) differs from conductivity (k) in that instead of being for 1 in. thickness it is for some other thickness. In column headed "Thickness, Inches" if the thickness shown is 1 in. the corresponding value in next column is "k"; if some thickness other than 1 in. is shown the corresponding value in the "k" column is really (C) and not (k) because the value is for the thickness specified and not for 1 in. thickness.

‡Values are for thickness of 1 in., reported by various laboratories but principally from tests at the Bureau of Standards. Tests at mean temperature of 90 F. mainly, but at 75 F. in a few cases.

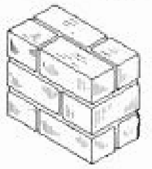
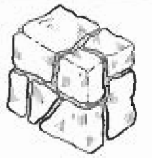
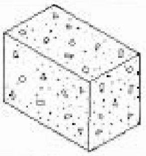
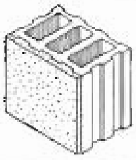
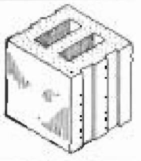
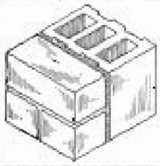
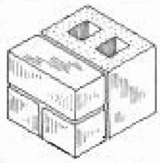
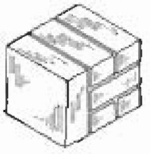
The table shown on this page lists the values of heat conductivity or conductance of common building materials, building surfaces, and materials used for heat insulation. These values when correctly used can be made to furnish a reasonably reliable indication of the amount of heat transmitted through practically any type of building construction. They have been gathered from a variety of sources and it is believed that they represent the consensus of current opinion. Not all types of materials whose resistance to heat flow has been measured are included in this list, some of the less common being omitted to keep table to a usable size.

The values listed should not be confused with those of the coefficient *U* which is the over-all heat conductance of a unit section of any wall. They are not values of *U*, but are values often used in calculating *U*.

HEAT TRANSMISSION TABLES (Values of U for Masonry Walls)*

MASONRY WALLS

Figures are in B t u per hour per square foot per degree temperature difference, air to air, still air inside, 15 m p h wind outside

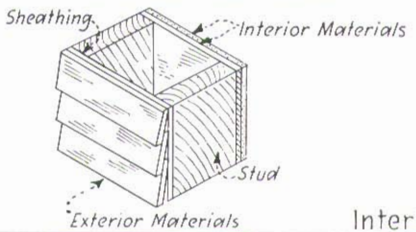
CONSTRUCTION NO.		INTERIOR MATERIALS → ↓ EXTERIOR MATERIALS	THICKNESS, INCHES	A	B	C	D	E	F	G	H	I
						FURRING STRIPS USED						
				Plain Wall, no Plaster	½" Plaster direct on Walls, no Furring	½" Plaster on Wood Lath or on Metal Lath or on ½" Plasterboard	½" Painted Plasterboard, no Plaster	½" Plaster on ½" Rigid Insulation	½" Plaster on 1" Rigid Insulation	½" Plaster on 2" Furring 1⅝" Rock Wool Fill	½" Plaster on 2" Furring ½" Flexible Insulation	½" Plaster on 1½" Corkboard set in ½" Cement Mortar
1		Common Brick Throughout	4	0.63	0.57	0.33	0.35	0.24	0.18	0.12	0.21	0.15
2			8	0.44	0.39	0.26	0.26	0.20	0.16	0.11	0.18	0.13
3			12	0.31	0.30	0.22	0.22	0.17	0.14	0.10	0.15	0.12
4			16	0.25	0.24	0.18	0.19	0.15	0.14	0.09	0.14	0.11
5		4" Face Brick, remainder Common Brick	4	0.81	0.72	0.35	0.40	0.26	0.19	0.13	0.22	0.16
6			8	0.49	0.46	0.29	0.30	0.22	0.16	0.12	0.19	0.14
7			12	0.35	0.34	0.24	0.24	0.20	0.15	0.10	0.16	0.12
8			16	0.28	0.26	0.20	0.20	0.16	0.13	0.09	0.15	0.11
9		Limestone or Sandstone	8	0.68	0.62	0.35	0.37	0.25	0.18	0.13	0.21	0.15
10			12	0.56	0.51	0.31	0.33	0.23	0.17	0.12	0.20	0.14
11			16	0.47	0.44	0.28	0.30	0.21	0.16	0.12	0.19	0.14
12			24	0.36	0.34	0.24	0.26	0.19	0.15	0.11	0.17	0.13
13		Concrete	8	0.68	0.62	0.35	0.37	0.25	0.19	0.13	0.21	0.15
14			10	0.62	0.56	0.33	0.35	0.24	0.18	0.12	0.20	0.15
15			16	0.47	0.44	0.28	0.30	0.21	0.16	0.12	0.19	0.14
16			20	0.41	0.38	0.26	0.27	0.20	0.15	0.11	0.17	0.13
17		Hollow Clay Tile With 1" Stucco Exterior Finish	4	0.53	0.49	0.30	0.32	0.23	0.17	0.12	0.19	0.14
18			8	0.39	0.37	0.25	0.26	0.20	0.15	0.11	0.17	0.13
19			10	0.35	0.33	0.23	0.24	0.18	0.14	0.11	0.16	0.12
20			12	0.31	0.29	0.21	0.22	0.17	0.13	0.10	0.15	0.12
21			16	0.24	0.23	0.18	0.19	0.15	0.12	0.09	0.14	0.11
22		Hollow Gypsum Tile, 1" Stucco Ext. Fin.	4	0.33	0.31	0.22	0.23	0.18	0.14	0.10	0.16	0.12
23		Cinder Blocks With one Air Cell Across Heat Flow	8	0.42	0.39	0.26	0.27	0.20	0.16	0.11	0.18	0.13
24			12	0.36	0.34	0.24	0.25	0.19	0.15	0.11	0.17	0.13
25		Cement Blocks With one Air Cell Across Heat Flow	8	0.56	0.52	0.31	0.33	0.23	0.17	0.12	0.20	0.14
26			12	0.49	0.46	0.29	0.30	0.22	0.16	0.12	0.19	0.14
27		4" Brick or 4" Cut Stone with Hollow Clay Tile Backing of This Thickness	6	0.37	0.35	0.24	0.25	0.19	0.16	0.11	0.17	0.13
28			8	0.36	0.34	0.24	0.25	0.19	0.15	0.11	0.17	0.13
29			10	0.32	0.31	0.22	0.23	0.18	0.14	0.10	0.16	0.12
30			12	0.28	0.26	0.20	0.20	0.16	0.13	0.10	0.15	0.11
31		4" Brick or 4" Cut Stone with Concrete Backing of This Thickness	6	0.61	0.56	0.33	0.35	0.24	0.18	0.13	0.20	0.15
32			10	0.51	0.47	0.30	0.31	0.22	0.17	0.12	0.19	0.14
33			16	0.41	0.38	0.26	0.27	0.20	0.15	0.11	0.17	0.13
34		4" Brick Veneer With Cinder Block Back'g of This Thickness	8	0.35	0.33	0.24	0.24	0.19	0.14	0.11	0.16	0.12
35			12	0.31	0.30	0.22	0.22	0.17	0.14	0.10	0.16	0.12
36		With Cement Block Back'g of This Thickness	8	0.45	0.42	0.27	0.29	0.21	0.16	0.12	0.18	0.13
37			12	0.40	0.38	0.26	0.27	0.20	0.15	0.11	0.17	0.13
38		4" Cut Stone With Common Brick Backing of This Thickness	8	0.37	0.35	0.24	0.25	0.19	0.15	0.11	0.17	0.13
39			12	0.28	0.27	0.20	0.21	0.16	0.13	0.10	0.15	0.12
40			16	0.23	0.22	0.17	0.18	0.15	0.12	0.09	0.13	0.11

*Courtesy of Publisher, "Heating and Ventilating Reference Data."

HEAT TRANSMISSION TABLES (Values of U for Wood Constructions)*

WOOD FRAME WALLS

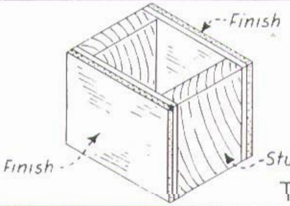
Figures are in B.t.u. per hour per square foot per degree temperature difference, air to air, still air inside, 15 m.p.h. wind outside



Construction No.	Exterior Material	Type Sheathing / Interior Materials	A	B	C	D	E	F	G	H
			Plain Wall - no Plaster ½" Plaster Board on Studding.	½" Plaster on Wood Lath on Metal Lath, or on ½" Plaster Board.	½" Plaster on ½" Rigid Insulation on Studding	½" Plaster on 1" Rigid Insulation on Studding	½" Plaster on 1½" Corkboard on Studding	Same as B, but with Stud faced one side with bright Aluminum Foil	Same as B plus ½" Flexible Insulation against Sheathing	Same as B plus 3⅝" Rock Wool between Studding
41	Wood Siding or Clapboard	Wood Sheathing - 1"	0.25	0.24	0.19	0.15	0.11	0.19	0.17	0.060
42		Rigid Insulator Sheathing - ½"	0.23	0.22	0.18	0.14	0.11	0.18	0.16	0.059
43		Plaster board Sheathing - ½"	0.28	0.27	0.20	0.16	0.12	0.21	0.18	0.062
44	Wood Shingles	Wood Sheathing - 1"	0 24	0.23	0.18	0.14	0.11	0.18	0.16	0.059
45		Rigid Insulator Sheathing ½	0.19	0.19	0.15	0.12	0.10	0.15	0.14	0.056
46		Plaster board Sheathing - ½"	0.24	0.24	0.19	0.15	0.11	0.19	0.16	0.060
47	Stucco	Wood Sheathing - 1"	0.31	0.29	0.22	0.16	0.12	0.22	0.19	0.063
48		Rigid Insulator Sheathing - ½"	0.27	0.26	0.20	0.16	0.12	0.20	0.18	0.062
49		Plasterboard Sheathing - ½"	0.40	0.38	0.26	0.19	0.14	0.27	0.22	0.066
50	Brick Veneer	Wood Sheathing - 1"	0.23	0.23	0.18	0.14	0.11	0.18	0.16	0.059
51		Rigid Insulator Sheathing - ½"	0.23	0.22	0.18	0.14	0.11	0.18	0.16	0.059
52		Plasterboard Sheathing - ½"	0.31	0.30	0.22	0.17	0.12	0.22	0.19	0.063

WOOD FRAME PARTITIONS

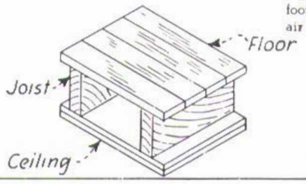
Figures are in B t u per hour per square foot per degree temperature difference air to air still air both sides



Construction No.	Finish / Type of Partitions	A	B	C	D	E	F
			Double Partitions				
		Single Partitions (one Side open)	Air Spaces Between Studding	Gypsum Fill Between Studding	One side of Stud Space Faced with bright Alum. Foil	½" Flexible Insulation Between Studding (in Air Space)	3⅝" Rock Wool Fill Between Studding
53	Plaster on Wood lath or ⅜" Plasterboard	0.60	0.33	0 11	0.24	0.21	0.063
54	Plaster on Metal Lath	0 68	0.38	0.11	0.27	0.22	0.065

WOOD FRAME FLOORS & CEILINGS

Figures are in B t u per hour per square foot per degree temperature difference, air to air still air both sides



Construction No.	Ceiling Material	Insulation Between Joists / Type of Flooring	A	B	C	D	E
			No Flooring	Flooring (Y. P.) on Joists	Flooring (Y. P.) on ½" Rigid Insulation on Joists	H. W. Flooring on Y. P. Sub-Flooring on Joists	¼" Battleship Linoleum on Y. P. Flooring
55	No Ceiling	No Insulation	—	0.45	0.27	0.34	0.34
56	Plaster on Wood Lath or ⅜" Plasterboard	No Insulation	0.60	0.28	0.20	0.24	0.24
57	Plaster on ½" Rigid Insulator	No Insulation	0.34	0.21	0.16	0.18	0.18
58	Metal Lath Plaster	½" Rigid or Flexible	0.25	0.17	0.14	0.15	0.15
59	Metal Lath Plaster	Bright Aluminum Foil	0.29	0.22	0 17	0.19	0.19
60	Metal Lath Plaster	3⅝" Rock Wool	0.066	0.062	0.056	0.059	0.059
61	1½" Corkboard & Plaster	No Insulation	0 16	0.12	0.10	0.11	0.11

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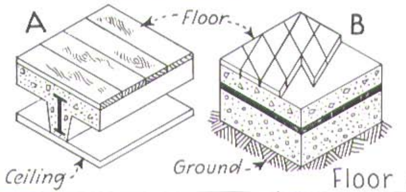
*Courtesy of Publisher, "Heating and Ventilating Reference Data."

CONCRETE FLOORS & CEILINGS

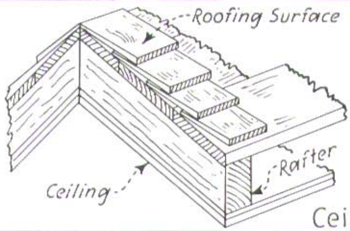
Figures are in B.t.u. per hour per square foot per degree temperature difference, air to air, still air both sides.



Construction No.	Construction Type	Floor Construction → Material under Floor ↓		Thickness of Concrete	A Bare Concrete Floor	B Y. P. Flooring on Wood Sleepers Embedded in Concrete	C H.W. Flooring on Y. P. Subflooring on Wood Sleepers Embedded in Concrete	D 1" Tile or Terrazzo on Concrete	E ¼" Battleship Linoleum on Concrete
62	A	Floor Slab Exposed. No Finished Ceiling Beneath.		4	0.63	0.39	0.30	0.60	0.43
63				6	0.57	0.37	0.28	0.55	0.40
64				8	0.52	0.35	0.27	0.50	0.38
65				10	0.48	0.33	0.26	0.46	0.35
66	A	½" Plaster. Direct on Under Surface of Concrete		4	0.58	0.37	0.28	0.55	0.40
67				6	0.52	0.35	0.27	0.50	0.38
68				8	0.48	0.33	0.26	0.47	0.36
69				10	0.45	0.31	0.25	0.43	0.34
70	A	¾" Plaster on Wood or Metal Lath. (Suspended or Furred Ceiling)		4	0.37	0.27	0.22	0.36	0.29
71				6	0.35	0.26	0.21	0.34	0.28
72				8	0.33	0.25	0.21	0.32	0.26
73				10	0.31	0.24	0.20	0.30	0.25
74	A	½" Plaster on ½" Rigid Insulation (Suspended or Furred Ceiling)		4	0.24	0.20	0.17	0.24	0.21
75				6	0.23	0.19	0.16	0.23	0.20
76				8	0.22	0.18	0.16	0.22	0.19
77				10	0.22	0.18	0.16	0.21	0.19
78	A	½" Plaster on 1½" Corkboard in 1½" Cement Mortar on Concrete		4	0.15	0.13	0.12	0.15	0.13
79				6	0.14	0.13	0.11	0.14	0.13
80				8	0.14	0.12	0.11	0.14	0.13
81				10	0.14	0.12	0.11	0.14	0.12
82	B	Stone Concrete Directly on Ground, no Insulation, no Cinder Concrete.		4	1.05	0.52	0.37	0.96	0.59
83				6	0.89	0.48	0.34	0.83	0.54
84				8	0.78	0.44	0.32	0.73	0.49
85				10	0.69	0.41	0.31	0.65	0.46
86	B	3" Cinder Concrete on Ground, Insulation on top of this, under Stone Concrete.	No Insulation	4	0.64	0.40	0.30	0.61	0.44
87			No Insulation	8	0.53	0.35	0.27	0.51	0.38
88			1" Rigid Insulat.	4 or 8	0.21	0.18	0.15	0.21	0.18
89			2" Corkboard	4 or 8	0.12	0.11	0.10	0.12	0.11

WOOD FRAME PITCHED ROOFS

Figures are in B.t.u. per hour per square foot per degree temperature difference, air to air, still air inside, 15 m.p.h. wind outside.



Construction No.	Roofing Surface ↓	Ceiling Surface → Type Insulation ↓	A No Ceiling	B ½" Plaster on Wood or Metal Lath or ⅜" Plasterboard	C ½" Rigid Insulation With or Without ½" Plaster	D ½" Plaster on 1" Rigid Insulation	E ½" Plaster on 1½" Corkboard	F ½" Plaster on 2" Corkboard
90	Wood Shingles on Wood Strips	No Insulation	0.45	0.28	0.22	0.17	0.12	0.10
91	Asphalt Composition, Tile or Slate on Wood Sheathing	No Insulation	0.54	0.31	0.23	0.17	0.13	0.10
92	Wood Shingles on Wood Strips, or Asphalt Shingles, Composition Roofing, or Slate or Tile Roofing on Wood Sheathing	½" Flexible	0.25	0.16	0.15	0.12	0.098	0.084
93		1" Flexible	0.17	0.12	0.12	0.10	0.083	0.073
94		Aluminum Foil on one side of Air Space	—	0.23	0.18	0.14	0.11	0.092
95		3⅝" Rockwool	—	0.063	0.059	0.054	0.050	0.045

MASONRY PARTITIONS

		No Plaster	Plastered One Side	Plastered Both Sides
96	4" Hollow Clay Tile	0.43	0.40	0.38
97	4" Common Brick	0.49	0.45	0.43
98	4" Hollow Gypsum Tile	0.29	0.28	0.27

Figures are in B.t.u. per hour per square foot per degree temperature difference, air to air, still air both sides.

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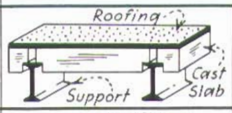
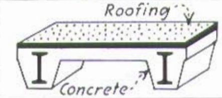
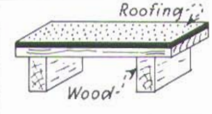
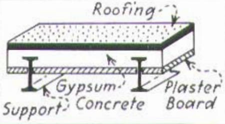
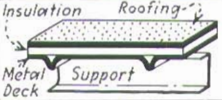
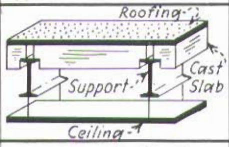
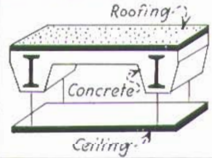
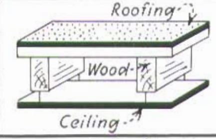
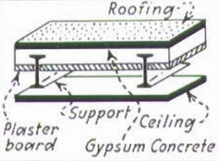
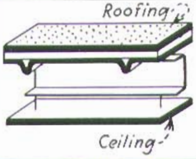
*Courtesy of Publisher, "Heating and Ventilating Reference Data."

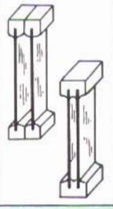
HEAT TRANSMISSION TABLES (Values of U for Roofs, Windows, Skylights, and Doors)*

FLAT & BUILT-UP ROOFS

Figures for all constructions on this sheet are in B.t.u per hour per square foot per degree temperature difference, air to air, still air inside, 15 m.p.h wind outside

Construct. No			Thickness of Roof Deck	A	B	C	D	E	F	G	H
				No Insulation	Rigid Insulation				Corkboard		
					½"	1"	1½"	2"	1"	1½"	2"
99		Precast	1⅝	0.83	0.37	0.23	0.17	0.14	0.22	0.16	0.13
100		Concrete	2	0.81	0.36	0.23	0.17	0.14	0.22	0.16	0.13
101			4	0.71	0.34	0.22	0.17	0.14	0.21	0.16	0.12
102			6	0.64	0.32	0.21	0.16	0.13	0.20	0.15	0.12
103		Wood	1	0.50	0.28	0.20	0.15	0.12	0.19	0.14	0.11
104			1½	0.37	0.24	0.17	0.14	0.11	0.17	0.13	0.11
105			2	0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10
106			4	0.18	0.14	0.12	0.10	0.085	0.11	0.094	0.081
107		2" Gypsum Fiber Concrete on ½" Plasterboard	2½	0.38	0.24	0.18	0.14	0.12	0.17	0.13	0.11
108			3½	0.31	0.21	0.16	0.13	0.11	0.15	0.12	0.10
109		Flat Metal	–	0.94	0.39	0.24	0.18	0.14	0.23	0.16	0.13
110		Precast	1⅝	0.42	0.26	0.18	0.14	0.12	0.18	0.14	0.11
111		Concrete	2	0.42	0.26	0.18	0.14	0.12	0.18	0.14	0.11
112			4	0.39	0.25	0.18	0.14	0.12	0.17	0.14	0.11
113			6	0.37	0.24	0.17	0.14	0.11	0.17	0.13	0.11
114		Wood	1	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
115			1½	0.26	0.23	0.15	0.12	0.10	0.14	0.11	0.095
116			2	0.24	0.21	0.14	0.11	0.097	0.13	0.11	0.091
117			4	0.15	0.12	0.10	0.088	0.078	0.10	0.085	0.074
118		2" Gypsum Fiber Concrete on ½" Plasterboard	2½	0.27	0.19	0.15	0.12	0.10	0.14	0.11	0.096
119			3½	0.23	0.17	0.13	0.11	0.096	0.13	0.11	0.090
120		Flat Metal	–	0.45	0.27	0.19	0.15	0.12	0.18	0.14	0.11

WINDOWS & SKYLIGHTS

121		Single Sash	Double Strength Window Glass ⅛ in Thick	1.24
122		Double Sash		0.58
123		Triple Sash		0.38
124		Double Glazed Single Sash		0.63
125		Plate Glass ⅜ in Thick		1.19

WOOD & METAL DOORS

126		Thin Wood Doors with Glass	1.24
127		1" Wood Doors	0.70
128		2" Wood Doors	0.45
129		3" Wood Doors	0.30
130		Metal and Asbestos Doors	0.65

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*Courtesy of Publisher, "Heating and Ventilating Reference Data."

HEAT LOSSES AND INFILTRATION THROUGH DOORS AND WINDOWS*

The purpose of this sheet is to enable the user rapidly and accurately to determine the heat losses and infiltration through doors and windows by the use of tables for standard size doors and windows, eliminating the laborious calculation of crack lengths. The method and the data were developed by Ralph A. Krauss, combustion engineer, Anthracite Industries Laboratory, and appear here with slight modifications.

Tables III, IV, and VII give the heat losses through standard-size windows and doors, based on a 70° F. temperature difference and 15 mile wind. Table I permits adjustment to other temperature differences, and Table II of other wind velocities.

The first two columns give the width and height of the window or door opening in inches. The opening refers to the outside dimensions of the window or door. The next column gives the area in square feet. This figure is not used in calculating window losses, but is subtracted from the gross wall area to obtain the actual or net area of the wall structure.

The transmission loss is given in Btu per hour for single and double glass, the latter referring to two separate thicknesses of glass with an air space between. The presence of storm sash fulfills this condition, but "double-strength" glass does not. For single glass, the transmission coefficient is 1.13 Btu per sq. ft. per hr. per degree F., and for double glass, 0.45 Btu.

Doors consisting largely of glass or thin wood panels are assumed to have the same transmission loss as windows of the same size. For solid wood doors, multiplying factors are given in Table VI.

Infiltration loss depends upon a number of factors, including the construction of the window and its fit. The tables are calculated on the basis of standard data.

Infiltration losses through types of windows not given in the tables may be calculated by multiplying the loss shown under "Weatherstripped, Poor," by the factors given in Table V.

Metal windows sometimes consist of part stationary and part movable sections. In this case, count the entire window for transmission loss and the movable part for infiltration loss.

The tables have been based upon a 15 mile wind. For other wind velocities, use the multiplying factors in Table II.

Infiltration is assumed to occur only on the windward half of the building, although it is safer to compute the total possible leakage of the entire structure, making sure that the infiltration loss is not less than half of this figure.

After the heat loss and infiltration have been determined for a given room from the tables for a 70° F. temperature, the total can be corrected for other design temperature differences by multiplying by the factors in Table I.

TABLE I. CORRECTION FACTORS FOR TEMPERATURE DIFFERENCE

Design temperature difference, F.	Multiplying factor
90	1.29
85	1.21
80	1.14
75	1.07
70	1.00
65	0.93
60	0.86
55	0.79
50	0.72
45	0.64
40	0.57
35	0.50
30	0.43

TABLE II. WIND CORRECTION FACTORS

Wind, M.p.h.	Factor
3	0.1
4	0.15
5	0.2
6	0.3
8	0.45
10	0.60
12	0.75
14	0.90
16	1.10
18	1.25
20	1.40
25	1.80

TABLE III. HEAT LOSSES, SINGLE CASEMENT WINDOWS

Size of opening (inches)		Area (sq. ft.)	Transmission loss at 70° F. diff. (B.t.u. per hr.)		Infiltration loss at 70° F. diff. 15 mile wind (B.t.u. per hr.)			
					Weather-stripped		Non-weather-stripped	
Width	Height		Single glass	Double glass	Aver.	Poor	Aver.	Poor
20½	21	3.0	237	95	205	296	346	970
20½	45	6.4	506	202	325	468	545	1530
20½	53	7.54	595	238	363	524	610	1710
22½	23	3.75	296	118	224	324	378	1060
24	24	4.0	316	126	248	343	400	1120
24	30	5.0	395	157	268	386	450	1260
24½	25	4.25	335	134	253	365	425	1190
24½	29	4.92	388	155	265	382	445	1250
24½	53	9.0	712	284	381	550	640	1800
25½	23	4.08	322	129	240	346	404	1130
28½	25	4.95	390	156	265	382	445	1250
28½	29	5.74	452	180	284	410	478	1340
28½	31	6.12	473	193	295	425	495	1390
28½	35	6.92	545	218	314	454	528	1480
28½	41	8.10	640	255	343	495	578	1620
28½	47	9.30	735	293	374	540	628	1760
28½	53	10.5	830	331	403	582	678	1900
31½	29	6.35	500	200	300	430	502	1410
31½	33	7.22	570	228	320	460	535	1500
31½	35	7.66	605	242	330	474	552	1550
31½	37	8.10	640	255	340	490	570	1600
31½	41	8.96	709	283	360	518	603	1690
31½	47	10.3	815	325	390	560	652	1830
31½	53	11.6	915	366	420	603	702	1970
34½	29	6.95	548	219	315	453	528	1480
34½	33	7.90	625	249	335	480	560	1570
34½	37	8.85	700	279	355	510	595	1070
34½	41	9.83	788	310	375	540	627	1760
34½	47	11.2	895	352	405	582	678	1900
34½	53	12.6	995	396	435	625	728	2040

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*Courtesy of Publisher, "Heating and Ventilating Reference Data."

TABLE IV. HEAT LOSSES, DOUBLE-HUNG WINDOWS

Window size (inches)		Area (sq. ft.)	Transmission loss at 70° F. diff. (B.t.u. per hr.)		Infiltration loss at 70° F. diff. 15 mile wind (B.t.u. per hr.)			
					Weather-stripped		Non-weather-stripped	
Width	Height		Single glass	Double glass	Aver.	Poor	Aver.	Poor
16	46	5.10	405	162	350	500	580	1620
20	36	5.00	400	160	330	477	545	1545
20	42	5.84	460	184	360	520	595	1685
20	46	6.38	505	202	380	540	625	1740
20	54	7.50	592	236	420	608	694	1970
20	58	8.06	636	255	440	635	725	2060
22	30	4.58	362	145	310	448	512	1450
22	46	7.03	556	221	390	565	650	1810
22	54	8.25	645	260	430	625	720	2020
22	58	8.85	700	280	455	658	750	2130
24	36	6.00	475	189	360	520	600	1680
24	42	7.00	555	222	390	560	645	1810
24	46	7.67	608	242	410	590	680	1920
24	50	8.33	660	262	430	620	715	2000
24	58	9.69	768	315	470	680	780	2180
24	62	10.33	820	326	490	700	810	2290
26	54	9.80	787	308	465	670	775	2170
26	58	10.45	829	329	480	700	800	2260
26	62	11.20	887	352	500	730	840	2350
28	30	5.85	462	185	360	520	595	1485
28	36	7.00	555	222	390	560	645	1810
28	38	7.40	587	233	400	575	660	1860
28	42	8.16	647	258	420	605	700	1950
28	46	8.95	710	283	435	630	725	2040
28	50	9.75	775	307	460	660	760	2160
28	54	10.50	833	331	480	690	800	2250
28	56	10.90	862	344	490	710	810	2300
28	58	11.25	894	354	500	720	830	2340
28	62	12.05	955	380	520	745	860	2420
28	66	12.80	1013	403	540	775	900	2520
30	42	8.75	695	276	430	625	720	2020
30	54	11.25	894	354	490	710	800	2280
30	58	12.05	955	380	510	740	850	2400
30	62	12.90	1020	406	540	780	900	2520
31	30	6.46	510	204	382	554	630	1790
31	36	7.75	612	244	413	596	680	1930
31	42	9.05	715	285	442	640	730	2070
31	46	9.90	782	312	462	670	763	2160
31	50	10.80	850	340	482	700	800	2260
31	54	11.60	915	365	503	725	830	2370
31	58	12.50	990	394	522	755	862	2450
31	62	13.42	1055	423	543	785	895	2540
31	66	14.25	1120	450	563	815	930	2640
32	54	12.00	950	378	510	740	850	2400
32	58	12.90	1020	406	530	770	880	2460
32	62	13.80	1090	435	550	800	920	2580
34	30	7.10	560	224	405	585	670	1895
34	36	8.50	670	268	435	630	720	2040
34	42	9.90	780	312	455	660	750	2130
34	46	10.85	855	342	475	685	785	2230
34	50	11.80	930	372	495	715	820	2320
34	54	12.75	1010	404	520	755	880	2460
34	58	13.70	1080	430	535	775	885	2500
34	62	14.60	1155	460	565	815	940	2640
34	66	15.60	1230	490	575	831	950	2690
36	46	11.50	910	362	500	723	825	2340
36	50	12.50	990	394	520	750	860	2440
36	54	13.50	1065	425	540	780	890	2530
36	56	14.00	1100	440	550	795	908	2580
36	58	14.50	1145	460	560	810	925	2620
36	62	15.50	1220	488	580	840	956	2720
36	66	16.50	1300	520	600	868	990	2810
40	46	12.75	1005	402	530	766	875	2480
40	50	13.90	1095	438	550	795	908	2575
40	54	15.00	1185	472	570	825	940	2670

TABLE V. WINDOW CORRECTION FACTORS

(Multipliers for Table IV—Weatherstripped, Poor)

Type	Multiplier
Double-hung metal	2
Same, weatherstripped	1
Industrial, pivoted, metal	5
Residential metal casement	1.5

TABLE VI.—DOOR CORRECTION FACTORS

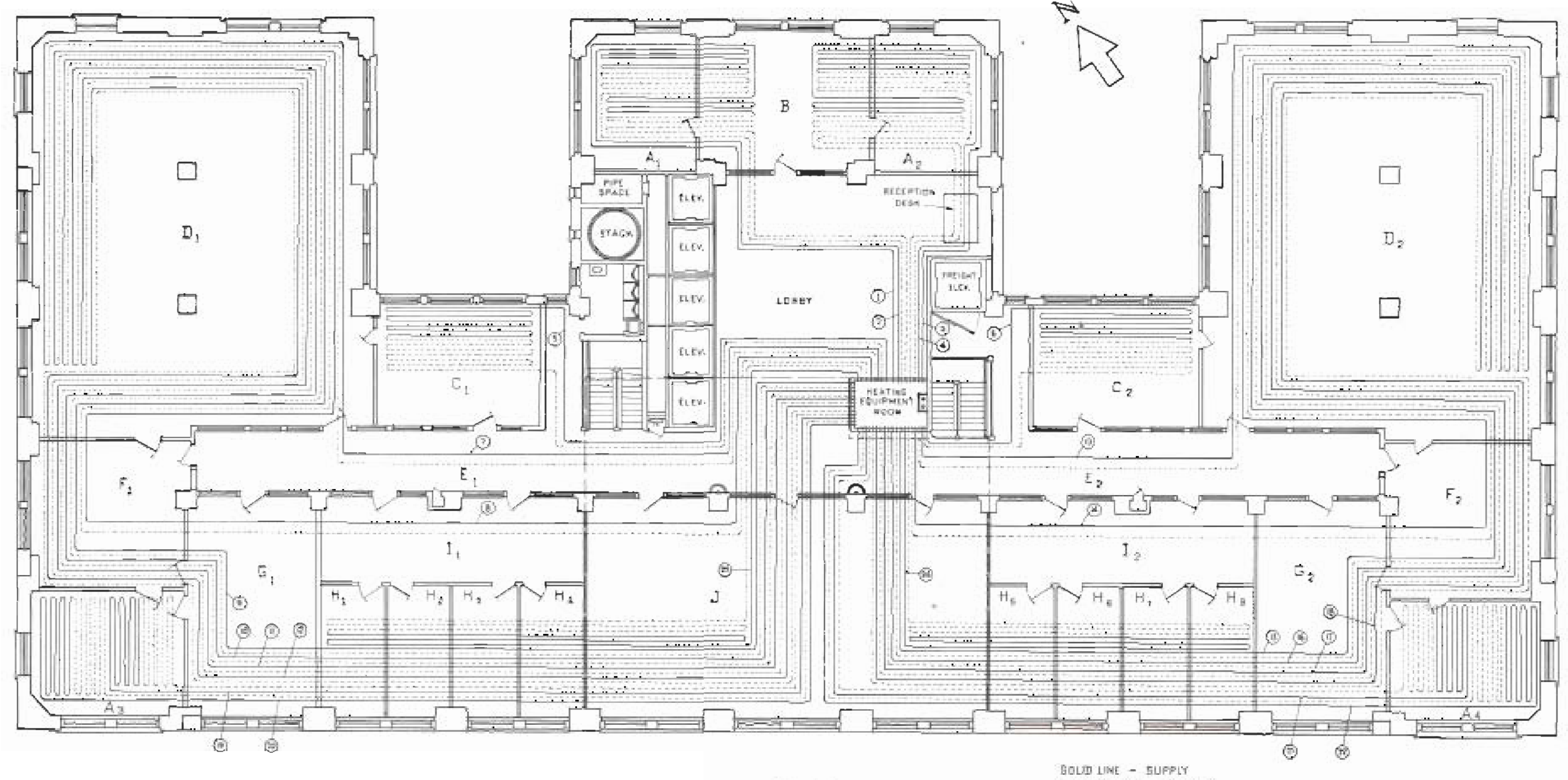
(Multipliers for Table VII—Double Glass)

Actual thickness of door, inches	Multiplying factor
25/32	1.5
1 1/16	1.3
1 5/16	1.2
1 3/8	1.1
1 5/8	1.0
2 1/8	0.85
2 5/8	0.75

TABLE VII.—DOORS AND DOUBLE CASEMENTS

Size of opening (inches)		Area (sq. ft.)	Transmission loss at 70° F. diff. (B.t.u. per hr.)		Infiltration loss at 70° F. diff. 15 mile wind (B.t.u. per hr.)			
					Weather-stripped		Non-weather-stripped	
Width	Height		Single glass	Double glass	Aver.	Poor	Aver.	Poor
SINGLE DOORS								
24	78	13.0	1025	410	1190	2380	2380	4760
24	80	13.3	1050	420	1215	2430	2430	4860
28	78	15.2	1200	480	1240	2480	2480	4960
28	80	15.6	1230	492	1250	2500	2500	5000
30	78	16.2	1280	512	1260	2520	2520	5040
30	80	16.7	1320	530	1285	2570	2570	5140
32	80	17.8	1405	562	1310	2620	2620	5240
34	82	19.3	1525	610	1355	2710	2710	5420
36	80	20.0	1580	632	1355	2710	2710	5420
36	84	21.0	1660	665	1400	2800	2800	5600
DOUBLE OR FRENCH DOORS AND WINDOWS								
32½	35½	7.8	615	246	430	620	715	2010
36½	41½	10.5	830	332	490	710	820	2300
36½	45¾	11.5	910	365	505	730	840	2360
36½	53¾	13.6	1070	428	585	845	975	2740
40½	35½	10.0	790	316	467	675	780	2190
40½	41½	11.7	925	370	515	740	855	2400
40½	45¾	12.8	1010	404	550	790	910	2550
40½	53¾	15.1	1190	475	605	875	1010	2830
44½	53¾	16.7	1320	530	625	900	1040	2920
48	78	26.0	2060	825	1925	3850	3850	7700
48	80	26.7	2110	844	1960	3920	3920	7840
48	84	28.0	2210	884	2025	4050	4050	8100
60	78	32.5	2570	1030	2065	4130	4130	8260
60	80	33.3	2640	1060	2100	4200	4200	8400
60	84	35.0	2770	1110	2165	4330	4330	8660

The above figures are for doors consisting of glass or thin wood panels. For solid wood doors, multiply the transmission-loss figures given for double glass by the factors in Table VI.



TOP FLOOR

Figure 44. Piping layout for the top floor of a multistory structure.

Supplementary Design Data

IN THE PRECEDING SECTIONS of this bulletin two design procedures are described so that the reader would have a choice of using two simplified methods of sizing the radiant heating panels. Simplified procedures are not always desirable however. There are times when extreme accuracy of design is an economic necessity. Take, for example, a housing project where a hundred or more similar houses are to be built. In such a case, even the slightest error would be significant because it would be multiplied a hundred times. Or, for another example, take a multistory building, where a dozen or more floors are identical, there too it is necessary to have exact engineering.

Generally, if the building floor area exceeds 20,000 square feet, either in one or more buildings, a heating engineer should be consulted, and for such cases we have prepared a special design procedure. This procedure is presented in the Supplement to this bulletin, but included here is an abstract of the theory and an example application.

This procedure is unique in that it allows the engineer to start his design by fixing the panel water temperature and varying the panel area. There are two reasons for this sequence, (1) only in rare cases does the designer have a separate controller on each panel, and (2) the system is thermally balanced in the initial design.

The data for this procedure, contained in 36 tables of the Supplement to this bulletin, were obtained from an extensive laboratory investigation. The laboratory work took three years to complete and the analysis took another year; in all, four years went into the preparation of the Supplement data. As a result of this work, heating engineers may save hours of design time. The following example will illustrate the use of the design tables, but the theory is explained in the Appendix of the Supplement. Copies of the Supplement and large size DESIGN DATA-FORMS may be obtained upon request.

EXAMPLE

The structure selected for the example is a multistory office building. In designing the heating system for such a building, the engineer may assume that partitions will be moved from time to time; consequently, individual room control is rarely possible.

The piping layout, as shown in Figure 21, page 56, is merely one possible solution. There are others, of course, but this layout illustrates a very simple, easy to assemble, plan. Notice that no extra coils are required for the corridor, and that no trench or cove is required for headers. Only ½-inch steel pipe is necessary for the horizontal lines. The only large size pipes are in the pipe shaft of the heating equipment room.

It is generally advantageous to spend a few more hours in design and layout work in order to reduce fabrication and installation costs. Very often a few minutes of an engineer's time will save hours of the mechanic's time. The hydraulic calculations are shown in Table I on page 61, and are based on data shown in Figure 21, page 56.

As a guide to the design procedure, we will follow the items in the Radiant Panel Heating Design Data-Form, page 58.

Required

1. Panel area, A_p (see Glossary, page 66)
2. Panel output, q

Design Conditions

1. Structure—office building (this establishes a: see Table II, page 61)
2. Wind – 10 mph, North
3. Design temperatures
   air, t_o = −10 F
   basement = . . .
   attic = 70 F (penthouse)

Rooms

1. Purpose—A (holds for A_1, A_2, A_3, and A_4)
2. Size = 16′×19′×10′, ceiling height = 10 ft.
3. Total area, A_t = 1308 sq. ft.
   10×19 = 190 sq. ft.—exposed wall and windows
   10×16 = 160 sq. ft.—exposed wall and windows
   16×19 = 304 sq. ft.—floor
   10×19 = 190 sq. ft.—inside wall
   10×16 = 160 sq. ft.—inside wall
   16×19 = 304 sq. ft.—ceiling
   Total 1308 sq. ft.
4. Panel location—ceiling
   Preferably on surface with greatest heat loss, for example, floor panel with slab floor construction, or ceiling panel with flat roof construction.

Assumptions

1. Mean water temperature, t_w = 135 F.
   This is an assumption for the first panel only, after that, t_w is fixed.
2. Assumed panel area, u = 0.220 (A_p = 288)
   It is generally sufficient to assume panel areas as given in Table III, page 64.

RADIANT PANEL HEATING DESIGN DATA – FORM

STRUCTURE Office Bldg. LOCATION Somewhere DESIGNER H.V. Engineer DATE Now

WIND VELOCITY 10 M.P.H. WIND DIRECTION North OUTSIDE AIR TEMP. -10 F GROUND TEMP. --

BASEMENT AIR TEMP. -- ATTIC TEMP. 70 F (Penthouse) ROOMS PARTIALLY HEATED Mach. Area Above Lobby @ +50 F.

ROOMS	PURPOSE	A	B	C	D	E & I	F	G	H	J	LOBBY
	SIZE	45'- 19'- 10'	24'- 19'- 10'	17' × 23' × 10'	46' × 56' × 10'	2-1 (ONE HALF)	20' × 21' × 10'	18' × 20' × 10'	9' × 18' × 10'	30' × 56' × 10'	28' × 29' × 10'
	TOTAL AREA	1308	1772	1582	7192	5756	1701	2068	864	5146	2802
	PANEL LOCATION	CLG	CLG	CLG	CLG.	CLG	CLG	CLG	CLG.	CLG.	CLG
ASSUMPTIONS	AVE. WATER TEMP.	136	135	135 *	135	135	135	135	135	135	136
	ASSUMED AREA u	.220 = 288	.131 = 240	.172 = 272	.180 = 1292	.0278 = 160	.131 = 222	.118 = 240	.116 = 100	.109 = 561	.093 = 262
	UNHEATED AREA	1020	1532	1310	5900	5596	1479	1828	764	4585	2540
	COMFORT CONSTANT a	140	140	140	140	140	140	140	140	140	140
	RESISTANCE R	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74	0.74
GLASS	NET AREA A_g	75	50	75	300	NONE	50	50	25	160	NONE
	ASHVE CODE		SINGLE	PANE		GLAZING		THROUGHOUT			
	U_g	1.13	1.13	1.13	1.13	—	1.13	1.13	1.13	1.13	—
	Δt_{rg}	1.0	1.0	1.0	1.0	—	1.0	1.0	1.0	1.0	—
	U_{ng}	.083	.032	.065	.058	—	.038	.031	.037	.037	—
DOORS	NET AREA A_d		NO	EXTERIOR		DOORS	ABOVE		GROUND	FLOOR	
	ASHVE CODE										
	U_d	—	—	—	—	—	—	—	—	—	—
	Δt_{rd}	—	—	—	—	—	—	—	—	—	—
	U_{nd}	—	—	—	—	—	—	—	—	—	—
EXPOSED WALLS	NET AREA A_w	275	190	155	1010	NONE	160	130	60	410	60
	ASHVE CODE	68 B	68 B	49 B	68 B	—	68 B	68 B	68 B	68 B	68 B
	U_w	.34	.34	.34	.34	—	.34	.34	.34	.34	.34
	Δt_{rw}	1.0	1.0	1.0	1.0	—	1.0	1.0	1.0	1.0	1.0
	U_{nw}	.092	.042	.048	.058	—	.037	.024	.029	.030	.008
CEILING	NET AREA A_c	16	226	179	1284	849	182	309	62	PENTHOUSE	564
	ASHVE CODE	12 H	SEE TABLE 16	12 H	12 H	12 H	12 H	12 H	12 H	ABOVE	8A TABLE 13
	U_c	.11	.11	.11	.11	.11	.11	.11	.11	2H 18	.35
	Δt_{rc}	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	ROOM	.75
	U_{nc}	.002	.016	.010	.024	.017	.014	.018	.009	—	.058
FLOOR	NET AREA A_f	HEAT	GAIN	THROUGH	ALL	FLOORS	TAKEN	AS	10 B hr^{-1} ft^{-2}	OF PANEL	
	ASHVE CODE										
	A_f q	2870	2000	1950	10,000	—	1800	1500	600	5610	7600
	$-\Delta t_{rf}$ $A_e \Delta t_o$	81,500	120,400	104,800	417,200	NO PANEL	110,700	144,000	61,000	36,700	202,000
	U_{nf}	-.035	-.016	-.019	-.021	BELOW	-.015	-.010	-.010	-.015	-.038
Ue	ΣU_n	.142	.079	.098	.118	.077	.074	.064	.066	.062	.058
VENTILATION	RATE CFH/FT. CRACK	19	19	19	19	—	19	19	19	19	—
	CRACK LENGTH	50	50	75	150	—	50	50	25	150	—
	TOTAL CFH	950	950	1425	2850	—	950	950	475	2850	—
	Δt_r	1.0	1.0	1.0	1.0	—	1.0	1.0	1.0	1.0	—
	V_r	1	$\frac{1}{2}$	1	$\frac{1}{3}$	—	$\frac{1}{2}$	$\frac{1}{2}$	1	1	—
RESULTS	u	.215	.133	.169	.177	.0224	.128	.113	.126	.109	.093
	PANEL AREA A_p	287	236	268	1273	430	216	233	109	561	261
	OUTPUT q	48.4	47.3	47.7	48.0	15.9	47.2	47.1	47.2	46.9	46.8
	ADJUSTED u	—	—	—	—	—			—	--	—
	ADJUSTED q	—	—	—	—	—	—	—	—	—	—
	EFFICIENCY e	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.79	0.60
	BOILER LOAD	16,430	17,400	16,200	61,900	7,300	11,200	12,900	6,720	33,300	20,400

ALL AREAS IN SQUARE FEET

$$\Delta t_r = \frac{t_a - t_x}{t_a - t_o} \text{ SEE GUIDEBOOK}$$

ASHVE CODE REFERS TO TYPE OF CONSTRUCTION AS LISTED IN ASHVE GUIDE

$$U_{ng} = \frac{A_g \Delta t_{rg} U_g}{A_p}$$

$$\text{BOILER LOAD} = \frac{A_p q}{e} \text{ B/HR}$$

3. Unheated area, $A_u = 1020$ sq. ft.
   $(1308-288=1020)$
4. Comfort constant, $a = 140$ F.
   (See Table II, page 64 for other values for a)
5. Resistance, $R = 0.71\,\mathrm{Fft^2hrB^{-1}}$.
   (If panels are constructed as shown in Figure 27, page 63, R may be taken as 0.71 for ceiling and wall panels, and 0.46 for floor panels.)

Exposed Surfaces

(See GLASS, DOORS, etc. on DESIGN DATA-FORM)

1. Net area, A_g, A_d, etc. is the area through which heat is passing. For example, for A_w use the wall area minus the glass and door area.
2. ASHVE Code. The ASHVE GUIDE lists several types of construction and the proper "U" value for that construction. For example, in the 1950 GUIDE, page 181, wall number 25 D is a brick veneer wall with an insulating board sheathing.
3. U_g, U_d, etc. is the coefficient of transmittance as given in a reputable source. For example, for wall 25 D of the GUIDE, $U_w = 0.21\,\mathrm{Bhr^{-1}ft^{-2}F^{-1}}$.
4. Δt_{r_g}, Δt_{r_d}, etc. Generally this is unity, but whenever the surface (wall, floor, etc.) is not between the outside air and the inside air, then Δt_r is not unity. (See Glossary, page 66.)
5. U_{n_g}, U_{n_d}, etc. is determined by the equation

$$U_{n_g} = \frac{A_g \times U_g \times \Delta t_{r_g}}{A_u} = \frac{75 \times 1.13 \times 1.0}{1020} = 0.083$$

Equivalent Transmittance, U_e

$$U_e = \Sigma U_n = U_{n_g} + U_{n_d} + U_{n_w} + U_{n_c} + U_{n_f}$$
$$= 0.083 + 0 + 0.092 + 0.002 - 0.035 = 0.142$$

Ventilation

1. Ventilation rate. The ventilation rate may be determined by (1) the crack method or (2) by the air-change method. The crack method is given here, but if the air-change method is used put the total ventilation in the space marked "CFH." In this case, 19 CFH ft. of crack.
2. Crack length—insert the length of crack permitting infiltration of air, in this case 50 ft.
3. Total CFH = the total amount of infiltration $= 19 \times 50 = 950$
4. Δt_r same as Δt_{r_g}, etc. $= 1.0$
5. $V_e = \dfrac{V}{A_t} = \dfrac{950}{1308}$ (in this case) $= 0.73$
   use $V_e = 1$

Results

1. n obtained from the Table, page 65—in this case $n = 0.219$ (page 58)
2. Panel area, $A_p = 0.219 \times 1308 = 287$ sq. ft.
3. Output, q—obtained from the Table, page 65—in this case, $q = 48.4$
4. Adjusted n—Since $R = 0.71$ (not changed) n and q need not be adjusted.
5. Adjusted q—same as (4) above.
6. Efficiency, e is the efficiency of the panel so that

$$e = \frac{\text{Panel output}}{\text{Panel output} + \text{back losses}}$$
$$= \frac{287 \times 48.4}{287 \times 48.4 + .10\,(287 \times 48.4)}$$
$= 0.90$ for 10% back loss

7. Boiler load—how much energy the boiler must supply the panel.

Boiler load, is $\dfrac{A_p q}{e} = \dfrac{287 \times 48.4}{0.90} = 15400$ B/hr

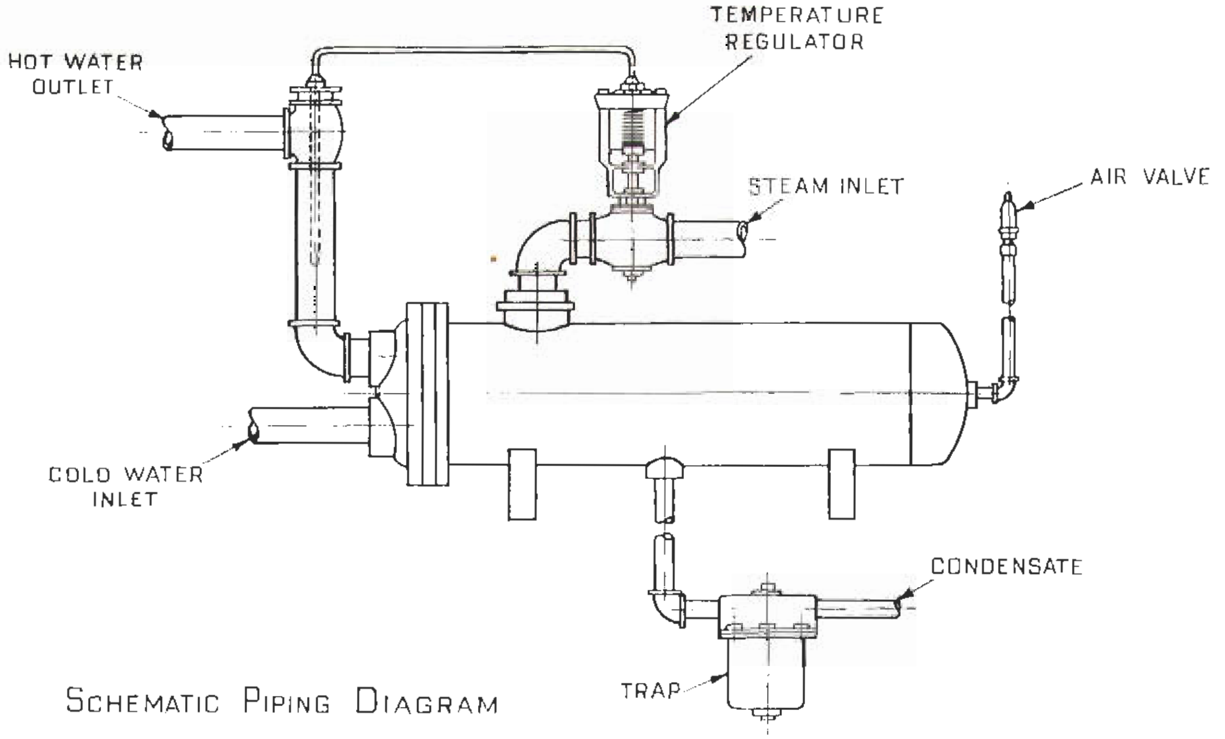
If the designer has a short cut for determining the total transmission load, Q, (total heat load minus the ventilation load) then

$$U_e = \frac{Q}{A_u \Delta t} = \frac{Q}{(A_p - A_t)\,(t_a - t_o)}$$

The control problem is not discussed in this bulletin. However a paper (*Simplifying Comfort Control for Radiant Heating*, W. P. Chapman and R. E. Fischer—Heating and Ventilating, June 1948) has been prepared covering the control problem, the gist of which is that an inside air thermostat is adequate to control a panel system if the thermal lag of the panel is less than the thermal lag of the structure. More complex problems arise when the thermal lag of the panel exceeds that of the structure. When the latter case is inevitable, a competent control engineer should be consulted.

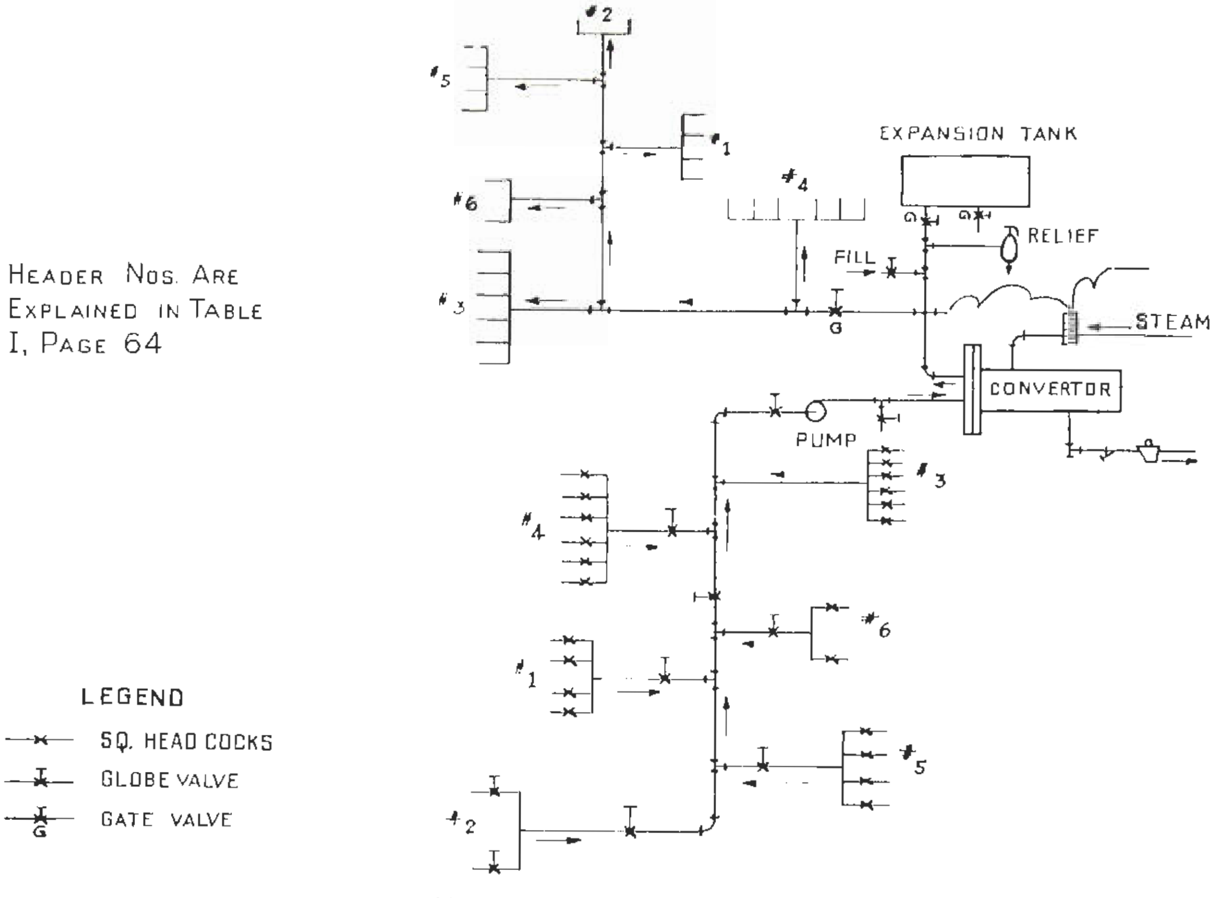
Safety Factors

In this design procedure there are the same safety factors as in any other comfort heating design, i.e., the ventilation factors and thermal characteristics are similar. However, there is an additional advantage in using this procedure in that the panel water temperature is included in the initial calculations. By selecting water temperatures 20 degrees below maximum, the system can operate at 115-120% above design without injury to the panels. Obviously, therefore, it is desirable to select panel water temperatures that will give panel surface temperatures within the ideal range (*Effect of Panel Location on Skin and Clothing Surface*



SCHEMATIC PIPING DIAGRAM

Figure 25



HEADER ARRANGEMENTS

Figure 26

Temperature, L. P. Herrington and R. J. Lorenzi—Heating, Piping and Air Conditioning, Journal Section, October 1949).

SPECIFICATIONS

These specifications are written for the TOP FLOOR only. Specifications for the entire building would be modified to provide for extra equipment, but the following will serve as a guide for any building, large or small:

1. CONVERTER: One converter shall be furnished in each heating equipment room. The converter shall be a shell and tube type, with a removable tube bundle and removable head at one end. The water shall flow through the tubes, and steam shall flow through the shell (see schematic sketches page 60, Figures 25 and 26).

The converter shall heat 40 gpm from 125 F to 145 F with a steam pressure of 30 psia (34.3 pph of condensate for 400,000 B/hr). The water circuit through the converter shall have a maximum pressure drop of 2 feet of water. The converter shell shall be of carbon steel and capable of standing a pressure of 125 psia. The converter shall conform to the Tubular Exchanger Manufacturers Association Standards, Secs. 1, 3, and 6, and shall be installed in accordance with TEMA Standards Sect. 8.

2. CIRCULATING PUMPS: A circulating pump of an approved make shall be furnished and installed for each floor. The pump must operate smoothly and quietly without vibration, and flexible connections shall be used if it is necessary to prevent transmission of sound. The pump shall conform to the Hydraulics Institute Standards, Sect. B, and shall be installed in accordance with Secs. 71-101 of these Standards.

The pump motors shall be designed to operate on 220-volt, 60-cycle, single-phase current in accordance with National Electrical Manufacturer's Association Standards MG1. The pump shall be capable of fulfilling the following requirements:

Deliver 40 gpm against a head of 50 feet of water, using a ¾-hp, 3450-rpm, direct-connected motor.

3. EXPANSION TANK: Furnish and install in the heating equipment room, one 24-gallon expansion tank of welded steel construction and so certified to withstand a static air pressure of 100 psi for a period of two hours without drop.

4. RADIANT HEATING PANELS: Panels shall be of the size and arrangement shown on the plans. The pipe shall be bent on a diameter not less than 12 times the nominal diameter. Any section of piping showing evidence of flattening or splitting shall not be repaired but shall be removed from the panel. No threaded joints shall be permitted in concealed mains, branches, or panels. Spacing shall be 12 inches for the floor panels and 8 inches for the ceiling and wall panels and as required to fit in the equipment room.

Floor Panels: A concrete subfloor shall be constructed by others at the desired grade as specified by this contractor. This panel shall be secured to this subfloor in such a manner that the piping shall not shift when covered. The piping shall be covered by concrete to a depth at the thinnest point of approximately 2 inches to the top of the pipe, as shown in Figure 27, page 63.

Ceiling Panels: Before the erection of the metal lath and plaster, the piping shall be securely fastened to the beams and the ceiling shall then be plastered. It is the responsibility of this contractor to see that plaster is forced through the lath to completely cover the back part of the pipe. See Figure 27, page 63 for illustration.

5. PIPING:

A. Cold Water and Drainage Piping

All piping shall be installed of the sizes and in the locations as shown on the plans. It shall be run parallel to the walls and ceilings in a neat and workmanlike manner.

All valves and accessories shall be installed as shown on the plans, and a minimum of fittings shall be used. The pipes shall be cut to a measured fit, and all threaded piping shall be reamed after cutting.

All piping shall be installed to grade in the direction of flow on the plans to avoid trapped conditions. In no case shall plumbing lines be run with a grade of less than ¼-inch to the foot.

All cold water, vent, condensate and condenser water piping shall be galvanized steel (ASTM A-120). This contractor shall furnish and install all water drainage and vent piping required to provide the proper operation and drainage from the equipment.

B. Steam Piping

All steam piping shall be standard weight, scale-free steel piping with malleable iron fittings and shall grade in the direction of flow or as indicated not less than 1-inch in 20 feet for mains and 1-inch in 10 feet for branches. Long runs shall be made with expansion loops, or with swing joints to take care of expansion without misalignment or damage.

C. Hot Water Piping

All panel heating piping shall be standard weight, scale-free black steel piping (ASTM A-120) and shall be installed as indicated on the plans.

6. CONTROLS:

A controller shall operate a motorized steam valve in conjunction with an outdoor temperature controller of the remote bulb type to maintain predetermined temperatures of water to the heating panels depending on the outside air temperature. This controller must be able to vary the water temperature from 75 F to 115 F when the outside temperature changes from 60 F to −10 F. The controller shall be provided with a manually operated device which for unusual temperature conditions will permit the system to be operated at a water temperature either above or below that which is considered normal for the existing outdoor temperature. This device shall have an effect on the water temperature of plus or minus 15% of the operating range of the water temperature controller.

When the outside temperature is below 60 F, the controller or auxiliary switch will keep the water circulator in continuous operation.

An "emergency" (OFF-ON) switch shall be provided which shall cut off all electrically operated heating equipment.

(If zone control is desired, three-way mixing valves can be installed on the headers. As many as nine zones per floor can be set up with this piping layout.)

7. PRESSURE REDUCING AND RELIEF VALVE: The water supply from the converter shall be equipped with a ¾-inch pressure reducing and relief valve. The drain from the relief valve shall discharge over the floor in the equipment room and flow to the floor drain.

8. AIR VENTS:

A. Steam Heating System

Vents of the vacuum atmospheric type shall be installed where indicated on the plans.

B. Hot Water Heating System

Vents shall be of the manual type (pet cocks) and shall be installed at the high point in the return line. The vents shall discharge through a ¼-inch tube to the floor drain in the equipment room on each floor.

9. BALANCING FITTINGS:

Furnish and install in the return pipe from each panel a square head cock for balancing the fluid flow. These fittings shall be designed for 125 psi water pressure and be of the size of the respective pipes. See Figure 26, page 60, Header Arrangement.

10. PRESSURE TESTS: (Hot Water) Before connecting to the equipment, concealing in floor, walls, or ceiling, and insulating, all water and heating piping shall be subjected to a hydrostatic test of 65 psi (or line pressure) for two hours, the reading to be taken near the converter when possible and proven tight against leaks. The piping should be tested in segments when it is found necessary to do so as in the case of piping concealed in concrete floors, etc.*

11. ADJUSTMENT AND TRIAL RUN: After the contractor has completed the heating distribution system, he shall put all parts in working order, and the system shall be given a run of sufficient duration, as determined by the engineer, to ascertain the proper operation of the equipment. The control system shall be tested in the presence of the engineer and under his direction in such a manner as to show that it is installed according to the specifications. The system shall be balanced by the adjustment of the square head cocks and flow control valves, to obtain the proper temperature in the respective return piping in all the rooms. This balancing shall be done on a day that is sufficiently cold to give a fair test of the system. Fuel shall be furnished by the owner.

The system shall be filled in such a manner as to vent the air from the system without leaving any air pockets. The system shall not be run at normal operating temperatures until the plaster and concrete are thoroughly dried out. The system shall not be started until the engineer has given his approval to start the initial operation.

SUGGESTIONS IN DESIGNING

1. In estimating the water temperature for a given room, use the equation

$$t_w - t_1 = \frac{Q}{500\,G}$$

where t_1 = inlet water temperature, F.
G = flow, gpm
Q = heat loss from fluid, Bhr^{-1}

For example, if a pipe carrying 5.0 gpm must run 100 feet through a corridor where the panel output is 50 Bhr^{-1}ft^{-1} (of pipe) and the water temperature was initially 120 F, then the temperature at the end of the corridor would be

$$t_w = 120 - \frac{50 \times 100}{500 \times 5} = 120 - 2 = 118 \text{ F.}$$

The temperature of 118 F would be used as the t_w term in the DESIGN TABLES.

2. In rooms requiring small supplementary panels, run the supply and return lines in the floor if ceiling panels are used, and in the ceiling if floor panels are used. The former is especially desirable in a structure where cold floors might obtain with a ceiling panel.

3. Install the panels in the surface which has the greatest heat loss, unless coil fabrication costs are

---

*As an alternative, the contractor may test the piping with air, nitrogen, or carbon dioxide prior to the hydrostatic test. This gas test should be at least 10 psig. If leaks are detected, it is possible to repair them without having to drain the system. The prescribed hydrostatic test should be performed after the gas test.

prohibitive. It is considered good practice to install ceiling panels in a room over a basement or over any partially heated room. Sometimes, however, the use of insulation will change the coldest surface; for example, in a ranch-type house, the use of insulation in the ceiling and walls makes the floor the surface with the greatest heat loss.

4. Keep down panel surface temperature for physiological reasons. (*Effect of Panel Location on Skin and Clothing Surface Temperature*, L. P. Herrington and R. J. Lorenzi—Heating, Piping and Air Conditioning, Journal Section, October 1949.) Panel surface temperatures are indicated by light face or bold face type in Design Table, page 65.

PANEL CONSTRUCTION

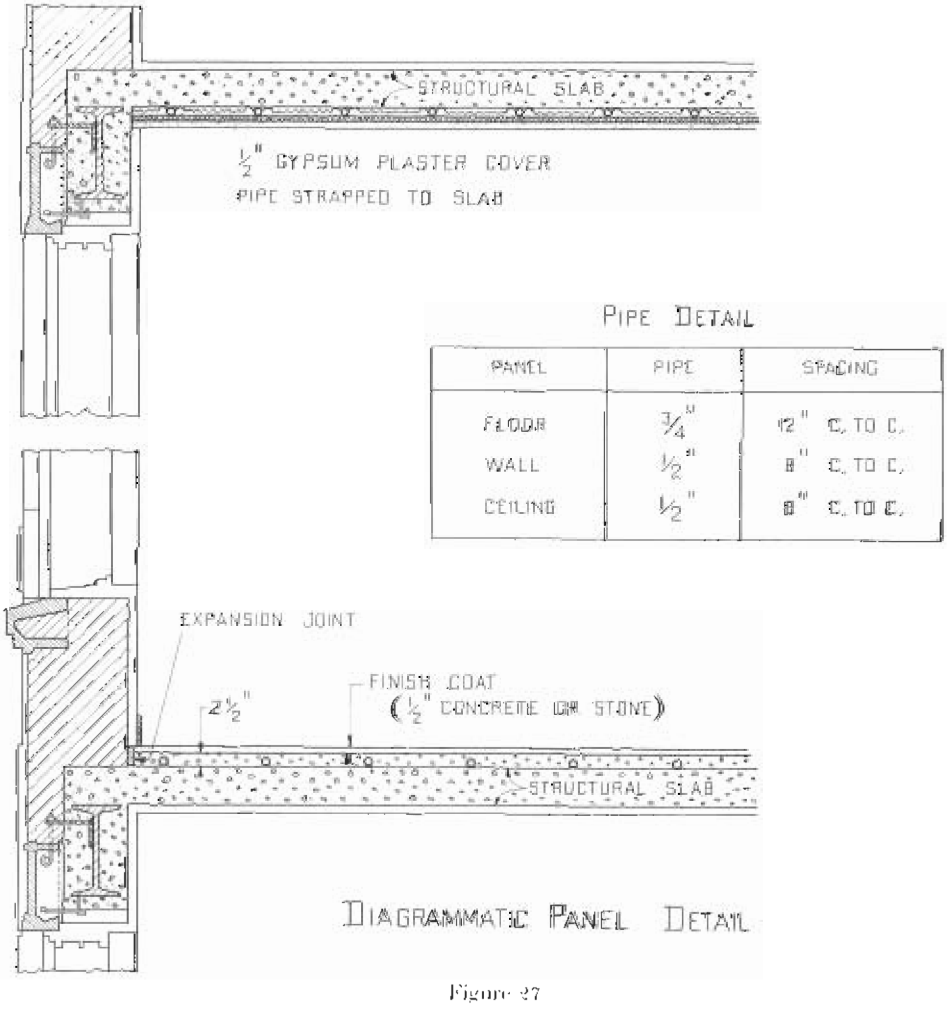
There are many ways of installing radiant panels, especially in floors. The two panels shown in Figure 27 below are representative of panels for multistory structures. For single-story structures, such as garages, hangars, etc., there are a few slight changes. For a floor panel in a single-story structure, it is important to have a solid slab of concrete beneath the pipe so that ground water cannot seep up and around the pipe. If economically possible, the subslab should be made of insulating concrete so that heat loss to the ground will be reduced. When a floor slab is to be used in a multistory building, there would be no need to have any insulation; the back losses from the panel would serve as a ceiling panel in the room below.

HYDRAULIC CALCULATIONS

In any hydraulic system it is necessary to determine hydraulic friction. In a panel heating system the hydraulic circuits must be balanced so that the water temperatures in the panels equal the water temperatures assumed on the DESIGN DATA-FORM.

The first step is to determine the flow in each circuit, (Column 3 of Table I, page 64). The required flow is set up for either a 10 F or 20 F drop in the circuit. For a 20 F drop one gpm equals 10,000 B/hr. Pipe number 1 in the table carries 1.76 gpm or 17,600 B/hr.

The second step is to determine the hydraulic



PANEL	PIPE	SPACING
FLOOR	¾"	12" C. TO C.
WALL	½"	8" C. TO C.
CEILING	½"	8" C. TO C.

Figure 27

resistance in each pipe when carrying the required volume of water (Column 6 of the Table). This is accomplished by determining the resistance in feet per unit length—from standard hydraulic tables—and the length of the coil.

The third step is to balance the circuits. This is explained in Columns 7 and 8. Groups of pipe are brought into several headers. The header is then put in series with a globe valve and each pipe is in series with a balance fitting, generally a square head cock. This piping arrangement is shown in Figure 26, page 60. Notice that pipe number 9 in the table has the greatest hydraulic friction with 37.5 feet, and pipe number 6 has the lowest resistance—9.1 feet of water, and yet when this system is in operation, all circuits will have the resistance of 37.5 feet of water. Pipe number 6 is controlled so that there are 2.2 feet of resistance on the balance valve and 26.2 feet on the header valve.

Maximum head in panel .... =37.5 feet
Head allowance for converter, fittings
and connecting pipe .... =12.5 feet
Total head .... =50.0 feet
Total required flow .... =39.8 gpm

TABLE I—HYDRAULIC CALCULATIONS

Pipe No.*	Header No.	Flow gpm	Resistance feet/100	Length feet	Total Resistance feet	Added Head	
						Header feet	Line feet
1	1	1.76	3.78	338	12.8	23.4	1.3
2	1	1.76	3.78	376	14.1		0.0
3	1	1.76	3.78	311	11.8		2.3
4	1	1.41	2.45	348	13.1		1.0
5	2	1.41	2.45	460	11.3	26.2	0.0
6	2	1.82	4.05	370	9.1		2.2
7	3	1.82	4.05	468	19.0	0.0	18.5
8	3	2.15	5.55	642	26.1		11.4
9	3	2.15	5.55	678	37.5		0.0
10	3	2.15	5.55	676	37.4		0.1
11	3	2.15	5.55	670	37.2		0.3
12	3	2.15	5.55	668	37.0		0.5
13	4	1.45	2.60	380	9.9	3.3	24.3
14	4	1.45	2.60	546	14.2		20.0
15	4	2.15	5.55	616	34.2		0.0
16	4	2.15	5.55	614	34.1		0.1
17	4	2.15	5.55	606	33.6		0.6
18	4	2.15	5.55	606	33.6		0.6
19	5	1.37	2.35	444	10.4	25.7	1.4
20	5	1.37	2.35	502	11.8		0.0
21	5	1.37	2.35	419	9.8		2.0
22	5	1.37	2.35	457	10.7		1.1
23	6	1.82	4.05	450	18.2	19.3	0.0
24	6	1.82	4.05	338	13.7		4.5

*See floor plan for key to pipe numbers, Figure 24, page 56.

TABLE II

Values for Comfort Constant, a Under Various Conditions
($a = t_a + MRT$)

Conditions	Comfort Constant, a
Foundry	98
Very Active Work	108
Gymnasiums	110
Light Manufacturing	128
Stores and Kitchens	130
Ballrooms	134
Class Rooms	138
Offices and Homes	140
Seated at Ease	142
Hospitals and Bathrooms	150
Swimming Pool	158
Reclining Clothed	162
Reclining Nude	170

TABLE III

Guide for Assumed Panel Areas

Exposed Surfaces	Per Cent of Floor or Ceiling Area Assumed as Panel
3 walls plus floor or ceiling	90–100
2 walls plus floor or ceiling	75–90
1 wall plus floor or ceiling	60–75
Floor or ceiling only	45–60

If assumed panel area is in error by 20% as compared to computed area, the error will not appreciably affect the design. If the error is in excess of 20%, then the calculation for U_c should be repeated using the first result as an estimate.

t_o = —10 F

Properties of Ceiling Panels

a = 140 F

PANEL TEMPERATURES

Light Face Type . . . . Up to 100 F
Bold Face Type . . . . 100 – 110 F

V_c	U_c		t_W = mean panel water temperature								
			100	110	115	120	125	130	135	140	145
0	0.05	u	.162	.127	.114	.104	.096	.088	**.082**	**.077**	**.072**
		q	21.6	28.8	32.4	36.0	39.6	43.2	**46.8**	**50.4**	**54.0**
	0.10	u	.269	.218	.199	.183	.169	.158	**.148**	**.139**	**.131**
		q	22.5	29.7	33.3	36.9	40.5	44.1	**47.7**	**51.3**	**54.9**
	0.15	u	.347	.289	.266	.247	.230	.216	.203	**.192**	**.182**
		q	23.2	30.4	34.0	37.6	41.2	44.8	48.4	**52.0**	**55.6**
	0.20	u	....	.358	.333	.311	.292	.276	.261	**.247**	**.235**
		q	....	30.9	34.5	38.1	41.7	45.3	48.9	**52.5**	**56.1**
	0.25	u	....	....	.385	.362	.341	.323	.307	**.292**	**.278**
		q	....	....	34.7	38.3	41.9	45.5	49.1	**52.7**	**56.3**
1	0.05	u	.204	.161	.146	.133	.123	.114	**.106**	**.099**	**.093**
		q	21.7	28.9	32.5	36.1	39.7	43.3	**46.9**	**50.5**	**54.1**
	0.10	u	.309	.253	.232	.214	.199	.186	**.174**	**.164**	**.155**
		q	22.6	29.8	33.4	37.0	40.6	44.2	**47.8**	**51.4**	**55.0**
	0.15	u	.380	.319	.295	.275	.257	.241	.228	**.215**	**.204**
		q	23.3	30.5	34.1	37.7	41.3	44.9	48.5	**52.1**	**55.7**
	0.20	u	....	.383	.357	.335	.315	.298	.282	**.268**	**.255**
		q	....	31.0	34.6	38.2	41.8	45.4	49.0	**52.6**	**56.2**
	0.25	u	....	....	....	.383	.362	.343	.326	**.311**	**.297**
		q	....	....	....	38.4	42.0	45.6	49.2	**52.8**	**56.4**
3	0.05	u	.291	.236	.216	.198	.184	.171	**.160**	**.151**	**.142**
		q	21.9	29.1	32.7	36.3	39.9	43.5	**47.1**	**50.7**	**54.3**
	0.10	u	.375	.342	.313	.290	.269	.251	.236	**.222**	**.210**
		q	22.8	30.0	33.6	37.2	40.8	44.4	48.0	**51.6**	**55.2**
	0.15	u	....	.376	.351	.328	.309	.291	.276	**.262**	**.249**
		q	....	30.7	34.3	37.9	41.5	45.1	48.7	**52.3**	**55.9**
	0.20	u	....	....	.401	.377	.356	.338	.321	**.306**	**.292**
		q	....	....	34.8	38.4	42.0	45.6	49.2	**52.8**	**56.4**
	0.25	u	....	....	....	....	.397	.378	.360	**.344**	**.329**
		q	....	....	....	....	42.2	45.8	49.4	**53.0**	**56.6**
7	0.05	u	....	.357	.331	.309	.289	.272	**.256**	**.243**	**.230**
		q	....	29.5	33.1	36.7	40.3	43.9	**47.5**	**51.1**	**54.7**
	0.10	u	....	....	.380	.356	.336	.317	.301	**.286**	**.272**
		q	....	....	34.0	37.6	41.2	44.8	48.4	**52.0**	**55.6**
	0.15	u	....	....	....	....	.392	.373	.355	**.339**	**.324**
		q	....	....	....	....	41.9	45.5	49.1	**52.7**	**56.3**
	0.20	u	....	....	....	....	....	....	.394	**.377**	**.362**
		q	....	....	....	....	....	....	49.6	**53.2**	**56.8**
	0.25	u	....	....	....	....	....	....	....	....	**.394**
		q	....	....	....	....	....	....	....	....	**57.0**

A_p = Panel Area = uA_t, sq. ft.

q = Panel Output, Btu per hr. per sq. ft.

GLOSSARY

TEMPERATURE TERMS

t_a = inside air temperature, F, generally assumed as $\frac{a}{2}$

t_p = panel temperature, F.

t_e = equivalent temperature of unheated surface areas, F.

t_o = outside air temperature, F.

t_w = mean water temperature in coils, F.

t_s = temperature of air entering room or the temperature of the air in adjacent room, F.

HEAT TRANSFER COEFFICIENTS

h_c = convection film coefficient, air to unheated areas, $Bhr^{-1}ft^{-2}F^{-1}$

h_p = convection film coefficient, panel to air, $Bhr^{-1}ft^{-2}F^{-1}$

h_r = radiation coefficient, panel to nonpanel areas, $Bhr^{-1}ft^{-2}F^{-1}$

C_e = equivalent conductance from nonpanel areas to outside air, $Bhr^{-1}ft^{-2}F^{-1}$

U_e = equivalent transmittance from inside air to outside air, $Bhr^{-1}ft^{-2}F^{-1}$

R = thermal resistance, pipe to panel surface, Fft^2hrB^{-1}

AREA TERMS

A_t = total surface area of room, ft²

A_p = surface area of panel, ft²

$A_e = A_t - A_p$, ft²

$u = A_p/A_t$, dimensionless

$r = 1 - u$, dimensionless

GENERAL TERMS

q = panel output, $Bhr^{-1}ft^{-2}$

a = comfort constant, F.

V = ventilation rate of air at t_s, CFH

$\Delta t' = t_a - t_s$, F.

$\Delta t = t_a - t_o$, F.

$\Delta t_g = t_a - t_g$ = air-to-air temperature difference with respect to glass area, F.

Similarly for Δt_c, Δt_w, Δt_f and Δt_i

$$V_e = \frac{V\Delta t'}{A_t\Delta t},\ ft^3hr^{-1}ft^{-2}$$

$\Delta t_v = \frac{\Delta t'}{\Delta t} = 1$ in all cases except where ventilation air is preheated.

SUBSCRIPTS

a = inside air

c = ceiling

e = equivalent

f = floor

g = glass

i = surface not losing heat

o = outside air

p = panel

t = total

w = water or exposed walls (see Δt_w above)

CONCLUSION

The work of many investigators has been reviewed to develop this straightforward, rational design procedure. Its four advantages are: (1) the system is thermally balanced in the design; (2) comfort conditions are considered in the analysis; (3) physiological limits are readily determined; and (4) it is based upon laboratory and theoretical analysis.

Of the four advantages only the first—thermal balance—is unique with this system. But it is this thermal balance characteristic that is an important addition to this system. It is rarely possible for a designer to control each panel independently, and yet this is necessary if the system is to have designed thermal balance. As can be seen from the example, the panels are designed according to the water temperature that will be in the panel. Yet this balance can be obtained from an inside air thermostat because when the comfort conditions are desirable, the water temperature will be as predicted (or determined) in the design. In other words, the balance lies in the design equations which are solved and tabulated for the designer.

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